Heating Ventilating Air Conditioning Guide 1952

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Air Conditioning
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HEATING VENTILATING AIR CONDITIONING GUIDE 1952

An Instrument of Service Prepared for the Profession containing

A TECHNICAL DATA SECTION OF REFERENCE MATERIAL ON THE DESIGN AND SPECIFICATION OF HEATING, VENTILATING AND AIR CONDITIONING SYSTEMS BASED ON—THE TRANSACTIONS—THE INVESTIGATIONS OF THE RESEARCH LABORATORY AND COOPERATING INSTITUTIONS—AND THE PRACTICE OF THE MEMBERS AND FRIENDS OF THE SOCIETY; A MANUFACTURERS' CATALOG DATA SECTION CONTAINING ESSENTIAL AND RELIABLE INFORMATION CONCERNING MODERN EQUIPMENT; COMPLETE INDEXES TO TECHNICAL AND CATALOG DATA SECTIONS.

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PREFACE TO THE 30th EDITION

The Heating Ventilating Air Conditioning Guide, 1952, shows an increase over previous editions in both usefulness and size. The Technical Data Section which comprises 1064 pages of technical and design information on 50 different subjects, represents an increase of 16 pages in space, in addition to an equal increase filling the equivalent of 16 pages derived from condensation of previous text.

The chapter arrangement of the last two editions has been retained, the chapters being grouped under the familiar section titles: I Fundamentals, II Human Reactions, III Heating and Cooling Loads, IV Combustion and Consumption of Fuels, V Systems and Equipment, VI Special Systems, VII Instruments and Codes.

While the entire book has been reviewed carefully and important revisions have been made throughout, special attention is called to the improvements in the following chapters:

Chapter 5—Heat Transfer. The tables of unit conductances for thermal convection and methods for computing radiant heat exchange for various conditions have been brought into agreement with latest research results.

Chapter 9—Heat Transmission Coefficients of Building Materials. The section on water vapor and condensation in building construction has been rewritten and enlarged. Particular attention has been given to the discussion of visible and concealed condensation and to methods of preventing moisture damage in buildings. A number of new materials are listed in the tables of conductivities and conductances, and revised values are shown for previously listed materials where new data have become available.

Chapter 11—Heating Load. Average winter temperatures for October to May were obtained from the *U. S. Weather Bureau* and the *Canadian Meteorological Service* and are listed in the table of Winter Climatic Conditions for 316 United States cities and 16 Canadian cities.

Chapter 12—Cooling Load. Design tables for heat gain through flat glass and glass block have been simplified, and design tables for figured rolled glass have been added.

Chapter 13—Fuels and Combustion. The sections dealing with oil and gas fuels have been rewritten and expanded to include more information on properties and combustion data for present day fuels.

Chapter 15—Heating Boilers, Furnaces, and Space Heaters. Boiler rating information has been brought up to date and a new abridged table showing current *I=B=R* boiler rating and sizing practice has been added.

Chapter 16—Chimneys and Draft Calculations. The section on residential chimneys has been rewritten with special emphasis on the performance and selection of low-height chimneys for basementless and other low buildings. The information is based on *National Bureau of Standards* test results. A new chart for determining gas appliance vent and chimney sizes is included to show recommendations of the *American Gas Association.*

Chapter 17—Estimating Fuel Consumption for Space Heating. The chapter was revised by the Technical Advisory Committee on Combustion. New constants were determined for use in fuel consumption formulas. All illustrative problems were reworked using the suggested constants.

Chapter 20—Steam Heating Systems. Several piping diagrams have been improved by changes which bring them up to date with recent practice in the industry.

Chapter 22—Radiators and Convectors. The information on baseboard type of radiation has been extended. Btu ratings have been added to the previous Equivalent Direct Radiation ratings of the currently manufactured sizes of radiators.

Chapter 24—Unit Heaters and Unit Ventilators. The unit ventilator section has been rewritten in conformity with present practice in rating and installation. Several unit heater piping diagrams have been revised.

Chapter 25—Unit Air Conditioners and Unit Air Coolers. Sections dealing with construction, ratings, operation and application of room coolers have been enlarged.

Chapter 32—Fans. A section has been added on application of fans for high temperature work. Nomenclature and designations for fans have been brought into accord with latest industry practice.

Chapter 33—Air Cleaning. The air filter section has been rewritten to provide more data on types of filters and dusts encountered. Information has been added describing the charged-media type of filter.

Chapter 36—Refrigeration. The section on the absorption system has been enlarged to include a diagram and description of the lithium bromide-water absorption system.

Chapter 44—Industrial Air Conditioning. The table showing recommended air conditions for manufacturing, storing, and handling various types of products has been greatly enlarged, and many important factors affecting the processes involved have been listed with the various products.

Chapter 50—Codes and Standards. The list of codes and standards has been enlarged to include new codes of interest, and latest editions of all codes have been indicated. Names and addresses of organizations which can supply the codes and standards are listed for convenience in obtaining copies.

A comprehensive cross index is included for the Technical Data Section.

As a convenience in finding available types of equipment, this volume contains a Catalog Data Section listing the names, addresses and brief descriptions of products of 277 manufacturers. The section is provided with an elaborate cross index.

In the preparation of this edition, the Guide Committee has been able to draw upon the resources of information represented by: the experiences of many Society members and other practicing engineers; the files of trade associations, government agencies, and various engineering publications; and the data collected and compiled by the Technical Advisory Committees and the Research Laboratory of the Society. It is impossible to give full credit to all of those who have shared in bringing THE GUIDE to its present position of usefulness to the heating ventilating air conditioning profession. To those who have contributed so much in the compiling of previous editions, THE GUIDE Committee is pleased to add the names of those who have been most active in preparing this volume:

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The 1952 edition of THE GUIDE is presented with confidence that it will serve to advance the arts and sciences of heating, ventilating, cooling, and air conditioning.

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CONTENTS

CONTENTS (CONCLUDED)

INDEX

Heating Ventilating
Air Conditioning
GUIDE 1952

●

TECHNICAL DATA SECTION

CHAPTERS 1–50 and PAGES 1–1064

●

Cross Reference to Subjects in Chapters 1–50 Alphabetically Listed

●

30th EDITION

INDEX

HEATING VENTILATING AIR CONDITIONING

GUIDE 1952

Technical Data Section

Chapters 1–50 and Pages 1–1064

Index to Technical Data Section

Index to Technical Data Section

Index to Technical Data Section

Index to Technical Data Section

Index to Technical Data Section

CHAPTER 1

TERMINOLOGY

Glossary of Physical and Heating, Ventilating, Refrigerating and Air Conditioning Terms Used in the Text

Absolute Zero: The zero from which absolute temperature is reckoned. Approximately −273.2 C or −459.8 F.

Absorbent: A sorbent which changes physically or chemically, or both, during the sorption process.

Absorption: The action of a material in extracting one or more substances present in an atmosphere or mixture of gases or liquids; accompanied by physical change, chemical change, or both, of the sorbent.

Acceleration: The time rate of change of velocity, *i.e.*, the derivative of velocity with respect to time. In the cgs system the unit of acceleration is the centimeter per (second) (second); in the fps system the unit is the foot per (second) (second), $a = \frac{dv}{dt}$.

Acceleration Due to Gravity: The rate of gain in velocity of a freely falling body, the value of which varies with latitude and elevation. The international gravity standard has the value of 980.665 cm per (sec) (sec) or 32.174 ft per (sec) (sec) which is the actual value of this acceleration at sea level and about 45 deg latitude.

Adiabatic: An adjective descriptive of a process such that no heat is added to, or taken from, a substance or system undergoing the process.

Adsorbent: A sorbent which does not change physically or chemically during the sorption process

Adsorption: The action, associated with surface adherence, of a material in extracting one or more substances present in an atmosphere or mixture of gases and liquids, unaccompanied by physical or chemical change. Commercial adsorbent materials have enormous internal surfaces.

Aerosol: An assemblage of small particles, solid or liquid, suspended in air. The diameters of the particles may vary from 100 microns down to 0.01 micron or less, *e.g.*, dust, fog, smoke.

Air Cleaner: A device designed for the purpose of removing airborne impurities such as dusts, gases, vapors, fumes and smokes. (Air cleaners include air washers, air filters, electrostatic precipitators and charcoal filters.)

Air Conditioning: The simultaneous control of all, or at least the first three, of those factors affecting both the physical and chemical conditions of the atmosphere within any structure. These factors include temperature, humidity, motion, distribution, dust, bacteria, odors and toxic gases, most of which affect in greater or lesser degree human health or comfort. (*See Comfort Air Conditioning.*)

Air, Dry: In psychrometry, air unmixed with, or containing no, water vapor.

Air, Saturated: A mixture of dry air and saturated water vapor, all at the same dry-bulb temperature.

Air, Standard: Air with a density of 0.075 lb per cu ft and an absolute viscosity of 1.22×10^{-5} lb mass per (ft) (sec). This is substantially equivalent to dry air at 70 F and 29.92 in. (Hg) barometer.

Air Washer: An enclosure in which air is drawn or forced through a spray of water in order to cleanse, humidify, or dehumidify the air.

Anemometer: An instrument for measuring the velocity of a fluid.

Aspect Ratio: In air distribution outlets, the ratio of the length of the core of a grille, face or register to the width.

In rectangular ducts, the ratio of the width to the depth.

Atmospheric Pressure: The pressure due to the weight of the atmosphere. It is the pressure indicated by a barometer. *Standard Atmospheric Pressure* or *Standard Atmosphere* is the pressure of 76 cm of mercury having a density of 13.5951 grams per cu cm, under standard gravity of 980.665 cm per (sec) (sec). It is equivalent to 14.696 lb psi or 29.921 in. of mercury at 32 F.

Baffle: A surface used for deflecting fluids, usually in the form of a plate or wall.

from the room. Such a device transfers its heat to the room largely by convection air currents.

Radiator, Direct: Same as Radiator.

Radiator, Recessed: A heating unit set back into a wall recess, but not enclosed.

Radiator, Tube or Tubular: A heating unit used as a radiator in which the heat transfer surfaces are principally tubes.

Refrigerant: A substance which produces a refrigerating effect by its absorption of heat while expanding or vaporizing.

Refrigeration, Ton of: The removal of heat at a rate of 200 Btu per min, 12,000 Btu per hr, or 288,000 Btu per 24 hr.

Resistance, Thermal: The reciprocal of thermal conductance. Symbol R.

Resistivity, Thermal: The reciprocal of thermal conductivity. Symbol r.

Resistor, Electric: A material used to produce heat by passing an electric current through it.

Return, Dry: A return pipe in a steam heating system which carries both water of condensation and air. The dry return is above the level of the water line in the boiler in a gravity system. (See *Return, Wet.*)

Return, Wet: That part of a return main of a steam heating system which is filled with water of condensation. The wet return usually is below the level of the water line in the boiler, although not necessarily so. (See *Return, Dry.*)

Return Mains: Pipes or conduits which return the heating or cooling medium from the heat transfer unit to the source of heat or refrigeration.

Reversed-Return System: A system in which the heating or cooling medium from several heat transfer units is returned along paths arranged so that all circuits composing the system or composing a major sub-division of it are of practically equal length.

Sabin: A unit of equivalent sound absorption equal to the equivalent absorption of one square foot of a surface of unit absorptivity (*i.e.*, of one square foot of surface which absorbs all incident sound energy).

Saturation: The condition for co-existence in stable equilibrium of a vapor and liquid or a vapor and solid phase of the same substance. Example: Steam over the water from which it is being generated.

Saturation, Degree of: The ratio of the weight of water vapor associated with a pound of dry air to the weight of water vapor associated with a pound of dry air saturated at the same temperature.

Smoke: An air suspension (aerosol) of particles, usually but not necessarily solid, often originating in a solid nucleus, formed from combustion or sublimation. Also defined as carbon or soot particles less than 0.1 micron in size which result from the incomplete combustion of carbonaceous materials such as coal, oil, tar, and tobacco.

Smokeless Arch: An inverted baffle placed in an up-draft furnace toward the rear to aid in mixing the gases of combustion, and thereby to reduce the smoke produced.

Solar Constant: The solar intensity incident on a normal surface located outside the earth's atmosphere at a distance from the sun equal to the mean distance between the earth and the sun. Its value is 415, 445, or 430 Btu per (hr) (sq ft) as the July, January, or mean value, respectively. At sea level in July the solar intensity value is about 300 Btu per (sq ft) (hr) since about 28 percent is absorbed in the earth's atmosphere.

Sorbent: A material which extracts one or more substances present in an atmosphere or mixture of gases or liquids with which it is in contact, due to an affinity for such substances.

Sorption: Adsorption or absorption.

Split System: A system in which the heating is accomplished by means of radiators or convectors supplemented by mechanical circulation of air (heated or unheated) from a central point. Ventilation may be provided by the same system.

Square Foot of Heating Surface (*Equivalent*)**:** This term is synonymous with Equivalent Direct Radiation (EDR).

Stack Height: The height of a gravity convector between the bottom of the heating unit and the top of the outlet opening.

Steam: Water in the vapor phase. *Dry Saturated Steam* is steam at the saturation temperature corresponding to the pressure, and containing no water in suspension. *Wet Saturated Steam* is steam at the saturation temperature corresponding to the pressure, and containing water particles in suspension. *Superheated Steam* is steam at a temperature higher than the saturation temperature corresponding to the pressure.

Steam Heating System: A heating system in which heat is transferred from the boiler or other source of heat to the heating units by means of steam at, above, or below atmospheric pressure.

Steam Trap: A device for allowing the passage of condensate, or air and condensate, and preventing the passage of steam.

Supply Mains: The pipes through which the heating medium flows from the boiler or source of supply to the run-outs and risers leading to the heating units.

Surface, Heating: The exterior surface of a heating unit. *Extended heating surface (or extended surface)*: Heating surface consisting of fins, pins or ribs which receive heat by conduction from the prime surface. *Prime Surface*: Heating surface having the heating medium on one side and air (or extended surface) on the other. (See also Boiler Heating Surface.)

Temperature: The thermal state of matter with reference to its tendency to communicate heat to matter in contact with it. If no heat flows upon contact, there is no difference in temperature.

Temperature, Absolute: Temperature expressed in degrees above absolute zero.

Temperature, Dew-Point: The temperature at which the condensation of water vapor in a space begins for a given state of humidity and pressure as the temperature of the vapor is reduced. The temperature corresponding to saturation (100 percent relative humidity) for a given absolute humidity at constant pressure.

Temperature, Dry-Bulb: The temperature of a gas or mixture of gases indicated by an accurate thermometer after correction for radiation.

Temperature, Effective: An arbitrary index which combines into a single value the effect of temperature, humidity, and air movement on the sensation of warmth or cold felt by the human body. The numerical value is that of the temperature of still, saturated air which would induce an identical sensation.

Temperature, Mean Radiant (MRT): The temperature of a uniform black enclosure in which a solid body or occupant would exchange the same amount of radiant heat as in the existing nonuniform environment.

Temperature, Wet-Bulb: Thermodynamic wet-bulb temperature is the temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature. Wet-bulb temperature (without qualification) is the temperature indicated by a wet-bulb psychrometer constructed and used according to specifications. (*ASME* Power Test Codes, Series 1932, Instruments and Apparatus, Part 18.)

Therm: A quantity of heat equivalent to 100,000 Btu.

Thermodynamics, Laws of: Two laws upon which rest the classical theory of thermodynamics. These laws have been stated in many different, but equivalent ways. *The First Law*: (1) When work is expended in generating heat, the quantity of heat produced is proportional to the work expended; and conversely, when heat is employed in the performance of work, the quantity of heat which disappears is proportional to the work done. (Joule)[a] (G.P.)[b]; (2) If a system is caused to change from an initial state to a final state by adiabatic means only, the work done is the same for all adiabatic paths connecting the two states. (Zemansky); (3) In any power cycle or refrigeration cycle the net heat absorbed by the working substance is exactly equal to the net work done. *The Second Law*: (1) It is impossible for a self-acting machine, unaided by any external agency, to convey heat from a body of lower to one of higher temperature. (Clausius) (G.P.); (2) It is impossible to derive mechanical work from heat taken from a body unless there is available a body of lower temperature into which the residue not so used may be discharged (Kelvin) (G.P.); (3) It is impossible to construct an engine that, operating in a cycle, will produce no effect other than the extraction of heat from a reservoir and the performance of an equivalent amount of work (Zemansky).

Thermostat: An instrument which responds to changes in temperature, and which directly or indirectly controls temperature.

Transmittance, Thermal: The time rate of heat flow, from the fluid on the warm side to the fluid on the cold side, per (square foot) (degree temperature difference between the two fluids). Sometimes called *Overall Coefficient of Heat Transfer*.

Common unit is Btu per (hour) (square foot) (Fahrenheit degree). Symbol U.

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[a] Names of authors who first stated laws are given in parentheses.
[b] From Glossary of Physics, by LeRoy Dougherty Weld (McGraw-Hill, 1937).

Two-Pipe System (*Steam or Water*): A heating system in which one pipe is used for the supply of the heating medium to the heating unit, and another for the return of the heating medium to the source of heat supply. The essential feature of a two-pipe system is that each heating unit receives a direct supply of the heating medium, which medium cannot have served a preceding heating unit.

Up-Feed System: A heating system in which the supply mains are below the level of the heating units which they serve.

Vacuum Heating System: A two-pipe steam heating system equipped with the necessary accessory apparatus which will permit operating the system below atmospheric pressure when desired.

Vane Ratio: In air distributing devices the ratio of depth of vane to shortest opening width between two adjacent grille bars.

Vapor: The gaseous form of substances which are normally in the solid or liquid state, and which can be changed to these states either by increasing the pressure or decreasing the temperature. Vapors diffuse. (*ASA* definition.)

Vapor Heating System: A steam heating system which operates under pressures at or near atmospheric and which returns the condensate to the boiler or receiver by gravity. Vapor systems have thermostatic traps or other means of resistance on the return ends of the heating units for preventing steam from entering the return mains; they also have a pressure-equalizing and air-eliminating device at the end of the dry-return.

Velocity: A vector quantity which denotes at once the time rate and the direction of a linear motion. $V = \frac{ds}{dt}$. For uniform linear motion $V = \frac{s}{t}$. Common units are: feet per second.

Ventilation: The process of supplying or removing air, by natural or mechanical means, to or from any space. Such air may or may not have been conditioned. (See *Air Conditioning*.)

Viscosity: That property of semi-fluids, fluids and gases by virtue of which they resist an instantaneous change of shape or arrangement of parts. It is the cause of fluid friction whenever adjacent layers of fluid move with relation to each other.

The coefficient of viscosity is the resistance offered by a layer of the fluid of unit area to the motion parallel to this area of another layer of the fluid at unit distance moving with unit velocity relative to the first layer. This coefficient is known as the *absolute viscosity*, and in cgs units is the force in dynes per square centimeter at a velocity of 1 cm per second at a distance of 1 cm. This unit of absolute viscosity is the *poise*. Values of absolute viscosity are frequently listed in centipoises, a centipoise being 1/100 of 1 *poise*.

In many formulas *absolute viscosity* is expressed in pounds per foot second or pounds per foot hour. Conversion from centipoises may be made as follows: viscosity in centipoises $\times$ 0.000672 = viscosity in pounds per foot second, or viscosity in centipoises $\times$ 2.42 = viscosity in pounds per foot hour.

Kinematic viscosity is the ratio of *absolute viscosity* to the density of a fluid. The unit is the stoke which equals one poise per cu cm per gram, or 1 sq cm per sec. For conversion to English units: kinematic viscosity in stokes $\times$ 0.001076 = kinematic viscosity in square feet per second.

Volume, Specific: The volume of a substance per unit mass; the reciprocal of density. Units: cubic feet per pound, cubic centimeters per gram, etc.

Warm Air Heating System: A warm air heating plant consists of a heating unit (fuel-burning furnace) enclosed in a casing, from which the heated air is distributed to various rooms of the building through ducts.

Warm Air Heating System, Forced: A warm air heating system in which circulation of air is effected by a fan. Such a system may include air cleaning devices.

Warm Air Heating System, Gravity: A warm air heating system in which the motive head producing flow depends on the difference in weight between the heated air leaving the casing and the cooler air entering the bottom of the casing.

Warm Air Heating System, Perimeter: A warm air heating system of the combination panel and convection type. Warm air ducts imbedded in the concrete slab of a basementless house, around the perimeter, receive heated air from a furnace and deliver it to the heated space through registers placed in or near the floor. Air is returned to the furnace from registers near the ceiling.

CHAPTER 2

ABBREVIATIONS AND SYMBOLS

Standard Abbreviations; Standard Symbols; Greek Alphabet; Conversion Equations; Graphical Symbols for Piping, Ductwork, Heating and Ventilating, Refrigerating; Identification of Piping by Color

THIS chapter contains information regarding abbreviations, symbols, and conversion equations, which are of particular interest to the engineer engaged in heating, ventilating, and air conditioning.

ABBREVIATIONS

Abbreviations are shortened forms of names and expressions employed in texts and tabulations, and should not generally be used as symbols in equations. Most of the following abbreviations have been compiled from a list of approved standards.[1] In general, the period has been omitted in all abbreviations, except where the omission results in the formation of an English word. Additional abbreviations applying to individual chapters will be found at the end of Chapters 3, 4, 12, 31, 35, and 46.

Absolute	abs
Air horsepower	air hp
Alternating-current (as adjective)	a-c
Ampere	amp
Ampere-hour	amp-hr
Atmosphere	atm
Average	avg
Avoirdupois	avdp
Barometer	bar.
Boiling point	bp
Brake horsepower	bhp
Brake horsepower-hour	bhp-hr
British thermal unit	Btu
British thermal units per hour	Btuh
Calorie	cal
Centigram	cg
Centimeter	cm
Centimeter-gram-second (system)	cgs
Cubic	cu
Cubic centimeter	cu cm or cc
Cubic foot	cu ft
Cubic feet per minute	cfm
Cubic feet per second	cfs
Decibel	db
Degree[2]	deg or °
Degree, Centigrade	C
Degree, Fahrenheit	F
Degree, Kelvin	K
Degree, Réaumur	R
Diameter	diam

[1] Abbreviations for Scientific and Engineering Terms, Z10.1-1941 (*American Standards Association*).

[2] It is recommended that the abbreviation for the temperature scale, F, C, K, R, be included in expressions for numerical temperatures but, wherever feasible, the abbreviation for *degree* be omitted, *e.g.*, 68 F.

Direct-current (as adjective)	d-c
Electromotive Force	emf
Feet per minute	fpm
Feet per second	fps
Foot	ft
Foot-pound	ft-lb
Foot-pound-second (system)	fps
Freezing point	fp
Gallon	gal
Gallons per minute	gpm
Gallons per second	gps
Gram	g
Gram-calorie	g-cal
Horsepower	hp
Horsepower-hour	hp-hr
Hour	hr
Inch	in.
Inch-pound	in.-lb
Indicated horsepower	ihp
Indicated horsepower-hour	ihp-hr
Kilogram	kg
Kilowatt	kw
Kilowatt-hour	kwhr
Mass	mass
Melting point	mp
Meter	m
Micron	μ (mu)
Miles per hour	mph
Millimeter	mm
Minute	min
Molecular weight	mol. wt
Mol	mol
Ounce	oz
Pound	lb
Pounds per square inch	psi
Pounds per square inch, gage	psig
Pounds per square inch, absolute	psia
Revolutions per minute	rpm
Revolutions per second	rps
Second	sec
Specific gravity	sp gr
Specific heat	sp ht
Square foot	sq ft
Square inch	sq in.
Watt	w
Watthour	whr

SYMBOLS

A letter symbol is a single character, with subscript or superscript if required, used to designate a physical magnitude in mathematical equations and expressions. Two or more symbols together always represent a product. The following have been compiled from a selected list of approved standards.[3] Additional symbols and variations in the standard symbols

[3] Letter Symbols for Mechanics of Solid Bodies, Z10.3-1942, and Letter Symbols for Heat and Thermodynamics, Z10.4-1943 (*American Standards Association*).

found necessary in the individual chapters will be found in a list at the end of Chapters 3, 4, 12, 31, 35, and 46.

Acceleration, due to gravity ... g
Acceleration, linear ... a
Area ... A
Change in specific volume during vaporization ... v_{fg}
Density, Weight per unit volume, Specific weight ... d or ρ (rho)

$$d = \frac{1}{v}$$

Distance, linear ... s
Dry saturated vapor, Dry saturated gas at saturation pressure and temperature, vapor in contact with liquid ... *Subscript* g

Efficiency ... η
Elevation above some datum ... z, Z
Emissivity ... ϵ
Energy in general; work, total; work, molal ... E
Entropy. (The capital should be used for any weight, and the small letter for unit weight) ... S or s

Force, total load ... F
Gas Constant, in equation $pV = nRT$ ... R
Head ... H or h
Heat content, Total heat, Enthalpy. (The capital should be used for any weight and the small letter for unit weight) ... H or h
Heat content of saturated liquid, Total heat of saturated liquid, Enthalpy of saturated liquid, sometimes called heat of the liquid ... h_f

Heat content of dry saturated vapor, Total heat of dry saturated vapor, Enthalpy of dry saturated vapor ... h_g
Heat of vaporization at constant pressure ... L or h_{fg}
Hydraulic radius ... R_H
Internal energy, Intrinsic energy. (The capital should be used for any weight and the small letter for unit weight) ... U or u
Length of path of heat flow, thickness ... L

Load, total ... W
Mechanical efficiency ... e_m
Mechanical equivalent of heat ... J
Power, Horsepower, Work per unit time ... P
Pressure, Absolute pressure, Gage pressure, Force per unit area ... p

Quantity (total) of fluid, water, gas, heat; Quantity by volume; Total quantity of heat transferred ... Q
Quality of steam, Pounds of dry steam per pound of mixture ... x
Reynolds Number ... N_{Re}
Saturated liquid at saturation pressure and temperature, Liquid in contact with vapor ... *Subscript* f
Specific heat ... c

Specific heat at constant pressure ... c_p
Specific heat at constant volume ... c_v
Specific volume, Volume per unit weight, Volume per unit mass ... v
Temperature (ordinary) F or C. (*Theta* is used preferably only when t is used for Time in the same discussion) ... t or θ (*theta*)
Temperature (absolute) F abs or K. (Capital *theta* is used preferably only when small *theta* is used for ordinary temperature) ... T or Θ (*capital theta*)
Thermal conductance:[4] heat transferred per (unit time) (degree) ... C

$$C = \frac{1}{R} = \frac{kA}{L} = \frac{q}{t_1 - t_2}$$

[4] Terms ending *ivity* designate properties independent of size or shape, sometimes called *specific properties*. Examples: conductivity, resistivity. Terms ending *ance* designate quantities depending not only on the material, but also upon size and shape, sometimes called *total quantities*. Examples: conductance, transmittance. Terms ending *ion* designate rate of heat transfer. Examples: conduction, transmission.

Thermal conductance per unit area, Unit conductance: heat transferred per (unit time) (unit area) (degree) . . . . . C_a

$$C_a = \frac{C}{A} = \frac{1}{RA} = \frac{q}{A(t_1 - t_2)} = \frac{k}{L}$$

Thermal conductivity: heat transferred per (unit time) (unit area) (degree per unit length) . . . . . k

$$k = \frac{\frac{q}{A}}{\frac{(t_1 - t_2)}{L}}$$

Surface coefficient of heat transfer, Film coefficient of heat transfer, Individual coefficient of heat transfer: heat transferred per (unit time) (unit area) (degree) . . . . . f

$$f = \frac{\frac{q}{A}}{t_1 - t_2}$$

(In general f is not equal to k/L, where L is the actual thickness of the fluid film.)

Overall coefficient of heat transfer, Thermal transmittance per unit area: heat transferred per (unit time) (unit area) (degree overall) . . . . . U

$$U = \frac{\frac{q}{A}}{t_1 - t_2}$$

Thermal transmission (heat transferred per unit time) . . . . . q

$$q = \frac{Q}{t}$$

Thermal resistance (degree per unit of heat transferred per unit time) . . . . . R

$$R = \frac{t_1 - t_2}{q} = \frac{L}{kA}$$

Thermal resistivity . . . . . $1/k$

Vaporization values at constant pressure, Differences between values for saturated vapor and saturated liquid at the same pressure . . . . . *Subscript* fg

Velocity V

Viscosity, absolute . . . . . μ

Viscosity, kinematic . . . . . μ/ρ

Volume (total) V

Volume per unit time, Rate at which quantity of material passes through a machine, Quantity of heat per unit time, Quantity of heat per unit weight . . . . . q

Weight of a major item, Total weight . . . . . W

Weight rate, Weight per unit of power, Weight per unit of time . . . . . w

Work (total) . . . . . W

THE GREEK ALPHABET

Α α	Alpha	Ι ι	Iota	Ρ ρ	Rho
Β β	Beta	Κ κ	Kappa	Σ σ ς	Sigma
Γ γ	Gamma	Λ λ	Lambda	Τ τ	Tau
Δ δ	Delta	Μ μ	Mu	Υ υ	Upsilon
Ε ε	Epsilon	Ν ν	Nu	Φ φ ϕ	Phi
Ζ ζ	Zeta	Ξ ξ	Xi	Χ χ	Chi
Η η	Eta	Ο ο	Omicron	Ψ ψ	Psi
Θ ϑ θ	Theta	Π π	Pi	Ω ω	Omega

CONVERSION EQUATIONS[5]

Heat, Power and Work

1 ton refrigeration	= 12,000 Btu per hour 200 Btu per minute
Latent heat of ice	= 143.4 Btu per pound
1 Btu	= 778.3 ft-lb 0.2930 Int. whr 252.0 I.T. calorie
1 Int. watthour	= 2656 ft-lb 3.413 Btu 3600 Int. joules 860 I.T. calories
1 Int. kilowatthour	= 3,413 Btu 3.517 lb water evaporated from and at 212 F
1 Int. kilowatt (1000 watts)	= 1.341 hp 56.88 Btu per minute 44,267 ft-lb per minute
1000 I.T. calories 1 I.T. Kilocalorie	= 3.968 Btu 3088 ft-lb 1.1628 Int. whr
1 horsepower	= 0.7455 Int. kw 42.40 Btu per minute 33,000 ft-lb per minute 550 ft-lb per second
1 boiler horsepower	= 33,475 Btu per hour 9.809 Int. kw

Weight and Volume

1 gal (U. S.)	= 231 cu in. 0.1337 cu ft
1 British or Imperial gallon	= 277.42 cu in.
1 cu ft	= 7.481 gal 1728 cu in.
1 cu ft water at 60 F (in vacuo)	= 62.37 lb
1 cu ft water at 212 F (" ")	= 59.83 lb
1 gal water at 60 F (" ")	= 8.338 lb
1 gal water at 212 F (" ")	= 7.998 lb
1 lb (avdp)	= 16 oz 7000 grains
1 bushel	= 1.244 cu ft
1 short ton	= 2000 lb

Pressure

1 lb per square inch	= 144 lb per square foot 2.0360 in. mercury at 32 F 2.0422 in. mercury at 62 F 2.309 ft water at 62 F 27.71 in. water at 62 F
1 oz per square inch	= 0.1276 in. mercury at 62 F 1.732 in. water at 62 F

[5] Checked in 1944 by *National Bureau of Standards*. Abbreviations *Int.* and *I.T.* refer to *International* and *International* (*Steam*) *Table*, respectively.

1 atmosphere	= 14.696 lb per square inch 2116 lb per square foot 33.94 ft water at 62 F 30.01 in. mercury at 62 F 29.921 in. mercury at 32 F
1 in. water at 62 F (in vacuo)	= 0.03609 lb per square inch 0.5774 oz per square inch 5.197 lb per square foot
1 ft. water at 62 F (in vacuo)	= 0.4330 lb per square inch 62.37 lb per square foot
1 in. mercury at 62 F (in vacuo)	= 0.4897 lb per square inch 7.835 oz per square inch 1.131 ft water at 62 F 13.57 in. water at 62 F
1 in. mercury at 32 F (in vacuo)	= 0.49115 lb per square inch

Metric Units

1 cm	= 0.3937 in. = 0.0328 ft
1 in.	= 2.540 cm
1 m	= 3.281 ft
1 ft	= 0.3048 m
1 sq cm	= 0.1550 sq in.
1 sq in.	= 6.452 sq cm
1 sq m	= 10.76 sq ft
1 sq ft	= 0.09290 sq m
1 cu cm	= 0.06102 cu in.
1 cu in.	= 16.39 cu cm
1 cu m	= 35.31 cu ft
1 cu ft	= 0.02832 cu m
1 liter	= 1000 cu cm = 0.2642 gal
1 kg	= 2.205 lb (avdp)
1 lb	= 0.4536 kg
1 metric ton	= 2205 lb (avdp)
1 gram	= 0.002205 lb (avdp)
1 kilometer per hour	= 0.6214 mph
1 gram per square centimeter	= 0.02905 in. mercury at 62 F 0.3942 in. water at 62 F
1 kg per sq cm (metric atmosphere)	= 14.22 lb per square inch
1 gram per cubic centimeter	= 0.03613 lb per cubic inch 62.43 lb per cubic foot
1 dyne	= 0.00007233 poundals
1 absolute joule	= 10,000,000 ergs 0.7376 ft-lb
1 Int. joule	= 0.7378 ft-lb
1 metric horsepower	= 75 kg-m per second 0.986 hp (U. S.)
1 I.T. kilocalorie per kilogram	= 1.8 Btu per pound
1 I.T. calorie per square centimeter	= 3.687 Btu per square foot
1 I.T. calorie per (second) (square centimeter) for a temperature gradient of 1 C deg per centimeter	= 2903 BTU per (hour) (square foot) for a temperature gradient of 1 F deg per inch of thickness.

GRAPHICAL SYMBOLS FOR DRAWINGS[6]

Graphical Symbols for Drawings — *Piping*

HEATING

1. High Pressure Steam
2. Medium Pressure Steam
3. Low Pressure Steam
4. High Pressure Return
5. Medium Pressure Return
6. Low Pressure Return
7. Boiler Blow Off
8. Condensate or Vacuum Pump Discharge
9. Feedwater Pump Discharge
10. Make-Up Water
11. Air Relief Line
12. Fuel Oil Flow
13. Fuel Oil Return
14. Fuel Oil Tank Vent
15. Compressed Air
16. Hot Water Heating Supply
17. Hot Water Heating Return

AIR CONDITIONING

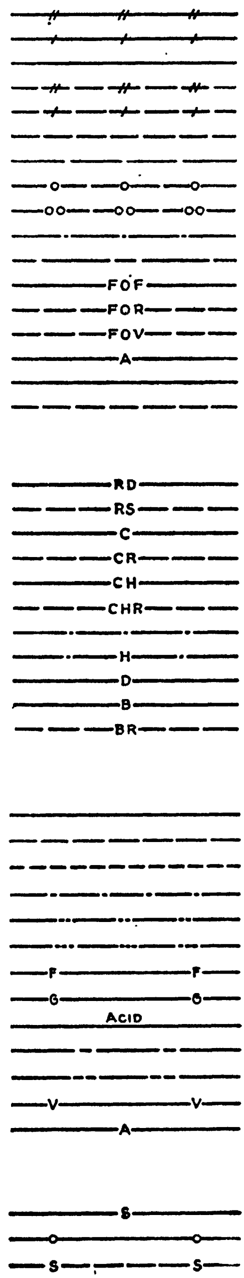
18. Refrigerant Discharge
19. Refrigerant Suction
20. Condenser Water Flow
21. Condenser Water Return
22. Circulating Chilled or Hot Water Flow
23. Circulating Chilled or Hot Water Return
24. Make-Up Water
25. Humidification Line
26. Drain
27. Brine Supply
28. Brine Return

PLUMBING

29. Soil, Waste or Leader (Above Grade)
30. Soil, Waste or Leader (Below Grade)
31. Vent
32. Cold Water
33. Hot Water
34. Hot Water Return
35. Fire Line
36. Gas
37. Acid Waste
38. Drinking Water Flow
39. Drinking Water Return
40. Vacuum Cleaning
41. Compressed Air

SPRINKLERS

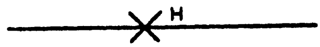
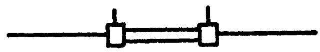
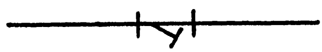
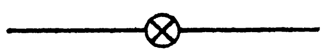
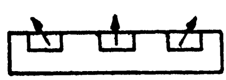
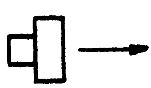
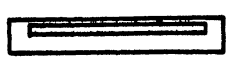
42. Main Supplies
43. Branch and Head
44. Drain



[6] Extracted from: American Standard Graphical Symbols for Pipe Fittings, Valves, and Piping (*ASA* Z32.2.3-1949) and American Standard Graphical Symbols for Heating, Ventilating, and Air Conditioning (*ASA* Z32.2.4-1949) with the permission of the publisher, *The American Society of Mechanical Engineers*, 29 West 39th St., New York 18, N. Y.

Graphical Symbols for Drawings

Heating

1. Air Eliminator	
2. Anchor	
3. Expansion Joint	
4. Hanger or Support	
5. Heat Exchanger	
6. Heat Transfer Surface, Plan (Indicate Type Such as Convector)	
7. Pump (Indicate Type Such as Vacuum)	
8. Strainer	
9. Tank (Designate Type)	
10. Thermometer	
11. Thermostat	
12. Traps	
12.1 Boiler Return	
12.2 Blast Thermostatic	
12.3 Float	
12.4 Float and Thermostatic	
12.5 Thermostatic	
13. Unit Heater (Centrifugal Fan), Plan	
14. Unit Heater (Propeller), Plan	
15. Unit Ventilator, Plan	

Graphical Symbols for Drawings *Heating*

16. Valves

16.1 Check

16.2 Diaphragm

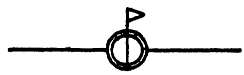
16.3 Gate

16.4 Globe

16.5 Lock and Shield

16.6 Motor Operated

16.7 Reducing Pressure

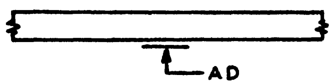


16.8 Relief (Either Pressure or Vacuum)

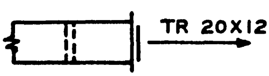
17. Vent Point

Graphical Symbols for Drawings *Ventilating*

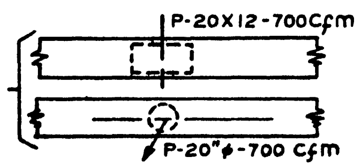
18. Access Door



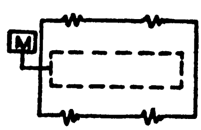
19. Adjustable Blank Off



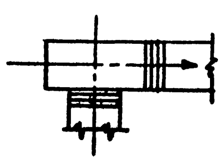
20. Adjustable Plaque



21. Automatic Dampers



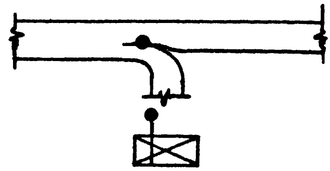
22. Canvas Connections



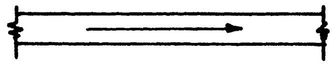
Graphical Symbols for Drawings

Ventilating

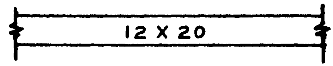
23. Deflecting Damper



24. Direction of Flow

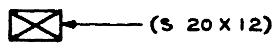


25. Duct (1st Figure, Side Shown; 2nd Side not Shown)

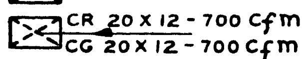


26. Duct Section (Exhaust or Return)

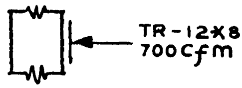
27. Duct Section (Supply)



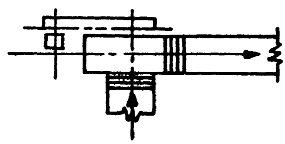
28. Exhaust Inlet Ceiling (Indicate Type)



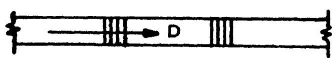
29. Exhaust Inlet Wall (Indicate Type)



30. Fan and Motor With Belt Guard

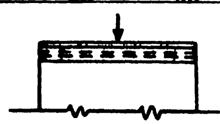


31. Inclined Drop in Respect to Air Flow

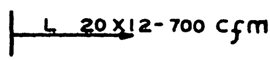


32. Inclined Rise in Respect to Air Flow

33. Intake Louvers on Screen

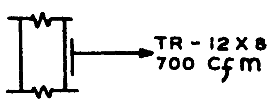


34. Louver Opening

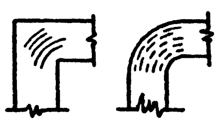


35. Supply Outlet Ceiling (Indicate Type)

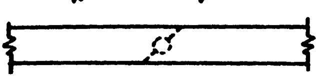
36. Supply Outlet Wall (Indicate Type)



37. Vanes



38. Volume Damper



Graphical Symbols for Drawings

Air Conditioning

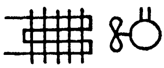
39. Capillary Tube 

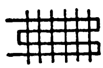
40. Compressor 

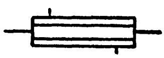
41. Compressor, Enclosed, Crankcase, Rotary, Belted 

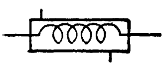
42. Compressor, Open Crankcase, Reciprocating, Belted 

43. Compressor, Open Crankcase, Reciprocating, Direct Drive 

44. Condenser, Air Cooled, Finned, Forced Air 

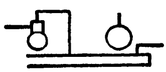
45. Condenser, Air Cooled, Finned, Static 

46. Condenser, Water Cooled, Concentric Tube in a Tube 

47. Condenser, Water Cooled, Shell and Coil 

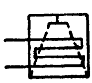
48. Condenser, Water Cooled, Shell and Tube 

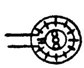
49. Condensing Unit, Air Cooled 

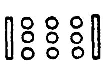
50. Condensing Unit, Water Cooled 

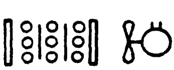
51. Cooling Tower 

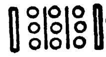
52. Dryer 

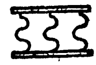
53. Evaporative Condenser 

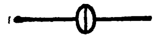
54. Evaporator, Circular, Ceiling Type, Finned 

55. Evaporator, Manifolded, Bare Tube, Gravity Air 

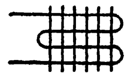
56. Evaporator, Manifolded, Finned, Forced Air 

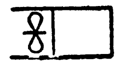
57. Evaporator Manifolded, Finned, Gravity Air 

58. Evaporator, Plate Coils, Headered or Manifold 

59. Filter, Line 

60. Filter & Strainer, Line 

61. Finned Type Cooling Unit, Natural Convection 

62. Forced Convection Cooling Unit 

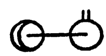
63. Gage 

64. High Side Float 

65. Immersion Cooling Unit 

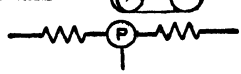
66. Low Side Float 

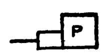
67. Motor-Compressor, Enclosed Crankcase, Reciprocating, Direct Connected 

68. Motor-Compressor, Enclosed Crankcase, Rotary, Direct Connected 

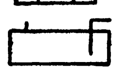
69. Motor-Compressor, Sealed Crankcase, Reciprocating 

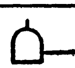
70. Motor-Compressor, Sealed Crankcase, Rotary 

71. Pressurestat 

72. Pressure Switch 

73. Pressure Switch With High Pressure Cut-Out 

74. Receiver, Horizontal 

75. Receiver, Vertical 

76. Scale Trap
77. Spray Pond
78. Thermal Bulb
79. Thermostat (Remote Bulb)
80. Valves
80.1 Automatic Expansion
80.2 Compressor Suction Pressure Limiting, Throttling Type (Compressor Side)
80.3 Constant Pressure, Suction
80.4 Evaporator Pressure Regulating, Snap Action
80.5 Evaporator Pressure Regulating, Thermostatic Throttling Type
80.6 Evaporator Pressure Regulating, Throttling Type (Evaporator Side)
80.7 Hand Expansion
80.8 Magnetic Stop
80.9 Snap Action
80.10 Suction Vapor Regulating
80.11 Thermo Suction
80.12 Thermostatic Expansion
80.13 Water
81. Vibration Absorber, Line

IDENTIFICATION OF PIPING SYSTEMS BY COLOR

The color scheme for identification of piping systems, based on material carried, as listed in the following table and shown in Fig. 1, is reprinted from Part V, Fourth Edition, of the Engineering Standards of the *Heating, Piping and Air Conditioning Contractors National Association.*[7]

Class	Color
F—Fire-protection	Red
D—Dangerous materials	Yellow or Orange
S—Safe Materials	Green (or the achromatic colors, white, black, gray or aluminum)
and, when required	
P—Protective materials	Bright blue
V—Extra valuable materials	Deep purple

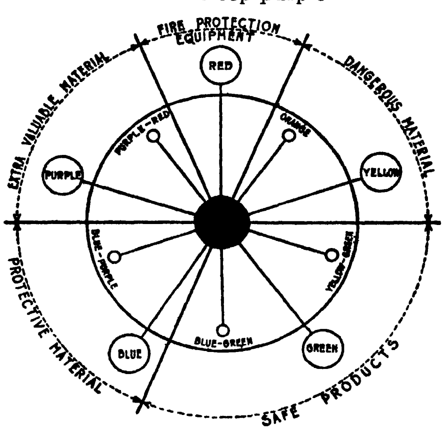


Fig. 1. Main Classification by Color

[7] See Scheme for Identification of Piping Systems, A13–1928, *American Standards Association.*

CHAPTER 3

THERMODYNAMICS

Mass and Energy Balances; Thermodynamic Properties of Moist Air; Formulas and Tables; Thermodynamic Properties of Water, Formulas and Tables; Degree of Saturation; A.S.H.V.E. Psychrometric Chart; Solution of Air Conditioning Problems by Use of Tables and Psychrometric Chart; U. S. Standard Atmosphere

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THERMODYNAMICS is that branch of natural science which deals with energy and its transformations into various forms. In this chapter the discussion will be limited to thermodynamics as it affects the arts of heating and air conditioning. This will necessarily presume some knowledge of the fundamentals of the science on the part of the reader who may also find it desirable to refer to a standard text on the subject, preferably one published after 1930.

MASS AND ENERGY BALANCES

The First Law of Thermodynamics is a statement of the Principle of Conservation of Energy. It may be stated as follows: *The energy added to a system is equal to the increase or decrease of the energy stored in the system, plus the energy which leaves the system.* For a completely contained, or non-flow system, this may be restated as: The heat added to a non-flow system is equal to the change in the internal energy of the system, plus the work done by the system.

$$ {}_1q_2 = U_2 - U_1 + w \tag{1}$$

For a constant pressure process

$$ {}_1q_2 = H_1 - H_2 \tag{2}$$

where

${}_1q_2$ = energy added between points 1 and 2.
U = internal energy of system.
w = work done by system.
H = enthalpy of system.

Subscripts 1 and 2 refer to sections of the system between which a change takes place.

For engineering problems, a more important application of the First Law is its use in cases in which, in addition to energy, one or more fluids are crossing the boundaries of the system. The most simple of these is the steady flow system, in which the rates of energy and mass flow across the boundaries of the system are constant, and no mass or energy is stored or released by the system.

Consider a system as illustrated in Fig. 1. The fluid crossing the boundaries of the system carries with it potential energy by reason of its elevation above some convenient datum, kinetic energy by reason of its velocity, and energy in the form of enthalpy. Additional energy may cross the boundaries of the system in the forms of heat or work. The various forms of energy crossing the boundaries between the sections under consideration may be equated by applying the First Law of Thermodynamics:

$$PE_1 + KE_1 + H_1 + {}_1q_2 = PE_2 + KE_2 + H_2 + w \tag{3}$$

where

PE = potential energy, Btu per pound dry air.
KE = kinetic energy, Btu per pound dry air.
H = enthalpy, Btu per pound dry air.
${}_1q_2$ = heat added between sections 1 and 2, Btu per pound dry air.
w = shaft work withdrawn between sections 1 and 2, Btu per pound of dry air.

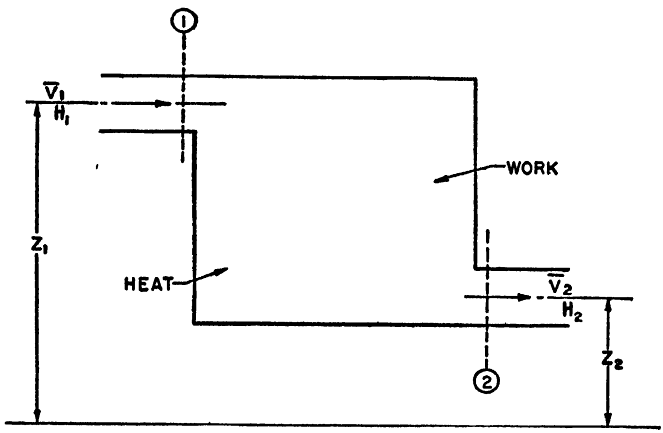

FIG. 1. ENERGY CHANGE BETWEEN TWO SECTIONS OF A SYSTEM

For most psychrometric problems, since the change in the potential energy and kinetic energy terms is negligible compared to the enthalpy change, Equation 3 may be simplified to

$$H_1 + {}_1q_2 = H_2 + w \tag{4}$$

where

H = enthalpy of the flowing medium, Btu per pound of dry air.

The enthalpy of the entire system may be broken down into constituent parts, thus:

$$GH = Gh + Lh_{wl} + Sh_{ws} \tag{5}$$

where

h = enthalpy of *moist* air, Btu per pound of dry air.
h_{wl} = enthalpy of liquid water, Btu per pound.
h_{ws} = enthalpy of solid water, Btu per pound.
L = flow rate of liquid water, pounds per hour.
S = flow rate of solid water, pounds per hour.
G = flow rate of dry air, pounds per hour.

Similarly, an equation expressing the conservation of mass may be written thus:

$$\sum_{\text{in}} [G(1 + W) + L + S] = \sum_{\text{out}} [G(1 + W) + L + S] \tag{6}$$

where

W = humidity ratio, pounds of water vapor per pound of dry air.

THERMODYNAMIC PROPERTIES OF MOIST AIR

The working substance of the air conditioning engineer is moist air. Air is actually a mixture of oxygen, nitrogen, carbon dioxide, water vapor, and traces of other gases.

The mixture consisting of the components other than water vapor is known as *dry air*. Its composition remains essentially constant under all conditions. In moist air the amount of water vapor varies considerably. To allow for this variation the specific properties of moist air are developed in terms of the relative amounts of water vapor and dry air. Accepted air conditioning practice is to express this in terms of the amount of water vapor per pound of dry air.

Terms frequently used in describing the condition of a mixture of air and water vapor are *humidity ratio, relative humidity, degree of saturation, dry-bulb temperature, thermodynamic wet-bulb temperature, and dew-point temperature.* These terms are defined in following paragraphs.

Humidity Ratio. Weight of water vapor associated with unit weight of dry air, pounds of water vapor per pound of dry air. Humidity ratio has also been called *specific humidity*, and this term is still used in many places.

Relative Humidity. Ratio of the mol fraction of water vapor in the actual mixture to the mol fraction of water vapor in saturated air at the same dry-bulb temperature and barometric pressure.

Degree of Saturation. Ratio of the actual humidity ratio to the humidity ratio of saturated air at the same dry-bulb temperature and barometric pressure.

Relative humidity and degree of saturation are related according to the identity:

$$\phi = \frac{\mu}{\left[1 - (1 - \mu) f_s \dfrac{P_s}{P_a}\right]} \tag{7}$$

where

Φ = relative humidity, expressed as a decimal.

μ = degree of saturation, expressed as a decimal.

P_a = observed (or barometric) pressure of the moist air.

P_s = saturation pressure of pure water at the prevailing temperature, expressed in the same units as P_a.

f_s = a dimensionless factor which may be regarded as accounting for influences arising when air and water are intermixed. Magnitudes of f_s have been reported by Goff and Gratch[1] and by Goff.[2] Table 1 gives values of f_s for a limited range of conditions.

Dry-Bulb Temperature. The temperature indicated by any type of thermometer or thermocouple not affected by the water vapor content of the air, or by radiation.

Thermodynamic Wet-Bulb Temperature. The temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature.

Consider an adiabatic system as shown in Fig. 2. Unsaturated air at the state h_1, W_1, enters the system at section 1, and saturated air at the state h^*, W^*, leaves the system at section 2. Liquid water at the state h_w^*, corresponding to the temperature of the saturated air leaving the system is supplied. Then, since no work is done and the system is strictly adiabatic, the energy equation becomes

$$h_1 + (W^* - W_1)h_w^* = h^* \tag{8}$$

where

* indicates condition at thermodynamic wet-bulb temperature.

TABLE 1. MAGNITUDES OF f_s FOR THE RANGE 0 TO 125 F
(Standard Barometric Pressure, 29.921 in. Hg)

TEMP. F	f_s	TEMP. F	f_s
0	1.0048	70	1.0045
10	1.0046	80	1.0047
20	1.0046	90	1.0048
30	1.0045	100	1.0050
40	1.0044	110	1.0053
50	1.0044	120	1.0055
60	1.0044	125	1.0057

Note: The original source[1] gives f_s to seven significant figures over the temperature range —208 F to + 202 F and over the pressure range 20 to 35 in. Hg.

The temperature corresponding to h^* for given values of h_1 and W_1 is called the thermodynamic wet-bulb temperature, or the temperature of adiabatic saturation.

The temperature indicated by an ordinary wet-bulb thermometer is affected by a number of factors not accounted for in Equation 8, and hence, may be quite different from the theoretical temperature obtained from its use. The measured wet-bulb temperature is influenced by (*a*) radiation from the surroundings to the wick; (*b*) conduction of heat along the stem of the thermometer; and (*c*) impact of the air on the wick or bulb of the thermometer. Arnold[3] has developed a theory which makes possible the calculation of the true thermodynamic wet-bulb temperature from observed data through the use of suitable corrections to be applied to the readings of the wet-bulb thermometer. However, unless extreme precision is required, the observed temperature may be taken equal to the theoretical temperature for most engineering problems, if no attempt is made to shield the wick from radiation and the air velocity past the wick is about 1000 fpm.

Dew-Point Temperature. The saturation temperature corresponding to a given combination of humidity ratio W and barometric pressure is called the dew-point temperature. It is the lowest temperature at which the given humidity ratio can exist at the corresponding barometric pressure. At this temperature condensation will first start to form when moist air is cooled.

Perfect Gas Relationships

Hypotheses, based on experimental observation of the physical behavior of gases, which were advanced by Boyle, Charles, Gay-Lussac, Dalton, Gibbs, Joule, Kelvin and others, were reduced to reasonably simple mathematical expressions and came to be regarded as physical laws. However, as scientific knowledge increased, and as more precise methods of measurement were developed, it became apparent that these simple equations did not describe the behavior of the mixtures of actual gases and vapors accurately.

The original statements have been found to be useful tools, nevertheless, in many cases. For example, the behavior of common diatomic and triatomic gases at low pressures follows these equations closely enough so that they may be used for some types of engineering problems. The practice of engineering is an art, and not an exact science, and many useful engineering works have been constructed through the use of approximations. The degree of approximation, however, which may be tolerated in engineering design must be decided by the engineer, based upon his study and experience in the field.

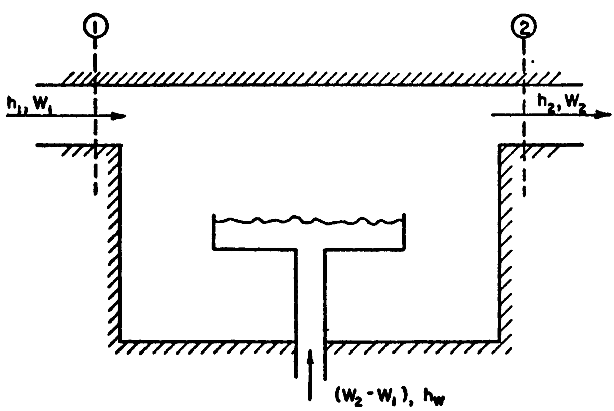

FIG. 2. ILLUSTRATION OF ADIABATIC SATURATION

Boyle's Law. One of the original observations of the physical behavior of gases was made by Robert Boyle who noted that, if a constant weight of gas is compressed with the temperature held constant, the volume V varied inversely as the absolute pressure P. Stated mathematically,

$$PV = \text{constant} \qquad \text{(temperature constant)} \tag{9}$$

Charles' Law. Experiments made independently by Charles and Gay-Lussac led to the formulation of what is now known as Charles' Law: If a constant weight of gas is heated or cooled at constant volume, the absolute pressure P varies as the absolute temperature T; if a constant weight of gas is heated or cooled at constant pressure, the volume V varies as the absolute temperature T. Stated mathematically,

$$\frac{P}{T} = \text{constant} \qquad \text{(volume constant)} \tag{10}$$

$$\frac{V}{T} = \text{constant} \qquad \text{(pressure constant)} \tag{11}$$

Boyle's Law and Charles' Law may be combined to form the equation of state for the ideal or perfect gas,

$$PV = RT \tag{12}$$

where

R is a constant whose value depends on the units selected for P, V, and T.

Dalton's Rule. Dalton stated that each gas in a mixture occupies the total volume of the mixture just as though the other gases were not present. Gibbs later expanded this statement for perfect gases into the following principles:

1. The pressure of a mixture of gases is the sum of the partial pressures of the individual gases when they exist at the total volume and temperature of the mixture.
2. The internal energy, enthalpy, and entropy of a mixture of gases are respectively equal to the sums of the individual internal energies, enthalpies, and entropies of the components when they exist at the total volume and temperature of the mixture.

While these relationships do not hold exactly for all systems of real gases, they may be used with a good degree of precision for many engineering applications at low pressures. Moreover, since water vapor very closely follows the perfect gas relationships in the range usually encountered in air conditioning, the Gibbs-Dalton Rule may frequently be applied to mixtures of dry air and water vapor.

Thus,

$$V_m = V_a = V_w \tag{13}$$

$$T_m = T_a = T_w \tag{14}$$

$$p_m = p_a + p_w \tag{15}$$

$$m_m h = m_a h_a + m_w h_w \tag{16}$$

where

Subscript m denotes mixture; subscript a denotes dry air; subscript w denotes water vapor.

Symbol m = weight of dry air crossing any duct section, pounds per minute.

Using Equations 12, 13, 14, and 15, the relation is obtained as follows:

$$v_T = \frac{n_a RT}{p_a} = \frac{n_w RT}{p_w} = \frac{(n_a + n_w)RT}{p} \tag{17}$$

where

V_T = total volume, cubic feet.

n_a = number of mols of dry air.

n_w = number of mols of water vapor.
R = universal gas constant, 1545 foot-pounds per (Fahrenheit degree) (mol).
p_a = partial pressure of dry air.
p_w = partial pressure of water vapor.
T = absolute temperature, Fahrenheit degrees.

The partial pressure of water vapor in the mixture is then

$$p_w = \frac{n_w}{n_a + n_w} p \tag{18}$$

or, the partial pressure of the water vapor in moist air is equal to the product of the mol fraction of the water vapor and the observed pressure of the mixture. A similar expression is obtained for the dry air.

Assuming that the perfect gas laws can be applied to water vapor at saturation at low pressures, the partial pressure of water vapor in a *saturated* mixture may be written as

$$p_s = \frac{n_s}{n_a + n_s} p \tag{19}$$

where

n_s = number of mols of water vapor at saturation.

The relative humidity may be obtained by combining Equations 18 and 19 and solving for the ratio of mol fractions. Thus, using perfect gas relationships,

$$\phi = \frac{p_w}{p_s} \tag{20}$$

The humidity ratio W may be obtained from Equation 17:

$$W = \frac{18\,016}{28.966} \frac{p_w}{p_a} = 0\,622 \frac{p_w}{p - p_w} \tag{21}$$

where 18.016 and 28.966 are the molecular weights of water and dry air, respectively.

Equation 16 may be rewritten as

$$h = h_a + Wh_w \tag{22}$$

where

h = enthalpy, Btu per pound of dry air.

In relating the enthalpy to the state of the moist air, the fact that in all applications only differences in enthalpy are involved, allows the arbitrary selection of a datum or zero enthalpy point. Accordingly, from perfect gas relationships, it is possible to write for any temperature of t, Fahrenheit greater than 0 F

$$h_a = 0.24\, t \tag{23}$$

where it is assumed that the same arbitrary datum of 0 F is used as in determining the properties of moist air in Table 2.

Tables of Thermodynamic Properties of Moist Air

Research work conducted at the *University of Pennsylvania* and at other institutions has shown that the Gibbs-Dalton Rule is inaccurate in varying degrees, depending on temperature, pressure and the amount of water vapor present. The probable reasons for this inaccuracy are due to the effect of:

1. Chemical solution of gas molecules in the water vapor.
2. The finite size of the molecules causing interference with the free passage of other molecules toward the boundaries of the system.
3. Intermolecular forces of attraction and repulsion.

Many attempts have been made to develop an equation of state which would predict the true states of real gases and vapors. The Van der Waal, Maxwell, and Beattie-Bridgman equations are probably the best known. Unfortunately, these expressions rapidly become much too complicated to be used in everyday calculations and, therefore, engineers find it more convenient to use tables of thermodynamic properties for specific working substances, as these can be prepared by physicists using the best laboratory equipment and all the refinements of mathematics.

Mechanical engineers have long been familiar with such tables for the properties of steam. Tables of the properties of moist air, as prepared by Goodenough and others, have been available for some time, but the latest and most precise of such tables are those which have resulted from a cooperative research agreement between the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *Towne Scientific School* of the *University of Pennsylvania.* These properties are published herein as Table 2, and are taken from a research report by Goff and Gratch.[4] Table 2, which experimentally and mathematically takes into account deviations from perfect gas behavior, such as those listed above, makes the application of the Gibbs-Dalton Rule a less frequent necessity.

In Table 2 there are 15 columns of figures, each column being headed by a suitable symbol. In the following sub-paragraphs brief explanations are given of the data in Table 2 under the appropriate column headings.

t(F) = Fahrenheit temperature defined in terms of absolute temperature T by the relation,

$$T = t + 459.69 \tag{24}$$

Absolute zero of temperature may be defined as the receiver temperature which will enable a Carnot Cycle engine to transform into work all the energy it receives in the form of heat.

W_s = humidity ratio at *saturation*. Saturation is the condition at which the vapor phase (moist air) may exist in equilibrium with a condensed phase (liquid or solid) at the given temperature and pressure (standard atmospheric pressure in the case of Table 2). At given values of temperature and pressure, the humidity ratio W can have any value from zero to W_s.

v_a = specific volume of dry air, cubic feet per pound.

$v_{as} = v_s - v_a$, the difference between the volume of moist air *at saturation*, per pound of dry air, and the specific volume of the dry air itself, cubic feet per pound of dry air.

v_s = specific volume of moist air *at saturation* per pound of dry air, cubic feet per pound of dry air.

h_a = specific enthalpy of dry air, Btu per pound of dry air. The specific enthalpy of dry air has been assigned the value zero at 0 F, standard atmospheric pressure. The energy unit Btu is related to the foot-pound by definition, as follows: 1 Btu = 778.3 ft-lb.

$h_{as} = h_s - h_a$, the difference between the enthalpy of moist air *at saturation*, per pound of dry air, and the specific enthalpy of the dry air itself, Btu per pound of dry air.

h_s = enthalpy of moist air *at saturation* per pound of dry air, Btu per pound of dry air.

s_a = specific entropy of dry air, Btu per (pound) (Fahrenheit degree). It will be noticed that the specific entropy of dry air has been assigned the value zero at 0 F and standard atmospheric pressure.

$s_{as} = s_s - s_a$, the difference between the entropy of moist air *at saturation*, per pound of dry air, and the specific entropy of the dry air itself, Btu per (pound of dry air) (Fahrenheit degree).

s_s = entropy of moist air *at saturation* per pound of dry air, Btu per (pound of dry air) (Fahrenheit degree).

h_w = specific enthalpy of condensed water (liquid or solid) at standard atmospheric pressure, Btu per pound of water. The specific enthalpy of liquid water has been assigned the value zero at 32 F, saturation pressure (0.088586 psia).

s_w = specific entropy of condensed water (liquid or solid) at standard atmospheric pressure, Btu per (pound of water) (Fahrenheit degree). The specific entropy of liquid water has been assigned the value zero at 32 F, saturation pressure (0.088586 psia).

p_s = saturation pressure of pure water vapor, pounds per square inch or inches of Hg (absolute pressure). At a given pressure, moist air can be saturated at any temperature, though this requires that it have a definite humidity ratio W_s and that the coexisting condensed phase contain a definite, but very small, quantity of dissolved air. On the other hand, pure water vapor (steam) below the critical temperature, can be saturated at only one temperature for a given pressure. The values of saturation pressure listed in Table 2 have been computed from the formulas of Goff and Gratch.

THERMODYNAMIC PROPERTIES OF WATER AT SATURATION

Since water vapor at low pressures acts almost as a perfect gas, the enthalpy of water vapor should also be a function only of the temperature within these limits. Therefore, the enthalpy of the water vapor may be expressed as being approximately equal to the enthalpy of saturated vapor at the dry-bulb temperature of the mixture. Substituting these values in Equation 22, the enthalpy of the mixture becomes

$$h = 0.24\, t + W h_g \tag{25}$$

where h_g is the value of the enthalpy of saturated vapor at the temperature t, and is obtained from Table 3.

Table 3 offers revisions to existing steam table data with extensions downward to -160 F. These revisions and extensions were a necessary preliminary to the construction of Table 2. A detailed explanation of the methods employed in the construction of Table 3 is given in a paper by John A. Goff and S. Gratch.[5]

As in Table 2, the temperature scale used as argument in Table 3 is the Fahrenheit scale defined in terms of absolute temperature T by Equation 24. The symbols used as column headings in Table 3 are the same as those used in steam tables, and have the same meanings.

Properties of water above 212 F from Keenan and Keyes[6] are given in Table 4.

DEGREE OF SATURATION

Degree of saturation has previously been defined as the ratio of the actual humidity ratio to the humidity ratio of saturated air at the same dry-bulb temperature and barometric pressure. This may be stated mathematically as

$$\mu = \frac{W}{W_s} \tag{26}$$

Obviously the degree of saturation μ can have any value from zero (dry air) to unity (moist air at saturation). The degree of saturation is con-

TABLE 2. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^5$	Volume cu ft/lb dry air v_a	v_{as}	v_s	Enthalpy Btu/lb dry air h_a	h_{as}	h_s	Entropy Btu per(°F) (lb dry air) s_a	s_{as}	s_s	Condensed Water Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg $p_s \times 10^3$	Fahr. Temp. t(F)
−160	0.0002120	7.520	0.000	7.520	−38.504	0.000	−38.504	−0.10300	0.00000	−0.10300	−222.00	−0.4907	0.0001009	−160
−155	0.0003869	7.647	0.000	7.647	−37.296	0.000	−37.296	−0.09901	0.00000	−0.09901	−220.40	−0.4853	0.0001842	−155
−150	0.0006932	7.775	0.000	7.775	−36.088	0.000	−36.088	−0.09508	0.00000	−0.09508	−218.77	−0.4800	0.0003301	−150
−145	0.001219	7.902	0.000	7.902	−34.881	0.000	−34.881	−0.09121	0.00000	−0.09121	−217.12	−0.4747	0.0005807	−145
−140	0.002109	8.029	0.000	8.029	−33.674	0.000	−33.674	−0.08740	0.00000	−0.08740	−215.44	−0.4695	0.001004	−140
−135	0.003586	8.156	0.000	8.156	−32.468	0.000	−32.468	−0.08365	0.00000	−0.08365	−213.75	−0.4642	0.001707	−135
−130	0.006000	8.283	0.000	8.283	−31.262	0.000	−31.262	−0.07997	0.00000	−0.07997	−212.03	−0.4590	0.002858	−130
−125	0.009887	8.411	0.000	8.411	−30.057	0.000	−30.057	−0.07634	0.00000	−0.07634	−210.28	−0.4538	0.004710	−125
−120	0.01606	8.537	0.000	8.537	−28.852	0.000	−28.852	−0.07277	0.00000	−0.07277	−208.52	−0.4485	0.007653	−120
−115	0.02571	8.664	0.000	8.664	−27.648	0.000	−27.648	−0.06924	0.00000	−0.06924	−206.73	−0.4433	0.01226	−115
−110	0.04063	8.792	0.000	8.792	−26.444	0.000	−26.444	−0.06577	0.00000	−0.06577	−204.92	−0.4381	0.01939	−110
−105	0.06340	8.919	0.000	8.919	−25.240	0.001	−25.239	−0.06234	0.00000	−0.06234	−203.09	−0.4329	0.03026	−105
−100	0.09772	9.046	0.000	9.046	−24.037	0.001	−24.036	−0.05897	0.00000	−0.05897	−201.23	−0.4277	0.04666	−100
−95	0.1489	9.173	0.000	9.173	−22.835	0.002	−22.833	−0.05565	0.00000	−0.05565	−199.35	−0.4225	0.07111	−95
−90	0.2242	9.300	0.000	9.300	−21.631	0.002	−21.629	−0.05237	0.00001	−0.05236	−197.44	−0.4173	0.1071	−90
−85	0.3342	9.426	0.000	9.426	−20.428	0.003	−20.425	−0.04913	0.00001	−0.04912	−195.51	−0.4121	0.1597	−85
−80	0.4930	9.553	0.000	9.553	−19.225	0.005	−19.220	−0.04595	0.00001	−0.04594	−193.55	−0.4069	0.2356	−80
−75	0.7196	9.680	0.000	9.680	−18.022	0.007	−18.015	−0.04280	0.00002	−0.04278	−191.57	−0.4017	0.3441	−75
−70	1.040	9.806	0.000	9.806	−16.820	0.011	−16.809	−0.03969	0.00003	−0.03966	−189.56	−0.3965	0.4976	−70
−65	1.491	9.932	0.000	9.932	−15.617	0.015	−15.602	−0.03663	0.00005	−0.03658	−187.53	−0.3913	0.7130	−65
−60	2.118	10.059	0.000	10.059	−14.416	0.022	−14.394	−0.03360	0.00006	−0.03354	−185.47	−0.3861	1.0127	−60
−55	2.982	10.186	0.000	10.186	−13.214	0.031	−13.183	−0.03061	0.00009	−0.03052	−183.39	−0.3810	1.4258	−55
−50	4.163	10.313	0.001	10.314	−12.012	0.043	−11.969	−0.02766	0.00012	−0.02754	−181.29	−0.3758	1.9910	−50
−48	4.747	10.364	0.001	10.365	−11.532	0.049	−11.483	−0.02649	0.00013	−0.02636	−180.44	−0.3738	2.2702	−48
−46	5.406	10.414	0.001	10.415	−11.051	0.056	−10.995	−0.02532	0.00014	−0.02518	−179.59	−0.3717	2.5854	−46
−44	6.149	10.465	0.001	10.466	−10.571	0.064	−10.507	−0.02416	0.00016	−0.02400	−178.73	−0.3696	2.9408	−44
−42	6.985	10.516	0.001	10.517	−10.090	0.073	−10.017	−0.02301	0.00019	−0.02282	−177.87	−0.3676	3.3408	−42
−40	7.925	10.566	0.001	10.567	−9.609	0.083	−9.526	−0.02186	0.00021	−0.02165	−177.01	−0.3655	3.7906	−40
−38	8.980	10.617	0.002	10.619	−9.129	0.094	−9.035	−0.02072	0.00024	−0.02048	−176.14	−0.3634	4.2958	−38
−36	10.16	10.668	0.002	10.670	−8.648	0.106	−8.542	−0.01958	0.00026	−0.01932	−175.27	−0.3614	4.8626	−36
−34	11.49	10.718	0.002	10.720	−8.168	0.121	−8.047	−0.01845	0.00030	−0.01815	−174.40	−0.3593	5.4980	−34
−32	12.98	10.769	0.002	10.771	−7.687	0.136	−7.551	−0.01733	0.00034	−0.01699	−173.52	−0.3573	6.2093	−32

[a] Compiled by John A. Goff and S. Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^4$	Volume cu ft/lb dry air v_a	v_{as}	v_s	Enthalpy Btu/lb dry air h_a	h_{as}	h_s	Entropy Btu per (°F) (lb dry air) s_a	s_{as}	s_s	Condensed Water Enthalpy Btu/Lb h_w	Entropy Btu/Lb (°F) s_w	Vap. Press In Hg $p_s \times 10^2$	Fahr. Temp. t(F)
−30	1.464	10.820	0.002	10.822	−7.207	0.154	−7.053	−0.01621	0.00038	−0.01583	−172.64	−0.3552	0.70046	−30
−28	1.649	10.870	0.003	10.873	−6.726	0.173	−6.553	−0.01509	0.00043	−0.01466	−171.75	−0.3531	0.78928	−28
−26	1.856	10.921	0.003	10.924	−6.246	0.196	−6.050	−0.01398	0.00048	−0.01350	−170.86	−0.3511	0.88838	−26
−24	2.087	10.972	0.004	10.976	−5.765	0.219	−5.546	−0.01287	0.00054	−0.01233	−169.97	−0.3490	0.99885	−24
−22	2.344	11.022	0.004	11.026	−5.285	0.246	−5.039	−0.01177	0.00061	−0.01116	−169.07	−0.3469	1.1219	−22
−20	2.630	11.073	0.005	11.078	−4.804	0.277	−4.527	−0.01067	0.00068	−0.00999	−168.17	−0.3449	1.2587	−20
−19	2.785	11.098	0.005	11.103	−4.564	0.293	−4.271	−0.01012	0.00072	−0.00940	−167.72	−0.3439	1.3327	−19
−18	2.948	11.124	0.005	11.129	−4.324	0.310	−4.014	−0.00958	0.00076	−0.00882	−167.26	−0.3428	1.4107	−18
−17	3.120	11.149	0.006	11.155	−4.083	0.328	−3.755	−0.00904	0.00080	−0.00824	−166.81	−0.3418	1.4929	−17
−16	3.301	11.174	0.006	11.180	−3.843	0.348	−3.495	−0.00850	0.00084	−0.00766	−166.35	−0.3408	1.5795	−16
−15	3.491	11.200	0.006	11.206	−3.603	0.368	−3.235	−0.00796	0.00089	−0.00707	−165.90	−0.3398	1.6706	−15
−14	3.692	11.225	0.007	11.232	−3.363	0.389	−2.974	−0.00743	0.00094	−0.00649	−165.44	−0.3387	1.7666	−14
−13	3.903	11.250	0.007	11.257	−3.123	0.412	2.711	−0.00689	0.00099	−0.00590	−164.98	−0.3377	1.8677	−13
−12	4.125	11.275	0.008	11.283	−2.882	0.436	−2.446	−0.00636	0.00104	−0.00532	−164.52	−0.3367	1.9740	−12
−11	4.359	11.301	0.008	11.309	−2.642	0.461	−2.181	−0.00582	0.00109	−0.00473	−164.06	−0.3357	2.0859	−11
−10	4.606	11.326	0.008	11.334	−2.402	0.487	−1.915	−0.00529	0.00115	−0.00414	−163.60	−0.3346	2.2035	−10
−9	4.865	11.351	0.008	11.359	−2.162	0.514	−1.648	−0.00475	0.00121	−0.00354	−163.14	−0.3336	2.3272	−9
−8	5.137	11.376	0.009	11.385	−1.922	0.543	−1.379	−0.00422	0.00128	−0.00294	−162.67	−0.3326	2.4573	−8
−7	5.423	11.401	0.010	11.411	−1.681	0.574	−1.107	−0.00369	0.00135	−0.00234	−162.21	−0.3316	2.5940	−7
−6	5.724	11.427	0.010	11.437	−1.441	0.606	−0.835	−0.00316	0.00142	−0.00174	−161.74	−0.3305	2.7377	−6
−5	6.040	11.452	0.011	11.463	−1.201	0.639	−0.562	−0.00263	0.00149	−0.00114	−161.28	−0.3295	2.8886	−5
−4	6.371	11.477	0.012	11.489	−0.961	0.675	−0.286	−0.00210	0.00157	−0.00053	−160.81	−0.3285	3.0472	−4
−3	6.720	11.502	0.013	11.515	−0.721	0.712	−0.009	−0.00157	0.00165	0.00008	−160.34	−0.3275	3.2137	−3
−2	7.085	11.528	0.013	11.541	−0.480	0.751	0.271	−0.00105	0.00174	0.00069	−159.87	−0.3264	3.3885	−2
−1	7.469	11.553	0.014	11.567	−0.240	0.792	0.552	−0.00052	0.00183	0.00131	−159.40	−0.3254	3.5720	−1
0	7.872	11.578	0.015	11.593	0.000	0.835	0.835	0.00000	0.00192	0.00192	−158.93	−0.3244	3.7645	0
1	8.295	11.604	0.015	11.619	0.240	0.880	1.120	0.00052	0.00202	0.00254	−158.46	−0.3234	3.9666	1
2	8.739	11.629	0.016	11.645	0.480	0.928	1.408	0.00104	0.00212	0.00316	−157.99	−0.3223	4.1785	2
3	9.204	11.654	0.017	11.671	0.721	0.977	1.698	0.00156	0.00223	0.00379	−157.52	−0.3213	4.4007	3
4	9.692	11.679	0.018	11.697	0.961	1.030	1.991	0.00208	0.00234	0.00442	−157.04	−0.3203	4.6337	4
5	10.20	11.705	0.019	11.724	1.201	1.085	2.286	0.00260	0.00246	0.00506	−156.57	−0.3193	4.8779	5
6	10.74	11.730	0.020	11.750	1.441	1.142	2.583	0.00312	0.00258	0.00570	−156.09	−0.3182	5.1339	6

[a] Compiled by John A. Goff and S. Gratch.

Table 2. Thermodynamic Properties of Moist Air[a] (Standard Atmospheric Pressure, 29.921 in. Hg) (Continued)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^3$	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg $p_s \times 10^2$	
7	1.130	11.756	0.021	11.777	1.681	1.202	2.883	0.00364	0.00271	0.00635	−155.61	−0.3172	5.4022	7
8	1.189	11.781	0.022	11.803	1.922	1.266	3.188	0.00415	0.00285	0.00700	−155.13	−0.3162	5.6832	8
9	1.251	11.806	0.024	11.830	2.162	1.332	3.494	0.00467	0.00299	0.00766	−154.65	−0.3152	5.9776	9
10	1.315	11.831	0.025	11.856	2.402	1.401	3.803	0.00518	0.00314	0.00832	−154.17	−0.3141	6.2858	10
11	1.383	11.857	0.026	11.883	2.642	1.474	4.116	0.00569	0.00330	0.00899	−153.69	−0.3131	6.6085	11
12	1.454	11.882	0.028	11.910	2.882	1.550	4.432	0.00620	0.00346	0.00966	−153.21	−0.3121	6.9462	12
13	1.528	11.907	0.029	11.936	3.123	1.630	4.753	0.00671	0.00363	0.01034	−152.73	−0.3111	7.2997	13
14	1.606	11.933	0.030	11.963	3.363	1.713	5.076	0.00721	0.00380	0.01101	−152.24	−0.3100	7.6696	14
15	1.687	11.958	0.032	11.990	3.603	1.800	5.403	0.00772	0,00399	0.01171	−151.76	−0.3090	8.0565	15
16	1.772	11.983	0.034	12.017	3.843	1.892	5.735	0.00822	0.00418	0.01240	−151.27	−0.3080	8.4612	16
17	1.861	12.009	0.035	12.044	4.083	1.988	6.071	0.00873	0.00438	0.01311	−150.78	−0.3070	8.8843	17
18	1.953	12.034	0.038	12.072	4.324	2.088	6.412	0.00923	0.00459	0.01382	−150.29	0.3059	9.3267	18
19	2.051	12.059	0.040	12.099	4.564	2.192	6.756	0.00973	0.00481	0.01454	−149.80	−0.3049	9.7889	19
20	2.152	12.084	0.042	12.126	4.804	2.302	7.106	0.01023	0.00504	0.01527	−149.31	−0.3039	10.272	20
21	2.258	12.110	0.044	12.154	5.044	2.416	7.460	0.01073	0.00528	0.01601	−148.82	−0.3029	10.777	21
22	2.369	12.135	0.046	12.181	5.284	2.536	7.820	0.01123	0.00553	0.01676	−148.33	−0.3018	11.305	22
23	2.485	12.160	0.049	12.209	5.525	2.661	8.186	0.01173	0.00579	0.01752	−147.84	−0.3008	11.856	23
24	2.606	12.186	0.051	12.237	5.765	2.792	8.557	0.01223	0.00607	0.01830	−147.34	−0.2998	12.431	24
25	2.733	12.211	0.054	12.265	6.005	2.929	8.934	0.01273	0.00635	0.01908	−146.85	−0.2988	13.032	25
26	2.865	12.236	0.057	12.293	6.245	3.072	9.317	0.01322	0.00665	0.01987	−146.35	−0.2977	13.659	26
27	3.003	12.262	0.059	12.321	6.485	3.221	9.706	0.01372	0.00696	0.02068	−145.85	−0.2967	14.313	27
28	3.147	12.287	0.062	12.349	6.726	3.377	10.103	0.01421	0.00728	0.02149	−145.36	−0.2957	14.966	28
29	3.297	12.312	0.065	12.377	6.966	3.540	10.506	0.01470	0.00761	0.02231	−144.86	−0.2947	15.709	29
30	3.454	12.338	0.068	12.406	7.206	3.709	10.915	0.01519	0.00796	0.02315	−144.36	−0.2936	16.452	30
31	3.617	12.363	0.071	12.434	7.446	3.887	11.333	0.01568	0.00832	0.02400	−143.86	−0.2926	17.227	31
32	3.788	12.388	0.075	12.463	7.686	4.072	11.758	0.01617	0.00870	0.02487	−143.36	−0.2916	18.035	32
32*	3.788	12.388	0.075	12.463	7.686	4.072	11.758	0.01617	0.00870	0.02487	0.04	0.0000	18.037	32*
33	3.944	12.413	0.079	12.492	7.927	4.242	12.169	0.01666	0.00904	0.02570	1.05	0.0020	18.778	33
34	4.107	12.438	0.082	12.520	8.167	4.418	12.585	0.01715	0.00940	0.02655	2.06	0.0041	19.546	34
35	4.275	12.464	0.085	12.549	8.407	4.601	13.008	0.01764	0.00977	0.02741	3.06	0.0061	20.342	35
36	4.450	12.489	0.089	12.578	8.647	4.791	13.438	0.01812	0.01016	0.02828	4.07	0.0081	21.166	36
37	4.631	12.514	0.093	12.607	8.887	4.987	13.874	0.01861	0.01056	0.02917	5.07	0.0102	22.020	37

[a] Compiled by John A. Goff and S. Gratch.
* Extrapolated to represent metastable equilibrium with undercooled liquid.

TABLE 2. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^3$	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg p_s	
38	4.818	12.540	0.097	12.637	9.128	5.191	14.319	0.01909	0.01097	0.03006	6.08	0.0122	0.22904	38
39	5.012	12.565	0.101	12.666	9.368	5.403	14.771	0.01957	0.01139	0.03096	7.08	0.0142	0.23819	39
40	5.213	12.590	0.105	12.695	9.608	5.662	15.230	0.02005	0.01183	0.03188	8.09	0.0162	0.24767	40
41	5.421	12.616	0.109	12.725	9.848	5.849	15.697	0.02053	0.01228	0.03281	9.09	0.0182	0.25748	41
42	5.638	12.641	0.114	12.755	10.088	6.084	16.172	0.02101	0.01275	0.03376	10.09	0.0202	0.26763	42
43	5.860	12.666	0.119	12.785	10.329	6.328	16.657	0.02149	0.01323	0.03472	11.10	0.0222	0.27813	43
44	6.091	12.691	0.124	12.815	10.569	6.580	17.149	0.02197	0.01373	0.03570	12.10	0.0242	0.28899	44
45	6.331	12.717	0.129	12.846	10.809	6.841	17.650	0.02245	0.01425	0.03670	13.10	0.0262	0.30023	45
46	6.578	12.742	0.134	12.876	11.049	7.112	18.161	0.02293	0.01478	0.03771	14.10	0.0282	0.31185	46
47	6.835	12.767	0.140	12.907	11.289	7.391	18.680	0.02340	0.01534	0.03874	15.11	0.0302	0.32386	47
48	7.100	12.792	0.146	12.938	11.530	7.681	19.211	0.02387	0.01591	0.03978	16.11	0.0321	0.33629	48
49	7.374	12.818	0.151	12.969	11.770	7.981	19.751	0.02434	0.01650	0.04084	17.11	0.0341	0.34913	49
50	7.658	12.843	0.158	13.001	12.010	8.291	20.301	0.02481	0.01711	0.04192	18.11	0.0361	0.36240	50
51	7.952	12.868	0.164	13.032	12.250	8.612	20.862	0.02528	0.01774	0.04302	19.11	0.0381	0.37611	51
52	8.256	12.894	0.170	13.064	12.491	8.945	21.436	0.02575	0.01839	0.04414	20.11	0.0400	0.39028	52
53	8.569	12.919	0.178	13.097	12.731	9.289	22.020	0.02622	0.01906	0.04528	21.12	0.0420	0.40492	53
54	8.894	12.944	0.185	13.129	12.971	9.644	22.615	0.02669	0.01976	0.04645	22.12	0.0439	0.42004	54
55	9.229	12.970	0.192	13.162	13.211	10.01	23.22	0.02716	0.02047	0.04763	23.12	0.0459	0.43565	55
56	9.575	12.995	0.200	13.195	13.452	10.39	23.84	0.02762	0.02121	0.04883	24.12	0.0478	0.45176	56
57	9.934	13.020	0.208	13.228	13.692	10.79	24.48	0.02809	0.02197	0.05006	25.12	0.0497	0.46840	57
58	10.30	13.045	0.216	13.261	13.932	11.19	25.12	0.02855	0.02276	0.05131	26.12	0.0517	0.48558	58
59	10.69	13.071	0.224	13.295	14.172	11.61	25.78	0.02902	0.02357	0.05259	27.12	0.0536	0.50330	59
60	11.08	13.096	0.233	13.329	14.413	12.05	26.46	0.02948	0.02441	0.05389	28.12	0.0555	0.52159	60
61	11.49	13.121	0.242	14.363	14.653	12.50	27.15	0.02994	0.02527	0.05521	29.12	0.0574	0.54047	61
62	11.91	13.147	0.251	13.398	14.893	12.96	27.85	0.03040	0.02616	0.05656	30.12	0.0594	0.55994	62
63	12.35	13.172	0.261	13.433	15.134	13.44	28.57	0.03086	0.02708	0.05794	31.12	0.0613	0.58002	63
64	12.80	13.197	0.271	13.468	15.374	13.94	29.31	0.03132	0.02803	0.05935	32.12	0.0632	0.60073	64
65	13.26	13.222	0.282	13.504	15.614	14.45	30.06	0.03177	0.02901	0.06078	33.11	0.0651	0.62209	65
66	13.74	13.247	0.292	13.539	15.855	14.98	30.83	0.03223	0.03002	0.06225	34.11	0.0670	0.64411	66
67	14.24	13.273	0.303	13.576	16.095	15.53	31.62	0 03269	0.03106	0.06375	35.11	0.0689	0.66681	67
68	14.75	13.298	0.315	13.613	16.335	16.09	32.42	0.03314	0.03213	0.06527	36.11	0.0708	0.69019	68
69	15.28	13.323	0.327	13.650	16.576	16.67	33.25	0.03360	0.03323	0.06683	37.11	0.0727	0.71430	69

[a] Compiled by John A. Goff and S. Gratch.

Table 2. Thermodynamic Properties of Moist Air[a] (Standard Atmospheric Pressure, 29.921 in. Hg) (Continued)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10^2$	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg p_s	
70	1.582	13.348	0.339	13.687	16.816	17.27	34.09	0.03405	0.03437	0.06842	38.11	0.0746	0.73915	70
71	1.639	13.373	0.351	13.724	17.056	17.89	34.95	0.03450	0.03554	0.07004	39.11	0.0765	0.76475	71
72	1.697	13.398	0.364	13.762	17.297	18.53	35.83	0.03495	0.03675	0.07170	40.11	0.0784	0.79112	72
73	1.757	13.424	0.377	13.801	17.537	19.20	36.74	0.03540	0.03800	0.07340	41.11	0.0803	0.81828	73
74	1.819	13.449	0.392	13.841	17.778	19.88	37.66	0.03585	0.03928	0.07513	42.10	0.0821	0.84624	74
75	1.882	13.474	0.407	13.881	18.018	20.59	38.61	0.03630	0.04060	0.07690	43.10	0.0840	0.87504	75
76	1.948	13.499	0.422	13.921	18.259	21.31	39.57	0.03675	0.04197	0.07872	44.10	0.0859	0.90470	76
77	2.016	13.525	0.437	13.962	18.499	22.07	40.57	0.03720	0.04337	0.08057	45.10	0.0877	0.93523	77
78	2.086	13.550	0.453	14.003	18.740	22.84	41.58	0.03765	0.04482	0.08247	46.10	0.0896	0.96665	78
79	2.158	13.575	0.470	14.045	18.980	23.64	42.62	0.03810	0.04631	0.08441	47.10	0.0914	0.99899	79
80	2.233	13.601	0.486	14.087	19.221	24.47	43.69	0.03854	0.04784	0.08638	48.10	0.0933	1.0323	80
81	2.310	13.626	0.504	14.130	19.461	25.32	44.78	0.03899	0.04942	0.08841	49.09	0.0952	1.0665	81
82	2.389	13.651	0.523	14.174	19.702	26.20	45.90	0.03943	0.05105	0.09048	50.09	0.0970	1.1017	82
83	2.471	13.676	0.542	14.218	19.942	27.10	47.04	0.03987	0.05273	0.09260	51.09	0.0989	1.1379	83
84	2.555	13.702	0.560	14.262	20.183	28.04	48.22	0.04031	0.05446	0.09477	52.09	0.1007	1.1752	84
85	2.642	13.727	0.581	14.308	20.423	29.01	49.43	0.04075	0.05624	0.09699	53.09	0.1025	1.2135	85
86	2.731	13.752	0.602	14.354	20.663	30.00	50.66	0.04119	0.05807	0.09926	54.08	0.1043	1.2529	86
87	2.824	13.777	0.624	14.401	20.904	31.03	51.93	0.04163	0.05995	0.10158	55.08	0.1062	1.2934	87
88	2.919	13.803	0.645	14.448	21.144	32.09	53.23	0.04207	0.06189	0.10396	56.08	0.1080	1.3351	88
89	3.017	13.828	0.668	14.496	21.385	33.18	54.56	0.04251	0.06389	0.10640	57.08	0.1098	1.3779	89
90	3.118	13.853	0.692	14.545	21.625	34.31	55.93	0.04295	0.06596	0.10890	58.08	0.1116	1.4219	90
91	3.223	13.879	0.716	14.595	21.865	35.47	57.33	0.04339	0.06807	0.11146	59.07	0.1135	1.4671	91
92	3.330	13.904	0.741	14.645	22.106	36.67	58.78	0.04382	0.07025	0.11407	60.07	0.1153	1.5135	92
93	3.441	13.929	0.768	14.697	22.346	37.90	60.25	0.04426	0.07249	0.11675	61.07	0.1171	1.5612	93
94	3.556	13.954	0.795	14.749	22.587	39.18	61.77	0.04469	0.07480	0.11949	62.07	0.1188	1.6102	94
95	3.673	13.980	0.822	14.802	22.827	40.49	63.32	0.04513	0.07718	0.12231	63.07	0.1206	1.6606	95
96	3.795	14.005	0.851	14.856	23.068	41.85	64.92	0.04556	0.07963	0.12519	64.06	0.1224	1.7123	96
97	3.920	14.030	0.881	14.911	23.308	43.24	66.55	0.04600	0.08215	0.12815	65.06	0.1242	1.7654	97
98	4.049	14.056	0.911	14.967	23.548	44.68	68.23	0.04643	0.08474	0.13117	66.06	0.1260	1.8199	98
99	4.182	14.081	0.942	15.023	23.789	46.17	69.96	0.04686	0.08741	0.13427	67.06	0.1278	1.8759	99
100	4.319	14.106	0.975	15.081	24.029	47.70	71.73	0.04729	0.09016	0.13745	68.06	0.1296	1.9333	100
101	4.460	14.131	1.009	15.140	24.270	49.28	73.55	0.04772	0.09299	0.14071	69.05	0.1314	1.9923	101
102	4.606	14.157	1.043	15.200	24.510	50.91	75.42	0.04815	0.09591	0.14406	70.05	0.1332	2.0528	102
103	4.756	14.182	1.079	15.261	24.751	52.59	77.34	0.04858	0.09891	0.14749	71.05	0.1350	2.1149	103
104	4.911	14.207	1.117	15.324	24.991	54.32	79.31	0.04900	0.1020	0.1510	72.05	0.1367	2.1786	104

[a] Compiled by John A. Goff and S. Gratch.

TABLE 2. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

Fahr. Temp. t(F)	Humidity Ratio $W_s \times 10$	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_c	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg p_s	
105	0.5070	14.232	1.155	15.387	25.232	56.11	81.34	0.04943	0.1052	0.1546	73.04	0.1385	2.2439	105
106	0.5234	14.258	1.194	15.452	25.472	57.95	83.42	0.04985	0.1085	0.1584	74.04	0.1403	2.3109	106
107	0.5404	14.283	1.235	15.518	25.713	59.85	85.56	0.05028	0.1118	0.1621	75.04	0.1421	2.3797	107
108	0.5578	14.308	1.278	15.586	25.953	61.80	87.76	0.05070	0.1153	0.1660	76.04	0.1438	2.4502	108
109	0.5758	14.333	1.321	15.654	26.194	63.82	90.03	0.05113	0.1189	0.1700	77.04	0.1456	2.5225	109
110	0.5944	14.359	1.365	15.724	26.434	65.91	92.34	0.05155	0.1226	0.1742	78.03	0.1472	2.5966	110
111	0.6135	14.384	1.412	15.796	26.675	68.05	94.72	0.05197	0.1264	0.1784	79.03	0.1491	2.6726	111
112	0.6333	14.409	1.460	15.869	26.915	70.27	97.18	0.05239	0.1302	0.1826	80.03	0.1508	2.7505	112
113	0.6536	14.435	1.509	15.944	27.156	72.55	99.71	0.05281	0.1342	0.1870	81.03	0.1525	2.8304	113
114	0.6746	14.460	1.560	16.020	27.397	74.91	102.31	0.05323	0.1384	0.1916	82.03	0.1543	2.9123	114
115	0.6962	14.485	1.613	16.098	27.637	77.34	104.98	0.05365	0.1426	0.1963	83.02	0.1560	2.9962	115
116	0.7185	14.510	1.668	16.178	27.878	79.85	107.73	0.05407	0.1470	0.2011	84.02	0.1577	3.0821	116
117	0.7415	14.536	1.723	16.259	28.119	82.43	110.55	0.05449	0.1515	0.2060	85.02	0.1595	3.1701	117
118	0.7652	14.561	1.782	16.343	28.359	85.10	113.46	0.05490	0.1562	0.2111	86.02	0.1612	3.2603	118
119	0.7897	14.586	1.842	16.428	28.600	87.86	116.46	0.05532	0.1610	0.2163	87.02	0.1629	3.3527	119
120	0.8149	14.611	1.905	16.516	28.841	90.70	119.54	0.05573	0.1659	0.2216	88.01	0.1646	3.4474	120
121	0.8410	14.637	1.968	16.605	29.082	93.64	122.72	0.05615	0.1710	0.2272	89.01	0.1664	3.5443	121
122	0.8678	14.662	2.034	16.696	29.322	96.66	125.98	0.05656	0.1763	0.2329	90.01	0.1681	3.6436	122
123	0.8955	14.687	2.103	16.790	29.563	99.79	129.35	0.05698	0.1817	0.2387	91.01	0.1698	3.7452	123
124	0.9242	14.712	2.174	16.886	29.804	103.0	132.8	0.05739	0.1872	0.2446	92.01	0.1715	3.8493	124
125	0.9537	14.738	2.247	16.985	30.044	106.4	136.4	0.05780	0.1930	0.2508	93.01	0.1732	3.9558	125
126	0.9841	14.763	2.323	17.086	30.285	109.8	140.1	0.05821	0.1989	0.2571	94.01	0.1749	4.0649	126
127	1.016	14.788	2.401	17.189	30.526	113.4	143.9	0.05862	0.2050	0.2636	95.00	0.1766	4.1765	127
128	1.048	14.813	2.482	17.295	30.766	117.0	147.8	0.05903	0.2113	0.2703	96.00	0.1783	4.2907	128
129	1.082	14.839	2.565	17.404	31.007	120.8	151.8	0.05944	0.2178	0.2772	97.00	0.1800	4.4076	129
130	1.116	14.864	2.652	17.516	31.248	124.7	155.9	0.05985	0.2245	0.2844	98.00	0.1817	4.5272	130
131	1.152	14.889	2.742	17.631	31.489	128.8	160.3	0.06026	0.2314	0.2917	99.00	0.1834	4.6495	131
132	1.189	14.915	2.834	17.749	31.729	133.0	164.7	0.06067	0.2386	0.2993	100.00	0.1851	4.7747	132
133	1.227	14.940	2.930	17.870	31.970	137.3	169.3	0.06108	0.2459	0.3070	101.00	0.1868	4.9028	133
134	1.267	14.965	3.029	17.994	32.211	141.8	174.0	0.06148	0.2536	0.3151	102.00	0.1885	5.0337	134
135	1.308	14.990	3.132	18.122	32.452	146.4	178.9	0.06189	0.2614	0.3233	103.00	0.1902	5.1676	135
136	1.350	15.016	3.237	18.253	32.692	151.2	183.9	0.06229	0.2695	0.3318	104.00	0.1918	5.3046	136
137	1.393	15.041	3.348	18.389	32.933	156.1	189.0	0.06270	0.2778	0.3405	105.00	0.1935	5.4446	137
138	1.439	15.066	3.462	18.528	33.174	161.2	194.4	0.06310	0.2865	0.3496	106.00	0.1952	5.5878	138
139	1.485	15.091	3.580	18.671	33.414	166.5	199.9	0.06350	0.2954	0.3589	107.00	0.1969	5.7342	139

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TABLE 2. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

Fahr. Temp. t(F)	Humidity Ratio W_s	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg p_s	
140	0.1534	15.117	3.702	18.819	33.655	172.0	205.7	0.06390	0.3047	0.3686	107.99	0.1985	5.8838	140
141	0.1584	15.142	3.829	18.971	33.896	177.7	211.6	0.06430	0.3142	0.3785	108.99	0.2002	6.0367	141
142	0.1636	15.167	3.961	19.128	34.136	183.6	217.7	0.06470	0.3241	0.3888	109.99	0.2018	6.1930	142
143	0.1689	15.192	4.098	19.290	34.377	189.7	224.1	0.06510	0.3343	0.3994	110.99	0.2035	6.3527	143
144	0.1745	15.218	4.239	19.457	34.618	196.0	230.6	0.06549	0.3449	0.4104	111.99	0.2051	6.5160	144
145	0.1803	15.243	4.386	19.629	34.859	202.5	237.4	0.06589	0.3559	0.4218	112.99	0.2068	6.6828	145
146	0.1862	15.268	4.539	19.807	35.099	209.3	244.4	0.06629	0.3672	0.4335	113.99	0.2084	6.8532	146
147	0.1924	15.293	4.698	19.991	35.340	216.4	251.7	0.06669	0.3790	0.4457	114.99	0.2101	7.0273	147
148	0.1989	15.319	4.862	20.181	35.581	223.7	259.3	0.06708	0.3912	0.4583	115.99	0.2117	7.2051	148
149	0.2055	15.344	5.033	20.377	35.822	231.3	267.1	0.06748	0.4038	0.4713	116.99	0.2134	7.3867	149
150	0.2125	15.369	5.211	20.580	36.063	239.2	275.3	0.06787	0.4169	0.4848	117.99	0.2150	7.5722	150
151	0.2197	15.394	5.396	20.790	36.304	247.3	283.6	0.06827	0.4304	0.4987	118.99	0.2167	7.7616	151
152	0.2271	15.420	5.587	21.007	36.545	255.9	292.4	0.06866	0.4445	0.5132	119.99	0.2183	7.9550	152
153	0.2349	15.445	5.788	21.233	36.785	264.7	301.5	0.06906	0.4591	0.5282	120.99	0.2200	8.1525	153
154	0.2430	15.470	5.996	21.466	37.026	273.9	310.9	0.06945	0.4743	0.5438	121.99	0.2216	8.3541	154
155	0.2514	15.496	6.213	21.709	37.267	283.5	320.8	0.06984	0.4901	0.5599	122.99	0.2232	8.5599	155
156	0.2602	15.521	6.439	21.960	37.508	293.5	331.0	0.07023	0.5066	0.5768	123.99	0.2248	8.7701	156
157	0.2693	15.546	6.675	22.221	37.749	303.9	341.7	0.07062	0.5237	0.5943	124.99	0.2265	8.9846	157
158	0.2788	15.571	6.922	22.493	37.990	314.7	352.7	0.07101	0.5415	0.6125	125.99	0.2281	9.2036	158
159	0.2887	15.597	7.178	22.775	38.231	326.0	364.2	0.07140	0.5600	0.6314	127.00	0.2297	9.4271	159
160	0.2990	15.622	7.446	23.068	38.472	337.8	376.3	0.07179	0.5793	0.6511	128.00	0 2313	9.6556	160
161	0.3098	15.647	7.727	23.374	38.713	350.1	388.8	0.07218	0.5994	0.6716	129.00	0.2329	9.8876	161
162	0.3211	15.672	8.020	23.692	38.954	363.0	402.0	0.07257	0.6204	0.6930	130.00	0.2345	10.125	162
163	0.3329	15.698	8.326	24.024	39.195	376.5	415.7	0.07296	0.6423	0.7153	131.00	0.2361	10.367	163
164	0.3452	15.723	8.648	24.371	39.436	390.5	429.9	0.07334	0.6652	0.7385	132.00	0.2377	10.614	164
165	0.3581	15.748	8.985	24.733	39.677	405.3	445.0	0.07373	0.6892	0.7629	133.00	0.2393	10.866	165
166	0.3716	15.773	9.339	25.112	39.918	420.8	460.7	0.07411	0.7142	0.7883	134.00	0.2409	11.123	166
167	0.3858	15.799	9.708	25.507	40.159	437.0	447.2	0.07450	0.7405	0.8150	135.01	0.2426	11.385	167
168	0.4007	15.824	10.098	25.922	40.400	454.0	494.4	0.07488	0.7680	0.8429	136.01	0.2441	11.652	168
169	0.4163	15.849	10.508	26.357	40 641	471.8	512.4	0.07527	0.7969	0.8722	137.01	0.2457	11.925	169
170	0.4327	15.874	10.938	26.812	40.882	490.6	531.5	0.07565	0.8273	0.9030	138.01	0.2473	12.203	170
171	0.4500	15.900	11.391	27.291	41.123	510.4	551.5	0.07603	0.8592	0.9352	139.01	0.2489	12.486	171
172	0.4682	15.925	11.870	27.795	41.364	531.3	572.7	0.07641	0.8927	0.9691	140.01	0.2505	12.775	172
173	0.4875	15.950	12.376	28.326	41.605	553.3	594.9	0.07680	0.9281	1.0049	141.01	0.2521	13.069	173
174	0.5078	15.975	12.911	28.886	41.846	576.5	618.3	0.07718	0.9654	1.0426	142.02	0.2537	13.369	174

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TABLE 2. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Concluded)

Fahr. Temp. t(F)	Humidity Ratio W_s	Volume cu ft/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu per (°F) (lb dry air)			Condensed Water			Fahr. Temp. t(F)
		v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	Enthalpy Btu/Lb h_w	Entropy Btu/(Lb)(°F) s_w	Vap. Press In. Hg p_s	
175	0.5292	16.001	13.475	29.476	42.087	601.1	643.2	0.07756	1.005	1.083	143.02	0.2553	13.675	175
176	0.5519	16.026	14.074	30.100	42.328	627.1	669.4	0.07794	1.047	1.125	144.02	0.2568	13.987	176
177	0.5760	16.051	14.710	30.761	42.569	654.7	697.3	0.07832	1.091	1.169	145.02	0.2584	14.304	177
178	0.6016	16.076	15.386	31.462	42.810	684.1	726.9	0.07870	1.137	1.216	146.03	0.2600	14.628	178
179	0.6288	16.102	16.104	32.206	43.051	715.2	758.3	0.07908	1.187	1.266	147.03	0.2616	14.958	179
180	0.6578	16.127	16.870	32.997	43.292	748.5	791.8	0.07946	1.240	1.319	148.03	0.2631	15.294	180
181	0.6887	16.152	17.689	33.841	43.534	783.9	827.4	0.07984	1.296	1.376	149.03	0.2647	15.636	181
182	0.7218	16.177	18.565	34.742	43.775	821.9	865.7	0.08021	1.357	1.437	150.04	0.2662	15.985	182
183	0.7572	16.203	19.504	35.707	44.016	862.5	906.5	0.08059	1.421	1.502	151.04	0.2678	16.340	183
184	0.7953	16.228	20.513	36.741	44.257	906.2	950.5	0.08096	1.490	1.571	152.04	0.2693	16.702	184
185	0.8363	16.253	21.601	37.854	44.498	953.2	997.7	0.08134	1.565	1.646	153.05	0.2709	17.071	185
186	0.8805	16.278	22.775	39.053	44.740	1004	1049	0.08171	1.645	1.727	154.05	0.2724	17.446	186
187	0.9283	16.304	24.047	40.351	44.981	1059	1104	0.08208	1.731	1.813	155.05	0.2740	17.828	187
188	0.9802	16.329	25.427	41.756	45.222	1119	1164	0.08245	1.825	1.907	156.06	0.2755	18.217	188
189	1.037	16.354	26.934	43.288	45.463	1184	1229	0.08283	1.928	2.011	157.06	0.2771	18.614	189
190	1.099	16.379	28.580	44.959	45.704	1255	1301	0.08320	2.039	2.122	158.07	0.2786	19.017	190
191	1.166	16.405	30.385	46.790	45.946	1332	1378	0.08357	2.161	2.245	159.07	0.2802	19.427	191
192	1.241	16.430	32.375	48.805	46.187	1418	1464	0.08394	2.296	2.380	160.07	0.2817	19.845	192
193	1.324	16.455	34.581	51.036	46.428	1513	1559	0.08431	2.444	2.528	161.08	0.2833	20.271	193
194	1.416	16.480	37.036	53.516	46.670	1619	1666	0.08468	2.609	2.694	162.08	0.2848	20.704	194
195	1.519	16.506	39.785	56.291	46.911	1737	1784	0.08505	2.794	2.879	163.09	0.2864	21.145	195
196	1.635	16.531	42.885	59.416	47.153	1871	1918	0.08542	3.002	3.087	164.09	0.2879	21.594	196
197	1.767	16.556	46.402	62.958	47.394	2022	2069	0.08579	3.238	3.324	165.10	0.2895	22.050	197
198	1.917	16.581	50.426	67.007	47.636	2195	2243	0.08616	3.507	3.593	166.10	0.2910	22.514	198
199	2.091	16.607	55.074	71.681	47.877	2395	2443	0.08653	3.817	3.904	167.11	0.2925	22.987	199
200	2.295	16.632	60.510	77.142	48.119	2629	2677	0.08689	4.179	4.266	168.11	0.2940	23.468	200

[a] Compiled by John A. Goff and S. Gratch.

Table 3. Thermodynamic Properties of Water at Saturation[a]

Fahr. Temp. t(F)	Absolute Pressure $p_s \times 10^5$		Specific Volume, cu ft pere lb			Enthalpy, Btu per lb			Entropy. Btu per (Lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Solid v_i	Evap. $v_{ig} \times 10^{-5}$	Sat. Vapor $v_g \times 10^{-5}$	Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	
−160	0.004949	0.01008	0.01722	36070	36070	−222.05	1212.43	990.38	−0.4907	4.0456	3.5549	−160
−155	0.009040	0.01840	0.01723	20080	20080	−220.44	1213.02	992.58	−0.4854	3.9812	3.4958	−155
−150	0.01620	0.03298	0.01723	11390	11390	−218.82	1213.62	994.80	−0.4801	3.9188	3.4387	−150
−145	0.02850	0.05803	0.01724	6577	6577	−217.17	1214.17	997.00	−0.4748	3.8583	3.3835	−145
−140	0.04928	0.1003	0.01724	3864	3864	−215.49	1214.70	999.21	−0.4695	3.7996	3.3301	−140
−135	0.08380	0.1706	0.01725	2308	2308	−213.80	1215.22	1001.42	−0.4643	3.7428	3.2785	−135
−130	0.1403	0.2856	0.01725	1400	1400	−212.08	1215.71	1003.63	−0.4590	3.6874	3.2284	−130
−125	0.2312	0.4708	0.01726	862.2	862.2	−210.34	1216.18	1005.84	−0.4538	3.6338	3.1800	−125
−120	0.3757	0.7649	0.01726	538.6	538.6	−208.58	1216.63	1008.05	−0.4485	3.5815	3.1330	−120
−115	0.6019	1.226	0.01727	341.1	341.1	−206.79	1217.05	1010.26	−0.4433	3.5308	3.0875	−115
−110	0.9517	1.938	0.01728	218.9	218.9	−204.98	1217.45	1012.47	−0.4381	3.4815	3.0434	−110
−105	1.486	3.025	0.01728	142.2	142.2	−203.14	1217.82	1014.68	−0.4329	3.4335	3.0006	−105
−100	2.291	4.664	0.01729	93.52	93.52	−201.28	1218.17	1016.89	−0.4277	3.3868	2.9591	−100
−95	3.491	7.108	0.01729	62.23	62.23	−199.40	1218.50	1019.10	−0.4225	3.3412	2.9187	−95
−90	5.260	10.71	0.01730	41.86	41.86	−197.49	1218.80	1021.31	−0.4173	3.2969	2.8796	−90
−85	7.841	15.96	0.01730	28.46	28.46	−195.56	1219.08	1023.52	−0.4121	3.2536	2.8415	−85
−80	11.57	23.55	0.01731	19.55	19.55	−193.60	1219.33	1025.73	−0.4069	3.2114	2.8045	−80
−75	16.89	34.39	0.01732	13.56	13.56	−191.62	1219.56	1027.94	−0.4017	3.1702	2.7685	−75
−70	24.43	49.74	0.01732	9.501	9.501	−189.61	1219.76	1030.15	−0.3965	3.1301	2.7336	−70
−65	35.01	71.28	0.01733	6.715	6.715	−187.58	1219.94	1032.36	−0.3914	3.0910	2.6996	−65
−60	49.72	101.2	0.01734	4.788	4.788	−185.52	1220.10	1034.58	−0.3862	3.0526	2.6664	−60
−55	70.01	142.6	0.01734	3.443	3.443	−183.44	1220.23	1036.79	−0.3810	3.0152	2.6342	−55
−50	97.76	199.0	0.01735	2.496	2.496	−181.34	1220.34	1039.00	−0.3758	2.9786	2.6028	−50
−48	111.5	227.0	0.01736	2.200	2.200	−180.49	1220.37	1039.88	−0.3738	2.9643	2.5905	−48
−46	127.0	258.5	0.01736	1.941	1.941	−179.64	1220.40	1040.76	−0.3717	2.9501	2.5784	−46
−44	144.4	294.0	0.01736	1.715	1.715	−178.78	1220.43	1041.65	−0.3696	2.9359	2.5663	−44
−42	164.1	334.0	0.01736	1.516	1.516	−177.92	1220.45	1042.53	−0.3676	2.9219	2.5543	−42
−40	186.1	379.0	0.01737	1.343	1.343	−177.06	1220.48	1043.42	−0.3655	2.9080	2.5425	−40
−38	211.0	429.5	0.01737	1.191	1.191	−176.19	1220.49	1044.30	−0.3634	2.8942	2.5308	−38
−36	238.8	486.2	0.01737	1.057	1.057	−175.32	1220.51	1045.19	−0.3614	2.8807	2.5193	−36
−34	270.0	549.7	0.01737	0.9391	0.9391	−174.45	1220.52	1046.07	−0.3593	2.8671	2.5078	−34
−32	304.9	620.8	0.01738	0.8355	0.8355	−173.57	1220.52	1046.95	−0.3573	2.8538	2.4965	−32

[a] Compiled by John A. Goff and S. Gratch.

TABLE 3. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (*Continued*)

Fahr. Temp. t(F)	Absolute Pressure $p_s \times 10^2$		Specific Volume, cu ft per lb			Enthalpy Btu per lb			Entropy, Btu per (Lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Solid v_i	Evap. $v_{ig} \times 10^{-4}$	Sat. Vapor $v_g \times 10^{-4}$	Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	
−30	0.3440	0.7003	0.01738	7.441	7.441	−172.68	1220.52	1047.84	−0.3552	2.8405	2.4853	−30
−28	0.3876	0.7891	0.01738	6.634	6.634	−171.80	1220.52	1048.72	−0.3532	2.8274	2.4742	−28
−26	0.4363	0.8882	0.01738	5.921	5.921	−170.91	1220.51	1049.60	−0.3511	2.8143	2.4632	−26
−24	0.4905	0.9987	0.01739	5.290	5.290	−170.01	1220.50	1050.49	−0.3490	2.8013	2.4523	−24
−22	0.5509	1.122	0.01739	4.732	4.732	−169.12	1220.49	1051.37	−0.3470	2.7885	2.4415	−22
−20	0.6181	1.259	0.01739	4.237	4.237	−168.21	1220.47	1052.26	−0.3449	2.7757	2.4308	−20
−19	0.6545	1.333	0.01739	4.011	4.011	−167.76	1220.46	1052.70	−0.3439	2.7695	2.4256	−19
−18	0.6928	1.410	0.01740	3.797	3.797	−167.31	1220.45	1053.14	−0.3429	2.7632	2.4203	−18
−17	0.7332	1.493	0.01740	3.596	3.596	−166.85	1220.43	1053.58	−0.3418	2.7568	2.4150	−17
−16	0.7757	1.579	0.01740	3.407	3.407	−166.40	1220.42	1054.02	−0.3408	2.7506	2.4098	−16
−15	0.8204	1.670	0.01740	3.228	3.228	−165.94	1220.41	1054.47	−0.3398	2.7444	2.4046	−15
−14	0.8676	1.766	0.01740	3.060	3.060	−165.48	1220.39	1054.91	−0.3388	2.7383	2.3995	−14
−13	0.9172	1.867	0.01740	2.901	2.901	−165.03	1220.38	1055.35	−0.3377	2.7320	2.3943	−13
−12	0.9694	1.974	0.01740	2.750	2.750	−164.57	1220.36	1055.79	−0.3367	2.7259	2.3892	−12
−11	1.024	2.086	0.01740	2.609	2.609	−164.11	1220.34	1056.23	−0.3357	2.7198	2.3841	−11
−10	1.082	2.203	0.01741	2.475	2.475	−163.65	1220.32	1056.67	−0.3347	2.7138	2.3791	−10
−9	1.143	2.327	0.01741	2.349	2.349	−163.18	1220.30	1057.12	−0.3336	2.7076	2.3740	−9
−8	1.207	2.457	0.01741	2.229	2.229	−162.72	1220.28	1057.56	−0.3326	2.7016	2.3690	−8
−7	1.274	2.594	0.01741	2.116	2.116	−162.26	1220.26	1058.00	−0.3316	2.6956	2.3640	−7
−6	1.344	2.737	0.01741	2.010	2.010	−161.79	1220.23	1058.44	−0.3306	2.6896	2.3590	−6
−5	1.419	2.888	0.01741	1.909	1.909	−161.33	1220.21	1058.88	−0.3295	2.6836	2.3541	−5
−4	1.496	3.047	0.01742	1.814	1.814	−160.86	1220.18	1059.32	−0.3285	2.6777	2.3492	−4
−3	1.578	3.213	0.01742	1.723	1.723	−160.39	1220.15	1059.76	−0.3275	2.6718	2.3443	−3
−2	1.664	3.388	0.01742	1.638	1.638	−159.92	1220.13	1060.21	−0.3264	2.6658	2.3394	−2
−1	1.754	3.572	0.01742	1.557	1.557	−159.45	1220.10	1060.65	−0.3254	2.6600	2.3346	−1
0	1.849	3.764	0.01742	1.481	1.481	−158.98	1220.07	1061.09	−0.3244	2.6541	2.3297	0
1	1.948	3.966	0.01742	1.408	1.408	−158.51	1220.04	1061.53	−0.3234	2.6483	2.3249	1
2	2.052	4.178	0.01742	1.340	1.340	−158.04	1220.01	1061.97	−0.3224	2.6425	2.3201	2
3	2.161	4.400	0.01743	1.275	1.275	−157.56	1219.97	1062.41	−0.3213	2.6367	2.3154	3
4	2.276	4.633	0.01743	1.214	1.214	−157.09	1219.94	1062.85	−0.3203	2.6309	2.3106	4
5	2.396	4.878	0.01743	1.155	1.155	−156.61	1219.90	1063.29	−0.3193	2.6252	2.3059	5
6	2.521	5.134	0.01743	1.100	1.100	−156.14	1219.88	1063.74	−0.3182	2.6194	2.3012	6

[a] Compiled by John A. Goff and S. Gratch.

Table 3. Thermodynamic Properties of Water at Saturation[a] (Continued)

Fahr. Temp t(F)	Absolute Pressure p_s		Specific Volume. cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (Lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Solid v_i	Evap. v_{ig}	Sat. Vapor v_g	Sat Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	
7	0.02653	0.05402	0.01743	10480	10480	−155.66	1219.84	1064.18	−0.3172	2.6138	2.2966	7
8	0.02791	0.05683	0.01743	9979	9979	−155.18	1219.80	1064.62	−0.3162	2.6081	2.2919	8
9	0.02936	0.05977	0.01744	9507	9507	−154.70	1219.76	1065.06	−0.3152	2.6025	2.2873	9
10	0.03087	0.06286	0.01744	9060	9060	−154.22	1219.72	1065.50	−0.3142	2.5969	2.2827	10
11	0.03246	0.06608	0.01744	8636	8636	−153.74	1219.68	1065.94	−0.3131	2.5912	2.2781	11
12	0.03412	0.06946	0.01744	8234	8234	−153.26	1219.64	1066.38	−0.3121	2.5857	2.2736	12
13	0.03585	0.07300	0.01744	7851	7851	−152.77	1219.59	1066.82	−0.3111	2.5801	2.2690	13
14	0.03767	0.07669	0.01744	7489	7489	−152.29	1219.55	1067.26	−0.3101	2.5746	2.2645	14
15	0.03957	0.08056	0.01744	7144	7144	−151.80	1219.50	1067.70	−0.3090	2.5690	2.2600	15
16	0.04156	0.08461	0.01745	6817	6817	−151.32	1219.46	1068.14	−0.3080	2.5635	2.2555	16
17	0.04363	0.08884	0.01745	6505	6505	−150.83	1219.41	1068.58	−0.3070	2.5581	2.2511	17
18	0.04581	0.09326	0.01745	6210	6210	−150 34	1219.36	1069.02	−0.3060	2.5526	2.2466	18
19	0.04808	0.09789	0.01745	5929	5929	−149.85	1219.31	1069.46	−0.3049	2.5471	2.2422	19
20	0.05045	0.1027	0.01745	5662	5662	−149.36	1219.26	1069.90	−0.3039	2.5417	2.2378	20
21	0.05293	0.1078	0.01745	5408	5408	−148.87	1219.21	1070.34	−0.3029	2.5364	2.2335	21
22	0.05552	0.1130	0.01746	5166	5166	−148.38	1219.16	1070.78	−0.3019	2.5310	2.2291	22
23	0.05823	0.1186	0.01746	4936	4936	−147.88	1219.10	1071.22	−0.3008	2.5256	2.2248	23
24	0.06105	0.1243	0.01746	4717	4717	−147.39	1219.05	1071.66	−0.2998	2.5203	2.2205	24
25	0.06400	0.1303	0.01746	4509	4509	−146 89	1218.98	1072.09	−0.2988	2.5150	2.2162	25
26	0.06708	0.1366	0.01746	4311	4311	−146.40	1218.93	1072.53	−0.2978	2.5097	2.2119	26
27	0.07030	0.1431	0.01746	4122	4122	−145.90	1218.87	1072.97	−0.2968	2.5045	2.2077	27
28	0.07365	0.1500	0.01746	3943	3943	−145 40	1218.81	1073.41	−0.2957	2.4991	2.2034	28
29	0.07715	0.1571	0.01747	3771	3771	−144.90	1218.75	1073.85	−0.2947	2.4939	2.1992	29
30	0.08080	0.1645	0.01747	3608	3608	−144.40	1218.69	1074.29	−0.2937	2.4887	2.1950	30
31	0.08461	0.1723	0.01747	3453	3453	−143.90	1218.63	1074.73	−0.2927	2.4835	2.1908	31
32	0.08858	0.1803	0.01747	3305	3305	−143.40	1218.56	1075.16	−0.2916	2.4783	2.1867	32
*32	0.088586	0.18036	0.01602	3304.6	3304.6	0.00	1075.16	1075.16	0.00000	2.1867	2.1867	32*

[a] Compiled by John A. Goff and S. Gratch.

* Extrapolated to represent metastable equilibrium with undercooled liquid.

TABLE 3. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	
*32	0.088586	0.18036	0.01602	3304.6	3304.6	0.00	1075.16	1075.16	0.00000	2.1867	2.1867	32*
33	0.092227	0.18778	0.01602	3180.5	3180.5	1.01	1074.59	1075.60	0.00205	2.1811	2.1831	33
34	0.095999	0.19546	0.01602	3061.7	3061.7	2.01	1074.03	1076.04	0.00409	2.1755	2.1796	34
35	0.099908	0.20342	0.01602	2947.8	2947.8	3.02	1073.46	1076.48	0.00612	2.1700	2.1761	35
36	0.10396	0.21166	0.01602	2838.7	2838.7	4.02	1072.90	1076.92	0.00815	2.1644	2.1726	36
37	0.10815	0.22020	0.01602	2734.1	2734.1	5.03	1072.33	1077.36	0.01018	2.1589	2.1691	37
38	0.11249	0.22904	0.01602	2633.8	2633.8	6.03	1071.77	1077.80	0.01220	2.1535	2.1657	38
39	0.11699	0.23819	0.01602	2537.6	2537.6	7.04	1071.20	1078.24	0.01422	2.1480	2.1622	39
40	0.12164	0.24767	0.01602	2445.4	2445.4	8.04	1070.64	1078.68	0.01623	2.1426	2.1588	40
41	0.12646	0.25748	0.01602	2356.9	2356.9	9.05	1070.06	1079.11	0.01824	2.1372	2.1554	41
42	0.13145	0.26763	0.01602	2272.0	2272.0	10.05	1069.50	1079.55	0.02024	2.1318	2.1520	42
43	0.13660	0.27813	0.01602	2190.5	2190.5	11.05	1068.94	1079.99	0.02224	2.1265	2.1487	43
44	0.14194	0.28899	0.01602	2112.3	2112.3	12.06	1068.37	1080.43	0.02423	2.1211	2.1453	44
45	0.14746	0.30023	0.01602	2037.3	2037.3	13.06	1067.81	1080.87	0.02622	2.1158	2.1420	45
46	0.15317	0.31185	0.01602	1965.2	1965.2	14.06	1067.24	1081.30	0.02820	2.1105	2.1387	46
47	0.15907	0.32387	0.01602	1896.0	1896.0	15.06	1066.68	1081.74	0.03018	2.1052	2.1354	47
48	0.16517	0.33629	0.01602	1829.5	1829.5	16.07	1066.11	1082.18	0.03216	2.0999	2.1321	48
49	0.17148	0.34913	0.01602	1765.7	1765.7	17.07	1065.55	1082.62	0.03413	2.0947	2.1288	49
50	0.17799	0.36240	0.01602	1704.3	1704.3	18.07	1064.99	1083.06	0.03610	2.0895	2.1256	50
51	0.18473	0.37611	0.01602	1645.4	1645.4	19.07	1064.42	1083.49	0.03806	2.0842	2.1223	51
52	0.19169	0.39028	0.01602	1588.7	1588.7	20.07	1063.86	1083.93	0.04002	2.0791	2.1191	52
53	0.19888	0.40492	0.01603	1534.3	1534.3	21.07	1063.30	1084.37	0.04197	2.0739	2.1159	53
54	0.20630	0.42003	0.01603	1481.9	1481.9	22.08	1062.72	1084.80	0.04392	2.0688	2.1127	54
55	0.21397	0.43564	0.01603	1431.5	1431.5	23.08	1062.16	1085.24	0.04587	2.0637	2.1096	55
56	0.22188	0.45176	0.01603	1383.1	1383.1	24.08	1061.60	1085.68	0.04781	2.0586	2.1064	56
57	0.23006	0.46840	0.01603	1336.5	1336.5	25.08	1061.04	1086.12	0.04975	2.0535	2.1033	57
58	0.23849	0.48558	0.01603	1291.7	1291.7	26.08	1060.47	1086.55	0.05168	2.0485	2.1002	58
59	0.24720	0.50330	0.01603	1248.6	1248.6	27.08	1059.91	1086.99	0.05361	2.0434	2.0970	59
60	0.25618	0.52160	0.01603	1207.1	1207.1	28.08	1059.34	1087.42	0.05553	2.0385	2.0940	60
61	0.26545	0.54047	0.01604	1167.2	1167.2	29.08	1058.78	1087.86	0.05746	2.0334	2.0909	61

[a] Compiled by John A. Goff and S. Gratch.
* Extrapolated to represent metastable equilibrium with undercooled liquid.

TABLE 3. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	
62	0.27502	0.55994	0.01604	1128.7	1128.7	30.08	1058.22	1088.30	0.05937	2.0284	2.0878	62
63	0.28488	0.58002	0.01604	1091.7	1091.7	31.08	1057.65	1088.73	0.06129	2.0235	2.0848	63
64	0.29505	0.60073	0.01604	1056.1	1056.1	32.08	1057.09	1089.17	0.06320	2.0186	2.0818	64
65	0.30554	0.62209	0.01604	1021.7	1021.7	33.08	1056.52	1089.60	0.06510	2.0136	2.0787	65
66	0.31636	0.64411	0.01604	988.63	988.65	34.07	1055.97	1090.04	0.06700	2.0087	2.0757	66
67	0.32750	0.66681	0.01605	956.76	956.78	35.07	1055.40	1090.47	0.06890	2.0039	2.0728	67
68	0.33900	0.69021	0.01605	926.06	926.08	36.07	1054.84	1090.91	0.07080	1.9990	2.0698	68
69	0.35084	0.71432	0.01605	896.47	896.49	37.07	1054.27	1091.34	0.07269	1.9941	2.0668	69
70	0.36304	0.73916	0.01605	867.95	867.97	38.07	1053.71	1091.78	0.07458	1.9893	2.0639	70
71	0.37561	0.76476	0.01605	840.45	840.47	39.07	1053.14	1092.21	0.07646	1.9845	2.0610	71
72	0.38856	0.79113	0.01606	813.95	813.97	40.07	1052.58	1092.65	0.07834	1.9797	2.0580	72
73	0.40190	0.81829	0.01606	788.38	788.40	41.07	1052.01	1093.08	0.08022	1.9749	2.0551	73
74	0.41564	0.84626	0.01606	763.73	763.75	42.06	1051.46	1093.52	0.08209	1.9701	2.0522	74
75	0.42979	0.87506	0.01606	739.95	739.97	43.06	1050.89	1093.95	0.08396	1.9654	2.0494	75
76	0.44435	0.90472	0.01606	717.01	717.03	44.06	1050.32	1094.38	0.08582	1.9607	2.0465	76
77	0.45935	0.93524	0.01607	694.88	694.90	45.06	1049.76	1094.82	0.08769	1.9560	2.0437	77
78	0.47478	0.96666	0.01607	673.52	673.54	46.06	1049.19	1095.25	0.08954	1.9513	2.0408	78
79	0.49066	0.99900	0.01607	652.91	652.93	47.06	1048.62	1095.68	0.09140	1.9466	2.0380	79
80	0.50701	1.0323	0.01607	633.01	633.03	48.05	1048.07	1096.12	0.09325	1.9419	2.0352	80
81	0.52382	1.0665	0.01608	613.80	613.82	49.05	1047.50	1096.55	0.09510	1.9373	2.0324	81
82	0.54112	1.1017	0.01608	595.25	595.27	50.05	1046.93	1096.98	0.09694	1.9328	2.0297	82
83	0.55892	1.1380	0.01608	577.34	577.36	51.05	1046.37	1097.42	0.09878	1.9281	2.0269	83
84	0.57722	1.1752	0.01608	560.04	560.06	52.05	1045.80	1097.85	0.10062	1.9236	2.0242	84
85	0.59604	1.2136	0.01609	543.33	543.35	53.05	1045.23	1098.28	0.10246	1.9189	2.0214	85
86	0.61540	1.2530	0.01609	527.19	527.21	54.04	1044.67	1098.71	0.10429	1.9144	2.0187	86
87	0.63530	1.2935	0.01609	511.60	511.62	55.04	1044.10	1099.14	0.10611	1.9099	2.0160	87
88	0.65575	1.3351	0.01610	496.52	496.54	56.04	1043.54	1099.58	0.10794	1.9054	2.0133	88
89	0.67678	1.3779	0.01610	481.96	481.98	57.04	1042.97	1100.01	0.10976	1.9008	2.0106	89
90	0.69838	1.4219	0.01610	467.88	467.90	58.04	1042.40	1100.44	0.11158	1.8963	2.0079	90
91	0.72059	1.4671	0.01610	454.26	454.28	59.03	1041.84	1100.87	0.11339	1.8919	2.0053	91

[a] Compiled by John A. Goff and S. Gratch.

TABLE 3. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s Lb/Sq In.	Absolute Pressure p_s In. Hg	Specific Volume, cu ft per lb Sat. Liquid v_f	Specific Volume Evap. v_{fg}	Specific Volume Sat. Vapor v_g	Enthalpy, Btu per lb Sat. Liquid h_f	Enthalpy Evap. h_{fg}	Enthalpy Sat. Vapor h_g	Entropy, Btu per (lb) (°F) Sat. Liquid s_f	Entropy Evap. s_{fg}	Entropy Sat. Vapor s_g	Fahr. Temp. t(F)
92	0.74340	1.5136	0.01611	441.10	441.12	60.03	1041.27	1101.30	0.11520	1.8874	2.0026	92
93	0.76684	1.5613	0.01611	428.38	428.40	61.03	1040.70	1101.73	0.11701	1.8830	2.0000	93
94	0.79091	1.6103	0.01611	416.07	416.09	62.03	1040.13	1102.16	0.11881	1.8786	1.9974	94
95	0.81564	1.6607	0.01612	404.17	404.19	63.03	1039.56	1102.59	0.12061	1.8741	1.9947	95
96	0.84103	1.7124	0.01612	392.65	392.67	64.02	1039.00	1103.02	0.12241	1.8698	1.9922	96
97	0.86711	1.7655	0.01612	381.51	381.53	65.02	1038.43	1103.45	0.12420	1.8654	1.9896	97
98	0.89388	1.8200	0.01612	370.73	370.75	66.02	1037.86	1103.88	0.12600	1.8610	1.9870	98
99	0.92137	1.8759	0.01613	360.30	360.32	67.02	1037.29	1104.31	0.12778	1.8566	1.9844	99
100	0.94959	1.9334	0.01613	350.20	350.22	68.02	1036.72	1104.74	0.12957	1.8523	1.9819	100
101	0.97854	1.9923	0.01614	340.42	340.44	69.01	1036.16	1105.17	0.13135	1.8480	1.9793	101
102	1.0083	2.0529	0.01614	330.96	330.98	70.01	1035.58	1105.59	0.13313	1.8437	1.9768	102
103	1.0388	2.1149	0.01614	321.80	321.82	71.01	1035.01	1106.02	0.13490	1.8394	1.9743	103
104	1.0700	2.1786	0.01614	312.93	312.95	72.01	1034.44	1106.45	0.13667	1.8351	1.9718	104
105	1.1021	2.2440	0.01615	304.34	304.36	73.01	1033.87	1106.88	0.13844	1.8309	1.9693	105
106	1.1351	2.3110	0.01615	296.02	296.04	74.01	1033.29	1107.30	0.14021	1.8266	1.9668	106
107	1.1688	2.3798	0.01616	287.96	287.98	75.00	1032.73	1107.73	0.14197	1.8224	1.9644	107
108	1.2035	2.4503	0.01616	280.14	280.16	76.00	1032.16	1108.16	0.14373	1.8182	1.9619	108
109	1.2390	2.5226	0.01616	272.58	272.60	77.00	1031.58	1108.58	0.14549	1.8140	1.9595	109
110	1.2754	2.5968	0.01617	265.24	265.26	78.00	1031.01	1109.01	0.14724	1.8098	1.9570	110
111	1.3128	2.6728	0.01617	258.14	258.16	79.00	1030.44	1109.44	0.14899	1.8056	1.9546	111
112	1.3510	2.7507	0.01617	251.25	251.27	80.00	1029.86	1109.86	0.15074	1.8015	1.9522	112
113	1.3902	2.8306	0.01618	244.57	244.59	80.99	1029.30	1110.29	0.15248	1.7973	1.9498	113
114	1.4305	2.9125	0.01618	238.10	238.12	81.99	1028.72	1110.71	0.15423	1.7932	1.9474	114
115	1.4717	2.9963	0.01618	231.82	231.84	82.99	1028.15	1111.14	0.15596	1.7890	1.9450	115
116	1.5139	3.0823	0.01619	225.73	225.75	83.99	1027.57	1111.56	0.15770	1.7849	1.9426	116
117	1.5571	3.1703	0.01619	219.83	219.85	84.99	1026.99	1111.98	0.15943	1.7809	1.9403	117
118	1.6014	3.2606	0.01620	214.10	214.12	85.99	1026.42	1112.41	0.16116	1.7767	1.9379	118
119	1.6468	3.3530	0.01620	208.54	208.56	86.98	1025.85	1112.83	0.16289	1.7727	1.9356	119
120	1.6933	3.4477	0.01620	203.16	203.18	87.98	1025.28	1113.26	0.16461	1.7687	1.9333	120
121	1.7409	3.5446	0.01621	197.93	197.95	88.98	1024.70	1113.68	0.16634	1.7647	1.9310	121

[a] Compiled by John A. Goff and S. Gratch.

Table 3. Thermodynamic Properties of Water at Saturation[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat Vapor h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	
122	1.7897	3.6439	0.01621	192.85	192.87	89.98	1024.12	1114.10	0.16805	1.7606	1.9286	122
123	1.8396	3.7455	0.01622	187.93	187.95	90.98	1023.54	1114.52	0.16977	1.7566	1.9264	123
124	1.8907	3.8496	0.01622	183.15	183.17	91.98	1022.96	1114.94	0.17148	1.7526	1.9241	124
125	1.9430	3.9561	0.01622	178.51	178.53	92.98	1022.39	1115.37	0.17319	1.7486	1.9218	125
126	1.9966	4.0651	0.01623	174.00	174.02	93.98	1021.81	1115.79	0.17490	1.7446	1.9195	126
127	2.0514	4.1768	0.01623	169.63	169.65	94.97	1021.24	1116.21	0.17660	1.7407	1.9173	127
128	2.1075	4.2910	0.01624	165.38	165.40	95.97	1020.66	1116.63	0.17830	1.7367	1.9150	128
129	2.1649	4.4078	0.01624	161.26	161.28	96.97	1020.08	1117.05	0.18000	1.7328	1.9128	129
130	2.2237	4.5274	0.01625	157.25	157.27	97.97	1019.50	1117.47	0.18170	1.7289	1.9106	130
131	2.2838	4.6498	0.01625	153.36	153.38	98.97	1018.92	1117.89	0.18339	1.7250	1.9084	131
132	2.3452	4.7750	0.01626	149.58	149.60	99.97	1018.34	1118.31	0.18508	1.7211	1.9062	132
133	2.4081	4.9030	0.01626	145.91	145.93	100.97	1017.76	1118.73	0.18676	1.7172	1.9040	133
134	2.4725	5.0340	0.01626	142.34	142.36	101.97	1017.18	1119.15	0.18845	1.7134	1.9018	134
135	2.5382	5.1679	0.01627	138.87	138.89	102.97	1016.59	1119.56	0.19013	1.7095	1.8996	135
136	2.6055	5.3049	0.01627	135.50	135.52	103.97	1016.01	1119.98	0.19181	1.7056	1.8974	136
137	2.6743	5.4450	0.01628	132.22	132.24	104.97	1015.43	1120.40	0.19348	1.7018	1.8953	137
138	2.7446	5.5881	0.01628	129.04	129.06	105.97	1014.85	1120.82	0.19516	1.6979	1.8931	138
139	2.8165	5.7345	0.01629	125.94	125.96	106.97	1014.26	1121.23	0.19683	1.6942	1.8910	139
140	2.8900	5.8842	0.01629	122.94	122.96	107.96	1013.69	1121.65	0.19850	1.6903	1.8888	140
141	2.9651	6.0371	0.01630	120.01	120.03	108.96	1013.11	1122.07	0.20016	1.6865	1.8867	141
142	3.0419	6.1934	0.01630	117.16	117.18	109.96	1012.52	1122.48	0.20182	1.6828	1.8846	142
143	3.1204	6.3532	0.01631	114.40	114.42	110.96	1011.94	1122.90	0.20348	1.6790	1.8825	143
144	3.2006	6.5164	0.01631	111.70	111.72	111.96	1011.35	1123.31	0.20514	1.6753	1.8804	144
145	3.2825	6.6832	0.01632	109.09	109.11	112.96	1010.77	1123.73	0.20679	1.6715	1.8783	145
146	3.3662	6.8536	0.01632	106.54	106.56	113.96	1010.18	1124.14	0.20845	1.6678	1.8763	146
147	3.4517	7.0277	0.01633	104.06	104.08	114.96	1009.59	1124.55	0.21010	1.6641	1.8742	147
148	3.5390	7.2056	0.01633	101.65	101.67	115.96	1009.01	1124.97	0.21174	1.6604	1.8721	148
149	3.6282	7.3872	0.01634	99.306	99.322	116.96	1008.42	1125.38	0.21339	1.6567	1.8701	149
150	3.7194	7.5727	0.01634	97.022	97.038	117.96	1007.83	1125.79	0.21503	1.6530	1.8680	150
151	3.8124	7.7622	0.01635	94.799	94.815	118.96	1007.24	1126.20	0.21667	1.6493	1.8660	151

[a] Compiled by John A. Goff and S. Gratch.

TABLE 3. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s		Specific Volume cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	
152	3.9074	7.9556	0.01635	92.635	92.651	119.96	1006.66	1126.62	0.21830	1.6457	1.8640	152
153	4.0044	8.1532	0.01636	90.528	90.544	120.97	1006.06	1127.03	0.21994	1.6421	1.8620	153
154	4.1035	8.3548	0.01636	88.477	88.493	121.97	1005.47	1127.44	0.22157	1.6384	1.8600	154
155	4.2046	8.5607	0.01637	86.480	86.496	122.97	1004.88	1127.85	0.22320	1.6348	1.8580	155
156	4.3078	8.7708	0.01637	84.536	84.552	123.97	1004.29	1128.26	0.22482	1.6312	1.8560	156
157	4.4132	8.9853	0.01638	82.642	82.658	124.97	1003.70	1128.67	0.22645	1.6276	1.8540	157
158	4.5207	9.2042	0.01638	80.798	80.814	125.97	1003.11	1129.08	0.22807	1.6239	1.8520	158
159	4.6304	9.4276	0.01639	79.001	79.017	126.97	1002.51	1129.48	0.22969	1.6204	1.8501	159
160	4.7424	9.6556	0.01639	77.251	77.267	127.97	1001.92	1129.89	0.23130	1.6168	1.8481	160
161	4.8566	9.8882	0.01640	75.546	75.562	128.97	1001.33	1130.30	0.23292	1.6133	1.8462	161
162	4.9732	10.126	0.01640	73.885	73.901	129.97	1000.74	1130.71	0.23453	1.6097	1.8442	162
163	5.0921	10.368	0.01641	72.267	72.283	130.98	1000.13	1131.11	0.23614	1.6062	1.8423	163
164	5.2134	10.615	0.01642	70.690	70.706	131.98	999.54	1131.52	0.23774	1.6027	1.8404	164
165	5.3372	10.867	0.01642	69.153	69.169	132.98	998.94	1131.92	0.23935	1.5990	1.8384	165
166	5.4634	11.124	0.01643	67.654	67.670	133.98	998.35	1132.33	0.24095	1.5956	1.8365	166
167	5.5921	11.386	0.01643	66.194	66.210	134.98	997.75	1132.73	0.24255	1.5920	1.8346	167
168	5.7233	11.653	0.01644	64.770	64.786	135.98	997.16	1133.14	0.24414	1.5887	1.8328	168
169	5.8572	11.925	0.01644	63.382	63.398	136.99	996.55	1133.54	0.24574	1.5852	1.8309	169
170	5.9936	12.203	0.01645	62.029	62.045	137.99	995.95	1133.94	0.24733	1.5817	1.8290	170
171	6.1328	12.487	0.01645	60.710	60.726	138.99	995.36	1134.35	0.24892	1.5782	1.8271	171
172	6.2746	12.775	0.01646	59.423	59.439	139.99	994.76	1134.75	0.25051	1.5748	1.8253	172
173	6.4192	13.070	0.01647	58.168	58.184	141.00	994.15	1135.15	0.25209	1.5713	1.8234	173
174	6.5666	13.370	0.01647	56.944	56.960	142.00	993.55	1135.55	0.25367	1.5679	1.8216	174
175	6.7168	13.676	0.01648	55.750	55.766	143.00	992.95	1135.95	0.25525	1.5644	1.8197	175
176	6.8699	13.987	0.01648	54.586	54.602	144.00	992.35	1136.35	0.25683	1.5611	1.8179	176
177	7.0259	14.305	0.01649	53.450	53.466	145.00	991.75	1136.75	0.25841	1.5577	1.8161	177
178	7.1849	14.629	0.01650	52.341	52.357	146.01	991.14	1137.15	0.25998	1.5543	1.8143	178
179	7.3469	14.959	0.01650	51.260	51.276	147.01	990.54	1137.55	0.26155	1.5508	1.8124	179
180	7.5119	15.295	0.01651	50.203	50.220	148.01	989.93	1137.94	0.26312	1.5475	1.8106	180
181	7.6801	15.637	0.01651	49.173	49.190	149.02	989.32	1138.34	0.26468	1.5442	1.8089	181

[a] Compiled by John A. Goff and S. Gratch.

TABLE 3. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Concluded)

Fahr. Temp. t(F)	Absolute Pressure p_s		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Liquid v_f	Evap. v_{fg}	Sat. Vapor v_g	Sat. Liquid h_f	Evap h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Evap. s_{fg}	Sat. Vapor s_g	
182	7.8514	15.986	0.01652	48.168	48.185	150.02	988.72	1138.74	0.26625	1.5408	1.8071	182
183	8.0258	16.341	0.01652	47.187	47.204	151.02	988.12	1139.14	0.26781	1.5375	1.8053	183
184	8.2035	16.703	0.01653	46.229	46.246	152.03	987.50	1139.53	0.26937	1.5341	1.8035	184
185	8.3845	17.071	0.01654	45.294	45.311	153.03	986.89	1139.92	0.27093	1.5308	1.8017	185
186	8.5688	17.446	0.01654	44.381	44.398	154.04	986.28	1140.32	0.27248	1.5275	1.8000	186
187	8.7565	17.829	0.01655	43.489	43.506	155.04	985.67	1140.71	0.27404	1.5242	1.7982	187
188	8.9476	18.218	0.01656	42.619	42.636	156.04	985.07	1141.11	0.27559	1.5209	1.7965	188
189	9.1422	18.614	0.01656	41.769	41.786	157.05	984.45	1141.50	0.27713	1.5176	1.7947	189
190	9.3403	19.017	0.01657	40.939	40.956	158.05	983.84	1141.89	0.27868	1.5143	1.7930	190
191	9.5420	19.428	0.01658	40.128	40.145	159.06	983.22	1142.28	0.28022	1.5111	1.7913	191
192	9.7473	19.846	0.01658	39.337	39.354	160.06	982.61	1142.67	0.28176	1.5078	1.7896	192
193	9.9563	20.271	0.01659	38.563	38.580	161.06	982.00	1143.06	0.28330	1.5045	1.7878	193
194	10.169	20.704	0.01659	37.807	37.824	162.07	981.38	1143.45	0.28484	1.5013	1.7861	194
195	10.386	21.145	0.01660	37.069	37.086	163.08	980.76	1143.84	0.28638	1.4980	1.7844	195
196	10.606	21.594	0.01661	36.348	36.365	164.08	980.15	1144.23	0.28791	1.4949	1.7828	196
197	10.830	22.050	0.01661	35.643	35.660	165.08	979.54	1144.62	0.28944	1.4917	1.7811	197
198	11.058	22.515	0.01662	34.954	34.971	166.09	978.91	1145.00	0.29097	1.4884	1.7794	198
199	11.290	22.987	0.01663	34.281	34.298	167.10	978.29	1145.39	0.29250	1.4852	1.7777	199
200	11.526	23.468	0.01663	33.623	33.640	168.10	977.68	1145.78	0.29402	1.4820	1.7760	200
201	11.767	23.957	0.01664	32.980	32.997	169.11	977.05	1146.16	0.29554	1.4789	1.7744	201
202	12.011	24.455	0.01665	32.351	32.368	170.11	976.43	1146.54	0.29706	1.4756	1.7727	202
203	12.260	24.961	0.01665	31.737	31.754	171.12	975.81	1146.93	0.29858	1.4725	1.7711	203
204	12.513	25.476	0.01666	31.136	31.153	172.12	975.19	1147.31	0.30010	1.4693	1.7694	204
205	12.770	26.000	0.01667	30.549	30.566	173.13	974.56	1147.69	0.30161	1.4662	1.7678	205
206	13.031	26.532	0.01667	29.974	29.991	174.14	973.94	1148.08	0.30312	1.4631	1.7662	206
207	13.297	27.074	0.01668	29.413	29.430	175.14	973.32	1148.46	0.30463	1.4600	1.7646	207
208	13.568	27.625	0.01669	28.863	28.880	176.15	972.69	1148.84	0.30614	1.4568	1.7629	208
209	13.843	28.185	0.01669	28.326	28.343	177.16	972.06	1149.22	0.30765	1.4536	1.7613	209
210	14.123	28.754	0.01670	27.801	27.818	178.17	971.43	1149.60	0.30915	1.4506	1.7597	210
211	14.407	29.333	0.01671	27.287	27.304	179.17	970.81	1149.98	0.31065	1.4474	1.7581	211
212	14.696	29.921	0.01671	26.784	26.801	180.18	970.17	1150.35	0.31215	1.4444	1.7565	212

[a] Compiled by John A. Goff and S. Gratch.

Table 4. Properties of Saturated Steam: Pressure Table[a]

Abs. Press. In. Hg p	Temp F t	Specific Volume: Sat. Liquid v_f	Specific Volume: Sat. Vapor v_g	Enthalpy: Sat. Liquid h_f	Enthalpy: Evap. h_{fg}	Enthalpy: Sat. Vapor h_g	Entropy: Sat. Liquid S_f	Entropy: Evap. S_{fg}	Entropy: Sat. Vapor S_g	Abs. Press. In. Hg p
0.25	40.23	0.01602	2423.7	8.28	1071.1	1079.4	0.0166	2.1423	2.1589	0.25
0.50	58.80	0.01604	1256.4	26.86	1060.6	1087.5	0.0532	2.0453	2.0985	0.50
0.75	70.43	0.01606	856.1	38.47	1054.0	1092.5	0.0754	1.9881	2.0635	0.75
1.00	79.03	0.01608	652.3	47.05	1049.2	1096.3	0.0914	1.9473	2.0387	1.00
1.5	91.72	0.01611	444.9	59.71	1042.0	1101.7	0.1147	1.8894	2.0041	1.5
2	101.14	0.01614	339.2	69.10	1036.6	1105.7	0.1316	1.8481	1.9797	2
4	125.43	0.01622	176.7	93.34	1022.7	1116.0	0.1738	1.7476	1.9214	4
6	140.78	0.01630	120.72	108.67	1013.6	1122.3	0.1996	1.6881	1.8877	6
8	152.24	0.01635	92.16	120.13	1006.9	1127.0	0.2186	1.6454	1.8640	8
10	161.49	0.01640	74.76	129.38	1001.4	1130.8	0.2335	1.6121	1.8456	10
12	169.28	0.01644	63.03	137.18	996.7	1133.9	0.2460	1.5847	1.8307	12
14	176.05	0.01648	54.55	143.96	992.6	1136.6	0.2568	1.5613	1.8181	14
16	182.05	0.01652	48.14	149.98	988.9	1138.9	0.2662	1.5410	1.8072	16
18	187.45	0.01655	43.11	155.39	985.7	1141.1	0.2746	1.5231	1.7977	18
20	192.37	0.01658	39.07	160.33	982.7	1143.0	0.2822	1.5069	1.7891	20
22	196.90	0.01661	35.73	164.87	979.8	1144.7	0.2891	1.4923	1.7814	22
24	201.09	0.01664	32.94	169.09	977.2	1146.3	0.2955	1.4789	1.7744	24
26	205.00	0.01667	30.56	173.02	974.8	1147.8	0.3014	1.4665	1.7679	26
28	208.67	0.01669	28.52	176.72	972.5	1149.2	0.3069	1.4550	1.7619	28
30	212.13	0.01672	26.74	180.19	970.3	1150.5	0.3122	1.4442	1.7564	30
Lb/Sq In.										Lb/Sq In.
14.696	212.00	0.01672	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566	14.696
16	216.32	0.01674	24.75	184.42	967.6	1152.0	0.3184	1.4313	1.7497	16
18	222.41	0.01679	22.17	190.56	963.6	1154.2	0.3275	1.4128	1.7403	18
20	227.96	0.01683	20.089	196.16	960.1	1156.3	0.3356	1.3962	1.7319	20
22	233.07	0.01687	18.375	201.33	956.8	1158.1	0.3431	1.3811	1.7242	22
24	237.82	0.01691	16.938	206.14	953.7	1159.8	0.3500	1.3672	1.7172	24
26	242.25	0.01694	15.715	210.62	950.7	1161.3	0.3564	1.3544	1.7108	26
28	246.41	0.01698	14.663	214.83	947.9	1162.7	0.3623	1.3425	1.7048	28
30	250.33	0.01701	13.746	218.82	945.3	1164.1	0.3680	1.3313	1.6993	30
32	254.05	0.01704	12.940	222.59	942.8	1165.4	0.3733	1.3209	1.6941	32
34	257.58	0.01707	12.226	226.18	940.3	1166.5	0.3783	1.3110	1.6893	34
36	260.95	0.01709	11.588	229.60	938.0	1167.6	0.3831	1.3017	1.6848	36
38	264.16	0.01712	11.015	232.89	935.8	1168.7	0.3876	1.2929	1.6805	38
40	267.25	0.01715	10.498	236.03	933.7	1169.7	0.3919	1.2844	1.6763	40
42	270.21	0.01717	10.029	239.04	931.6	1170.7	0.3960	1.2764	1.6724	42
44	273.05	0.01720	9.601	241.95	929.6	1171.6	0.4000	1.2687	1.6687	44
46	275.80	0.01722	9.209	244.75	927.7	1172.4	0.4038	1.2613	1.6652	46
48	278.45	0.01725	8.848	247.47	925.8	1173.3	0.4075	1.2542	1.6617	48
50	281.01	0.01727	8.515	250.09	924.0	1174.1	0.4110	1.2474	1.6585	50
52	283.49	0.01729	8.208	252.63	922.2	1174.8	0.4144	1.2409	1.6553	52
54	285.90	0.01731	7.922	255.09	920.5	1175.6	0.4177	1.2346	1.6523	54
56	288.23	0.01733	7.656	257.50	918.8	1176.3	0.4209	1.2285	1.6494	56
58	290.50	0.01736	7.407	259.82	917.1	1176.9	0.4240	1.2226	1.6466	58
60	292.71	0.01738	7.175	262.09	915.5	1177.6	0.4270	1.2168	1.6438	60
62	294.85	0.01740	6.957	264.30	913.9	1178.2	0.4300	1.2112	1.6412	62
64	296.94	0.01742	6.752	266.45	912.3	1178.8	0.4328	1.2059	1.6387	64
66	298.99	0.01744	6.560	268.55	910.8	1179.4	0.4356	1.2006	1.6362	66
68	300.98	0.01746	6.378	270.60	909.4	1180.0	0.4383	1.1955	1.6338	68
70	302.92	0.01748	6.206	272.61	907.9	1180.6	0.4409	1.1906	1.6315	70
72	304.83	0.01750	6.044	274.57	906.5	1181.1	0.4435	1.1857	1.6292	72
74	306.68	0.01752	5.890	276.49	905.1	1181.6	0.4460	1.1810	1.6270	74
76	308.50	0.01754	5.743	278.37	903.7	1182.1	0.4484	1.1764	1.6248	76
78	310.29	0.01755	5.604	280.21	902.4	1182.6	0.4508	1.1720	1.6228	78
80	312.03	0.01757	5.472	282.02	901.1	1183.1	0.4531	1.1676	1.6207	80
82	313.74	0.01759	5.346	283.79	899.7	1183.5	0.4554	1.1633	1.6187	82
84	315.42	0.01761	5.226	285.53	898.5	1184.0	0.4576	1.1592	1.6168	84
86	317.07	0.01762	5.111	287.24	897.2	1184.4	0.4598	1.1551	1.6149	86
88	318.68	0.01764	5.001	288.91	895.9	1184.8	0.4620	1.1510	1.6130	88
90	320.27	0.01766	4.896	290.56	894.7	1185.3	0.4641	1.1471	1.6112	90
92	321.83	0.01768	4.796	292.18	893.5	1185.7	0.4661	1.1433	1.6094	92
94	323.36	0.01769	4.699	293.78	892.3	1186.1	0.4682	1.1394	1.6076	94
96	324.87	0.01771	4.606	295.34	891.1	1186.4	0.4702	1.1358	1.6060	96
98	326.35	0.01772	4.517	296.89	889.9	1186.8	0.4721	1.1322	1.6043	98
100	327.81	0.01774	4.432	298.40	888.8	1187.2	0.4740	1.1286	1.6026	100
150	358.42	0.01809	3.015	330.51	863.6	1194.1	0.5138	1.0556	1.5694	150
200	381.79	0.01839	2.288	355.36	843.0	1198.4	0.5435	1.0018	1.5453	200
300	417.33	0.01890	1.5433	393.84	809.0	1202.8	0.5879	0.9225	1.5104	300
400	444.59	0.0193	1.1613	424.0	780.5	1204.5	0.6214	0.8630	1.4844	400
500	467.01	0.0197	0.9278	449.4	755.0	1204.4	0.6487	0.8147	1.4634	500

[a] Reprinted by permission from *Thermodynamic Properties of Steam*, by J. H. Keenan and F. G. Keyes published by John Wiley and Sons, Inc., 1936 edition.

veniently used to interpolate values for the enthalpy, specific volume, and entropy of moist air from the data of Table 2. Within the estimated precision of the data of Table 2, at temperatures below 150 F, the volume v, enthalpy h, and entropy s, of moist air *per pound of dry air* at any degree of saturation μ may be computed from the simple relations:

$$v = v_a + \mu v_{as} \tag{27}$$

$$h = h_a + \mu h_{as} \tag{28}$$

$$s = s_a + \mu s_{as} \tag{29}$$

Thus, the degree of saturation is used in conjunction with Table 2 in the same manner as the quality is used with tables of the thermodynamic properties of steam.

Correction of Table 2 for Temperatures Above 150 F

The simple relations expressed in Equations 27, 28, and 29 give the properties of unsaturated air with satisfactory precision for most engineering design problems. Above 150 F, when greater precision than obtained by Table 2 is required, these simple relations can be adjusted by the addition of supplementary terms.

To correct the volume, it is necessary to add a correction term $\bar{v}$ which is defined as

$$\bar{v} = \frac{\mu(1 - \mu)A}{1 + aW_s\mu} \tag{30}$$

where a denotes the ratio of the apparent molecular weight of dry air (28.966) to the molecular weight of water (18.016) and is equal to 1.6078. Table 5 gives the values of the coefficient A for several higher temperatures, the value of μ at which the correction term $\bar{v}$ attains its maximum value, and the maximum value of $\bar{v}$ term there attained.

The correction term for the enthalpy is

$$\bar{h} = \frac{\mu(1 - \mu)B}{1 + aW_s\mu} \tag{31}$$

Table 5. Coefficients A, B, C Appearing in Equations 30, 31, 32, Maximum Values of Corrections Defined by Equations 30, 31, 32. Degree of Saturation at Which These Maxima Occur, $\bar{\mu}_m$. Maximum Value of Correction Defined by Equation 33. Degree of Saturation at Which This Maximum Occurs, $\bar{\bar{\mu}}_m$.

(*Standard Atmospheric Pressure*)

t (F)	A (ft^3/lb_a)	B (Btu/lb_a)	C ($Btu/F/lb_a$)	$\bar{v}_{max}$ (ft^3/lb_a)	$\bar{h}_{max}$ (Btu/lb_a)	$\bar{s}_{max}$ ($Btu/F/lb_a$)	$\bar{\mu}_m$	$\bar{\bar{s}}_{max}$ ($Btu/F/lb_a$)	$\bar{\bar{\mu}}_m$
96	0.0018	0.0268	0.00004	0.0004	0.0069	0.00001	0.4925	0.0015	0.3650
112	0.0042	0.0650	0.00009	0.0010	0.0155	0.00002	0.4878	0.0025	0.3632
128	0.0096	0.1439	0.00020	0.0022	0.0332	0.00005	0.4805	0.0040	0.3602
144	0.0215	0.3149	0.00042	0.0047	0.0693	0.00009	0.4691	0.0065	0.3557
160	0.0487	0.6969	0.00091	0.0099	0.1418	0.00019	0.4511	0.0106	0.3485
176	0.1169	1.636	0.00207	0.0207	0.2903	0.00037	0.4213	0.0179	0.3363
192	0.3363	4.608	0.00567	0.0451	0.6180	0.00076	0.3662	0.0333	0.3129

Table 5 gives the values of the coefficient B and maximum values of h, the maximum values occurring at the same degree of saturation as $\bar{v}$.

Corrections for the entropy consist of two terms: $\bar{s}$ which is defined as

$$\bar{s} = \frac{\mu(1 - \mu)C}{1 + aW_s\mu} \tag{32}$$

and $\bar{\bar{s}}$, the so-called *mixing entropy*, which contributes the larger part of the error. The *mixing entropy* is defined as

$$\begin{aligned} \bar{\bar{s}} = 0.1579\,[(1 + \mu a W_s) \log_{10}(1 + \mu a W_s - \mu a W_s \log_{10} (\mu) \\ - \mu(1 + aW_s) \log_{10}(1 + aW_s)] \end{aligned} \tag{33}$$

Table 5 lists the values of the coefficient C, the maximum values of $\bar{s}$ and $\bar{\bar{s}}$ and the values of μ at which they occur. The maximum $\bar{\bar{s}}$ occurs at the same degree of saturation as the maximum values of $\bar{v}$ and $\bar{h}$.

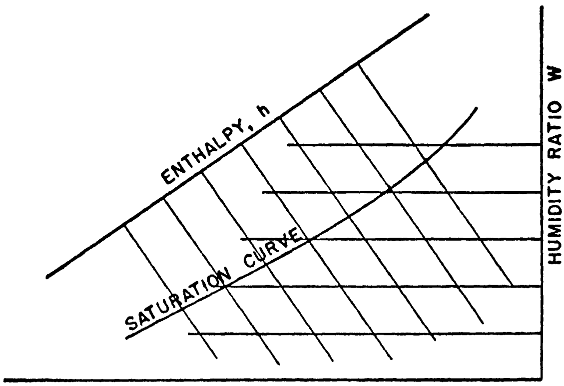

FIG. 3. BASIC COORDINATES OF A.S.H.V.E. PSYCHROMETRIC CHART

THE ASHVE PSYCHROMETRIC CHART

A psychrometric chart is a graphical representation of the thermodynamic properties of moist air. To be of real value in the solution of engineering problems, it must have distinctive features which aid in problem analysis.

An examination of psychrometric mass and energy balances shows that no properties other than *enthalpy* and *humidity ratio* are required for the solution of problems. It is logical, then, to use these properties as the coordinates of a psychrometric chart. This was initially done by Mollier in 1923,[7, 8] and is the arrangement followed in the chart included with THE GUIDE. The A.S.H.V.E. Psychrometric Chart is based on the best thermodynamic data available today, namely, those of Goff and Gratch, as given in Table 2. The chart is plotted on oblique coordinates of enthalpy and humidity ratio; the enthalpy axis making an angle of approximately 40 deg with the humidity ratio axis. This is shown in Fig. 3.

For practical use, the inclusion of dry-bulb and wet-bulb temperatures, volumes, and indices of the condition of the air in relation to saturation, is necessary for locating and describing states on the chart. The arrangement of the families of curves of constant dry-bulb temperature, wet-bulb temperature, volume, relative humidity and degree of saturation, are shown in Fig. 4.

Mass and energy balances deal only with net changes between definite states; the detailed history of a change is not involved. A chart used to facilitate such calculations must primarily aid in clearly establishing states. Lines drawn on the chart to connect different states need have

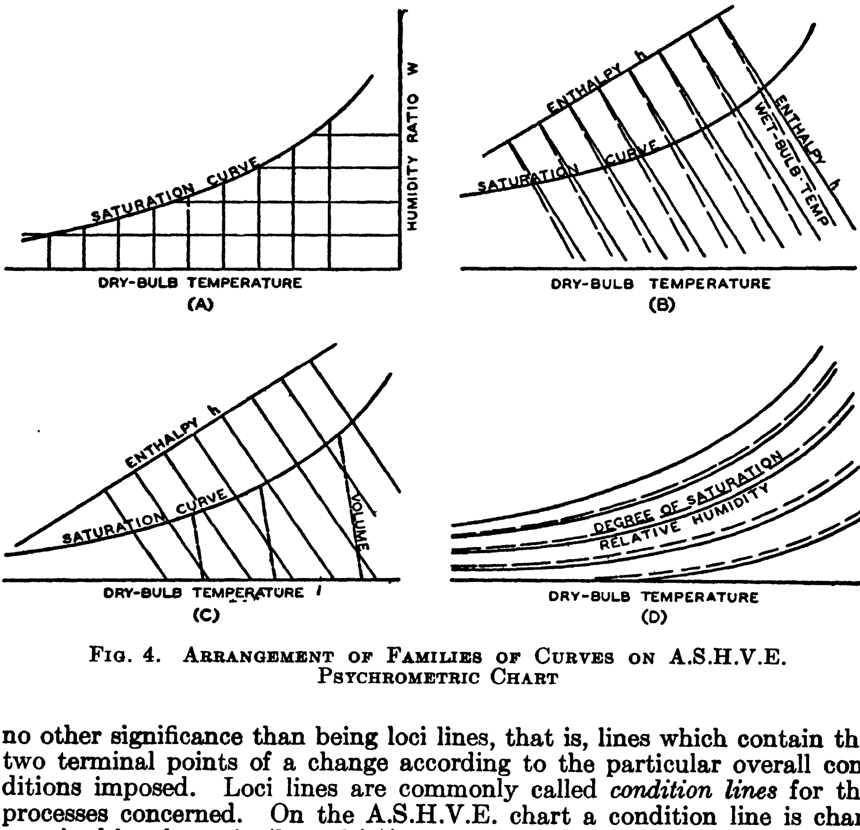

FIG. 4. ARRANGEMENT OF FAMILIES OF CURVES ON A.S.H.V.E. PSYCHROMETRIC CHART

no other significance than being loci lines, that is, lines which contain the two terminal points of a change according to the particular overall conditions imposed. Loci lines are commonly called *condition lines* for the processes concerned. On the A.S.H.V.E. chart a condition line is characterized by the ratio $(h_2 - h_1)/(W_2 - W_1)$.

An abridgment of the A.S.H.V.E. Psychrometric Chart appears in Fig. 5. A large size (24 x 32) chart will be found on the inside of the back cover.

The chart is drawn for standard atmospheric pressure. Steady flow changes commonly involve pressure drops with the flow, but so long as these pressure drops remain a small fraction of the barometric pressure, no appreciable errors need arise from using a constant pressure chart. For many design problems, the use of the standard pressure chart will not incur any undue error up to about 2000 ft above sea level.

The region above the saturation curve is a two-phase region giving equilibrium states for water in both the liquid and vapor phases. The ordinate of a point in this region is the weight of water in both phases per pound of dry air in the vapor phase, neglecting any dissolved air

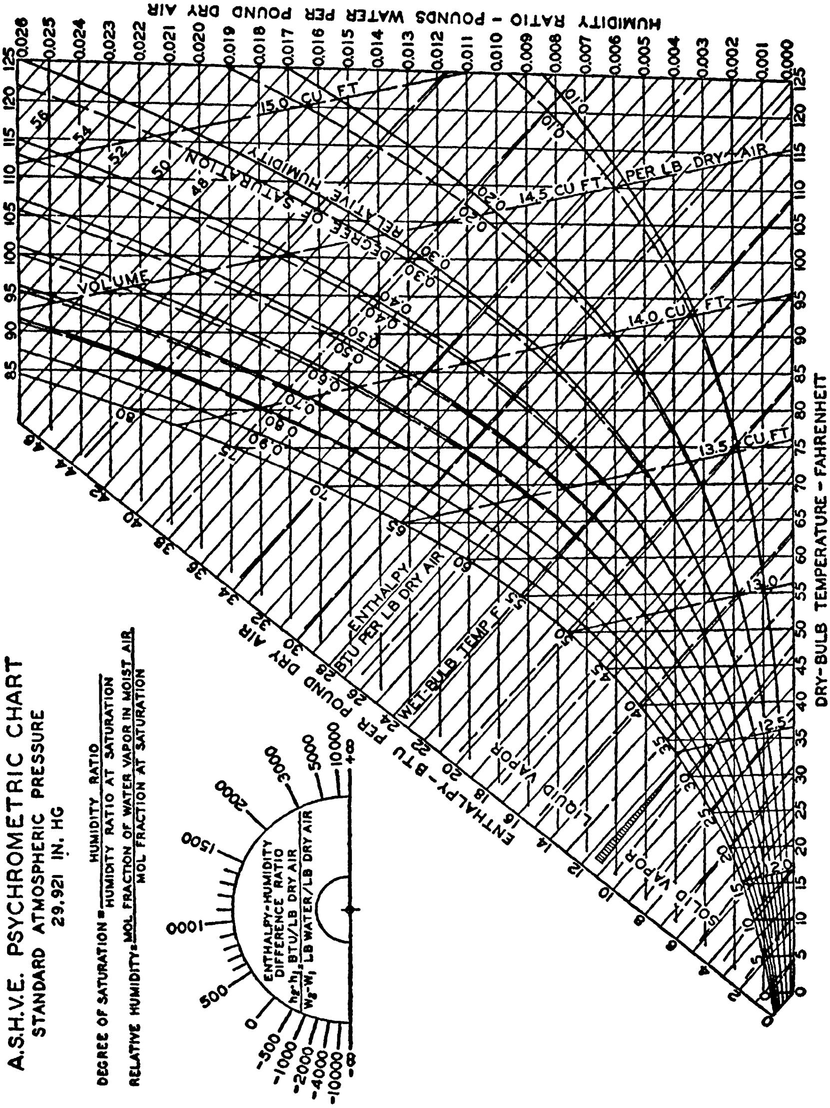

Fig. 5. Abridgment of A.S.H.V.E. Psychrometric Chart

in the condensed phase. The ordinate at the saturation curve, or the point at which the break in the isotherm through the point in question occurs, is the weight of water per pound of dry air in the vapor phase. Consequently, the difference between the two ordinates is the weight of condensed moisture per pound of dry air in the vapor phase.

The shaded solid-liquid-vapor region at 32 F is an isothermal three-phase zone, and separates the liquid-vapor zone from the solid-vapor zone. The temperature is 32 F throughout the shaded area.

It is possible to obtain two values for the wet-bulb temperature when this temperature is below 32 F. If the bulb of a thermometer is dipped into water at a temperature slightly above 32 F and held in a stream of air whose wet-bulb temperature is below 32 F, the temperature indicated by the thermometer will drop rapidly until a minimum is reached below 32 F. This will be accomplished without the formation of ice on the bulb of the thermometer. After reaching this minimum temperature, the reading will jump back to 32 F and remain there until the water on the bulb is frozen, after which it will slowly drop again until equilibrium is reached. The final temperature may be higher or lower than the first minimum reading, or it may be the same depending on the amount of moisture present in the mixture. In the absence of reliable data on the wet-bulb temperature over sub-cooled water, the chart, below 32 F, has been drawn for the equilibrium condition, that is, the values plotted on the A.S.H.V.E. chart are for the condition where the minimum temperature is reached with ice on the bulb of the thermometer.

USE OF TABLE 2 AND THE A.S.H.V.E. PSYCHROMETRIC CHART

The use of Table 2 and the A.S.H.V.E. Psychrometric Chart in analyzing typical air conditioning problems, is best explained by means of illustrative examples. In each of the following it is to be understood that the processes in question take place at a constant pressure of 29.921 in. Hg, *i.e.*, standard atmospheric pressure.

Example 1: Determine the enthalpy of moist air at 80 F dry-bulb temperature and 0.40 degree of saturation.

Solution a: From the data of Table 2, at 80 F, h_a = 19.221 Btu per lb of dry air and h_{as} = 24.47 Btu per lb of dry air. Then h at the specified conditions is 19.221 + 0.40(24.47) = 29.01 Btu per lb of dry air.

Solution b: From the A.S.H.V.E. Chart. Follow the 80 F dry-bulb line vertically until it intersects the 0.40 degree of saturation line. From this intersection, follow the line of constant enthalpy to the enthalpy scale and read 29.00 Btu per lb of dry air.

Example 2: Determine the thermodynamic wet-bulb temperature of moist air at the conditions of *Example 1*.

Solution a: From the data of Table 2. Applying Equation 8, h_1 = 29.01 Btu per lb of dry air (*Example 1*). As a first approximation this is h^*, the enthalpy at saturation at the thermodynamic wet-bulb temperature which is, therefore, approximately 63.5 F. W^* at 63.5 F is 12.57×10^{-3} lb of water vapor per lb of dry air, and W_1 is $0.02233 \times 0.40 = 0.00893$ lb of water vapor per lb of dry air. The specific enthalpy of liquid water at 63.5 F is 31.58 Btu per lb of water. As a second approximation, $h^* = 29.01 + (0.01257 - 0.00893)(31.58) = 29.12$ Btu per lb of dry air. Interpolation in Table 2 gives as the final answer $t^* = 63.64$ F.

Solution b: From the A.S.H.V.E. Chart. At the intersection of the 80 F dry-bulb temperature line and the 0.40 degree of saturation line, read the thermodynamic wet-bulb temperature.

Heating of Moist Air at Constant Pressure Without Addition of Moisture

Example 3: Air initially at 20 F, 0.80 degree of saturation, is heated to 120 F. Find the quantity of heat required to process 20,000 cfm of heated air.

The process is diagrammatically illustrated in Fig. 6. The energy equation for the process is

$$Gh_1 + {}_1q_2 = Gh_2$$

or

$${}_1q_2 = G(h_2 - h_1)$$

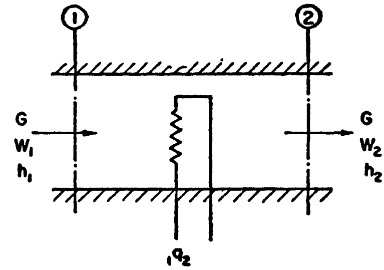

FIG. 6. ILLUSTRATION OF PROCESS OF EXAMPLE 3

Solution a: From the data of Table 2. The initial humidity ratio, which is the same as the final humidity ratio, is 0.80(0.002152) = 0.001722 lb of water vapor per lb of dry air; the initial enthalpy is 4.804 + 0.80(2.302) = 6.646 Btu per lb of dry air; the final degree of saturation is 0.001722/0.08149 = 0.02113; the final enthalpy is 28.841 + 0.02113(90.70) = 30.757 Btu per lb of dry air; the final volume is 14.611 + 0.02113(1.905) = 14.651 cu ft per lb of dry air. Since 20,000 cfm of heated air are to be processed, the total quantity of heat required is

$$_1q_2 = (20{,}000/14.651) \times 24.111 = 32{,}914 \text{ Btu per min.}$$

Solution b: From the A.S.H.V.E. Chart. The process is represented by the horizontal line 1–2, Fig. 7. The initial enthalpy, at 20 F dry-bulb temperature and 0.80 degree of saturation, is 6.65 Btu per lb of dry air. Since the final humidity ratio is the same as the initial humidity ratio, the ratio $(h_2 - h_1)/(W_2 - W_1) = \infty$. The horizontal line 1–2, Fig. 7, then represents the condition line for the process, and the final state of the moist air must lie on this line. The final state is located at the point at which the 120 F dry-bulb temperature line crosses the condition line, and is labeled point 2 on the figure. At this condition the final enthalpy is 30.8 Btu per lb of dry air and the final specific volume is 14.65 cu ft per lb of dry air. Substituting these values in the energy equation,

$$_1q_2 = (20{,}000/14.65) \times (30.8 - 6.65) = 32{,}950 \text{ Btu per min.}$$

Cooling of Moist Air at Constant Pressure with Condensation of Water

Referring to Fig. 8, moist air cooled from state 1 passes through successive states along the line $W = W_1$ = constant until the saturation line is intersected. The

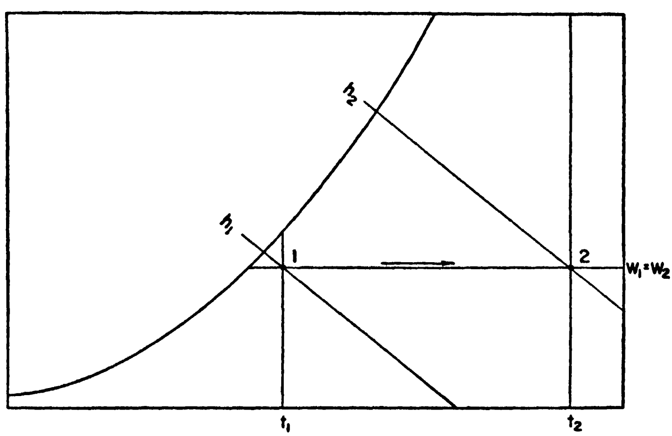

FIG. 7. SOLUTION OF EXAMPLE 3 ON A.S.H.V.E. PSYCHROMETRIC CHART

temperature at this point of intersection is by definition the dew-point temperature for state 1.

Further cooling through successive equilibrium states is accompanied by condensation. The succession of states for the total system, moist air and liquid water, is represented by a continuation of the $W = W_1$ line into the liquid vapor region. (Temperatures below 32 F would involve the solid-vapor region). Consider that the final temperature is t_3. The final enthalpy is then h_3; the liquid water formed is $(W_1 - W_3)$, where point 3 is at the intersection of the isotherm through 2 and the saturation curve; the final humidity ratio of the moist air is W_3; and this final moist air has dew-point, wet-bulb and dry-bulb temperatures all equal to t_3.

Example 4: How much heat must be removed from 20,000 cfm of air at 95 F dry-bulb temperature and 0.50 degree of saturation to cool the air to 70 F, saturated?

Solution a: From the data of Table 2. The initial humidity ratio is 0.50(0.03763) = 0.01837 lb of water vapor per lb of dry air; the initial enthalpy is 22.827 + 0.50(40.49) = 43.072 Btu per lb of dry air; the humidity ratio at saturation at the

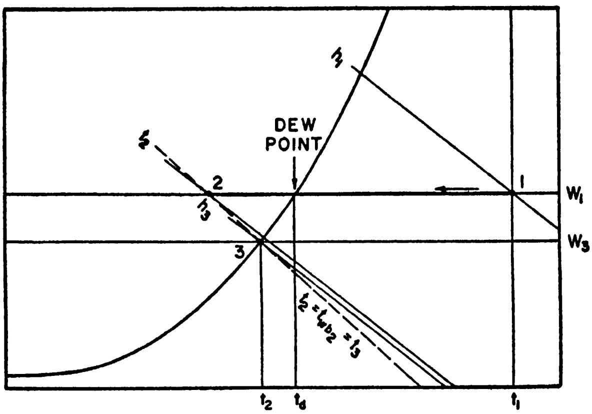

FIG. 8. COOLING OF AIR AT CONSTANT PRESSURE SHOWN ON A.S.H.V.E. PSYCHROMETRIC CHART

final temperature is 0.01582 lb of water vapor per lb of dry air; the quantity of liquid formed is 0.01837 − 0.01582 = 0.00255 lb of water vapor per lb of dry air; h_{w3} at 70 F is 38.11 Btu per lb of water; the initial specific volume is 13.980 + 0.50(0.822) = 14.391 cu ft per lb of dry air.

Fig. 9 illustrates the process diagrammatically. The energy equation for the process is

$$Gh_1 = Gh_3 + G(W_1 - W_3)h_{w3} + {}_1q_3$$

or

$${}_1q_3 = G[h_1 - h_3 - (W_1 - W_3)h_{w3}]$$

$$= \frac{20{,}000}{14.391} \times (43.072 - 34.09 - 0.00255 \times 38.07)$$

$$= 12{,}350 \text{ Btu per min.}$$

Solution b: From the A.S.H.V.E. Chart. Two methods may be used to solve the problem by use of the psychrometric chart. The simpler is to use the region to the left of the saturation line (Fig. 8). From point 1 draw a horizontal line on the chart until it intersects the constant temperature line in the liquid-vapor region corresponding to the final temperature, 70 F. This is shown as point 2 on the diagram. Then,

$${}_1q_3 = G(h_1 - h_2)$$

The initial enthalpy is 43 Btu per lb of dry air; the initial specific volume is 14.4

cu ft per lb of dry air; and the final enthalpy is 34.2 Btu per lb of dry air. The solution of the problem is

$$_1q_3 = \frac{20{,}000}{14.4} \times (43 - 34.2) = 12{,}200 \text{ Btu per min.}$$

The other method is to use an energy balance,

$$_1q_3 = G[h_1 - h_3 - h_{w3}(W_1 - W_3)]$$

The initial humidity ratio is 0.0183 lb of water vapor per lb of dry air, and the final humidity ratio is 0.0158 lb of water vapor per lb of dry air. Therefore, the heat to be removed is

$$_1q_3 = \frac{20{,}000}{14.4} \times (43 - 34.1 - 0.0025 \times 38.07)$$

$$= 12{,}130 \text{ Btu per min.}$$

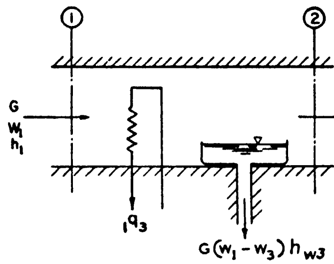

FIG. 9. ILLUSTRATION OF PROCESS OF EXAMPLE 4

Adiabatic Mixing of Two Steady Flow Air Streams at Constant Pressure

The process is diagrammed in Fig. 10. By applying the principles of the conservation of mass and energy, three equations may be written:

Mass balance for the dry air,

$$G_1 + G_2 = G_3$$

Energy balance for the process,

$$G_1h_1 + G_2h_2 = G_3h_3$$

Mass balance for the water vapor,

$$G_1W_1 + G_2W_2 = G_3W_3$$

Eliminating G_3 and combining the three equations yield the equation,

$$\frac{h_2 - h_3}{h_3 - h_1} = \frac{W_2 - W_3}{W_3 - W_1} = \frac{G_1}{G_2}. \tag{34}$$

Example 5: Outside air at 0 F dry-bulb temperature and 0.80 degree of saturation is to be mixed adiabatically with recirculated inside air at 70 F dry-bulb temperature and 0.20 degree of saturation, in the ratio of one pound of dry air in the former to four in the latter. Find the temperature and degree of saturation in the resulting mixture.

Solution a: From the data of Table 2. The only unknown properties are the humidity ratio W_3 and the enthalpy h_3 of the resulting mixture. These may be determined from Equation 34. Thus,

$$\frac{20.270 - h_3}{h_3 - 0.668} = \frac{0.003164 - W_3}{W_3 - 0.000630} = \frac{1}{4}$$

from which $h_3 = 16.350$ and $W_3 = 0.002657$ The enthalpy of the final mixture may also be expressed by Equation 28:

$$h_3 = h_a + \mu h_{as}$$

Since μ by definition is W_3/W_s, Equation 28 may be rewritten as

$$16.350 = h_a + (0.002657/W_s) \times h_{as}$$

At 56 F the right side of the equation is 16.332, and at 57 F it is 16.582. Interpolation gives as the final dry-bulb temperature of the mixture 56.07 F. At this temperature the humidity ratio at saturation is 0.00960 lb of water vapor per lb of dry air. Therefore, the final degree of saturation is

$$\mu = 0.002657/0.00960 = 0.277$$

Solution b. From the A.S.H.V.E. Chart. Equation 34 indicates that the state point of the resulting mixture lies on a straight line connecting the state points of

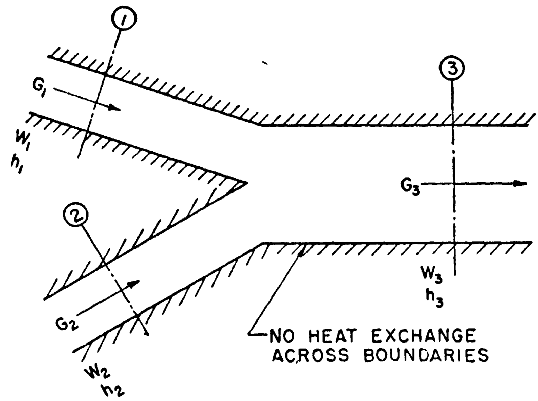

FIG. 10. ILLUSTRATION OF MIXING OF TWO STEADY FLOW STREAMS AT CONSTANT PRESSURE

the two streams being mixed, and divides this line into two segments whose respective lengths are inversely proportional to the rates of dry air flow in the corresponding streams. This is illustrated in Fig. 11. Points 1 and 2 are located and connected by a straight line. The state of the final mixture is set so that

$$\frac{G_1}{G_2} = \frac{D_{2-3}}{D_{3-1}} = \frac{1}{4}$$

Scaling the distances on the chart, the required solution to *Example 5* is 56 F dry-bulb temperature and 0.28 degree of saturation.

Addition of Moisture to an Adiabatic Stream

Consider a stream of moist air flowing adiabatically between two sections, 1 and 2, as in Fig. 12, with moisture addition at the rate $G_1(W_2 - W_1)$ and the moisture having the enthalpy h_w Btu per pound of moisture.

An energy balance yields

$$G_1 h_1 + G_1(W_2 - W_1)h_w = G_1 h_2 \tag{35}$$

Example 6: Liquid water chilled to 40 F is injected into an air stream initially at

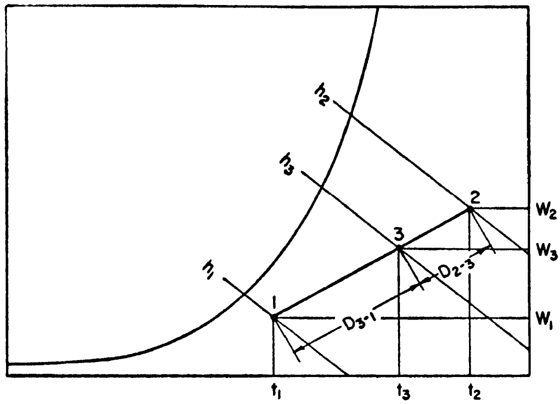

FIG. 11. SOLUTION OF EXAMPLE 5 ON A.S.H.V.E. PSYCHROMETRIC CHART

95 F dry-bulb temperature and 80 F thermodynamic wet-bulb temperature. At what temperature will saturation be reached? How much water must be evaporated to reach saturation?

Solution a: From the data of Table 2. The solution of Equation 35 for h_2 yields

$$h_2 = h_1 + (W_2 - W_1)h_w$$

The initial enthalpy of the moist air h_1 must be found from Equation 8,

$$h_1 = h^* - (W^* - W_1)h_w^*$$

$$22.827 + \mu 40.49 = 43.69 - (0.02233 - 0.03673\mu)\ (48.05)$$

from which $\mu = 0.511$.

Hence, $h_1 = 22.827 + 0.511(40.49)$

$= 43.52$ Btu per lb of dry air

and $W_1 = 0.03673(0.511)$

$= 0.01877$ lb per lb of dry air.

The solution of Equation 35 is

$$h_2 = 43.52 + (W_2 - 0.01877)(8.09)$$

By trial and error, this equation will be satisfied at the temperature 79.87 F. At this temperature the humidity ratio W_2 is 0.02223. The weight of water evaporated is therefore $0.02223 - 0.01877 = 0.00346$ lb per lb of dry air.

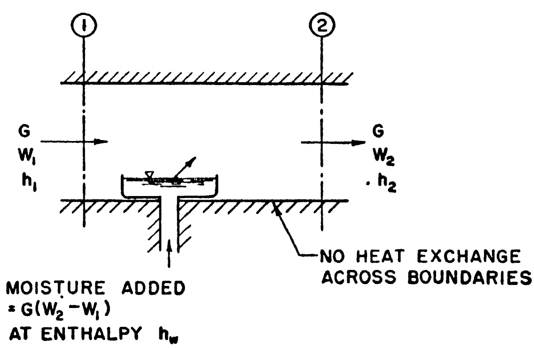

FIG. 12. ILLUSTRATION OF ADDITION OF MOISTURE TO AN ADIABATIC STREAM

Solution b: From the A.S.H.V.E. Chart. Solution of Equation 35 for the ratio $(h_2 - h_1)/(W_2 - W_1)$ yields

$$\frac{h_2 - h_1}{W_2 - W_1} = h_w \tag{36}$$

The slope of the condition line is therefore determined by the enthalpy of the water which is supplied. This slope is established on the chart by connecting the center of the protractor on the psychrometric chart with the value of h_w on the protractor. Draw a line parallel to this reference line through the initial point 1 (Fig. 13). The second line is the condition line for the process. Since the conditions of the problem require the final point to lie on the saturation line, the intersection of the condition line with the saturation line gives the desired solution.

Adiabatic Saturation

Adiabatic saturation is the designation given any process in which the state of moist air is changed from some initial unsaturated condition to a saturated one without the addition or removal of heat. According to this definition, the addition of moisture to an adiabatic stream may become an adiabatic saturation process. *Example 6* is an illustration.

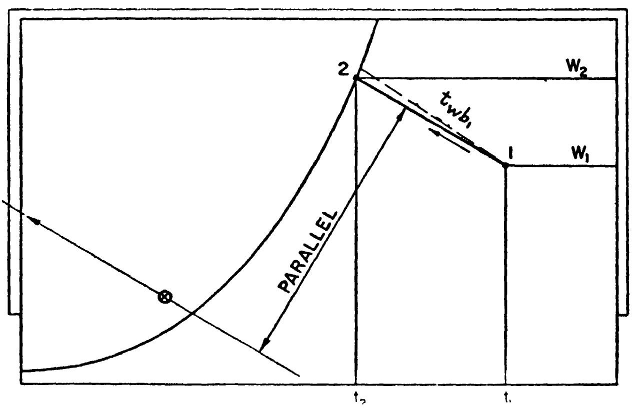

FIG. 13. SOLUTION OF EXAMPLE 6 ON A.S.H.V.E. PSYCHROMETRIC CHART

A type of adiabatic saturation of further practical interest is the use of continually recirculated spray water in a saturating air washer. Here the spray water will ultimately come to the same temperature as the saturated leaving air; this temperature is, by definition, the thermodynamic wet-bulb temperature. Hence, adiabatic saturation in this manner will have the final state point on the saturation curve, with the same thermodynamic wet-bulb temperature as the original state point.

In a process such as this, the moist air enthalpy changes very slightly. The moisture added and temperature change may be obtained from the psychrometric chart as sketched in Fig. 14.

Example 7: Moist air at 75 F dry-bulb temperature and 0.60 degree of saturation, is saturated adiabatically with recirculating spray water. Find the amount of water added and the change in enthalpy.

Solution a: From the data of Table 2. As previously stated, the final temperature of the mixture will be the thermodynamic wet-bulb temperature at the initial state. This must first be determined by the method of *Example 2,* and is 65.51 F. At this temperature the humidity ratio at saturation and enthalpy at saturation are 0.01350 and 30.45, respectively. The humidity ratio at the initial state is 0.60(0.01882) = 0.01129; the initial enthalpy is 0.60(20.59) + 18.018 = 30.372. The weight of water

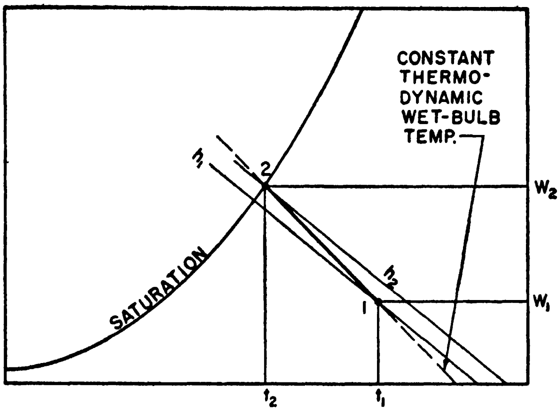

FIG. 14. MOISTURE ADDED AND TEMPERATURE CHANGE

added is therefore 0.01350 − 0.01129 = 0.00221 lb per lb of dry air; the enthalpy change is 30.45 − 30.372 = 0.078 Btu per lb of dry air.

Solution b: From the A.S.H.V.E. Chart. Since the initial and final states have the same thermodynamic wet-bulb temperature, the results may be read directly from the chart.

Addition of Heat and Water Vapor to an Air Stream in Steady Flow

Fig. 15 is a schematic representation of a system operating at constant pressure, where

G_1 is the rate of flow of dry air, pounds per minute.

G_w is the rate of evaporation of the water, pounds per minute.

h_w is the enthalpy of the liquid water entering, Btu per pound.

Q is the rate of heat addition, Btu per minute.

An energy balance for the system gives

$$G_1h_1 + Q + G_wh_w = G_1h_2 \tag{37}$$

A mass balance gives

$$G_1 + G_1W_1 + G_w = G_1 + G_1W_2$$

or

$$G_1W_1 + G_w = G_1W_2 \tag{38}$$

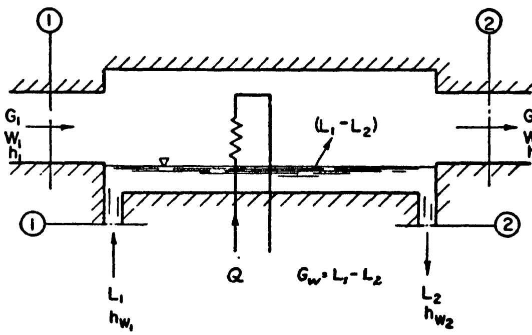

FIG. 15. ILLUSTRATION OF ADDITION OF HEAT AND WATER VAPOR TO AN AIR STREAM IN STEADY FLOW

Combining equations 37 and 38 and solving for the ratio $(h_2 - h_1)/(W_2 - W_1)$,

$$\frac{h_2 - h_1}{W_2 - W_1} = \frac{Q}{G_w} + h_w \tag{39}$$

Example 8: Moist air at 20 F dry-bulb temperature and 0.80 degree of saturation is heated and humidified until it is at 120 F dry-bulb temperature and 71.5 F thermodynamic wet-bulb temperature. Water at 55 F is supplied. If the air flow rate is 20,000 cfm at the initial conditions, how much heat is required?

Solution a: From the data of Table 2. The initial humidity ratio is 0.80(0.002152) = 0.00172; the initial enthalpy is 4.802 + 0.80(2.302) = 6.6456; the initial specific volume is 12.084 + 0.80(0.042) = 12.118. The degree of saturation at the final state may be determined from Equation 8 which may be rewritten as

$$h_{a2} + \mu h_{as2} + h_w^*(W^* - W_{s2}) = h^*$$

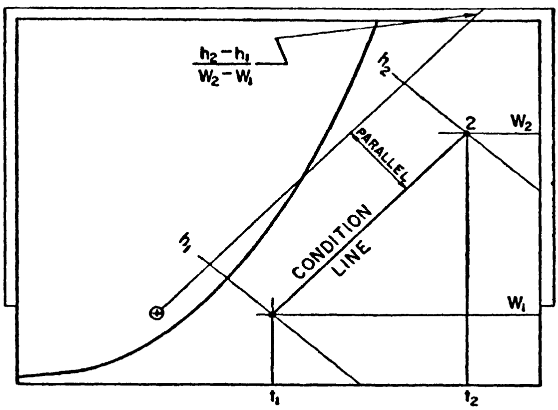

FIG. 16. SOLUTION OF EXAMPLE 8 ON A.S.H.V.E. PSYCHROMETRIC CHART

The values of these properties are: $h^* = 35.39$; $W^* = 0.01668$; $h_w^* = 39.61$; $h_{as2} = 90.70$; $W_{s2} = 0.08149$; $h_{a2} = 28.84$.

Making the proper substitutions and solving for degree of saturation,

$$\mu = 0.0681.$$

The final humidity ratio is therefore 0.0681(0.08149) = 0.005549; the final enthalpy is 28.84 + 0.0681(90.70) = 35.02 Btu per lb dry air.

The rate of water addition is obtained from Equation 38.

$$G_w = \frac{20{,}000}{12.12}(0.005549 - 0.00172)$$

$$= 6.82 \text{ lb per min.}$$

The heat supplied is obtained from Equation 37.

$$Q = G_1(h_2 - h_1) - G_w h_w$$

$$= \frac{20{,}000}{12.12}(35.02 - 6.65) - 6.32(28.08)$$

$$= 46{,}667 \text{ Btu per min.}$$

Solution b: From the A.S.H.V.E. Chart. Locate the initial and final states on the chart and connect them with a straight line. Through the reference point on the chart, draw a line parallel to the line connecting the initial and final state points, the condition line, and read the value of the ratio $(h_2 - h_1)/(W_2 - W_1)$ as 7500 from the protractor on the chart (Fig. 16). From Equation 39

$$\frac{h_2 - h_1}{W_2 - W_1} = \frac{Q}{G_w} + h_w = 7500$$

The rate of water supply was determined in *Solution a,* but will be found from the chart. It is

$$G_w = \frac{20{,}000}{12.1}(0.0055 - 0.0017)$$

$$= 6.28 \text{ lb per min}$$

$$Q = G_w(7500 - h_w)$$

$$= 6.28(7500 - 23)$$

$$= 46.900 \text{ Btu per min.}$$

TABLE 6. PRESSURE AND TEMPERATURE FOR ALTITUDES IN U. S. STANDARD ATMOSPHERE

ALTITUDE FEET Z	PRESSURE IN. OF HG P	TEMP F t
− 1,000	31.02	+62.6
− 500	30.47	+60.8
0	29.921	+59.0
+ 500	29.38	+57.2
+ 1,000	28.86	+55.4
+ 5,000	24.89	+41.2
10,000	20.58	+23.4
15,000	16.88	+ 5.5
20,000	13.75	−12.3
25,000	11.10	−30.1
30,000	8.88	−47.9
35,000	7.04	−65.8
40,000	5.54	−67.0
45,000	4.36	−67.0
50,000	3.436	−67.0

U. S. STANDARD ATMOSPHERE

The definition of the U. S. Standard Atmosphere is important to the air conditioning engineer as an essential standard of reference. The basic assumptions in defining the Standard Atmosphere are:

1. There is a linear decrease in temperature T with altitude up to the limit of the isothermal atmosphere at 35,332 ft. Thus,

$$T = T_o - 0.003566\, Z \tag{40}$$

2. The air is dry.
3. Air is a perfect gas obeying the laws of Charles and Boyle:

$$PV = RT$$

4. Gravity is constant at all altitudes with the standard value.
5. The temperature of the isothermal atmosphere is −66 F.

Standard values at sea level, which are part of the definition of the Standard Atmosphere, are:

Pressure	29.921 in. Hg
Temperature	59 F

Absolute Temperature	518.4 F Abs
Gravity	32.1740 ft per (sec) (sec)
Density	0.07651 lb per cu ft

Values of pressure and temperature are listed in Table 6 for altitudes in the standard atmosphere from −1000 to 50,000 ft above sea level. For further explanation, References 9 and 10 should be consulted.

LETTER SYMBOLS USED IN CHAPTER 3

μ = degree of saturation (decimal).

ρ = density of fluid, pounds per cubic foot.

φ = relative humidity (decimal).

a = ratio of apparent molecular weight of dry air (28.966) to the molecular weight of water (18.016) = 1.6078.

A = coefficient from Table 5 for use in Equation 30 (obtained from Table 5).

B = coefficient to be used in Equation 31 (obtained from Table 5).

C = coefficient for use in Equation 32 (obtained from Table 5).

f_s = factor accounting for effect of mixing air and water, dimensionless.

G = flow rate of dry air, pounds per hour.

G_1 = flow rate of dry air, pounds per minute.

G_w = rate of evaporation of water, pounds per minute.

H = enthalpy of the system.

H = enthalpy of the flowing medium, Btu per pound of dry air.

h = enthalpy of moist air, Btu per pound of dry air.

$\bar{h}$ = enthalpy correction term to be added above 150 F, to enthalpy.

h_a = specific enthalpy of dry air, Btu per pound.

h_{as} = $h_s - h_a$ = the difference between the enthalpy of moist air *at saturation* per pound of dry air, and the specific enthalpy of the dry air itself, Btu per pound of dry air.

h_g = enthalpy of saturated water vapor, Btu per pound.

h_s = enthalpy of moist air at saturation per pound of dry air, Btu per pound of dry air.

h_s^* = enthalpy of moist air at saturation at *thermodynamic wet-bulb temperature*, Btu per pound of dry air.

h_w = specific enthalpy of condensed water (liquid or solid) at standard pressure, Btu per pound water.

h_w^* = specific enthalpy of water as added at the *thermodynamic wet-bulb temperature* t^*, Btu per pound of dry air.

h_{wl} = enthalpy of liquid water, Btu per pound.

h_{ws} = enthalpy of solid water, Btu per pound.

KE = average kinetic energy, Btu per pound.

$\overline{KE}$ = average kinetic energy, Btu per pound.

L = flow rate of liquid water, pounds per hour.

m = weight of dry air crossing any duct section, pounds per minute.

n_a = mols dry air.

n_s = mols of water vapor at saturation.

n_w = mols of water vapor.

P = absolute pressure.

P_a = atmospheric pressure, inches Hg.

LETTER SYMBOLS (Continued)

P_o = standard atmospheric pressure by definition 29.921 in. Hg.
P_s = saturation pressure of pure water at prevailing temperature.
p = total pressure of a mixture of air and water vapor, pounds per square inch or inches Hg.
p_a = partial pressure of dry air, pounds per square inch or inches Hg.
p_s = saturation pressure of pure water vapor, pounds per square inch or inches Hg.
p_w = partial pressure of water vapor in mixture of air and water vapor, pounds per square inch or inches Hg.
PE = potential energy, Btu per pound dry air.
$\overline{PE}$ = average potential energy, Btu per pound dry air.
Q = total heat added or subtracted, between sections, Btu per minute.
q = ratio of energy added (or removed) to water added (or removed), Btu per pound. Also called specific enthalpy of *water added.*
$_1\mathrm{q}_2$ = energy added between points 1 and 2.
$_1q_2$ = heat added between sections 1 and 2, Btu per pound dry air.
R = universal gas constant, 1545 foot-pounds per (Fahrenheit degree) (mol).
R_a = gas constant for dry air.
R_w = gas constant for water vapor.
S = flow rate of solid water, pounds per hour.
s = entropy of moist air per pound of dry air, Btu per (pound) (Fahrenheit degree).
$\bar{s}$ = correction to be added to entropy of moist air obtained from Table 2.
$\bar{\bar{s}}$ = additional correction to be added to entropy because of "mixing entropy" (obtained from Table 5). Correction to be added to value of s obtained from Table 2.
s_a = specific entropy of dry air, Btu per (pound) (Fahrenheit degree, absolute).
s_{as} = the difference between the entropy of moist air *at saturation* per pound of dry air, and the specific entropy of the dry air itself, Btu per (pound of dry air) (Fahrenheit degree, absolute).
s_s = entropy of moist air at saturation per pound of dry air, Btu per (pound of dry air) (Fahrenheit degree, absolute).
s_w = specific entropy of condensed water (liquid or solid) at standard atmospheric pressure, Btu per (pound of water) (Fahrenheit degree, absolute).
T = absolute temperature, Fahrenheit degrees.
T_o = standard atmospheric temperature, by definition 518.4 F absolute.
t = temperature, Fahrenheit degrees.
t^* = thermodynamic wet-bulb temperature, Fahrenheit degrees.
U = internal energy of system.
U = internal energy.
V = volume.
$\bar{V}$ = average velocity, feet per minute.
v = volume of moist air per pound of dry air, cubic feet per pound.
$\bar{v}$ = correction to be added to volume of moist air per pound of dry air, above 150 F.
v_a = specific volume of dry air, cubic feet per pound.

LETTER SYMBOLS (Concluded)

v_{as} = $v_s - v_a$, the difference between volume of moist air *at saturation*, per pound of dry air, and the volume of the dry air itself, cubic feet per pound of dry air.

v_s = volume of moist air at saturation per pound of dry air, cubic feet per pound of dry air.

v_T = total volume, cubic feet.

W = humidity ratio, of moist air, pounds of water per pound of dry air.

W_s = humidity ratio, *at saturation*, weight of water vapor per pound of dry air, pound per pound.

W_s^* = humidity ratio corresponding to thermodynamic wet-bulb temperature t^*, pounds of water per pound of dry air.

w = work done by system.

w = shaft work withdrawn between sections 1 and 2, Btu per pound of air.

Z = elevation above any datum, feet.

$\bar{Z}$ = average elevation, feet.

Subscripts with symbols have following meanings: 1, 2, 3 indicate section of flow; a = air, w = water, wl = liquid water, ws = solid water, s = saturation, m = mixture; * indicates that the value is at thermodynamic wet-bulb temperature.

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[1] The Humidity Ratio of Moist Air at Saturation, by J. A. Goff and S. Gratch (Special Report of *University of Pennsylvania Thermodynamic Research Laboratory*, March 1948).

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CHAPTER 4

FLUID FLOW

Theory of Fluid Flow, Pressure Loss in Circular Pipes, Pressure Loss in Non-Circular Pipes; Flow of Compressible Fluids, Ideal Flow Through Nozzle or Orifice; Flow Measurement, Head Meters, Installation of Head Meters, Pitot Tube, Variable Area Flow Meters

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THE flow of fluids is part of the branch of engineering science known as fluid mechanics, which will be discussed here insofar as it applies to the work of engineers in the fields of heating, ventilating, and air conditioning. Probably air is the most frequently handled fluid, but other gases and liquids are often involved. Compressible fluids (gases) and incompressible fluids (liquids) vary somewhat in behavior, though in cases where pressure and density changes are small, the gases may be treated as incompressible fluids.

THEORY OF FLUID FLOW

The following energy equation for one dimensional steady flow processes will serve as a basis for the theory of the flow of fluids. This equation is presented in several ways in various texts, but a suitable form is

$$\frac{V_1^2}{2g_c} + Ju_1 + p_1v_1 + Jq + \frac{g}{g_c} z_1 = \frac{V_2^2}{2g_c} + Ju_2 + p_2v_2 + W + \frac{g}{g_c} z_2 \qquad (1)$$

where

V = velocity in feet per second.
g = gravitational acceleration, in feet per (second) (second).
g_c = gravitational conversion factor = 32.174 (pounds mass per pound force) $\times$ ft per (second) (second).
J = mechanical equivalent of heat = 778 foot pounds per Btu.
u = internal energy, in Btu per pound of fluid.
p = pressure in pounds per square foot.
v = specific volume, in cubic feet per pound.
W = mechanical work done by the fluid in foot pounds per pound of fluid.
q = heat transferred to the fluid in Btu per pound of fluid flowing.
z = elevation above some arbitrary datum, in feet.
Subscript 1 refers to the entrance, subscript 2 to the exit.

Introducing the enthalpy h, which by definition is $u + \frac{pv}{J}$, expressed in Btu per pound of fluid, Equation 1 becomes

$$\frac{V_1^2}{2g_c} + Jh_1 + Jq + \frac{g}{g_c} z_1 = \frac{V_2^2}{2g} + Jh_2 + W + \frac{g}{g_c} z_2 \qquad (2)$$

The equivalent differential form for energy Equation 1 is

$$\frac{1}{2g_c}\, dV^2 + J\, du + d(pv) + \frac{g}{g_c}\, dz - J\, dq + dW = 0 \qquad (3)$$

Replacing v by its equal $g/g_c\rho$ (where ρ is density in pounds weight per cubic foot) and rearranging, Equation 3 becomes

$$\frac{1}{2g}\,dV^2 + \frac{1}{\rho}\,dp + dz + \frac{g_c}{g}\,[J\,du + p\,dv - J\,dq + dW] = 0 \qquad (4)$$

In the case of flow through a pipe, no outside work is performed so that $dW = 0$. Furthermore,

$$J\,du + p\,dv = JT\,ds = J\,dq + JT\,ds' \qquad (5)$$

where

ds = total change in entropy.

ds' = change in entropy due to internal irreversibility from turbulence and friction.

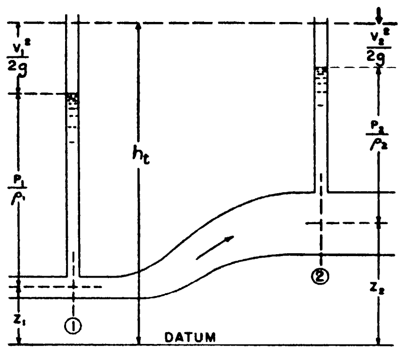

FIG. 1. RELATION OF VARIOUS FACTORS IN BERNOULLI EQUATION

Accordingly, Equation 4 may be written

$$\frac{1}{2g}\,dV^2 + \frac{dp}{\rho} + dz + \frac{g_c}{g}\,JT\,ds' = 0^* \qquad (6)$$

In cases where there is no internal irreversibility, $ds' = 0$, and Equation 6 may be integrated to give

$$\frac{V_1^2}{2g} + \frac{p_1}{\rho_m} + z_1 = \frac{V_2^2}{2g} + \frac{p_2}{\rho_m} + z_2 \qquad (7)$$

where ρ_m is the proper mean density.

This is commonly called the Bernoulli equation, named after the Swiss mathematician and physician who first propounded the theory. $\frac{V^2}{2g}$ is known as the velocity head, $\frac{p}{\rho}$ is the pressure head, and z is the elevation head, all in feet of the fluid; the total head, h_t is the sum of the other three heads. Fig. 1 shows diagrammatically the relation of the various factors. The pressure at point 2 is lower than at point 1 because of the elevation of point 2 over point 1, and the velocity at point 2 is lower than at point 1 because of the larger pipe diameter at point 2. If the

* In the analysis of subsequent portions of this chapter the distinction between g and g_c will be omitted. Aside from the dimensional consistency the factor, g/g_c, is not in general significant in fluid flow analysis.

pipe diameter were the same throughout, the velocity, and consequently the velocity head, would be the same at both points, but the higher elevation at point 2 would still be responsible for a loss in pressure. The utility of the equation is evident, though it should be remembered that in it the *effects of friction and turbulence are neglected*, and that Fig. 1 represents ideal conditions. It should also be noted that care must be taken in determining the proper mean density. Accordingly, the Bernoulli equation is applied most conveniently to incompressible fluids for which density is constant.

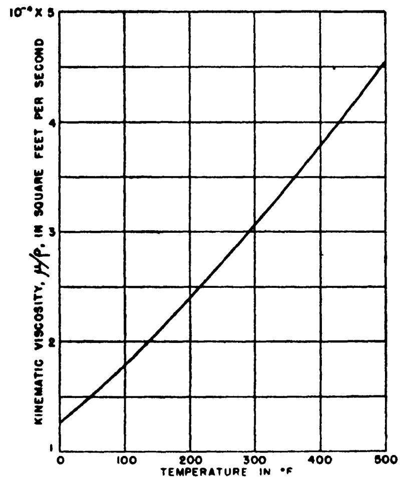

FIG. 2. RELATION OF KINEMATIC VISCOSITY TO TEMPERATURE OF AIR

Pressure Loss in Circular Pipes

The pressure loss in circular pipes is customarily expressed by the formula:

$$h_f = \frac{flV^2}{2g\,\mathrm{d}} \tag{8}$$

where

h_f = the loss in head of the fluid under conditions of flow, in feet.
l = the length of the pipe, in feet.
V = the velocity, in feet per second.
g = the acceleration due to gravity = 32.174 ft per (second) (second).
d = the internal diameter of the pipe, in feet.
f = a dimensionless friction coefficient.

The formula is generally known by the name of Darcy or Fanning, though it seems to have been originated by d'Aubisson de Voisins in 1834.

The factor f is a function of the Reynolds number,

$$N_{Re} = \frac{\mathrm{d}\,V\rho}{\mu} \tag{9}$$

where

N_{Re} = Reynolds number.
ρ = the density in pounds per cubic foot.
μ = the absolute viscosity in pounds per foot-second.

Both f and the Reynolds number are dimensionless. To aid in computing the Reynolds number, values of $\frac{\mu}{\rho}$, the kinematic viscosity, are shown as a function of temperature for air in Fig. 2, and for water in Fig. 3.

Fig. 4 shows the relation between f and the Reynolds number, adapted from a review by Moody.[1] The straight line sloping downward at the left of the chart supplies the values of f for laminar flow determined by the formula:

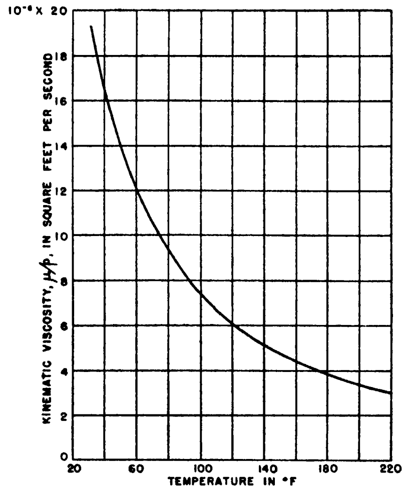

FIG. 3. RELATION OF KINEMATIC VISCOSITY TO TEMPERATURE OF WATER

$$f = \frac{64}{N_{Re}} \tag{10}$$

With laminar flow, the velocity profile is a parabola, having the formula:

$$V = \frac{\rho h_f}{4\mu l}(r^2 - L^2) \tag{11}$$

where

r = the radius of the pipe in feet.
L = distance perpendicularly from the axis of the pipe, in feet.

Accordingly, the maximum velocity occurs at the center of the pipe and is twice the average velocity; the average velocity is found when $L = 0.707\ r$. It is worth noting that roughness of the pipe wall has no effect on the loss in head for laminar flow.

Between values of the Reynolds number of 2000 and 4000, there is an

[1] Superior numbers refer to the references at the end of chapter.

unstable region where the flow changes from laminar to turbulent, or vice versa. The actual value is impossible of prediction for any conditions of flow, though in general it may be said that the prevailing type of flow persists into the unstable region; however, once the change starts, it proceeds very rapidly.

When the flow is turbulent, the velocity profile is essentially parabolic over four-fifths of the pipe diameter, but near the pipe walls, the effect of friction becomes evident, and in the boundary layer at the pipe wall the flow is laminar. Fig. 5 compares the velocity profiles for three different Reynolds numbers, but for the same average velocity.

The lower curve in the turbulent region in Fig. 4 represents the relation of f to the Reynolds number for smooth pipe, such as drawn brass tubing

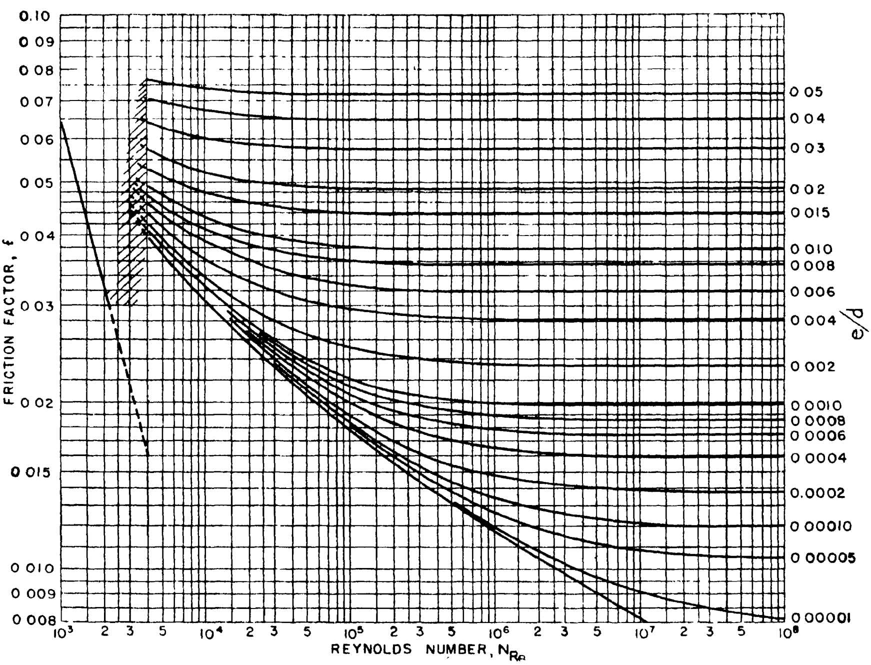

FIG. 4. RELATION BETWEEN FRICTION FACTOR AND REYNOLDS NUMBER
NOTE: The straight line at left shows values of Friction Factor for laminar flow.
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or glass tubing. The effect of roughness on f, an effect which is considerable in turbulent flow, is open to some conjecture; artificially roughened pipes, for instance, give results at variance with actual tests. The curves above the smooth pipe curve of Fig. 4 represent a summary of tests on rough pipe, each of them identified by a value of e/d, with e signifying the absolute roughness in feet. Values of e for different pipes are given in Table 1.

To find the friction loss for any pipe, follow the curve with the proper value of e/d, to the pertinent value of N_{Re}, and from this point proceed horizontally to left margin to find the value of f for use in Equation 8.

The curves in Fig. 4 may be approximated very closely by the empirical formula:[1]

$$f = .0055\left[1 + \left(20{,}000\,\frac{e}{d} + \frac{10^6}{N_{Re}}\right)^{1/3}\right] \tag{12}$$

Equation 8 is applicable to all liquids, and to gases when the pressure loss is less than 10 percent of the initial pressure. When the loss in head is high, the formula to be used for gases is

$$\frac{p_1^2 - p_2^2}{p_1^2} = \frac{flV_1^2}{gd\,p_1v_1} \tag{13}$$

which may be arranged to give the loss in pressure,

$$p_1 - p_2 = p_1\left[1 - \sqrt{1 - \frac{flV_1^2}{gd\,p_1v_1}}\right] \tag{14}$$

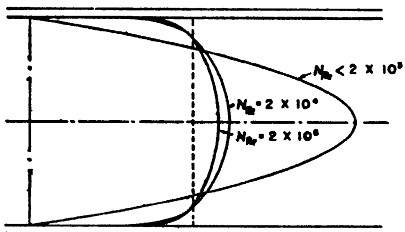

FIG. 5. COMPARISON OF VELOCITY PROFILES FOR 3 DIFFERENT REYNOLDS NUMBERS BUT FOR SAME AVERAGE VELOCITY

Pressure Loss in Non-Circular Pipes

The formulas for friction loss in pipes are based on the use of pipes of circular cross-section. The same formulas may be extended to non-circular sections, by suitable modification. In the basic formula, Equation 8, the internal diameter d is to be replaced by the hydraulic diameter d_H defined by the equation:

$$d_H = \frac{4 \times \text{area of cross-section}}{\text{wetted perimeter of cross-section}} \tag{15}$$

For example, in a rectangular duct, 1 ft by 2 ft, the cross-section area is 2 sq ft, and the perimeter 6 ft. Then the hydraulic diameter will be $d_H = (4 \times 2)/6 = 1^1/_3$ ft.

In the case of a round pipe,

$$d_H = \frac{4 \times \pi d^2/4}{\pi d} = d \tag{16}$$

In computing the Reynolds number, and from that the friction factor, the hydraulic diameter is not to be used. A better approximate procedure is to replace the length in the Reynolds number by the shortest dimension plus one-fourth of the hydraulic diameter. Thus, in a duct of dimension a x b where $a < b$, N_{Re}, for the purposes of calculating friction factors, is

$$N_{Re} = (a + 0.25d_H)V\rho/\mu \tag{17}$$

TABLE 1. VALUES OF e FOR DIFFERENT KINDS OF PIPE

TYPE OF PIPE	e
Smooth drawn tubing	0.000005
Commercial steel or wrought iron	0.00015
Asphalted cast-iron	0.0004
Galvanized iron	0.0005
Cast-iron	0.00085
Wood stave	0.0006 to 0.003
Concrete	0.001 to 0.01
Riveted steel	0.003 to 0.03

This value of N_{Re} may be used in Equation 10 for laminar flow, and in Equation 12 or Fig. 4 for turbulent flow. The error in the approximation is somewhat greater for laminar than for turbulent flow. In the former case, the relative error may be as much as 10 percent, while in the latter it almost always is less than 3 percent.

FLOW OF COMPRESSIBLE FLUIDS

In the flow of compressible fluids, the large density variations make impracticable the use of the Bernoulli equation, (Equation 7). In certain special cases, however, the exact equations for compressible flow may be stated. If flow occurs with no friction or other internal irreversibility, Equation 6 becomes

$$\frac{1}{2g}\,dV^2 + \frac{dp}{\rho} = 0 \tag{18}$$

If, in addition, the flow is adiabatic,

$$p\rho^{-k} = p_1\rho_1^{-k} \tag{19}$$

so that Equation 18 becomes

$$\frac{1}{2g}\,dV^2 + \frac{p_1^{1/k}}{\rho_1}\frac{dp}{p^{1/k}} = 0 \tag{20}$$

or by integration,

$$\frac{1}{2g}(V_2^2 - V_1^2) + \frac{k}{k-1}\frac{p_1}{\rho_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1\right] = 0 \tag{21}$$

This extension to compressible flow of Bernoulli's equation reduces to the more familiar form if the pressure change is small.

The ratio of specific heats, k, is used extensively in fluid dynamics; values of k for various gases are given in Table 2.

TABLE 2. RATIO OF SPECIFIC HEAT AT CONSTANT PRESSURE TO SPECIFIC HEAT AT CONSTANT VOLUME FOR COMPRESSIBLE FLUIDS

COMPRESSIBLE FLUID	RATIO $k = c_p/c_v$
Helium and other monatomic gases	1.66
Air and other diatomic gases	1.40
Ammonia and hydrogen sulfide	1.34
Carbon dioxide, methane, natural gas, superheated steam, moist steam down to a quality of 97 percent	1.28 to 1.32
Sulfur dioxide, ethylene, acetylene	1.24 to 1.26

It is convenient in the analysis of compressible flow to introduce the velocity of propagation of pressure impulses or, more familiarly, the sonic velocity, a. For perfect gases this is given by the equation:

$$a^2 = kgp/\rho = kgRT \tag{22}$$

Accordingly, Equation 21 may be written

$$\tfrac{1}{2}(V_2^2 - V_1^2) + \frac{a_1^2}{k-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} - 1\right] = 0 \tag{23}$$

or, by rearrangement,

$$\frac{p_2}{p_1} = \left[1 - \frac{k-1}{a_1}\left(\frac{V_2^2 - V_1^2}{2}\right)\right]^{\frac{k}{k-1}} \tag{24}$$

which permits the calculation of the ratio of pressures at entrance and exit of the steady flow device—pipe, orifice, or nozzle. From Equations 19 and 22 it follows that

$$\frac{a_2^2}{a_1^2} = \frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} \tag{25}$$

so that

$$\frac{k-1}{2}(V_2^2 - V_1^2) + a_2^2 - a_1^2 = 0 \tag{26}$$

and

$$\frac{a_2^2}{a_1^2} = \frac{1 + \frac{k-1}{2}\frac{V_1^2}{a_1^2}}{1 + \frac{k-1}{2}\frac{V_2^2}{a_2^2}} \tag{27}$$

The ratio of flow velocity to sonic velocity is known as the Mach number,

$$M = V/a$$

This parameter is particularly useful in compressible flow analysis. In general, if $M \leqq 0.1$ the flow may be considered to be incompressible. This is generally true in heating and ventilating air ducts.

In terms of the Mach number

$$\frac{a_2^2}{a_1^2} = \frac{T_2}{T_1} = \frac{1 + \frac{k-1}{2}M_1^2}{1 + \frac{k-1}{2}M_2^2} \tag{28}$$

and

$$\frac{p_2}{p_1} = \left(\frac{1 + \frac{k-1}{2}M_1^2}{1 + \frac{k-1}{2}M_2^2}\right)^{\frac{k}{k-1}} \tag{29}$$

The quantity,

$$p^0 = p\left(1 + \frac{k-1}{2}M^2\right)^{\frac{k}{k-1}} \tag{30}$$

is called the stagnation pressure, and gives a measure of pressure energy. For incompressible flow

$$p^0 = p + \frac{1}{2g}\rho V^2 = p + q \tag{31}$$

where

$$q = \frac{1}{2g}\rho V^2 \tag{32}$$

and is the dynamic pressure. A total head tube measures stagnation pressure directly.

From Equation 29 it follows that for frictionless, adiabatic flow

$$p_2^0 = p_1^0 \tag{33}$$

This represents another extension of the Bernoulli equation to compressible flow. Friction will cause a loss in pressure energy.

Ideal Flow through Nozzle or Orifice

The majority of low measuring systems depend upon a correlation between pressure drop, area, and quantity of flow. The basic formulas may be stated on the assumption that the flow is frictionless and adiabatic. Designating the main stream by station 1, and flow at some measuring restriction by station 2, the flow in pounds per second is

$$w = \rho_1 A_1 V_1 = \rho_2 A_2 V_2 \tag{34}$$

or, in terms of Mach number,

$$w = A_2 M_2 \sqrt{kgp_2\rho_2} \tag{35}$$

According to Equation 29

$$\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} = \frac{1 + \frac{k-1}{2} M_2^2}{1 + \frac{k-1}{2} M_1^2} \tag{36}$$

from which

$$M_2^2 = \frac{2}{k-1}\left(\frac{1 + \frac{k-1}{2} M_1^2}{1 - M_1^2/M_2^2}\right)\left[\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} - 1\right] \tag{37}$$

so that

$$w = A_2 \sqrt{\frac{1 + \frac{k-1}{2} M_1^2}{1 - M_1^2/M_2^2}} \sqrt{\frac{2kgp_2\rho_2}{k-1}\left[\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} - 1\right]} \tag{38}$$

If the initial velocity is sufficiently small, M_1^2 will be negligible so that

$$w = A_2 \sqrt{\frac{2kgp_2\rho_2}{k-1}\left[\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} - 1\right]} = A_2 \sqrt{\frac{2kgp_1\rho_1}{k-1}\left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}\left[\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} - 1\right]} \tag{39}$$

If this is computed and the figures are plotted, the curved line (partly solid and partly broken) of Fig. 6 is found. The maximum value of $\frac{p_2}{p_1}$ may be computed by differentiating w with respect to p_2 and equating the result to zero. This operation produces the formula:

$$\frac{p_2}{p_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \tag{40}$$

For air, with $k = 1.40$, $\frac{p_2}{p_1} = 0.53$.

Actually, the broken part of the curve is not attained for the flow in the nozzle. If the ratio of p_2 to p_1 is decreased from unity, the mass rate of discharge, as well as the volume, increases from zero to a maximum, as shown by the solid section of the curve in Fig. 6; thereafter, as p_2/p_1 is decreased further, the discharge is constant, as indicated by the horizontal line. The value of p_2 at the maximum point is called the critical pressure, or p_c, and it is seen that p_c is approximately 53 percent of p_1 when air is flowing.

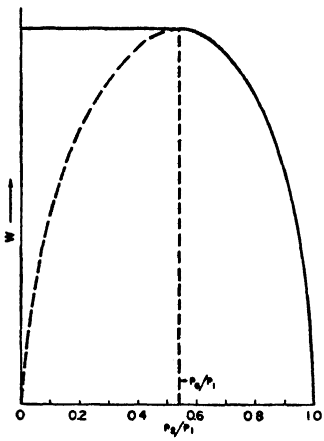

FIG. 6. RELATION OF FLOW OF GAS TO PRESSURE DROP IN A CONVERGING TUBE

To find the velocity at the critical pressure, it is assumed that the upstream velocity V_1 is so small as to be negligible. Using the subscript c to indicate conditions at the critical point, from Equation 29

$$\frac{p_c}{p_1} = \frac{1}{\left(1 + \frac{k-1}{2} M_c^2\right)^{\frac{k}{k-1}}} \tag{41}$$

or

$$M_c = \sqrt{\frac{2}{k-1}\left[\left(\frac{p_1}{p_c}\right)^{\frac{k-1}{k}} - 1\right]} \tag{42}$$

Substituting the critical pressure ratio from Equation 40 it follows that

$$M_c = 1 \tag{43}$$

or that the velocity at the throat is equal to the local sonic velocity.

In developing the working equations for orifices and nozzles, it is customary to start with the incompressible form of the flow Equation 38. In this case both M_1 and M_2 are small quantities, $\rho_1 = \rho_2$, and $(p_1 - p_2)/p_2 = \Delta p/p_2$ is small. Retaining only first order terms, it follows from Equation 35 that

$$\frac{M_1}{M_2} = \frac{A_2}{A_1} \tag{44}$$

so that

$$\sqrt{\frac{1 + \frac{k-1}{2} M_1^2}{1 - M_1^2/M_2^2}} \cong \frac{1}{\sqrt{1 - (A_2/A_1)^2}} = \frac{1}{\sqrt{1 - \beta^4}} \tag{45}$$

where $\beta = D_2/D_1$. The quantity $1/\sqrt{1 - \beta^4}$ is the velocity of approach

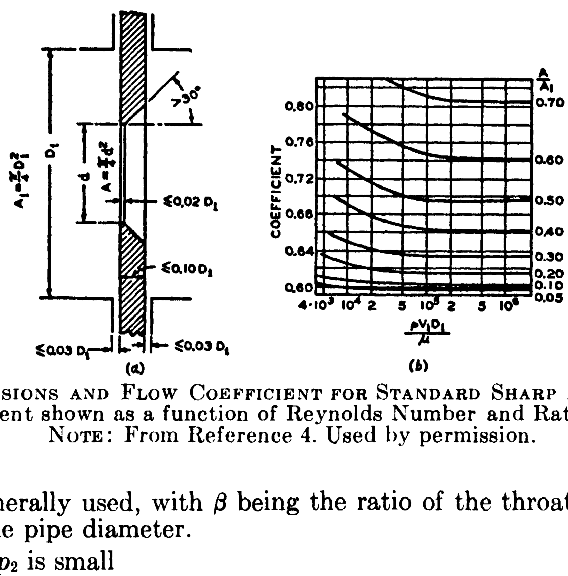

Fig. 7. Dimensions and Flow Coefficient for Standard Sharp Edged Orifice (Coefficient shown as a function of Reynolds Number and Ratio, A/A.)
Note: From Reference 4. Used by permission.

factor as generally used, with β being the ratio of the throat or orifice diameter to the pipe diameter.

Since $\Delta p/p_2$ is small

$$\frac{2k}{k-1}\left[\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} - 1\right] \cong \frac{2\Delta p}{p_2} \tag{46}$$

and the mass flow is

$$w = \frac{A_2}{\sqrt{1 - \beta^4}} \sqrt{2g\rho\Delta p} \tag{47}$$

The volume flow is then

$$Q = A_2 \frac{1}{\sqrt{1 - \beta^4}} \sqrt{2g\Delta p/\rho} = A_2 \frac{1}{\sqrt{1 - \beta^4}} \sqrt{2gh_f} \tag{48}$$

FLOW MEASUREMENT

The measurement of quantity of flow of fluids is generally accomplished either by making observations on the change of state due to flow system configuration, such as the pressure drop across a metering orifice, or by the displacement of some device, such as a rotating vane system. The selection of the metering system will be determined by the type of fluid, the precision of measurement desired, the cost of equipment and installation, the range of flow quantity, the ease of maintenance, and the method of observation and recording.

Fluid meters may be classified as follows: (1) *Head Meters* (Pressure Sensitive), which may be of *Venturi, flow nozzle, orifice plate*, or *Pitot tube* types; (2) *Area Meters*, which may be of *gate, tapered tube*, or *tapered plug* types; (3) *Force Meters*, which may be of *vane, propellor*, or *turbine* types; (4) *Quantity Meters*, which may be of *weighing tank, reciprocating piston*, or *geared impeller* types.

Rate meters are generally of the first three groups, although rates are

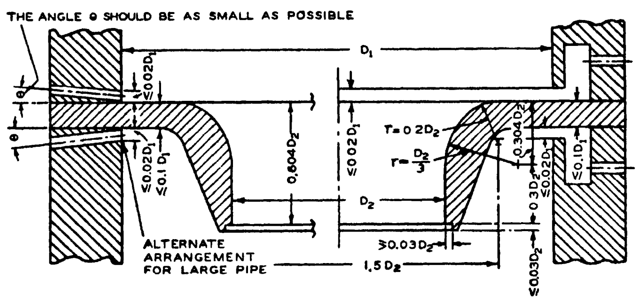

FIG. 8. DIMENSIONS FOR INTERNATIONAL STANDARDS ASSOCIATION FLOW NOZZLE
NOTE: From Reference 3. Used by permission of *A.S.M.E.*

obtainable from timed observations of quantity meters. Similar quantity measurements can be obtained by suitable integration of rate meter indications.

Head Meters

The head meter is of sufficient flexibility so that almost any type of flow measuring problem can be handled. For this reason, standard and reference measurements are usually made in this way. With proper care, extreme precision can be obtained. Also, an inexpensive installation can be made to give moderate precision. In general, a head meter requires fairly competent installation and maintenance to give satisfactory service.

Among the types of head meters, the Venturi has the advantage of having a low pressure loss, but requires considerable length of space. The orifice plate reduces the required length to a minimum at the cost of considerable pressure drop. The flow nozzle represents a compromise, but must be calibrated individually, or made with extreme care from standard specifications, to obtain a good precision of measurement. Standard configurations, for representative orifice plates and flow nozzles, are shown in Figs. 7 and

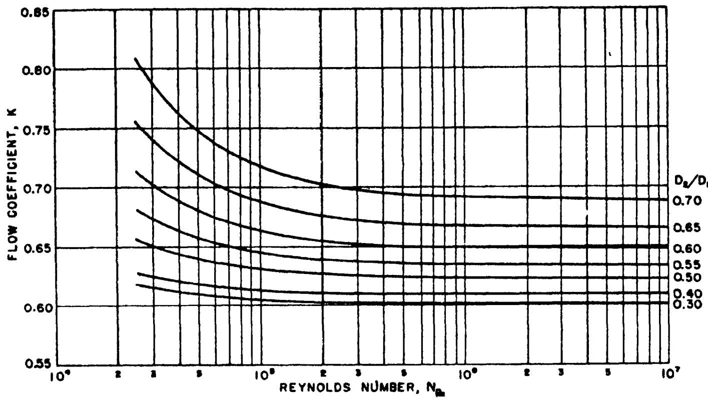

FIG. 9. FLOW COEFFICIENTS, K, FOR SQUARE-EDGED ORIFICE PLATES AND FLANGE TAPS IN SMOOTH PIPE

NOTE: From Table 6 of Reference 2.

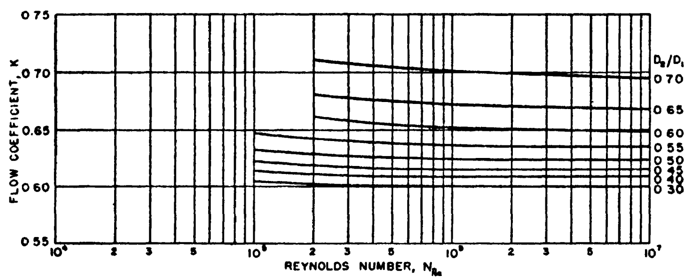

FIG. 10. FLOW COEFFICIENTS, K, FOR SQUARE-EDGED ORIFICE PLATES AND RADIUS TAPS IN SMOOTH PIPE

NOTE: From Fig. 36d of Reference 4.

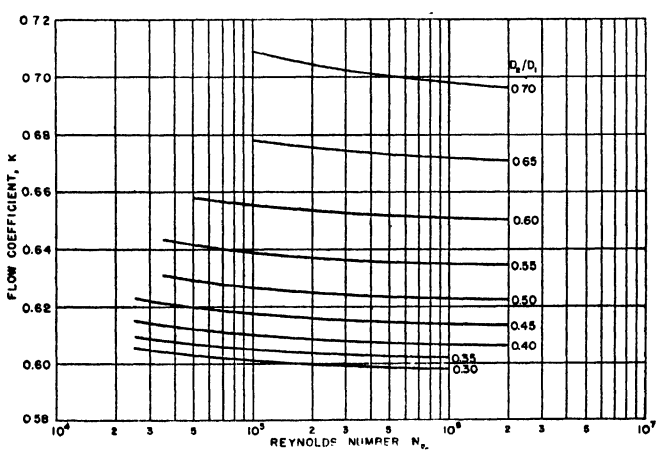

FIG. 11. FLOW COEFFICIENTS, K, FOR SQUARE-EDGED ORIFICE PLATES AND VENA CONTRACTA TAPS, IN SMOOTH PIPE

NOTE: From Table 7 of Reference 2.

8, respectively. Additional specifications are given in References 2, 3, and 4 at end of chapter.

The measurement of flow in head meters is dependent upon observations of a static pressure difference between two parts of the system. In general, there exists a reduction in area, either by a smooth contour as in the case of a Venturi or flow nozzle, or by a vena contracta following an orifice. In either case, the flow will be given by a suitable modification of Equations 47 and 48. Adding a correction term, these become

$$w = \frac{A_2 C}{\sqrt{1 - \beta^4}} \sqrt{2g\rho\Delta p} \tag{49}$$

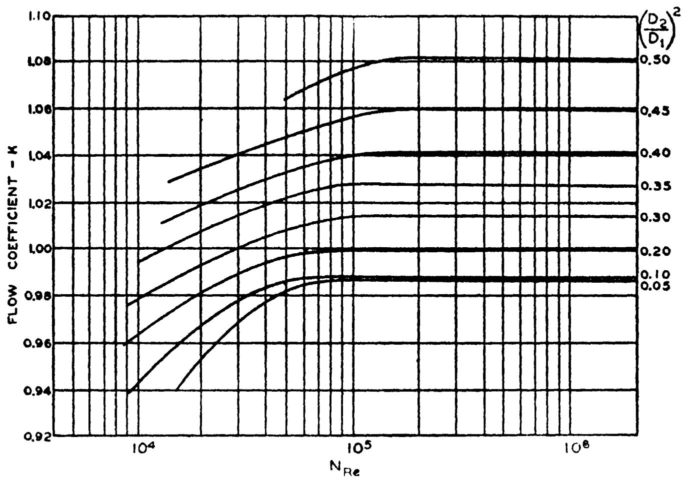

FIG. 12. FLOW COEFFICIENT FOR INTERNATIONAL STANDARDS ASSOCIATION FLOW NOZZLE AS A FUNCTION OF AREA RATIO $(D_2/D_1)^2$ AND THE REYNOLDS NUMBER N_{Re}
NOTE: From Reference 3. Used by permission of *A.S.M.E.*

and

$$Q = \frac{A_2 C}{\sqrt{1 - \beta^4}} \sqrt{2gh_f} \tag{50}$$

In all cases, A_2 refers to the minimum area of the Venturi, nozzle, or orifice. The correction factor C is introduced to account for any loss due to departure from isentropic flow, and for any deviation between the downstream measured pressure and the actual pressure at the minimum section. Since these corrections are usually dependent on configuration, as described by the ratio of upstream to minimum area A_1/A_2, it is often convenient to introduce a combined flow coefficient, K which is

$$K = \frac{C}{\sqrt{1 - \beta^4}} \tag{51}$$

so that

$$w = KA_2\sqrt{2g\rho\Delta p} \tag{52}$$

and

$$Q = KA_2\sqrt{2gh_l} \tag{53}$$

The coefficients K and C will be determined by the area ratio A_1/A_2, or by the diameter ratio D_1/D_2 and the Reynolds number. Extensive data are available from various sources.[3,4] Figs. 9, 10, 11, 12 show representative values of K based on N_{Re}, the Reynolds number, and the ratio of diameter of the orifice or nozzle throat to pipe diameter.

In any specific application, numerical values for areas, densities, and pressure units can be substituted in Equations 52 and 53 to obtain compact working formulas. In cases where the tests are run under non-standard

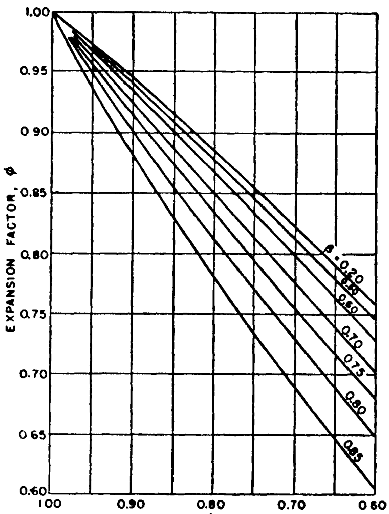

FIG. 13. RELATION OF EXPANSION FACTOR, ϕ, FOR NOZZLES TO DIAMETER RATIO AND PRESSURE LOSS FOR AIR AND OTHER DIATOMIC GASES

conditions, care must be taken that proper adjustment be made in computing flow rates.

In measuring the flow of compressible fluids, the approximate Equations 52 and 53 must be corrected for density variations. The need for such correction is evident by comparing Equations 35 and 38 with Equation 47. Introducing a multiplicative correction factor ϕ the equation for flow with no loss becomes:

$$w = \frac{A_2\phi}{\sqrt{1-\beta^4}}\sqrt{2g\rho_1(p_1 - p_2)} \tag{54}$$

By comparison with Equation 35,

$$\phi = \sqrt{\frac{k}{2}\frac{p_2}{p_1}\frac{M_2^2(1-\beta^4)}{(p_1/p_2) - 1}} \tag{55}$$

From Equations 34 and 35,

$$M_1^2/M_2^2 = \beta^4(p_2/p_1)^{2/k} \tag{56}$$

Hence, by Equation 38 for a small value of M_1,

$$\phi = \sqrt{(p_2/p_1)^{1/k}\left(\frac{k}{k-1}\right)\frac{(p_1/p_2)^{(k-1)/k} - 1}{(p_1/p_2) - 1}\cdot\frac{1 - \beta^4}{1 - \beta^4(p_2/p_1)^{2k}}} \tag{57}$$

From Equations 37 and 56, if M_2 is not too large,

$$\frac{2}{k}\frac{1}{M_2^2}\left(\frac{p_1}{p_2} - 1\right) \simeq 1 - \beta^4 + \tfrac{1}{4}[1 - 2(k-2)\beta^4]M_2^2 \tag{58}$$

so that, approximately,

$$\phi \simeq 1 - \frac{3}{8}\left[1 + \left(1 - \frac{2k}{3}\right)\beta^4\right]M_2^2 \tag{59}$$

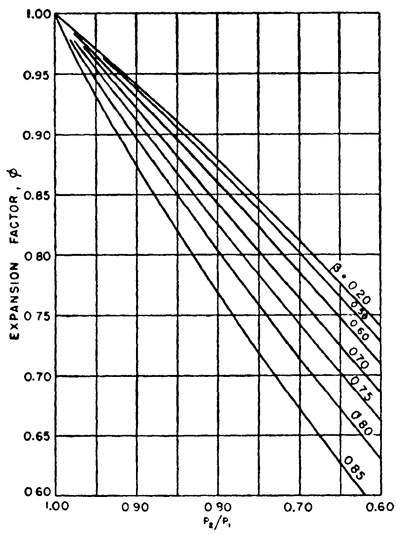

FIG. 14. RELATION OF EXPANSION FACTOR, ϕ, FOR NOZZLES TO DIAMETER RATIO AND PRESSURE LOSS FOR STEAM, CARBON DIOXIDE AND NATURAL GAS

where to the same degree of approximation from Equation 36

$$M_2^2 = 2\frac{p_1 - p_2}{kp_1} \tag{60}$$

Values of ϕ are also given by Figs. 13 and 14.

While ϕ may be used for smooth nozzles and Venturi tubes, it is necessary in the case of orifices, where the departure from ideal flow is significant, to replace ϕ by an empirical factor Y obtained by the equation

$$Y = 1 - (0.41 + 0.35\beta^4)\left(\frac{p_1 - p_2}{p_1}/k\right) \tag{61}$$

When this correction is used, the flow will be given by the equation

$$w = KA_2Y\sqrt{2g\rho_1(p_1 - p_2)} \tag{62}$$

Installation of Head Meters

In the installation of Venturis, orifices, and nozzles, care must be taken in regard to upstream and downstream flow conditions. Recommended practice, with reference to fittings and valves, is shown by Fig. 15. If these conditions cannot be met, some flexibility is possible by introducing straightening vanes, as described more fully in References 2 and 5.

The static pressure difference across the head meter is measured by suitably located static pressure taps. For orifices, the different sets of pressure taps are called flange taps, radius taps, vena contracta taps, and pipe or full-flow taps. The relative locations of the first three of these are shown in Fig. 16. The need for applying suitable coefficients to measure-

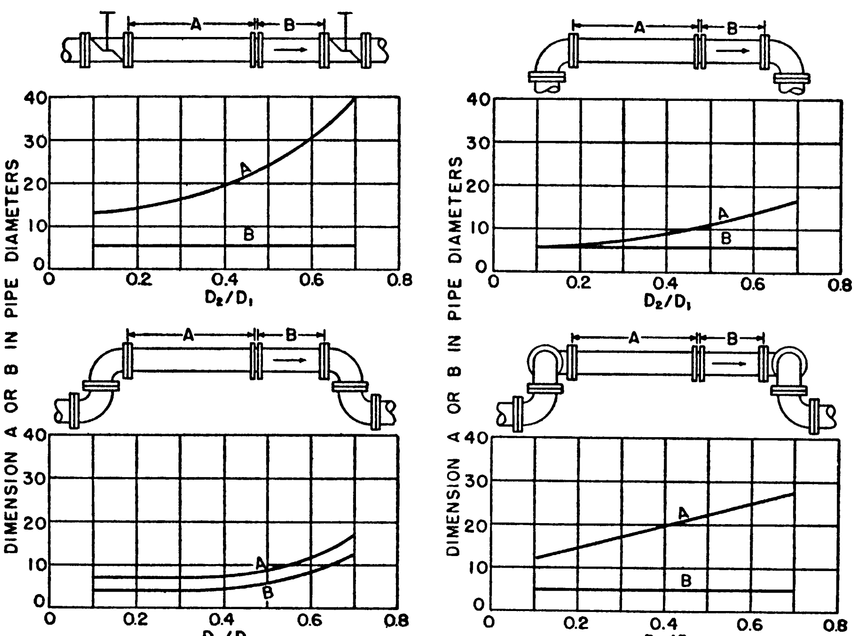

FIG. 15. MINIMUM CONDITIONS TO BE OBSERVED WHEN INSTALLING ORIFICES AND NOZZLES BETWEEN FITTINGS AND VALVES

ments obtained by means of the different taps, is indicated by the curve of change, in pressure of the flowing fluid, shown in the lower part of Fig. 16. Pipe taps are located $2\frac{1}{2}$ pipe diameters upstream and 8 pipe diameters downstream (both distances being measured from the upstream face of the orifice plate) so that pressures are measured before the orifice plate has had any effect on the flow, and after the recovery in pressure has been completed. The use of pipe or full-flow taps has been limited to the metering of natural fuel gas in certain areas. As they are not suited to use in heating and ventilating work, no data for them are given in this chapter.

Still another type of pressure tap, the corner tap, is used in European practice. Pressures are taken from recesses in the flange connected to annular slits in the corners formed by the pipe wall and the orifice plate.

Coefficients for these taps have been adopted by the *International Standards Association*, but are not used commercially in America.

It will be noted that the location of the downstream pressure tap of the vena contracta arrangement is variable. Vena contracta is the term applied to the minimum cross-section of the jet from the orifice, where the static pressure is at a minimum. Its location, and the location of the downstream vena contracta tap, vary with the ratio of orifice to pipe diameter, and with rate of flow, as shown in Fig. 17; the tap is generally located in accordance with the mean curve in the figure.

Similar precautions are required when using flow nozzles and Venturi tubes. In many cases, these will be supplied ready for installation. In such cases, the manufacturer's instructions should be followed with care

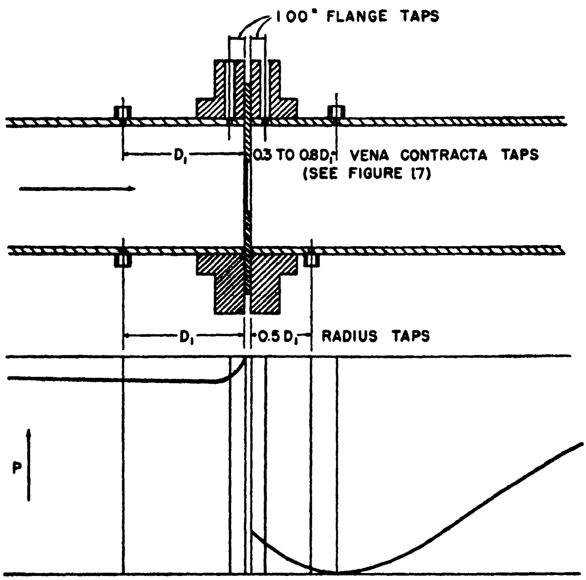

FIG. 16. RELATIVE LOCATION OF FLANGE, RADIUS AND VENA CONTRACTA TAPS

in order to avoid serious errors. Further information on such systems is given in References 2, 3, and 5.

Pitot Tube

In certain cases, such as in rectangular ducts, it is impracticable to use standard orifices, and consequently, either a specially designed orifice must be calibrated, or an independent flow device must be used. In either case, the Pitot tube is useful. It consists essentially of an inner bent tube with its open end pointing upstream so as to measure total pressure, and an outer tube having small holes on the side for communicating static pressure to a manometer. (See Fig. 3, Chapter 49). The difference in liquid level in the manometer will be proportional to the square of the velocity, for incompressible flow, so that in general

$$V = \sqrt{2gh_t} \tag{63}$$

For compressible flow, the differential head is to be divided by the correction F_o from Equation 64,

$$F_o = 1 + \tfrac{1}{4}M^2 + \tfrac{1}{40}M^4 \tag{64}$$

where

$$M^2 = 2\,\frac{p_0 - p}{kp} \tag{65}$$

In the use of a Pitot tube system, care is required in obtaining correct total and static pressures. The total pressure tube must be smoothly constructed, and should point directly upstream. The static pressure tap must be located so that local flow interferences will not reduce the value. In any case, it is better to obtain an independent calibration, or use a specially designed and manufactured probe. A number of such Pitot tube probes are available, and can be used without calibration.

In using Pitot tubes to obtain flow rates, it is necessary to make a traverse of the pipe and thereby to obtain one of the profiles of Fig. 5.

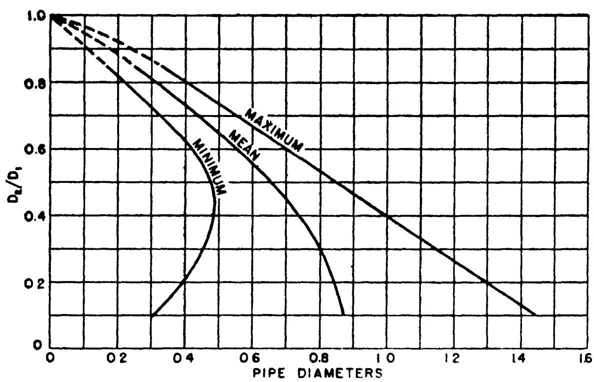

FIG. 17. LOCATION OF VENA CONTRACTA IN RELATION TO RATIO OF ORIFICE TO PIPE DIAMETER AND TO RATE OF FLOW

In rectangular ducts or near valves or fittings, a disturbed flow pattern would be obtained, and therefore, in such cases, a fairly complete survey should be made. The flow in such cases will be computed from the average of the local velocities, as obtained by Equation 63.

Variable Area Flow Meters

For permanent installations where high precision, ruggedness, and ease of operation are important, the variable area flow meter has proved very satisfactory. Its most frequent use is in measurement of liquids or gases in small diameter pipes. For ducts or pipes over 6 in. in diameter, the expense of this meter may not be warranted. In large systems, however, the meter might be placed in a by-pass line and used in conjunction with an orifice.

In its most common form, the variable area meter, Fig. 18, consists essentially of a float which is free to move vertically in a transparent tapered tube. The fluid to be metered enters at the narrow bottom end of the tube and moves upward, passing at some point through the annulus formed between the float and inside wall of the tube. At any particular rate of flow, the float assumes a definite position in the tube, its location being indicated by means of a calibrated scale on the tube.

The position of the float is established by a balance between the fluid pressure forces across the annulus and the weight of the float itself. The buoyant force which must support the float, $v_f(\rho_f - \rho)$, is balanced by the pressure difference acting on the cross-section area of the float, $A_f\Delta p$, where ρ_f, A_f, v_f, are, respectively, the float density, float cross-section area, and float volume. Accordingly, the difference in head across the annulus is given by

$$h_f = \frac{\Delta p}{\rho} = \frac{v_f(\rho_f - \rho)}{A_f\rho} \tag{66}$$

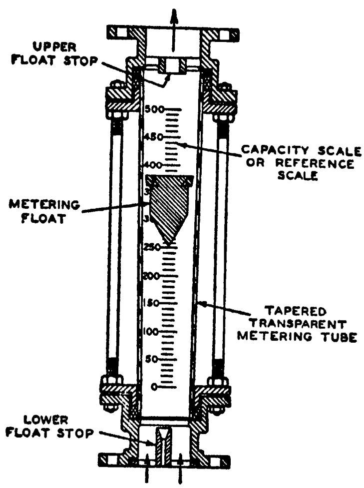

FIG. 18. SCHEMATIC DIAGRAM OF VARIABLE AREA FLOW METER

The volume flow follows from equation (53) as

$$Q = KA_2\sqrt{2gv_f(\rho_f - \rho)/\rho A_f} \tag{67}$$

and the mass flow as

$$w = \rho Q = KA_2\sqrt{2gv_f(\rho_f - \rho)\cdot\rho/A_f} \tag{68}$$

The flow for any selected fluid is, accordingly, very nearly proportional to the area, so that a convenient calibration of the tube may be obtained. The behavior of the flow coefficient, K, has been investigated[6] and the action of the flow meter as just outlined, experimentally confirmed. The flow coefficient variation for any float must be known in order to use the meter for different fluids. Some developments have been carried on in the design of the float to reduce the variation of the flow coefficient with Reynolds number, and also with regard to float materials, to reduce the dependence of mass flow calibration on fluid density.

This type of flow meter is usually furnished in standard sizes calibrated

for specific fluids by the manufacturer. The compactness, reliability, and ease of installation are particularly advantageous when many measurements of essentially the same type are to be made.

LETTER SYMBOLS USED IN CHAPTER 4

β = ratio, throat or orifice diameter to pipe diameter.
μ = absolute viscosity, pounds per foot second.
μ/ρ = kinematic viscosity, square feet per second.
ρ = density of flowing fluid, pounds per cubic foot.
ρ_m = proper mean density.
ρ_w = density of water at 60 F (62.37 lb per cubic foot).
ρ_f = density of float in variable area meters.
ϕ = expansion factor for nozzles.
a = velocity of sound, feet per second.
A = cross-sectional area of flow, square feet.
C = correction factor (coefficient of discharge) for flow through orifice, nozzle or Venturi.
c_p = specific heat of gas at constant pressure.
c_v = specific heat of gas at constant volume.
D = diameter of fluid stream, feet.
d = internal diameter of pipe, feet.
d_h = hydraulic diameter, feet.
e = absolute roughness of pipe surface, feet.
F_c = correction factor for differential head in compressible flow.
f = dimensionless friction coefficient.
g = gravitational acceleration, feet per (second) (second).
g_c = gravitational conversion factor = 32.174 (pounds mass per pound force) × feet per (second) (second).
h = enthalpy, Btu per pound of fluid.
h_f = loss of head, feet of fluid.
h_t = total head, feet of fluid.
J = mechanical equivalent of heat = 778 foot pounds per Btu.
K = flow coefficient (correction factor), including velocity of approach correction factor, for flow through orifice, nozzle or Venturi.
k = ratio of specific heat at constant pressure to specific heat at constant volume.
L = perpendicular distance from axis of pipe, feet.
l = length of pipe, feet.
M = Mach number.
N_{Re} = Reynolds number.
p = pressure, pounds per square foot.
p^0 = stagnation pressure.
p_c = critical pressure.
Q = discharge rate, cubic feet per second.
q = heat transferred to the fluid per pound of fluid flowing.
R = gas constant.
r = radius of pipe in feet.
s = entropy of fluid in Btu per (pound) (Fahrenheit degree).
T = temperature, Fahrenheit degrees, absolute.
u = internal energy, Btu per pound of fluid.
V = velocity, feet per second.
V_c = critical velocity, feet per second.
v = specific volume, cubic feet per pound.
W = mechanical work, foot pounds per pound of fluid flowing.
w = mass flow of gas, pounds per second.
Y = expansion factor—correcting for expansion of gas under reduced downstream pressure.
z = elevation above some arbitrary datum, feet.

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CHAPTER 5

HEAT TRANSFER

Conduction, Convection, Radiation; Equations for Conduction, Convection, Radiation and Combined Convection and Radiation; Heat-Flow Resistance, in Series and Parallel; Practical Heat Transfer Problems; Periodic and Transient Heat Flow

HEAT is the form of energy that is transferred by virtue of an existing temperature difference. The temperature difference is the potential which causes the transfer, the latter in turn being resisted by the thermal properties of the material combined in a single term known as the resistance. Energy exchange associated with evaporation, condensation, etc., is treated elsewhere such as in the section on *cooling tower design* in Chapter 34. The objectives of this chapter are to:

1. Describe the mechanisms and present the rate equations for the different modes of heat transfer.
2. Illustrate the application of the basic concepts to steady-state problems (temperature independent of time or a cyclic variable thereof) by means of several typical solutions of heat transfer systems.
3. Present concise summaries of available methods of analysis for transient and periodic heat transfer problems.

Further applications to specific systems will be found throughout THE GUIDE.

CONDUCTION, CONVECTION AND RADIATION

Thermal conduction is the term applied to the mechanism of heat transfer whereby the molecules of higher kinetic energy transmit part of their energy to adjacent molecules of lower kinetic energy by direct molecular action. Since the temperature is proportional to the average kinetic energy of the molecules, thermal transfer will occur in the direction of decreasing temperature. The motion of the molecules is random; there is no net material flow associated with the conduction mechanism. In the case of flowing fluids, thermal conduction is significant in the region very close to a solid boundary or wall, for in this region the flow is laminar, parallel with the wall surface, and there are practically no cross currents in the direction of the heat transfer across the solid fluid boundary. In solid bodies the significant mechanism of heat transfer is always thermal conduction.

Contrasted to the thermal conduction mechanism, *thermal convection* involves energy transfer by eddy mixing and diffusion[1] in addition to conduction. This is shown schematically in Fig. 1 which exhibits transfer from a pipe wall at surface temperature t_s to a colder fluid at a bulk temperature t_f. (Bulk temperature is that which would be attained if the fluid stream were drawn off at a certain section and mixed. It is therefore somewhat higher than the lowest temperature in the stream.) In the *laminar* sublayer, immediately adjacent to the wall, the heat transfer occurs by thermal conduction; in the *transition* region, which is called the buffer layer, eddy mixing as well as conduction effects are significant;

in the *eddy* or *turbulent* region the major fraction of the transfer occurs by eddy mixing.

In most commercial equipment the main body of the fluid is in turbulent flow, and the laminar film exists at the solid walls only, as shown in Fig. 1, but in cases of low-velocity flow in small tubes, or with viscous liquids such as heavy oil (low Reynolds numbers), the entire flow may be laminar. In these latter cases there is no transition or eddy region.

When the fluid currents are produced by sources external to the heat transfer region, as for example by a pump, the described solid to fluid heat transfer is termed *forced convection*. In contrast, if the fluid currents are generated internally, as a result of non-homogeneous densities arising from the temperature variations, the heat transfer is termed *free convection*.

In the conduction and convection mechanisms, the transfer of heat is associated with matter. For *radiant heat transfer*, however, a change in

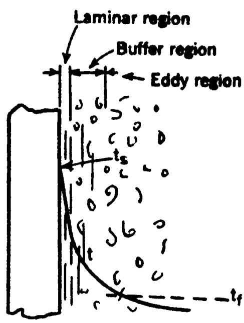


FIG. 1. THERMAL CONVECTION CONDITIONS

energy form takes place, from internal energy at the source to electromagnetic energy for transmission, then back to internal energy at the receiver.

The rate of heat transfer, corresponding to the three transfer mechanisms previously described, may be expressed by three rate equations. These are similar to Ohm's Law for electrical flow, the current flow through a resistance being proportional to the potential. The convection and radiation flow rate expressions may be approximated by a potential (temperature difference) and a resistance in order that heat transfer calculations may be effected more conveniently and rapidly.

Thermal Conduction Equation

Equation 1 states symbolically that the thermal conduction per unit transfer area normal to the flow, q/A, Btu per (hour) (square foot), is proportional to the temperature gradient $(dt)/(dL)$, Fahrenheit degrees per foot. The proportionality factor is termed the *thermal conductivity*, k, Btu per (hour) (square foot) (Fahrenheit degree per foot of thickness).

$$\frac{q}{A} = -k\frac{dt}{dL} \tag{1}$$

The minus sign on the right side of the equation is introduced to indicate positive transfer in the direction of decreasing temperature. Fig. 2 shows the physical significance of the indicated quantities.

It should be emphasized that the thermal conductivity used should be expressed in consistent units; either using the inch or foot throughout.

Expressions of conductivity used in the heating field are usually inconsistent in this sense, in that it is customary to refer to the conductivity *per square foot* but for one inch of thickness. This custom has been adopted for the reason that wall thicknesses are usually expressed in inches, whereas if expressed in feet, decimal or fractional thicknesses would result. When dealing with flat walls, no complication is involved in using the inconsistent expression of conductivity. However, where curved or spherical walls are concerned, considerable complication is involved. Therefore, in this discussion the consistent units of conductivity expressed in *Btu per* (*hour*) (*square foot*) (*Fahrenheit degrees per one foot thickness*) are used throughout. *Conductivity values obtained from Chapter 9 or Table* 1 *in this chapter, must therefore be converted for use in the calculations of this chapter by dividing by 12.* As an example, the conductivity of brick listed as 5.0 in Table 2 of Chapter 9, becomes 0.42 when used in the calculations of this chapter.

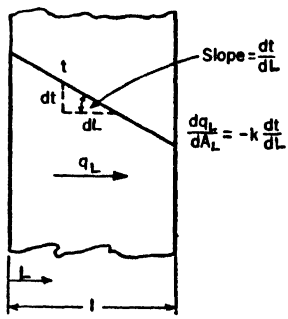


FIG. 2. THERMAL CONDUCTION IN A FLAT SLAB

Also, it should be emphasized that in order to make the calculations and applications consistent in this chapter, *all dimensions of thickness must be expressed in feet.*

Thermal Convection Equation

$$\frac{q}{A} = h_c(t_s - t_f) \tag{2}$$

This rate equation states that the thermal convection per unit transfer area (q/A), Btu per (hour) (square foot) is proportional to the tem-

TABLE 1. APPROXIMATE UNIT THERMAL CONDUCTIVITIES[a]

Conductivity, k = Btu per (hr) (sq ft) (F deg per in.)

MATERIAL	k	MATERIAL	k
Air	0.168	Lead	240.0
Aluminum	1416.0	Nickel	408.0
Brass (70 − 30)	720.0	Soil	2.4–12.0
Cast-Iron	336.0	Steel, mild	312.0
Copper	2640.0	Water, liquid	4.08
Glass	3.6–7.32		

[a] Thermal conductivities depend to some extent on temperature. The above magnitudes are approximate only. Refer to Chapter 9, and Reference 4 for additional data.

perature difference $(t_s - t_f)$ which is the temperature of the surface less that of the fluid. The particular fluid temperature to use for a given system will be noted under the discussion of that system. The proportionality factor is termed the *unit thermal convective conductance* (sometimes called the film coefficient for convection), h_c, Btu per (hour) (square foot) (Fahrenheit degree). Fig. 1 shows the conditions associated with convection.

The heat transmission by free or natural convection for objects surrounded by air can be conveniently expressed as in Equation 2a:

$$\frac{q}{A} = C\left(\frac{1}{D}\right)^{0.2}\left(\frac{1}{T_{av}}\right)^{0.181}(t_s - t_f)^{1.27} \quad (2a)$$

where

$\frac{q}{A}$ = heat transmission by convection, Btu per (square foot) (hour).
C = a constant depending upon the shape of the surface.
D = diameter of pipe or circular duct or height of vertical wall, inches. (Effect of diameter or height becomes constant at 24 in.).
T_{av} = average of wall surface and surrounding air temperature, Fahrenheit degrees absolute.
$t_s - t_f$ = temperature excess between wall surface and surrounding air, Fahrenheit degrees.

For horizontal cylinders, the value of C = 1.02 has been well established by various investigations. For vertical plates, the value of C = 1.39 has been fairly well established. Suggested values[2] of C for horizontal plates warmer than the surrounding air are 1.79 when facing upward, and 0.89 when facing downward.

Problems in either forced convection or natural convection may be solved by the simple first-power equation if the convection coefficient is expressed as a unit conductance:

$$q = h_c A (t_1 - t_2) \quad (2b)$$

where

q = heat transmission by convection, Btu per hour.
A = surface area, square feet.
$t_1 - t_2$ = temperature difference between the surface and the fluid, Fahrenheit degrees.
h_c = unit convective conductance, from Table 2, Btu per (square foot) (hour) (Fahrenheit degree temperature difference).

Thermal Radiation Equation

The relation given by Equation 3 is applicable to systems in which radiant exchange takes place between the surfaces of solids, as schemati-

$$q_r = \sigma A_1 F_A F_E (T_1^4 - T_2^4) \quad (3)$$

cally shown in Fig. 3. Gaseous and luminous radiation are not considered in this discussion. Equation 3 states that the net radiation per unit transfer area of surface 1, q_r/A Btu per (hour) (square foot), which *sees* surface 2 through a non-absorbing medium, is proportional to the difference of the

Table 2. Approximate Unit Conductances for Thermal Convection for Several Flow Systems

Expressed in Convenient Empirical Form

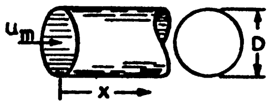
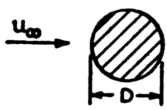
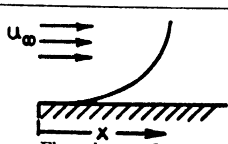
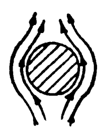
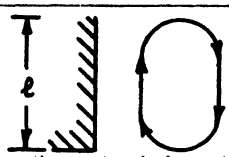
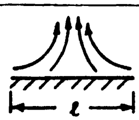
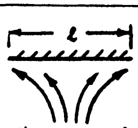
Case	System	Heat Transfer Equation[a] and its Limits of Application	Reference
		Forced Convection	
1	 Longitudinal flow in a circular cylinder[b].	*General Equation* $\frac{h_cD}{k} = 0.0225 \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{C_p\mu}{k}\right)^{0.33}$ where $x > 4.4D$ and $\left(\frac{DG}{\mu}\right) > 2200$	3
2	Same as Case 1.[b]	*Equation for Air* $h_c = 5.4 \times 10^{-4}(T_f)^{0.3} \frac{G^{0.8}}{D^{0.2}}$ where $x > 4.4D$ and $\left(\frac{DG}{\mu}\right) > 2200$	3
3	Same as Case 1.[b]	*Equation for Liquid Water* (32–400 F) $h_c = 13.5(t_f)^{0.54} \frac{u_m^{0.8}}{D^{0.2}}$ where $x > 4.4D$ and $\left(\frac{DG}{\mu}\right) > 2200$	3
4	 Flow normal to a single cylinder.	*General Equation* $\frac{h_cD}{k} = 0.26 \left(\frac{DG}{\mu}\right)^{0.6} \left(\frac{c_p\mu}{k}\right)^{0.3}$ where $1000 < \frac{DG}{\mu} < 50{,}000$	4
5	Same as Case 4.	*Equation for Air* $h_{c(\text{average})} = 0.211(T_f)^{0.43} \frac{(u_\infty\rho)^{0.8}}{D^{0.4}}$ where $1000 < \frac{DG}{\mu} < 50{,}000$	4
6	Same as Case 4.	*Equation for Liquid Water* (32–400 F) $h_{c(\text{average})} = 34.0(t_f)^{0.36} \left(\frac{(u_\infty)^{0.6}}{D^{0.4}}\right)$ where $1000 < \frac{DG}{\mu} < 50{,}000$	4
7	 Flow along a flat plate.	*General Equation* (Turbulent Flow) $\frac{h_{cx}x}{k} = 0.0296 \left(\frac{xG}{\mu}\right)^{0.8} \left(\frac{c_p\mu}{k}\right)^{0.33}$ where $\left(\frac{xG}{\mu}\right) > 500{,}000$ and $h_c = 1.25h_{cx}$	3
8	Same as Case 7.	*Equation for Air* (Turbulent Flow) $h_{cx} = 0.51(T_f)^{0.3} \frac{(u_\infty\rho)^{0.8}}{(x)^{0.2}}$ where $\left(\frac{xG}{\mu}\right) > 500{,}000$ $h_{c(\text{average})} = 1.25h_{cx}$	3

Table 2. Approximate Unit Conductances for Thermal Convection for Several Flow Systems (*Concluded*)

Case	System	Heat Transfer Equation[a] and its Limits of Application	Reference
		Forced Convection	
9	Same as Case 7.	*General Equation* (Laminar Flow) $\frac{h_{cx}x}{k} = 0.332 \left(\frac{xG}{\mu}\right)^{0.5} \left(\frac{c_p\mu}{k}\right)^{0.33}$ For $\left(\frac{xG}{\mu}\right) < 500{,}000$ $h_{c\,(\text{average})} = 2h_{cx}$	3
10	Same as Case 7.	*Equation for Air* (Laminar Flow) $h_{cx} = 0.0562(T_f)^{0.5} \frac{(u_\infty \rho)^{0.5}}{x^{0.5}}$ For $\left(\frac{xG}{\mu}\right) < 500.000$	3
		Free Convection[c]	
11	 Free convection past a heated horizontal cylinder.	*Equation for Air* $h_c = 0.271 \left(\frac{P}{P_0}\right)^{0.5} \left(\frac{\Delta t}{D}\right)^{0.25}$ $10^3 < N_{Gr} < 10^7$	5
12	 Free convection past a single vertical surface.	*Equation for Air* $h_c = 0.354 \left(\frac{P}{P_0}\right)^{0.50} \left(\frac{\Delta t}{l}\right)^{0.25}$ $10^3 < N_{Gr} < 10^7$ $h_c = 0.420 \left(\frac{P}{P_0}\right)^{0.50} \left(\frac{\Delta t}{l}\right)^{0.25}$ $10^8 < N_{Gr} < 10^{10}$	6
13	 Free convection past a heated horizontal surface (face up).	*Equation for Air* $h_c = 0.478 \left(\frac{P}{P_0}\right)^{0.50} \left(\frac{\Delta t}{l}\right)^{0.25}$ $10^3 < N_{Gr} < 10^7$	5
14	 Free convection past a heated horizontal surface (face down).	*Equation for Air* $h_c = 0.239 \left(\frac{P}{P_0}\right)^{0.50} \left(\frac{\Delta t}{l}\right)^{0.25}$ $10^3 < N_{Gr} < 10^7$	5

[a] Fluid properties should be evaluated at the arithmetic mean fluid temperature, $t_f = (t_{\text{surface}} + t_{\text{fluid}})$ divided by 2.

[b] These expressions are suitable approximations to longitudinal flow in other than right circular cylinders, provided the hydraulic diameter is employed as the conduit dimension parameter. For non-circular cross-sections, the hydraulic diameter is equal to four times the cross-sectional area divided by the wetted perimeter.

[c] For low rates of heat transfer by free convection the exponent decreases towards zero, and for higher rates, increases towards 0.33. The above equations employing an exponent equal to 0.25 are applicable in the intermediate range indicated.

NOMENCLATURE AND DIMENSIONS FOR TABLE 2

c_p = heat capacity at constant pressure, Btu per (pound) (Fahrenheit degree).
D = cylinder diameter, feet.
f = subscript denoting film.

g = body force per unit mass, feet per hour per hour. (For static system on earth, $g = 32.2 \times 3600^2$ feet per hour per hour.)
G = 3600 $u_m\rho$ = mass flow per unit cross-sectional area normal to flow, pounds per (hour) (square foot of flow cross-section).
N_{Gr} = Grashof modulus, dimensionless ($N_{Gr} = D^3\rho^2\beta\Delta t g/\mu^2$).
h_c = average unit thermal convective conductance from the leading edge of surface to the position x, Btu per (hour) (square foot) (Fahrenheit degree).
h_{cx} = local unit thermal convective conductance, at the position x from the leading edge of surface, Btu per (hour) (square foot) (Fahrenheit degree).
k = thermal conductivity, Btu per (hour) (square foot) (Fahrenheit degree per foot thickness).
l = a dimension of the system, feet.
m = a subscript denoting *mean*.
P = pressure, atmospheres.
P_o = pressure (atmospheric) atmospheres.
t = temperature, Fahrenheit.
T = temperature, Fahrenheit, absolute.
u = fluid velocity, feet per second.
V = volume, cubic feet.
x = a dimension of the system, feet.
β = coefficient of cubical expansion ($\beta = \frac{1}{V}(\frac{dV}{dT})_P$); for perfect gases $\beta = 1/T$.
Δt = difference between wall and fluid temperatures, Fahrenheit degrees.
μ = fluid viscosity, pounds per (hour) (foot).
ρ = density, pounds per cubic foot.
∞ = infinity, referring the quantity to a point not directly affected by the phenomenon in question.

TABLE 3. RADIATION FACTORS OR EMISSIVITIES, ϵ*

For the determination of factor F_E in Equation 3

Class	Surfaces	Fraction of Black-Body Radiation At 50–100 F	Fraction of Black-Body Radiation At 1000 F	Absorptivity for Solar Radiation
1	A small hole in a large box, sphere, furnace, or enclosure	0.97 to 0.99	0.97 to 0.99	0.97 to 0.99
2	Black non-metallic surfaces such as asphalt, carbon, slate, paint, paper	0.90 to 0.98	0.90 to 0.98	0.85 to 0.98
3	Red brick and tile, concrete and stone, rusty steel and iron, dark paints (red, brown, green, etc.)	0.85 to 0.95	0.75 to 0.90	0.65 to 0.80
4	Yellow and buff brick and stone, firebrick, fire clay	0.85 to 0.95	0.70 to 0.85	0.50 to 0.70
5	White or light-cream brick, tile, paint or paper, plaster, whitewash	0.85 to 0.95	0.60 to 0.75	0.30 to 0.50
6	Window glass	0.90 to 0.95		Transparent[a]
7	Bright aluminum paint; gilt or bronze paint	0.40 to 0.60		0.30 to 0.50
8	Dull brass, copper, or aluminum; galvanized steel; polished iron	0.20 to 0.30	0.30 to 0.50	0.40 to 0.65
9	Polished brass, copper, monel metal	0.02 to 0.05	0.05 to 0.15	0.30 to 0.50
10	Highly polished aluminum, tin plate, nickel, chromium	0.02 to 0.04	0.05 to 0.10	0.10 to 0.40

* Emissivities of other materials may be found in Reference 4. [a] Reflects about 8 percent.

fourth powers of the absolute surface temperatures ($T_1^4 - T_2^4$). The proportionality factor ($\sigma F_A F_E$) may be conveniently separated into three parts (excepting in some problems involving interreflections, where it is not possible to divide the product ($F_A F_E$) into separate terms):

σ = the Stefan-Boltzmann radiation constant = 1730×10^{-12} Btu per (hour) (square foot) (Fahrenheit degree absolute temperature to the fourth power).
F_A = the geometrical factor which is dimensionless and $\leqq 1$. This factor accounts for the shape and relative position of the two surfaces. The value of $F_A = 1$ may be used in the cases of large parallel planes, long concentric cylinders or smaller bodies in large enclosures.
F_E = the emissivity factor which is also dimensionless and $\leqq 1$. This factor accounts for the absorption and emission characteristics of the surfaces for the

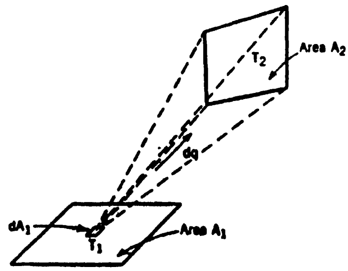


FIG. 3. RADIATION BETWEEN SURFACES

radiation which exists. Emissivities or absorptivities (ϵ) for many common surfaces, are given in Table 3. The value of F_E for large parallel planes, long concentric cylinders, or large enclosed bodies is $1 \div (1/\epsilon_1 + 1/\epsilon_2 - 1)$.

The radiation under black-body conditions, or for an emissivity of 1.0, is given in Table 4[22] for cold surfaces as low as −39 F to warmer surfaces as high as 139 F. Some net radiation exchange solutions for several common radiation systems are given in Table 5.

There are several methods by which the geometrical factors F_A can be determined. One method involves the use of a mechanical geometrical integrator (Reference 7). Photographic and other methods are given in References 8 and 9.

Equivalent Conductance for Radiation

Although Equation 3 is a suitable equation for describing radiant exchange, it is not convenient for computations where other modes of energy transfer are operative. For such cases, it is convenient to define an *equivalent conductance for radiation* by the equation:

$$q_r = h_r\, A\, (t_1 - t_2) \tag{4}$$

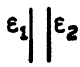
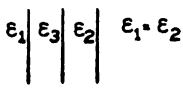
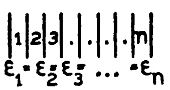
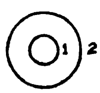
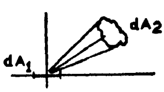
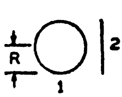
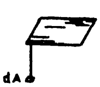
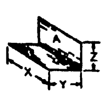
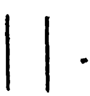
TABLE 4. HEAT TRANSMISSION BY RADIATION FOR BLACK-BODY CONDITIONS[a]

Expressed in Btu per (square foot) (hour)

TEMP F DEG	0	−1	−2	−3	−4	−5	−6	−7	−8	−9
−30	59.3	58.7	58.2	57.7	57.2	56.7	56.2	55.7	55.2	54.7
−20	65.2	64.7	64.1	63.5	62.9	62.3	61.7	61.1	60.5	59.9
−10	71.4	70.8	70.1	69.5	68.9	68.3	67.7	67.1	66.4	65.8
0	78.0	77.4	76.7	76.0	75.4	74.7	74.0	73.4	72.7	72.1
	0	+1	+2	+3	+4	+5	+6	+7	+8	+9
0	78.0	78.7	79.4	80.1	80.8	81.5	82.2	82.9	83.6	84.3
10	85.0	85.7	86.5	87.2	88.0	88.7	89.4	90.2	90.9	91.7
20	92.4	93.3	94.0	94.8	95.6	96.4	97.2	98.0	98.8	99.6
30	100	101	102	103	104	105	105	106	107	108
40	109	110	111	112	112	113	114	115	116	117
50	118	119	120	121	122	123	123	124	125	126
60	127	128	129	130	131	132	133	134	135	136
70	137	138	139	140	142	143	144	145	146	147
80	148	149	150	151	152	153	154	155	156	157
90	159	160	161	162	163	164	166	167	168	169
100	170	171	173	174	175	176	178	179	180	182
110	183	184	185	187	188	189	191	192	193	195
120	196	197	199	200	201	203	204	206	207	209
130	211	212	214	215	217	218	220	221	222	224

[a] *Example:* Radiation from walls of room at 32 F to surface at −25 F for effective emissivity of 0.95 = (102 − 62.3) 0.95 = 37.7 Btu per (square foot) (hour).

Table 5. Net Radiation Solutions

System	Solution	Remarks
 Two infinite parallel planes.	$\frac{q_r}{A} = \frac{1}{\frac{1}{\epsilon_1}+\frac{1}{\epsilon_2}-1}\sigma(T_1^4 - T_2^4)$	Considering interreflections. (Reference 5)
 One radiation shield between two infinite parallel planes.	$\frac{q_r}{A} = \left(\frac{1}{2}\right)\frac{1}{\frac{1}{\epsilon_1}+\frac{1}{\epsilon_2}-1}\sigma(T_1^4 - T_2^4)$	Considering interreflections. (Reference 5)
 n radiation shields between two infinite parallel planes.	$\frac{q_r}{A} = \frac{1}{n+1}\left(\frac{q_r}{A}\right)_0$ where $\left(\frac{q_r}{A}\right)_0$ is the net radiation exchange without the shields.	Considering interreflections. (Reference 5)
 Two concentric spheres or two infinitely long cylinders.	$\frac{q_r}{A_1} = \frac{1}{\frac{1}{\epsilon_1}+\frac{A_1}{A_2}\left(\frac{1}{\epsilon_2}-1\right)}\sigma(T_1^4 - T_2^4)$	Considering interreflections and diffuse surfaces. (Reference 5)
 Two areas dA_1 and dA_2	$\frac{dq_r}{dA_1} = \epsilon_1\epsilon_2\sigma F_A(T_1^4 - T_2^4)$	Surface diffuse, neglecting interreflection. (Reference 5)
 Tube of infinite length parallel to an infinite wall.	$\frac{q_r}{2\pi RN} = \left(\frac{1}{2}\right)\epsilon_1\epsilon_2\sigma(T_1^4 - T_2^4)$ where N is the length of cylinder from which q_r is exchanged	Neglecting interreflections. (Reference 5)
 Surfaces are perfect radiators. Surface element dA and rectangle above and parallel to it, with one corner of rectangle contained in normal to dA.	See Fig. 4	(Reference 4)
 Surfaces are perfect radiators. Adjacent rectangles in perpendicular planes.	See Fig. 5	(Reference 4)
 Surfaces are perfect radiators. Opposed parallel rectangles and disks of equal size.	See Fig. 6	(Reference 4)

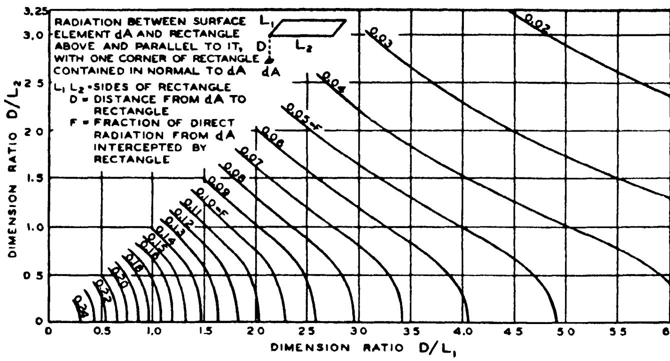


FIG. 4. GEOMETRICAL FACTOR F FOR DIRECT RADIATION BETWEEN AN ELEMENT dA AND A PARALLEL RECTANGLE*

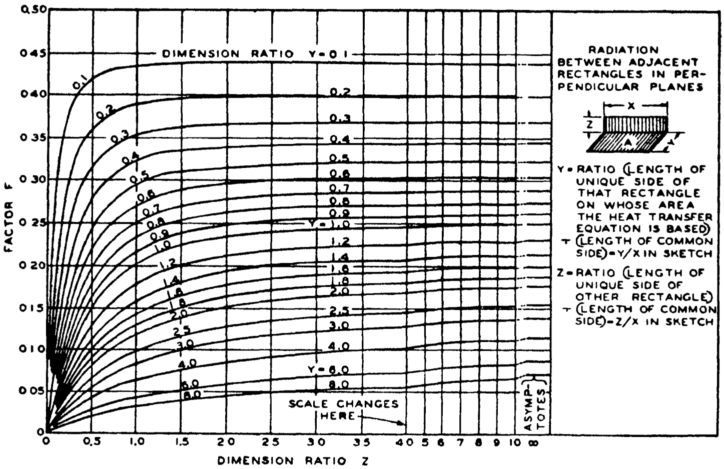


FIG. 5. GEOMETRICAL FACTOR F FOR DIRECT RADIATION BETWEEN ADJACENT RECTANGLES IN PERPENDICULAR PLANES*

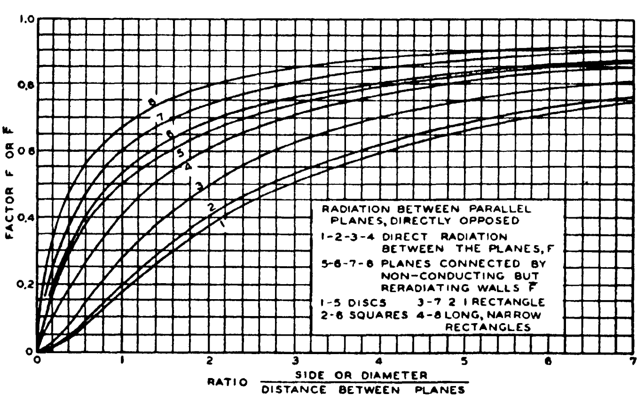


FIG. 6. GEOMETRICAL FACTOR F FOR DIRECT RADIATION BETWEEN OPPOSED PARALLEL RECTANGLE AND DISCS OF EQUAL SIZE*

* From Radiant Heat Transmission, by H. C. Hottel (*Mechanical Engineering*, July 1930, pp. 700 to 702).

The conductance h_r thus defined is a function of the shape-emissivity factor, as well as the temperatures of the radiator and receiver. Fig. 7 shows a plot of the equivalent conductance for two *black bodies* (*i.e.*, with emissivities equal to unity) which exchange energy only with one another.

Combined Convection and Radiation

It should be noted that the previous equations and tables give the heat transfer by convection and by radiation computed separately. In many practical cases it is desirable to treat convection and radiation as a single combined process, using a first-power equation:

$$q_{rc} = h_{rc} A \ (t_1 - t_2) \tag{5}$$

where q_{rc} is the total heat flow due to radiation and convection, in Btu per hour. Values of h_{rc}, the surface or film conductance for combined

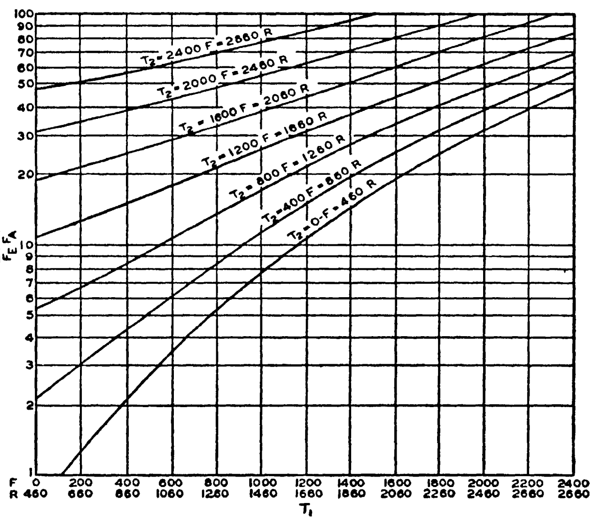


FIG. 7. EQUIVALENT CONDUCTANCE FOR RADIATION BETWEEN TWO BLACK BODIES EXCHANGING ENERGY ONLY WITH ONE ANOTHER

radiation and convection, are given in Chapter 9, (Table 1 and Fig. 4). Complete tables for the combined heat transfer of steam and hot water radiators, pipes, coverings, etc., will be found in the appropriate chapters.

HEAT-FLOW RESISTANCE

In most of the steady-state heat transfer problems encountered in air conditioning applications, more than one of the heat transfer mechanisms are effective, and the thermal current flows through several resistances in series or in parallel. In using the resistance concept, the calculations involved are analogous to the application of Ohm's Law in electricity, *viz.*, the heat flow or thermal current is directly proportional to the thermal potential or temperature difference, and inversely proportional to the thermal resistance:

$$q_{ro} = \frac{t_1 - t_2}{R} \tag{6}$$

Following the electrical analogy, when there is a thermal current flowing through several *resistances in series*, the resistances are additive:

$$R_T = R_1 + R_2 + R_3 + \cdots + R_n \tag{7}$$

Similarly, conductance is the reciprocal of resistance, and for heat flow through several *resistances in parallel*, the conductances are additive:

$$C_T = \frac{1}{R_T} = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} + \cdots + \frac{1}{R_n} \tag{8}$$

Practical Heat Transfer Problems

The use of these relations for resistance and conductance makes possible the solution of many practical heat transfer problems. As discussed in Chapters 9, 27 and 35, the practical analyses of heat transfer in building walls, in fin-tube coils and in pipe coverings, are usually computed by this method. The same resistance analysis may be applied to complicated

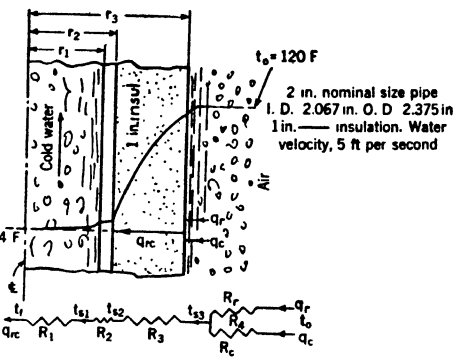


FIG. 8. HEAT TRANSFER CONDITIONS IN AN INSULATED COLD WATER LINE

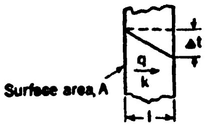
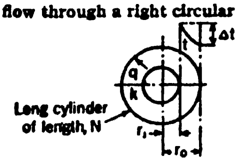
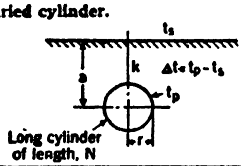
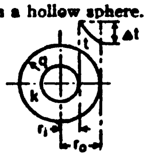
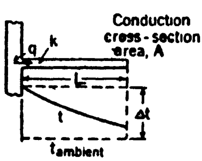
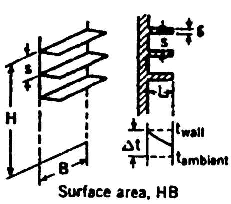
steady-state *conduction* problems. Table 6 gives the resistances in six common cases of steady-state conduction.

A complete analysis by the resistance method is well illustrated by considering the heat transfer from the air outside to the cold water inside of an insulated pipe. The temperature gradients and the nature of the resistance analysis are indicated by the two sketches of Fig. 8.

Since air is sensibly transparent to radiation, there will be some heat transfer by both radiation and convection to the outer insulation surface. The mechanisms act in parallel on the air side. The total transfer by radiation and convection then passes through the insulating layer and the pipe wall by thermal conduction, and thence by convection and radiation into main cold water streams. (Radiation is not significant on the water side as liquids are sensibly opaque to radiation, although water transmits energy in the visible region). The contact resistance between the insulation and the pipe wall is presumed to be equal to zero.

Referring to Fig. 8, the heat transferred for a given length N of pipe, q_{rc}, Btu per hour, may be thought of as flowing through the parallel resistances R_r and R_c, associated with the insulation surface radiation and convection transfer. Then the flow is through the resistance offered to thermal conduction by the insulation, R_3, through the pipe wall resistance,

TABLE 6. SOLUTIONS FOR SOME STEADY-STATE THERMAL CONDUCTION PROBLEMS[a, b]

No.	SYSTEM	Expressions for the resistance R entering into the equation: $q = \Delta t/R$ (Btu per hour)
1.	Flat wall or curved wall if curvature is small (wall thickness less than 0.1 of inside diameter). 	$R = \dfrac{L}{kA}$
2.	Radial flow through a right circular cylinder. 	$R = \dfrac{\log_e \frac{r_o}{r_i}}{2\pi kN}$ (See footnote c).
3.	The buried cylinder. 	$R = \dfrac{\log_e\left(\frac{a+\sqrt{a^2+r^2}}{r}\right)}{2\pi kN} = \dfrac{\cosh^{-1}\left(\frac{a}{r}\sqrt{1+\frac{r^2}{a^2}}\right)}{2\pi kN}$ For $\frac{a}{r} \geq 3$, a satisfactory approximation is: $R = \dfrac{\log_e \frac{2a}{r}}{2\pi kN} = \dfrac{\cosh^{-1}\frac{a}{r}}{2\pi kN}$
4.	Radial flow in a hollow sphere. 	$R = \dfrac{\frac{1}{r_i} - \frac{1}{r_o}}{4\pi k}$
5.	The straight fin or rod heated at one end. 	$R = \dfrac{m}{h_a p \tanh mL}$ (see footnotes d and e). For $ml > 2.3$, $\tanh mL \approx 1$; $m = \sqrt{h_a p/kA}$; A = conduction cross-section area. p = perimeter of cross-section A. h_a = unit conductance to the surroundings from the fin surface. k = thermal conductivity fin material. Δt = wall temperature—ambient temperature
6.	Finned surface of area HB. 	$R = \dfrac{(s+\delta)}{h_a\left(\frac{2}{m}\tanh ml + s\right)HB}$ $m = \sqrt{\dfrac{h_a p}{kA}} = \sqrt{\dfrac{2h_a}{k\delta}}$ Δt defined as in Case 5 above.

[a] The dimensions to be employed in these solutions are: length of dimension p, L, r = feet; units of k = Btu per (hour) (square foot) (Fahrenheit degree for *one foot thickness*); units of h, Btu per (hour) (square foot) (Fahrenheit degree); units of area, A = square feet.

[b] The thermal conductivity, k, in these solutions should be taken at the average material temperature.

[c] $\log_e x = 2.303 \log_{10} x$.

[d] This expression can also be employed as an approximation for tapered fins or of annular fins by employing average magnitudes of A and p.

[e] tanh is the hyperbolic tangent.

R_3, and into the water stream through the convection resistance, R_1. Note the analogy to the direct current electrical circuit problem. A temperature (potential) drop is required to overcome these resistances to the flow of thermal current. The total resistance to heat transfer, R_T, hour Fahrenheit degrees per Btu, is the summation of the individual resistances:

$$R_T = R_1 + R_2 + R_3 + R_4 \tag{9}$$

where the resultant parallel resistance R_4 is obtained from:

$$\frac{1}{R_4} = \frac{1}{R_r} + \frac{1}{R_c} \tag{10}$$

Provided the individual resistances may be evaluated, the total resistance can be obtained from this relation. Then the heat transfer for the length of pipe (N, ft) can be established by the relation:

$$q_{rc} \text{ (Btu per hour)} = \frac{(t_o - t_l)}{R_T} \tag{11}$$

For a unit length of the pipe the heat transfer rate is:

$$\frac{q_{rc}}{N} \text{ Btu per (hour) (foot)} = \frac{(t_o - t_l)}{R_T N} \tag{12}$$

The temperature drop, Δt, through an individual resistance may then be calculated from the relation:

$$\Delta t = R\, q_{rc} \tag{13}$$

where R is the resistance in question.

The problem is now reduced to one of evaluating the individual resistances of the system. This entails suitable manipulation of the rate Equations 1, 2 and 3 to produce expressions of the form:

$$q = \frac{\Delta t}{R} \tag{14}$$

where q is the heat transfer rate, and Δt is the potential drop or temperature difference through the resistance R. Table 6 lists such solutions for six different conduction systems. Table 2 in Chapter 9 and Table 1 of this chapter indicate the magnitudes of the thermal conductivities, k, to be employed in the expressions of Table 6, after dividing k by 12.

The solution applicable to the problem depicted in Fig. 8, for the calculation of R_2 and R_3, is case 2 in Table 6. Thus for a *1 ft length* of 2 in. nominal size pipe (I. D. = 2.067 in., O. D. = 2.375 in.) insulated with 1 in. of material having a conductivity of 0.025:

$$R_2 = \frac{\log_e \dfrac{1.188}{1.033}}{2\pi \times 26 \times 1} = 8.5 \times 10^{-4} \text{ (hr) (F deg) per Btu.}$$

$$R_3 = \frac{\log_e \dfrac{2.188}{1.188}}{2\pi \times 0.025 \times 1} = 3.9 \text{ (hr) (F deg) per Btu.}$$

The convection resistances to heat transfer from the pipe wall to the cold water, R_1, and from the air to the surface of the insulating material, R_c, are dependent on the flow conditions prevailing at these surfaces, and on the thermal properties of the fluids. These resistances are also directly dependent upon the temperature distribution, and for this reason it is necessary first, to guess on the basis of the problem statement, a temperature distribution upon which to base the initial calculations. Since the values of h_c for heat transfer between water and pipe walls are relatively high in this temperature range, it is logical to assume only a small temperature difference between the temperature of the fluid body and the temperature of the pipe wall. For the purpose of an initial guess, this temperature difference will be assumed to be 2 deg. On the other hand, h_c for heat transfer from air to a body is relatively small, and a higher temperature difference would be expected between these masses. The value initially assumed here will be 20 deg. In summary, the temperature distribution in the system is assumed as follows:

Fluid temperature = 34 F.
Inner pipe wall temperature = 36 F.
Outer insulation surface temperature = 100 F.
Ambient air temperature = 120 F.

With these assumptions and the problem statement, it is now possible to calculate values for the convective resistances. If it is found in the ultimate solution of the problem that the temperature distribution is different from that assumed, it will then be necessary to repeat the solution procedure.

If reference now be made to Table 2, it is found that Case 3 of this table is a system similar to that encountered in the convection between the water and the pipe wall. The equation for this case is the following:

$$h_c = 13.9(t_f)^{0.548}\frac{u_m^{0.8}}{D^{0.2}} \tag{15}$$

where

$$u_m = 5 \text{ fps}$$

$$D = \frac{2.067}{12} = 0.1725 \text{ ft}$$

$$t_f = \frac{34 + 36}{2} = 35\text{F}$$

$$h_c = \frac{13.9 \times 6.9 \times 3.62}{0.703} = 494 \text{ Btu per (hr) (sq ft) (F deg).}$$

This heat transfer rate is through the inner surface of the pipe and it is, therefore, this area that determines the resistance R_1.

$$A = \pi D = 0.542 \text{ sq ft per unit length of pipe,}$$

and therefore

$$R_1 = \frac{1}{h_c A} = \frac{1}{494 \times 0.542}$$

$$R_1 = 3.73 \times 10^{-3} \text{ (hr) (F deg) per Btu.}$$

R_3, and into the water stream through the convection resistance, R_1. Note the analogy to the direct current electrical circuit problem. A temperature (potential) drop is required to overcome these resistances to the flow of thermal current. The total resistance to heat transfer, R_T, hour Fahrenheit degrees per Btu, is the summation of the individual resistances:

$$R_T = R_1 + R_2 + R_3 + R_4 \tag{9}$$

where the resultant parallel resistance R_4 is obtained from:

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For a unit length of the pipe the heat transfer rate is:

$$\frac{q_{rc}}{N} \text{ Btu per (hour) (foot)} = \frac{(t_o - t_f)}{R_T N} \tag{12}$$

The temperature drop, Δt, through an individual resistance may then be calculated from the relation:

$$\Delta t = R\, q_{rc} \tag{13}$$

where R is the resistance in question.

The problem is now reduced to one of evaluating the individual resistances of the system. This entails suitable manipulation of the rate Equations 1, 2 and 3 to produce expressions of the form:

$$q = \frac{\Delta t}{R} \tag{14}$$

where q is the heat transfer rate, and Δt is the potential drop or temperature difference through the resistance R. Table 6 lists such solutions for six different conduction systems. Table 2 in Chapter 9 and Table 1 of this chapter indicate the magnitudes of the thermal conductivities, k, to be employed in the expressions of Table 6, after dividing k by 12.

The solution applicable to the problem depicted in Fig. 8, for the calculation of R_2 and R_3, is case 2 in Table 6. Thus for a *1 ft length* of 2 in. nominal size pipe (I. D. = 2.067 in., O. D. = 2.375 in.) insulated with 1 in. of material having a conductivity of 0.025:

$$R_2 = \frac{\log_e \frac{1.188}{1.033}}{2\pi \times 26 \times 1} = 8.5 \times 10^{-4} \text{ (hr) (F deg) per Btu.}$$

$$R_3 = \frac{\log_e \frac{2.188}{1.188}}{2\pi \times 0.025 \times 1} = 3.9 \text{ (hr) (F deg) per Btu.}$$

The convection resistances to heat transfer from the pipe wall to the cold water, R_1, and from the air to the surface of the insulating material, R_c, are dependent on the flow conditions prevailing at these surfaces, and on the thermal properties of the fluids. These resistances are also directly dependent upon the temperature distribution, and for this reason it is necessary first, to guess on the basis of the problem statement, a temperature distribution upon which to base the initial calculations. Since the values of h_c for heat transfer between water and pipe walls are relatively high in this temperature range, it is logical to assume only a small temperature difference between the temperature of the fluid body and the temperature of the pipe wall. For the purpose of an initial guess, this temperature difference will be assumed to be 2 deg. On the other hand, h_c for heat transfer from air to a body is relatively small, and a higher temperature difference would be expected between these masses. The value initially assumed here will be 20 deg. In summary, the temperature distribution in the system is assumed as follows:

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With these assumptions and the problem statement, it is now possible to calculate values for the convective resistances. If it is found in the ultimate solution of the problem that the temperature distribution is different from that assumed, it will then be necessary to repeat the solution procedure.

If reference now be made to Table 2, it is found that Case 3 of this table is a system similar to that encountered in the convection between the water and the pipe wall. The equation for this case is the following:

$$h_c = 13.9(t_f)^{0.543} \frac{u_m^{0.8}}{D^{0.2}} \tag{15}$$

where

$$u_m = 5 \text{ fps}$$

$$D = \frac{2.067}{12} = 0.1725 \text{ ft}$$

$$t_f = \frac{34 + 36}{2} = 35\text{F}$$

$$h_c = \frac{13.9 \times 6.9 \times 3.62}{0.703} = 494 \text{ Btu per (hr) (sq ft) (F deg).}$$

This heat transfer rate is through the inner surface of the pipe and it is, therefore, this area that determines the resistance R_1.

$$A = \pi D = 0.542 \text{ sq ft per unit length of pipe,}$$

and therefore

$$R_1 = \frac{1}{h_c A} = \frac{1}{494 \times 0.542}$$

$$R_1 = 3.73 \times 10^{-3} \text{ (hr) (F deg) per Btu.}$$

Case 11 of Table 2 fits the conditions of the problem if only free convection heating of the pipe is assumed. The equation in this case is as follows:

$$h_c = 0.271\left(\frac{P}{P_o}\right)^{0.5}\left(\frac{\Delta t}{D}\right)^{0.25} \tag{16}$$

where

$$\Delta t = 20 \text{ F}$$
$$D = 0.364 \text{ ft}$$
$$P = P_o = \text{one atmosphere}$$

therefore,

$$h_c = 0.271\left(\frac{20}{0.364}\right)^{0.25}$$

$$h_c = 0.737 \text{ Btu per (hr) (sq ft) (F deg).}$$

Using the surface area of the insulation, the value of the resistance per unit length is determined.

$$A = \pi\left(\frac{4.375}{12}\right) \times 1 = 1.14 \text{ sq ft}$$

$$R_c = \frac{1}{h_c A} = \frac{1}{0.737 \times 1.14}$$

$$R_c = 1.19 \text{ (hr) (F deg) per Btu.}$$

This result may not be deemed conservative inasmuch as the expression is for still air. If, however, the air is not still, but flows at approximately 5 mph or 7 fps, the heat transfer equation for forced convection would apply. This equation is Case 5 of Table 2.

$$h_{c(\text{average})} = 0.211(T_f)^{0.43}\frac{(u_\infty \rho)^{0.8}}{D^{0.4}} \tag{17}$$

$$T_f = \frac{100 + 120}{2} + 460 = 570 \text{ Rankine (Fahrenheit abstract)}$$

$$u_\infty = 7 \text{ fps}$$

$$\rho = 0.076\left(\frac{520}{570}\right) = 0.0694 \text{ lb per cu ft}$$

$$D = 0.364 \text{ ft}$$

$$h_{c(\text{average})} = \frac{0.211(570)^{0.43}(7 \times 0.0694)^{0.8}}{(0.364)^{0.4}}$$

$$h_{c(\text{average})} = 2.73 \text{ Btu per (hr) (sq ft) (F deg)}$$

and

$$R_c \text{ (Forced Convection)} = \frac{1}{h_c A} = \frac{1}{2.73 \times 1.14}$$

$$R_c = 0.321 \text{ (hr) (F deg) per Btu.}$$

The radiation resistance, R_r, which acts in parallel with the resistance just calculated, can be computed with the aid of Fig. 7. The pipe wall, assumed at 100 F *sees* the surroundings at 120 F. If these two temperatures are used with Fig. 7, a value for $\frac{h_r}{F_A F_E}$ is determined directly.

$$\frac{h_r}{F_A F_E} = 1.4 \text{ Btu per (hr) (F deg) (sq ft).}$$

The angle factor, F_A, is unity, and for an estimated surface emissivity of 0.95 (see Table 3), $F_E = 0.95$. Therefore,

$$h_r = 1.4\ F_A F_E = 1.4 \times 1 \times 0.95$$

$$h_r = 1.33 \text{ Btu per (hr) (F deg) (sq ft)}$$

and the radiation resistance, R_r, is then the following:

$$R_r = \frac{1}{h_r A} = \frac{1}{1.33 \times 1.14}$$

$$R_r = 0.659 \text{ (hr) (F deg) per Btu.}$$

The resultant resistance of R_c and R_r acting in parallel (see Fig. 8) can now be evaluated as:

$$\frac{1}{R_4} = \frac{1}{R_c} + \frac{1}{R_r} = \frac{1}{0.321} + \frac{1}{0.659} = 4.54 \text{ Btu per (hr) (F deg)}$$

$$R_4 = 0.216 \text{ (hr) (F deg) per Btu.}$$

The overall resistance, R_T, surroundings to cold water, is the sum of $R_1 + R_2 + R_3 + R_4 = 4.12$ (hr) (F deg) per Btu for 1 ft length of pipe. Note that the controlling resistances are R_3 and R_4, and that neglect of both R_1 and R_2 would not significantly influence the total resistance, R_T.

On the basis of this resistance calculation, the heat transfer from the surroundings to the cold water may be evaluated as:

$$\frac{q_{rc}}{N} = \frac{t_o - t_f}{R_T} = \frac{120 - 34}{4.12} = 20.8 \text{ Btu per (hr) (ft)}$$

or about 0.175 tons of refrigeration per 100 ft of pipe.

Since the calculation is based on a 1-ft pipe length:

$$q_{rc} = 20.8 \text{ Btu per hr.}$$

The temperature drops through the various resistances are now readily evaluated by Equation 14 as:

$$\Delta t = qR$$

$t_o - t_{s3}$ (air to insulation surface) $= qR_4 = 20.8 \times 0.216 = 4.49$ F deg

$t_{s3} - t_{s2}$ (through the insulation) $= qR_3 = 20.8 \times 3.9 = 81.2$ F deg

$t_{s2} - t_{s1}$ (through the pipe wall) $= qR_2 = 20.8 \times 8.5 \times 10^{-4} = 0.018$ F deg

$t_{s1} - t_f$ (pipe wall to cold water) $= qR_1 = 20.8 \times 3.73 \times 10^{-3} = 0.078$ F deg

This solution was obtained on the temperature distribution assumptions initially made. It is apparent that a better solution could be obtained if the whole problem were reiterated using the temperature distribution just calculated.

PERIODIC AND TRANSIENT HEAT FLOW

The foregoing data and examples dealt with steady-state heat transfer (not varying with time). In most practical heat transfer problems the heat flow depends upon time. Such cases can usually be divided into two classes: periodic and transient. Periodic heat transfer repeats periodically in time. Transient heat transfer exhibits no periodicity. Graphical, analytical and numerical methods are available for solving transient or periodic heat flow problems.[4,5,9,10,11] Graphical and numerical methods are the most versatile, and can be applied with minimum mathematical training.

A large number of analytical solutions for the case of heat conduction in variously shaped solids are available in the literature. Table 7 gives a

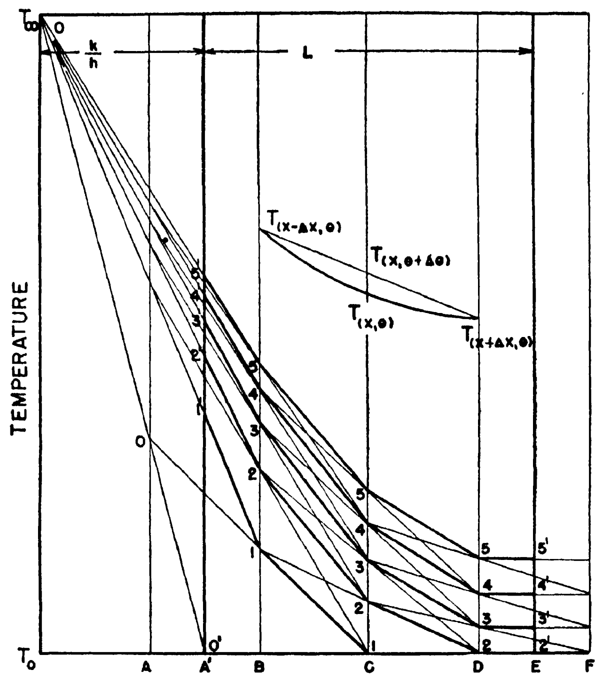


FIG. 9. EXAMPLE OF A GRAPHICAL SOLUTION TO A PROBLEM IN TRANSIENT HEAT CONDUCTION.

summary of the cases reported and tabulated. Many more analytical solutions are available in the form of infinite series,[10,11,17,18] but are not tabulated. Certain complex cases may be treated by combining the simple analytical solutions as discussed in Reference 16. (See also Reference 24).

Frequently, transient heat flow problems in one dimension have boundary conditions which make the problem difficult to treat analytically. In such cases, recourse may be made to a graphical method of solution sometimes called the Schmidt method.[4,9,18,19,20] This method will be briefly outlined for the case of transient heat flow in a slab insulated on one face, and suddenly exposed on the other face through a fixed thermal resistance to a higher temperature. The technique is general, however, and methods may be devised for any boundary conditions,[5,19] and also, for one dimensional (radial) heat flow in spheres and cylinders.[20,23]

TABLE 7. ANALYTICAL SOLUTIONS FOR HEAT CONDUCTION IN VARIOUSLY SHAPED SOLIDS

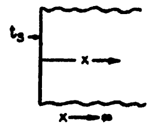
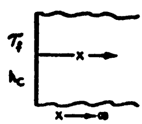
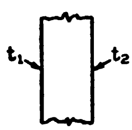
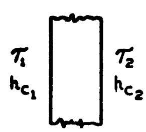
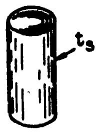
SHAPE OF SOLID	BOUNDARY CONDITIONS	DATA AVAILABLE IN GRAPHS
Semi-infinite. 	Surface temperature suddenly changed.	Temperature distribution in solid as a function of time. References: (4) p. 37, (5) p. V-28; (9) p. 254; (12) p. 46.
		Heat flow from surface as a function of time. References: (9) pp. 256, 267; (12) p. 47.
	A steady flow of heat is suddenly applied to the surface.	Temperature distribution as a function of time. Reference: (9) p. 257.
	The surface temperature has been varying sinusoidally with time for a long time.	Temperature distribution as a function of time. Reference: (9) p. 296.
		Heat flow from surface as a function of time. Reference: (9) p. 296.
Semi-infinite with fluid at free surface. 	The temperature of the fluid in contact with the surface has a sudden change in temperature. (The surface conductance is constant).	Temperature distribution as a function of time. References: (4) p. 37; (5) pp. V-45, 46, 47.
	The temperature of the fluid in contact with the surface has been varying sinusoidally with time for a long time. (The surface conductance is constant).	Temperature distribution as a function of time. Reference: (9) p. 298.
		Heat flow from the surface as a function of time. Reference: (9) p. 298
Slab. 	The temperatures t_1 and t_2 are suddenly changed from the initial uniform slab temperature to a new temperature. (The case where the surface on one side is insulated is treated by taking the case of a slab of twice the given thickness since the midplane has no heat flow due to symmetry).	Temperature distribution as a function of time. References: (5) p. V-12; (9) p. 265.
	The temperature t_1 and t_2 suddenly begin to increase as linear functions of time. The slab is initially at uniform temperature. (The case where one surface is insulated against heat flow is treated as noted above).	Temperature distribution as a function of time. Reference: (9) p. 268.
	The temperature at both surfaces has been varying sinusoidally for a long time.	Temperature distribution as a function of time. Reference: (9) p. 300.
		Heat flow from the surface. Reference (9) p. 303.
Slab immersed in a fluid with constant conductance between fluid and slab surface. 	The temperature of the fluid is suddenly changed from the initial uniform slab temperature. (If one surface is insulated against heat flow, see above).	Temperature distribution as a function of time. References: (4) pp. 32, 33, 34, 35; (5) pp. V-9, 10, 35, 42; (9) pp. 274, 284; (12) p. 106.
		Heat flow from the surface as a function of time. References: (5) p. V-10; (9) p. 274; (12) p. 107.
	The temperature of the fluid at one surface varies as a periodic function of time while the temperature of the fluid at the other surface is constant. The conductances need not be the same on both sides. (The variations in temperature are expressible as a Fourier series).	Temperature distribution as a function of time. Reference 13.
		Heat flow at the surface as a function of time. Reference 13.

TABLE 7. ANALYTICAL SOLUTIONS FOR HEAT CONDUCTION IN VARIOUSLY SHAPED SOLIDS (*Concluded*).

SHAPE OF SOLID	BOUNDARY CONDITIONS	DATA AVAILABLE IN GRAPHS
Cylinder of infinite axial dimension 	The surface temperature is suddenly changed from the initial (uniform) temperature.	Temperature distribution as a function of time. Reference: (9) p. 265.
		Heat flow from surface as a function of time. Multiply temperature difference between surface and fluid by surface conductance.
	The surface temperature suddenly begins to increase linearly with time.	Temperature distribution as a function of time. Reference: (9) p. 269.
Cylinder of infinite axial dimension immersed in a fluid. 	The surrounding fluid suddenly changes from the initial (uniform) temperature of the cylinder.	Temperature distribution as a function of time. References: (4) p. 36; (5) pp. V-16, V-35, V-43, V-48; (9) pp. 278, 286; (14).
		Heat flow from the surface as a function of time. References. (5) p. V-16; (9) p. 278.
	The temperature of the surrounding fluid changes sinusoidally.	Temperature distribution as a function of time. Reference: (5) p. VI-34.
		Heat flow from the surface as a function of time. Reference: (5) p. VI-36.
Sphere 	The temperature of the surface is suddenly changed from the initial uniform temperature.	Temperature distribution as a function of time. References: (5) p. V-23; (9) pp. 264 265.
	The temperature at the surface suddenly begins to change as a linear function of time.	Temperature distribution as a function of time. Reference: (9) p. 269.
Sphere immersed in fluid. 	The temperature of the surrounding fluid suddenly changes from the initial uniform sphere temperature.	Temperature distribution as a function of time. References: (4) p. 36; (5) pp. V-21, V-35, V-44; (9) pp. 281, 282; (4).
		Heat flow as a function of time. References: (5) p. V-21; (9) p. 281.
Rectangular bar of infinite length.	Any of the above noted boundary conditions for a slab.	Temperature distribution as a function of time. Combine solutions as indicated in Refs. 15 and 16.
Parallelopiped (rectangular).	Any of the above noted boundary conditions for a slab.	Temperature distribution as a function of time. Combine solutions as indicated in Refs. 15 and 16.
Cylinder of finite length.	Any of the boundary conditions given above for a cylinder and a slab.	Temperature distribution as a function of time. Combine solutions as indicated in Refs. 15 and 16
Hollow cylinder of infinite exterior radius.	The temperature of the surface suddenly changes from the initial (uniform) temperature.	Temperature distribution as a function of time. Combine solutions as indicated in Refs. 15 and 16.
		Heat flow at the surface as a function of time. Reference: (9) p. 267.

Consider the slab to be divided, as shown in Fig. 9, by n equidistant planes parallel to the slab surface and a distance Δx apart. Let the temperature of the slab at any plane and any time (θ) be denoted by $T_{(x,\theta)}$. Then the temperature of the slab at the two adjacent planes at the same time will be denoted as $T_{(x+\Delta x,\theta)}$ and $T_{(x-\Delta x,\theta)}$. In a similar manner the temperature of the x plane at a time $\Delta\theta$ later will be $T_{(x,\theta+\theta\Delta)}$.

In accordance with this nomenclature, the temperature at any plane x and time $\theta + \Delta\theta$ is given as

$$T_{(x,\theta+\Delta\theta)} = \frac{T_{(x+\Delta x,\theta)} + T_{(x-\Delta x,\theta)}}{2} \tag{18}$$

which may be interpreted as follows. The temperature of the slab at any plane, x, and any time, θ, is equal to the average temperature of the two adjacent planes obtained at the time $(\theta - \Delta\theta)$.

The time interval $\Delta\theta$ is determined by the equation

$$\Delta\theta = \frac{2\Delta x^2}{a} \tag{19}$$

Omitting the graphical construction at the slab boundaries, reference to Fig. 9 demonstrates the graphical method by means of which the temperature at each plane is determined at successive intervals of time in accordance with Equation 18.

For the problem stated, the boundary condition at the insulated surface is specified by the equation

$$\frac{\partial T}{\partial x} = 0$$

and at the uninsulated face by the equation

$$h(T_\infty - T) = -k\frac{\partial T}{\partial x}$$

In terms of finite differences these two equations (employing nomenclature established by Fig. 9) become

$$\frac{T_F - T_E}{\Delta x} = 0 \quad \text{or} \quad T_F = T_E \quad \text{at} \quad x = L$$

and

$$h(T_\infty - T_{A'}) = -k\frac{(T_B - T_A)}{\Delta x} \quad \text{at} \quad x = 0$$

or

$$\frac{T_\infty - T_{A'}}{k/h} = -\frac{T_A - T_B}{\Delta x}$$

The details of the graphical construction are best obtained by inspection of Fig. 9. Note that the line (0,0,0′) used to initiate the graphical construction, is the only one drawn to the slab boundary A'. The numbered points indicate temperatures at the sub-slab boundaries at 1,2,3, etc., time intervals ($\Delta\theta$) after the slab is exposed to the high temperature.

For transient heat flow in two dimensions, and also, for steady state conduction, numerical methods of solution are available in the literature.[5,9,11,18] These numerical methods are applicable to three dimensional problems, although the calculations involved normally become too tedious for most applications of the method. An additional technique of solution for one and two dimensional problems in transient conduction results from the analogy of electrical resistance-capacitance networks to thermal systems.[21]

For two dimensional problems in steady state conduction, additional techniques of solution are found in flux plotting,[5,18] and in the use of a potential tank.[5,18] These methods are of most direct use in problems in which the boundaries of the two dimensional shape are made up of isothermal and adiabatic surfaces.

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CHAPTER 6

PHYSIOLOGICAL PRINCIPLES

Chemical Vitiation of Air, Physical Impurities in Air, Thermal Interchanges Between the Body and Its Environment, High Temperature Hazards, Acclimatization, Upper Limits of Heat for Men at Work, Application of Physiologic Principles to Air Conditioning Problems, Effective Temperature Index and Comfort Zones

VENTILATION is defined in part as *the process of supplying air to, or removing air from, any space by natural or mechanical means.* The word in itself implies quantity, but air must be of the proper quality also. The term *air conditioning* in its broadest sense implies control of any or all of the physical or chemical qualities of the air. The A.S.H.V.E. Code of Minimum Requirements for Comfort Air Conditioning[1] defines it "as the process by which simultaneously the temperature, moisture content, movement and quality of the air in enclosed spaces intended for human occupancy may be maintained within required limits. If an installation cannot perform all of these functions, it shall be designated by a name that describes only the function or functions performed."

CHEMICAL VITIATION OF AIR

People living indoors bring about certain physical and chemical changes in the air about them. The oxygen content of the air diminishes and the carbon dioxide increases, but these changes are too slight to be significant except in air tight spaces as in submarines. Organic matter which is usually perceived as odors, comes from the body or clothes. Moisture and heat are given off by the body. There is no evidence of any toxic volatile material given off by man to the ambient air. Stale air may be offensive because of odors and may induce loss of appetite and loss of energy. Objectionable body odors have the same effects. These reasons, whether esthetic or physiological, usually make it desirable in the design of air conditioning systems to provide for the elimination or control of odors arising from occupancy, cooking, or other sources. This may be accomplished by introducing odor-free air in sufficient quantities to reduce odor concentrations by dilution to a level which is not objectionable. Odor-free air may be outdoor air or air which has been cleared of odors by sorption, washing, or other appropriate means.

In the case of vitiation by a few hazardous gases such as carbon monoxide from heating, cooking, and certain industrial processes, no satisfactory chemical treatment for the elimination of the impurity has been found. The only satisfactory solution is elimination at the source by local exhaust ventilation; or, if this is impossible, reduction to a safe concentration by dilution. (See Chapter 8.) In the case of contamination by other matter, including volatile vapors and gases, chemical treatment for the removal or reduction of the impurities has been made available through air cleaning methods, which are discussed in Chapter 33.

When the only source of contamination is the human occupant, and overheating is not a problem, the minimum quantity of outdoor air needed appears to be that required to remove objectionable body odors, or tobacco smoke. The concentration of body odor in a room, in turn, depends maintains the body temperature well above that of the surrounding air in a cool or cold environment. At the same time, heat is constantly lost from the body by radiation, convection and evaporation. Since, under ordinary conditions, the body temperature is maintained at its normal level of about 98.6 F, the heat production must be balanced by the heat loss. During work, the body temperature may rise; in fact, afternoon temperatures of normal persons average 1 deg above the resting value of the morning whether working or not.

The fundamental thermodynamic processes concerned in heat interchanges between the body and its environment may be described by the equation:

$$M = \pm S + E \pm R \pm C \tag{1}$$

where

M = rate of metabolism, heat produced within the body.
S = rate of storage, change in intrinsic body heat.
E = rate of evaporative heat loss.
R = rate of radiative heat loss or gain.
C = rate of convective heat loss or gain.

The rate of metabolism, M, is always positive. The storage, S, may be either positive or negative, depending upon whether heat is being stored or depleted owing to a rise or fall in body temperature. Under ordinary circumstances (when the dew-point of the air is below the body surface temperature) the evaporation loss, E, is always positive; that is, heat from metabolism supplies this loss. R and C are positive when body surface temperature is above that of walls and air, and negative when it is below.

DuBois,[7] after careful calorimeter studies on a fasting, nude man, plotted the partition of body heat loss and heat production as a function of temperature. Fig. 1 shows some disparity between heat production and heat loss. This disparity is S in Equation 1. In the central range of the experiments S was quite low and no increase in heat loss by vaporization was apparent.

Within the range of 81–86 F air temperature, with still air, there is, for a resting nude man, a point at which his body has to take no particular action to maintain its heat balance. If he is clothed, or if he is active, this point will naturally lie at a lower level. At this point, which may be termed the *neutral point* for that individual, conditions are neither too hot nor too cold. If, through a fall in air temperature, or a rise in air movement, the rate of heat loss from the skin to the environment is increased, then the body must do something to counteract this heat loss. Over a certain range, the body can achieve this by decreasing the flow of blood through the skin. This will result in some cooling of the skin and subjacent tissues, but the temperature of the deep tissues will be preserved. The range of external conditions over which this may be achieved, may be termed the *zone of vaso-motor regulation against cold.* Beyond this range, the temperature of the superficial tissues will fall still further, and that of the deep tissues will fall as well, unless some other steps are taken. The body normally does react; it increases heat production by increasing muscular tension, by shivering, or by spontaneous increase in activity.[8] As long as these are adequate to meet the increased rate of heat loss to the environment, a fall in deep body temperature may be prevented. Such conditions may be said to lie in the *zone of metabolic regulation against cold.* Beyond this point, the body enters the *zone of inevitable body cooling.* Once body temperature starts to fall, man is headed for disaster.

It will be seen that, in man, deep body temperature is preserved over an important range of cold external conditions, at the expense of (1) a fall in the temperature of the peripheral tissues, and (2) an increased expenditure of energy. As regards the first of these, the farther away superficial tissue lies from the central body mass, the more readily will its temperature fall.

On the hot side of the neutral point, there exists a *zone of vaso-motor regulation against heat,* corresponding to that against cold. The blood flow through the skin is increased when the opportunities for heat loss to the environment are restricted.[9] This increase in blood flow may double the conductance of the superficial tissues over that characteristic of the neutral point, and the temperature of the skin surface may rise until it is only three degrees below that of the deep tissues. If this increase in blood flow is

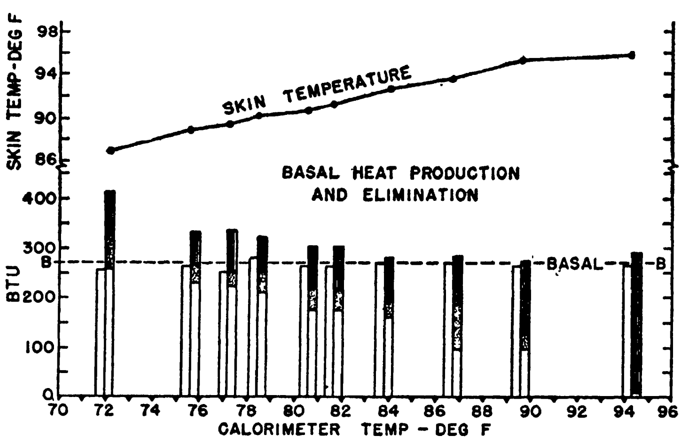

FIG. 1. HEAT LOSS FROM HUMAN BEING BY EVAPORATION, RADIATION, AND CONVECTION[a]

[a] Normal control, naked, in calorimeter at temperatures from 72.8 to 94.1 F. First column in each experiment represents heat production as determined by indirect calorimetry, the second column, hea t elimination The portion marked with vertical lines represents vaporization; the dotted area, convection; the unmarked area, radiation. The skin temperature represents the average reading of 18 spots on the surface.

unable to balance the restriction in heat loss, the body has entered the next zone. Once again, the normal body takes steps to prevent a change in its deep temperature; but they are not the counterpart of the steps taken under similar circumstances on the cold side. There is, in fact, very little change in heat production, beyond that resulting from a disinclination for exertion. The second line of defense, on the hot side, is a new and powerful method of promoting heat loss—the provision of water, by the operation of the sweat glands, for evaporative cooling. As long as evaporation is adequate to restore the desired heat loss, the body is in the *zone of evaporative regulation against heat.* When this ceases to be adequate, the body is in the *zone of inevitable body heating.* The body enjoys a little more latitude in this zone than it did in the corresponding zone on the cold side, but when the deep temperature rises more than 4 deg F, it loses its efficiency. All factors which affect the evaporation of water from the skin affect heat regulation in the zone of evaporative regulation. Atmospheric vapor pressure and air motion are most important. With dry-bulb temperature above body temperature, air motion facilitates evaporative heat loss by removing hot humid air from contact with the skin and replacing it with relatively drier air.

Heat regulation in man requires an intact set of sensory nerves, a normal sympathetic nerve supply to sweat glands and blood vessels, a great many sweat glands, and a circulatory system capable of carrying heat from muscles and viscera to the skin by circulation of the blood.

Some of the phenomena of body temperature control are shown graphically in Fig. 2. The dotted curves, from a study at the John B. Pierce Laboratory of Hygiene,[10] are for subjects lightly clothed in a semi-reclining position, and give the relation between the dry-bulb temperature of the environment (with about 45 percent relative humidity) and the metabolic

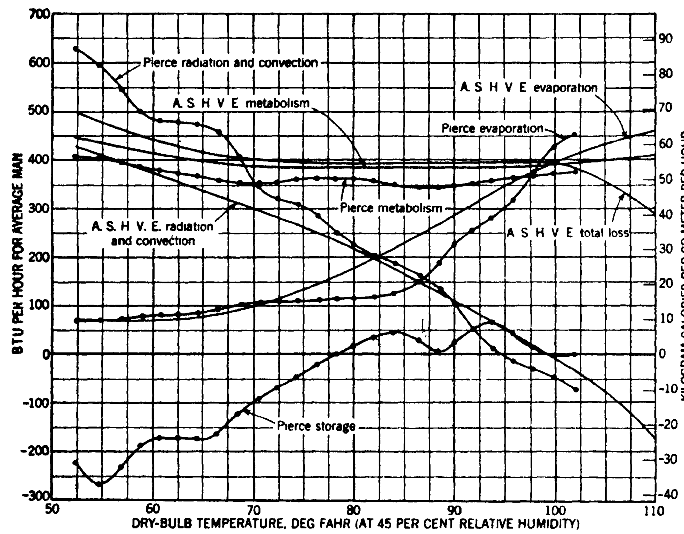

FIG. 2. RELATION BETWEEN METABOLISM, STORAGE, EVAPORATION, RADIATION PLUS CONVECTION, AND TEMPERATURE FOR THE CLOTHED SUBJECT

rate (heat production), the rate of heat dissipation by radiation and convection combined, and the latent heat loss due to evaporation from the skin and the respiratory tract. The smooth line curves from the work of the A.S.H.V.E. Research Laboratory[11] give the same relationships for healthy, male subjects (18 to 24 years of age), seated at rest and dressed in customary winter indoor clothing. The Pierce Laboratory data for the semi-reclining subjects also include the rate of heat storage (either positive or negative) due to a rise or fall in body temperature. For the normally clothed subjects, a curve gives the total heat loss (that is, the sum of the radiation, convection and evaporative losses). Here, storage is given by the difference between the metabolism and total heat loss.

The small difference between the metabolic rates for the two groups of subjects may be accounted for by difference in activity. Heat exchange between the body and the environment by radiation and convection is greater for the lightly clothed subject, both for cool conditions where there is excessive heat loss, and for very warm conditions where there is transfer of heat from the atmosphere to the body. The two curves for evaporative loss serve to show how physiological control uses evaporation of sweat to maintain equilibrium at high temperatures. Below 75 F, for the normally clothed subject, and below 85 F for the lightly clothed subject, evaporation loss is minimal and constant. Burch[12] has shown that this insensible perspiration reflects the permeability of the skin to the moisture of the body. Above these temperatures, control is obtained by the availability of sweat for evaporation. The difference in the curves above 75 F is probably largely determined by the difference in clothing and activity.

In the zone of evaporative heat regulation, air movement facilitates heat loss if the temperature of the air is not above that of the skin.[13] Under hot, dry conditions air movement may be of little advantage, or even of disadvantage, if it increases the addition of heat to the skin by conduction more than it promotes the loss of heat from the skin by evaporation.

TABLE 2. PHYSIOLOGICAL RESPONSES TO HEAT OF MEN AT REST AND AT WORK[a]

Effective Temp	Actual Cheek Temp (Fahr Deg)	Men at Rest			Men at Work 90,000 ft-lb of Work per Hour			
		Rise in Rectal Temp (Fahr Deg per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Weight by Perspiration (Lb per Hr)	Total Work Accomplished (Ft-Lb)	Rise in Body Temp (Fahr Deg per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Wt by Perspiration (Lb per Hr)
60	......	......	......	....	225,000	0.0	6	0.5
70	......	0.0	0	0.2	225,000	0.1	7	0.6
80	96.1	0.0	0	0.3	209,000	0.3	11	0.8
85	96.6	0.1	1	0.4	190,000	0.6	17	1.1
90	97.0	0.3	4	0.5	153,000	1.2	31	1.5
95	97.6	0.9	15	0.9	102,000	2.3	61	2.0
100	99.6	2.2	40	1.7	67,000	4.0[b]	103[b]	2.7[b]
105	104.7	4.0	83	2.7	49,000	6.0[b]	158[b]	3.5[b]
110	....	5.9[b]	137[b]	4.0[b]	37,000	8.5[b]	237[b]	4.4[b]

[a] Data by A.S.H.V.E. Research Laboratory.
[b] Computed value from exposures lasting less than one hour.

HIGH TEMPERATURE HAZARDS

Studies at the A.S.H.V.E. Research Laboratory[14] and elsewhere during the past two decades have made available much information dealing with the physiological effects of hot atmospheres on workers, and means of alleviating the distress and hazards associated therewith. Table 2 gives some of the physiological responses of men, at rest and at work, to hot environments. Frequent and continued exposure of workers to hot environments results in physiological derangement affecting the leucocyte count of the blood, and other factors dealing with man's mechanism of defense against infection.

Wherever S (Equation 1) becomes strongly positive and body temperature rises progressively, men will continue to work until body temperature reaches 103 F. When these body temperatures are exceeded, men work with declining efficiency and may be subject to heat stroke.

Heat exhaustion is a circulatory failure in which the venous return to the heart is reduced so that fainting results.[15] Early symptoms of heat exhaustion may include fatigue, headache, dizziness when erect, loss of appetite, nausea, abdominal distress, vomiting, shortness of breath, flushing of face and neck, pulse rate above 150, glazed eyes, and mental dis-

TABLE 3. UPPER LIMITS OF ENVIRONMENTAL CONDITIONS FOR ACCLIMATIZED, HEALTHY, YOUNG MEN IN MILITARY SERVICE

Environment	Reactions at the end of 4 hr	
	Rectal Temp F	Pulse rate
Relatively Easy	Below 101	Below 130
Difficult	101 to 102	130 to 145
Impossible	Above 102	Over 145

turbances as apathy, poor judgment, and irritability which usually precede fainting (syncope). Recovery is usually prompt when the man is removed to a cool place and kept lying down for a time, unless he has some other illness such as heart disease.

Heat cramps are painful muscle spasms in extremities, back and abdomen due, at least in part, to excessive loss of salt in sweating. Formerly common in hot industries, this manifestation of illness due to heat is now greatly reduced by drinking water containing 0.1 percent salt, or by proper use of salt tablets. Heat cramps are readily alleviated by administration of salt solution intravenously.

Heat stroke is a serious effect of exposure to great heat. The body temperature climbs rapidly to excessive levels often above 105 F when for unknown reasons free sweating suddenly stops. At such high temperatures, coma appears and death may be imminent. Emergency measures are required to reduce the excessive body temperature by cooling quickly to avoid irreparable damage to the brain.[16]

The deleterious physiologic effects of high temperatures exert a powerful influence upon physical activity, accidents, sickness and mortality. Both laboratory and field data show that physical work in warm atmospheres is a great effort, and that production falls progressively as the temperature rises.

ACCLIMATIZATION

When men move to deserts or to jungles some adaptation to the climate takes place. If work is gradually increased day by day, and if the men can get plenty of water and salt, and can sleep each night, acclimatization may be complete in 7 to 10 days. The acclimatized man works with a lower heart rate, lower skin and rectal temperature, and more stable blood pressure than when unacclimatized. The process of acclimatization requires work in the heat.[17] During the recent war, white troops lived and did hard physical work for long periods in tropical conditions when disease hazards were controlled.

In recent tests made at the A.S.H.V.E. Research Laboratory,[18] subjects were required to perform light work under very hot conditions for a 4-hr period each day. It was found that the ability of a new subject to endure these conditions showed daily improvement for a period of at least 2 weeks. However, after acclimatization was completed, a recess of several days had no effect on the endurance of the subject. Individuals differ widely in their capacity to acclimatize. Acclimatized men lose most of these improvements in a few weeks of temperate climate, even though they are vigorously active. In the course of acclimatization, the sweat glands come to secrete fluid less rich in salt.[19] For all except those carrying out really hard work in hot, dry atmospheres, this effects an important saving in

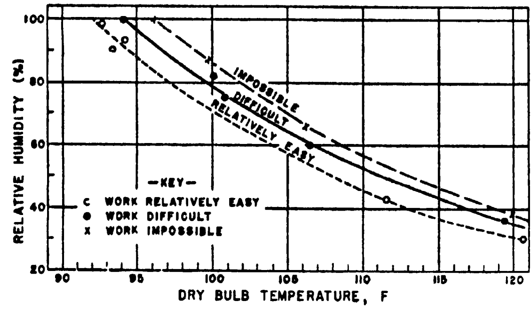

FIG. 3. HEAT ENDURANCE OF ACCLIMATIZED SUBJECTS WORKING AT A SPECIFIC RATE[20]

salt loss, and makes all the difference between being exposed and not being exposed to the risk of heat cramps.

The adaptive level changes somewhat with the season. There are also marked differences between the sexes. In the cold zone the thickness of thermal insulating tissues of women is almost double that of men, although the sensory responses to cold are similar. In the hot zone, the threshold of sweating is higher for women. The thickness and insulating value of the clothing worn are also important factors in the determination of the comfort level.

UPPER LIMITS OF HEAT FOR MEN AT WORK

In very hot conditions humidity is the limiting factor, and the wet-bulb temperature assumes great importance. In 1905 Haldane recognized that 88 F wet-bulb was the limit of endurance for coal miners, and later observers have concurred.

A study was made at the Armored Medical Research Laboratory[20] to determine the upper limits of environmental conditions under which a man can perform certain work. Thirteen enlisted men, thoroughly acclimatized to the hot conditions, served as subjects. During each test, the subjects were required to march for 4 hr at the rate of 3 mph, carrying

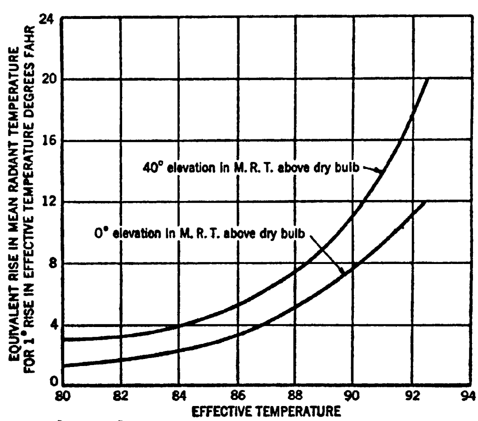

FIG. 4. EFFECT OF MRT ELEVATION IN TERMS OF EFFECTIVE TEMPERATURE

20 lb packs under a wide range of environmental conditions which were rated as relatively easy, difficult, and impossible, on the basis of the physiological reactions of the subjects at the end of the 4-hr period as shown in Table 3 and Fig. 3.

Recognition of the need of air conditioning for workers in hot industries is growing rapidly. The choice of the type of system to be used in any given instance, must be determined by the air conditioning engineer after a study of conditions. In some hot industries where few workers are

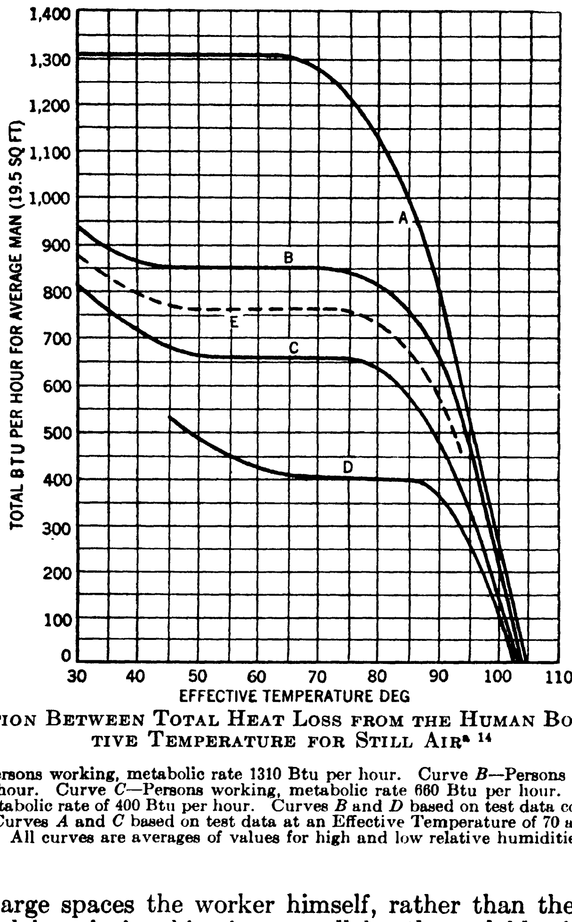

FIG. 5. RELATION BETWEEN TOTAL HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE FOR STILL AIR[a] [14]

[a] Curve *A*—Persons working, metabolic rate 1310 Btu per hour. Curve *B*—Persons working, metabolic rate 850 Btu per hour. Curve *C*—Persons working, metabolic rate 660 Btu per hour. Curve *D*—Persons seated at rest, metabolic rate of 400 Btu per hour. Curves *B* and *D* based on test data covering a wide temperature range. Curves *A* and *C* based on test data at an Effective Temperature of 70 and extrapolation of Curves *B* and *D*. All curves are averages of values for high and low relative humidities; variation due to humidity is small.

engaged in large spaces the worker himself, rather than the atmosphere, can be cooled by placing him in a small booth, and blowing cooled air over him, or by circulating cooled air through a loose-fitting suit.[21]

The A.S.H.V.E. Laboratory has studied the effects of walls of higher temperature than the air.[18] The findings are in part shown in Fig. 4. It will be seen that the importance of mean radiant temperature, as compared with that of the effective temperature, decreases as the effective temperature rises; and also, to a certain extent, as the mean radiant temperature itself rises. The lower of the two curves relates to conditions in which the MRT was kept approximately at the level of the DBT. If this curve is followed, it will be seen that, at 80 ET, a little more than 1 deg rise in MRT produces the same effect as 1 deg rise in ET; but when the ET is 92 deg, it takes a rise of 11 F deg in the MRT to produce the same effect as 1 deg rise in ET. The upper curve relates to conditions in which the MRT was kept about 40 F deg higher than the DBT. It will be seen that under these conditions, a rise in MRT is less effective, even at low values of ET; and that it loses its relative effectiveness more rapidly as the ET rises. It should not be assumed, however, that MRT does not matter much. All that these comparisons indicate is that unit rise in MRT becomes less important *as compared with unit rise in ET*, as conditions get hotter. This may be due more to a growing importance of unit rise in ET than to a diminishing importance of unit rise in MRT. Under ordinary *still air* conditions the effects of air temperature and MRT appear to be interdependent. Various authorities give 0.3 to 1 deg increase of room temperature to compensate for 1 deg depression of the MRT.

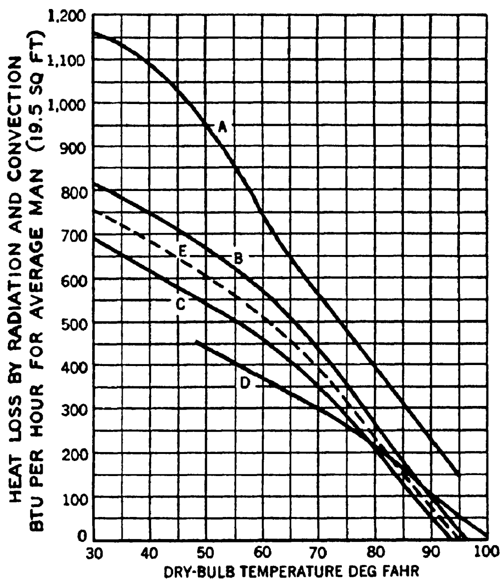

FIG. 6. RELATION BETWEEN RADIATION AND CONVECTION LOSS FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AIR[a] [14]

[a] Loc. Cit. See footnote a, Fig. 5.

APPLICATION OF PHYSIOLOGIC PRINCIPLES TO AIR CONDITIONING PROBLEMS

In order to estimate cooling loads in occupied spaces it is necessary to know the metabolic rate (heat production) of man. This has been studied extensively, and found to remain relatively constant per unit of body surface area in a subject fasting and resting quietly after a good night's sleep. The rate is high in children, and diminishes gradually with age; it increases in certain diseases and in the presence of fever. The metabolic rate is somewhat lower in women. Heat production goes up sharply with work and varies widely in different persons doing the same work. Figs. 5, 6, and 7 and Table 28 of Chapter 12 give sufficient basic data for estimating heat production and heat loss under various conditions.

EFFECTIVE TEMPERATURE INDEX AND COMFORT ZONES

There is no precise physiologic observation by which comfort can be evaluated. Mean skin temperature offers some promise. The zone of thermal neutrality differs with clothing, season, activity, and all the other factors controlling heat production (Table 4). The comfort zone is very similar to the zone of thermal neutrality.

Sensations of warmth or cold depend not only on the temperature of the surrounding air as registered by a dry-bulb thermometer, but also upon the temperature indicated by a wet-bulb thermometer, upon air movement, and upon radiation effects. Dry air at a relatively high temperature may feel cooler than air of lower temperature with a high moisture content. Air motion makes any moderate condition feel cooler. Radiation to cold or from warm surfaces is another important factor under certain conditions affecting the comfort reaction of the individual.

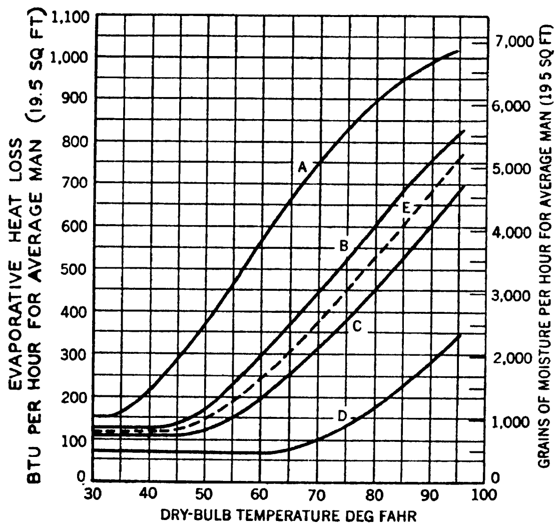

FIG. 7. EVAPORATIVE HEAT AND MOISTURE LOSS FROM THE HUMAN BODY IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS[a] [14]

[a] Loc. Cit. See footnote a, Fig. 5.

Combinations of temperature, humidity, and air movement which induce the same feeling of warmth are called thermo-equivalent conditions. A series of studies[22] at the A.S.H.V.E. Research Laboratory established the equivalent conditions for practical use. This scale of thermo-equivalent conditions not only indicates the sensation of warmth, but also to a considerable degree determines the physiological *effects* on the body induced by heat or cold. For this reason, it is called the *effective temperature* scale or index, and it denotes sensory heat level.

Effective temperature is an empirically determined index of the degree of warmth perceived on exposure to different combinations of temperature, humidity, and air movement. It was determined by trained subjects who compared the relative warmth of various air conditions in two adjoining conditioned rooms by passing back and forth from one room to the other.

The numerical value of the index for any given air conditions is fixed by the temperature of slowly moving (15 to 25 fpm air movement) saturated air which induces a like sensation of warmth or cold. Thus, any air condition has an effective temperature of 60 deg when it induces a sensation of warmth like that experienced in slowly moving air at 60 F saturated with moisture. The *effective temperature index* cannot be measured directly, but is determined from dry- and wet-bulb temperatures and air motion observations by reference to an Effective Temperature Chart (see Figs. 8, 9, and 10) or tables.

Fig. 8 gives the effective temperature for any combination of dry- and wet-bulb temperatures for *still air* (15 to 25 fpm) conditions. Charts similar to Fig. 8 for air velocities of 300 and 500 fpm have been presented in some of the earlier editions of the GUIDE. Fig. 9 is another form of effective temperature chart embodying all three variables: dry-bulb and wet-bulb temperatures, and air velocity.

Table 4. Comparison of Comfort Ranges With Zone of Thermal Neutrality

Investigators	Effective Temperature		Operative Temp	Remarks
	Optimum Line	Range	Range	
Comfort Zone				
Houghten and Yaglou.	66	63–71		Winter non-basal; at rest, normally clothed. Men and women.
Yaglou and Drinker..	71	66–75		Summer non-basal; at rest and normally clothed. Men.
Yaglou	72.5	66–82		Entire year; non-basal; at rest and stripped to waist. Men.
Keeton et al. ...	75	74–76		Entire year;basal,nude.Steady state (9 hr exposure). Men and women.
Zone of Thermal Neutrality				
DuBois and Hardy....	75	73.2–76.9		Basal; nude; men.
	71.8	64.8–76.0	....	Basal; clothed; men.
Winslow, Herrington and Gagge ..			84.0–87.8	Non-basal; at rest; nude; men.
	....	....	74 –84	Non-basal;at rest;clothed;men.

As stated previously, effective temperature is an index of the degree of warmth experienced by the body. An effective temperature line is, therefore, a line defining the various combinations of conditions which will induce like sensations of warmth. It does not necessarily follow that like sensations of comfort will also be experienced along the entire length of an effective temperature line. Some degree of discomfort is likely to be experienced at very high or very low relative humidities, regardless of the effective temperature. It has also been found that the optimum effective temperature varies with the season, and is lower in winter than in summer.

Tests[14] made at the A.S.H.V.E. Research Laboratory in very hot conditions, with subjects doing light work, were in very close agreement with the effective temperature chart. Other work[20] under similar environmental conditions, but with subjects walking 3 mph and carrying 20 lb packs, indicated that the effective temperature lines should be more nearly horizontal. It therefore appears that the slope of the ET lines may vary, depending upon the rate of work being performed.

Fig. 10 shows the A.S.H.V.E. Comfort Chart[22] modified in several respects from the chart previously shown. The former areas and arrows indicating the summer and winter comfort zones have been removed. The summer comfort zone was removed because it extended to temperatures where too large a percentage of the people would be uncomfortable. The winter comfort zone was removed for the same reason, and because of inadequate data in later studies.

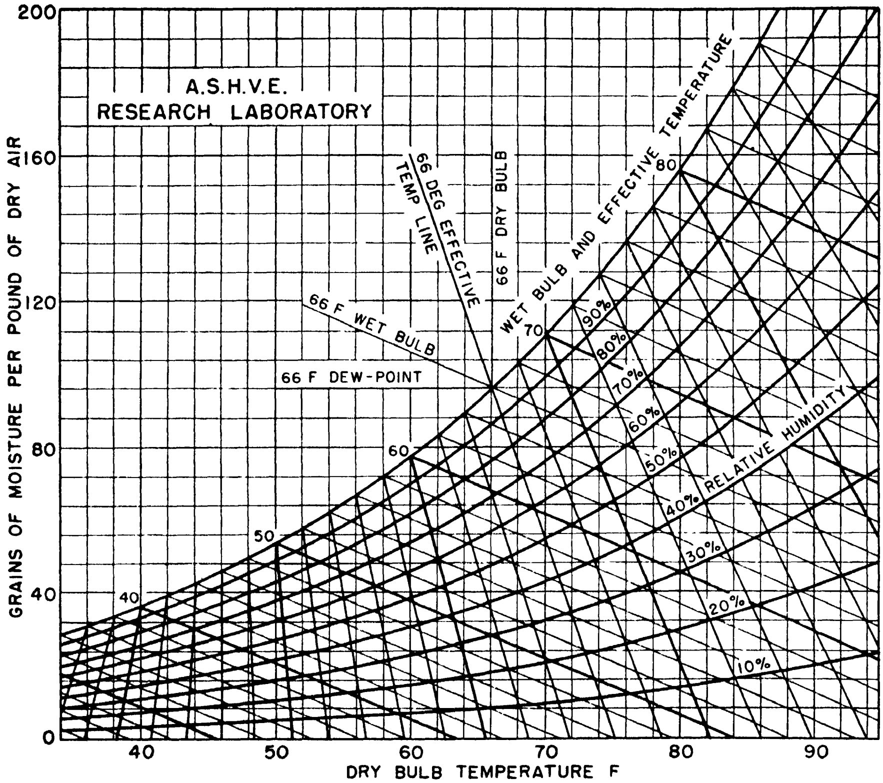

FIG. 8. PSYCHROMETRIC CHART, PERSONS AT REST, NORMALLY CLOTHED, IN STILL AIR

The distribution curve, showing the percent of people feeling comfortable at various effective temperatures in *summer*, indicates that a maximum of 98 percent of the people were comfortable at 71 ET. The study was conducted with relative humidities between 30 and 70 percent.

The distribution curve shown on the previously used chart, showing the percent of people feeling comfortable at various effective temperatures in the *winter*, was based on research prior to 1932. This curve indicated that at 66 ET a maximum number of people were comfortable. Later studies[23] by the A.S.H.V.E. Research Laboratory indicated that a maximum of 97.7 percent of the people were comfortable at 68 ET, and this finding has been confirmed by current practice.[24] However, adequate data from the later studies were available only for the ET range of 65 to 69,

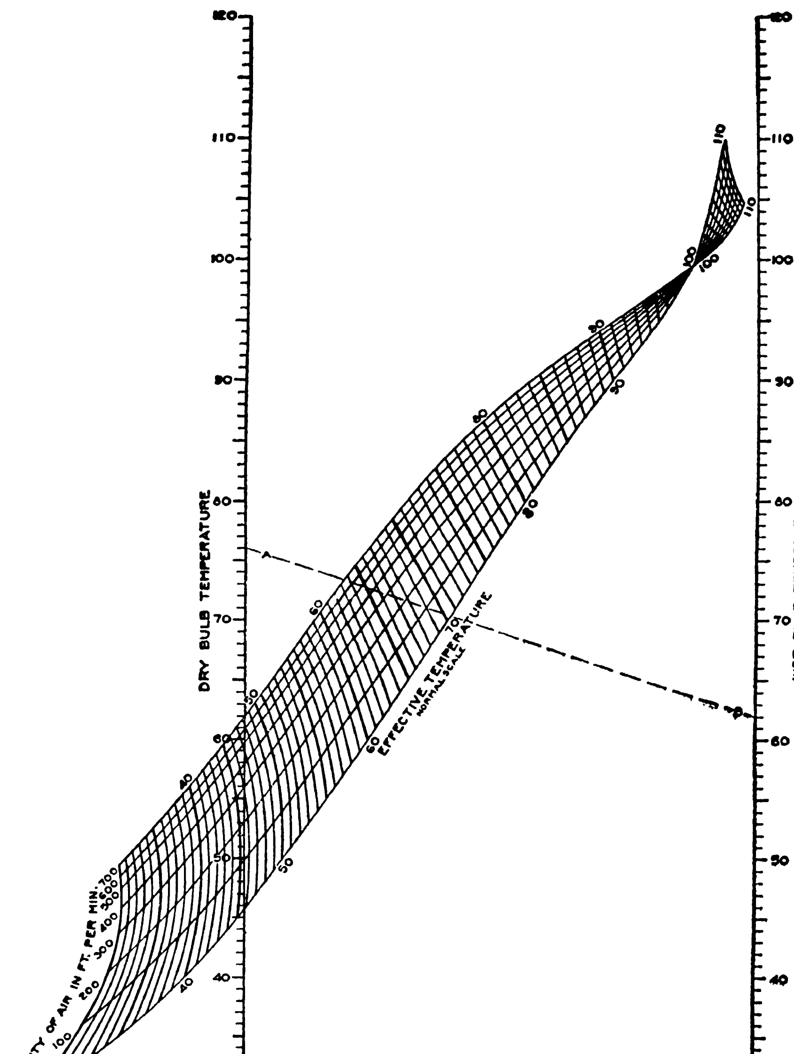

Fig. 9. Effective Temperature Chart Showing Normal Scale of Effective Temperature, Applicable to Inhabitants of the United States Under Following Conditions:

A. Clothing: Customary indoor clothing. *B. Activity:* Sedentary or light muscular work. *C. Heating Methods:* Convection type, *i.e.*, warm air, direct steam or hot water radiators, plenum systems.

as presented in Fig. 10. The studies should be extended to cover a wider range. The lighter weight clothing, probably worn in the later studies, accounts for the higher desirable ET.

Radiation from occupants to room surfaces, and between the occupants, has an important bearing on the feeling of warmth, and may alter to some measurable degree the optimum conditions for comfort previously indicated. Since the mean radiant temperature of a space is affected by cold walls and windows, as well as by the warm surfaces of heating units placed within the room or imbedded in the walls, these factors must be

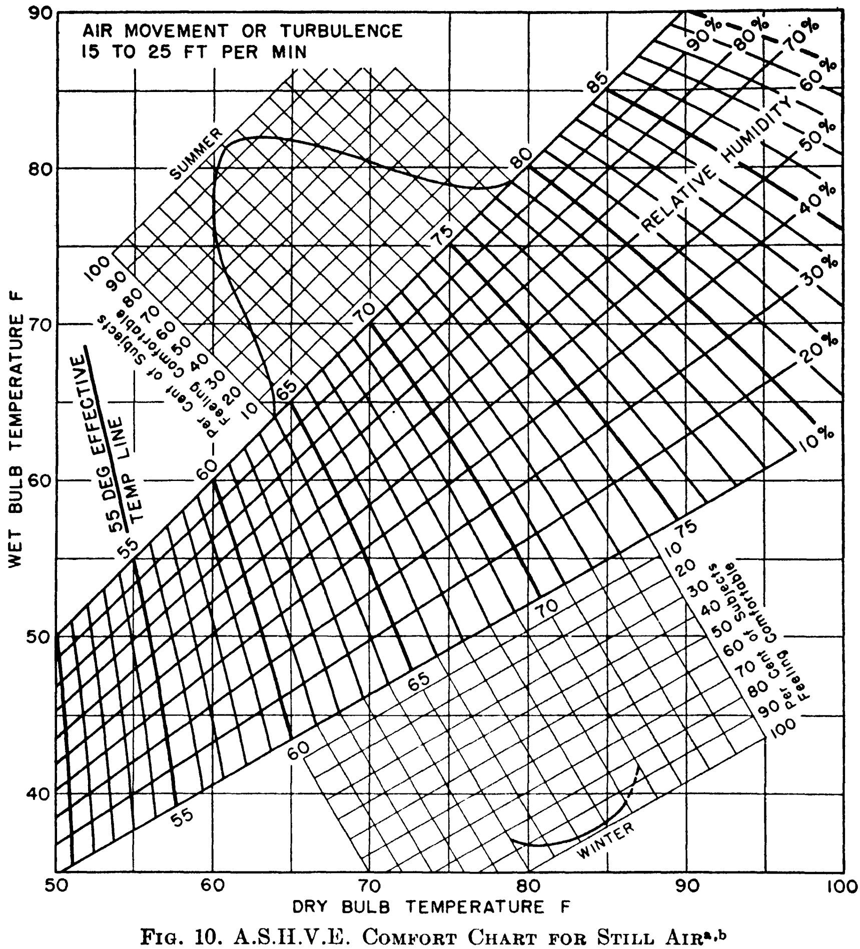

FIG. 10. A.S.H.V.E. COMFORT CHART FOR STILL AIR[a,b]

[a] *Note.*—Both summer and winter comfort lines apply to inhabitants of the United States only. Application of winter comfort line is further limited to rooms heated by central systems of the convection type. The line does not apply to rooms heated by radiant methods. Application of summer comfort line is limited to homes, offices and the like, where the occupants become fully adapted to the artificial air conditions. The line does not apply to theaters, department stores, and the like where the exposure is less than 3 hours. The summer comfort line shown pertains to Pittsburgh and to other cities in the northern portion of the United States and Southern Canada, and at elevations not in excess of 1000 ft above sea level. An increase of one deg ET should be made approximately per 5 deg reduction in north latitude.

[b] Dotted portion of winter comfort line was extrapolated beyond test data.

compensated. Likewise, in densely occupied spaces, such as classrooms, theaters and auditoriums, temperatures somewhat lower than those indicated by the comfort line may be desirable because of counter-radiation between the bodies of occupants in close proximity to each other. Such radiation will also elevate the mean radiant temperature of the room.

Many field studies[23] have been made to determine the optimum indoor effective temperature for both winter and summer in several metropolitan districts of the United States and Canada, in cooperation with the managements of offices employing large numbers of workers (Fig. 11). On the whole, women of all age groups studied prefer an effective temperature for comfort 1.0 deg higher than men. All men and women over 40 years of age prefer a temperature 1 deg ET higher than that desired by persons below this age. The persons serving in all of these studies were representative of office workers dressed for air conditioned spaces in the summer season, and engaged in the customary office activity.

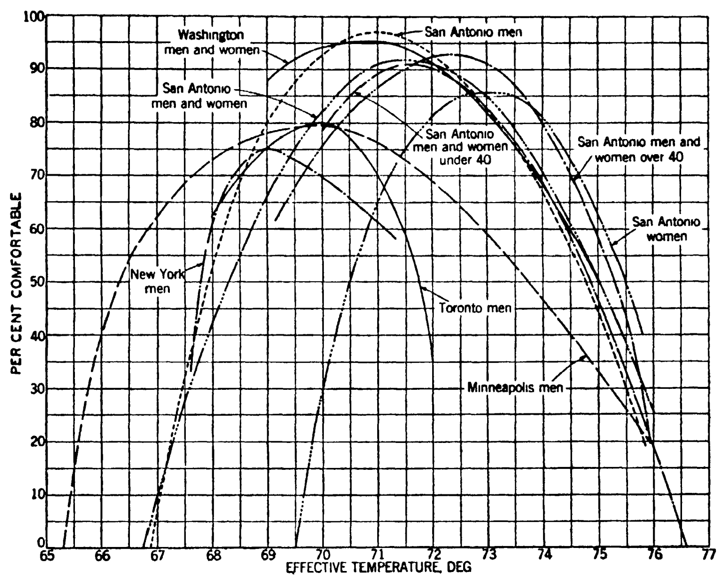

FIG. 11. RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OBSERVATIONS INDICATING COMFORT

On the basis of present knowledge, for different geographical regions and age groups, the most popular temperature varies from a low of 66 ET in winter to a high of 73 ET in summer. The spread for summer comfort is 69 to 73 ET.

A spread of 3 deg in the optimum effective temperature for summer cooling is ascribed to geographical location. However, variations in sensation of comfort among individuals may be greater for any given location than variations due to a difference in geographical location. The available information indicates that changes in weather conditions over a period of a few days do not alter the optimum indoor temperature.

Sudden chilling (shock) of persons entering a cooled and air conditioned space during the summer months, may at times be important. It is due to the rapid evaporation of perspiration which accumulated on the skin and in the clothing during previous subjection to hot and humid outside conditions. While studies[25] have shown that for healthy individuals this *shock* is not harmful, under some conditions it may be unpleasant or even harmful. People entering and remaining in cooled spaces for short periods, 15 min or less, may be satisfied with less cooling. For long occupancy very little deviation from the optimum effective temperature is indicated.

An exit shock upon re-entering a warm atmosphere is equally plausible. Experiments at the A.S.H.V.E. Research Laboratory[26] indicated no demonstrable harm to a healthy individual. Adaptation occurred as soon as normal perspiration was established. Mild exercise shortened the adaptation time.

A great number of persons seem to be fairly content in summer with a higher plane of indoor temperature. Studies by the University of Illinois[27] in cooperation with the A.S.H.V.E. Committee on Research indicate that effective temperatures as high as 74.5 ET are acceptable in the living quarters of a residence, and while this condition is not representative of optimum comfort, it provides sufficient relief in hot weather to be acceptable to the majority of users, in the interest of economy. Individual minority differences can be counteracted by clothing.

Satisfactory comfort conditions for persons at work[28] vary depending upon the rate of work and the amount of clothing worn. In general, the greater the degree of activity, the lower the effective temperature necessary for comfort. Clothing has been evaluated for its overall insulation effects by a physical unit, the *clo* which equals 0.116 C deg per (kilogram calorie) (square meter) (hour).[29] Yaglou[30] criticizes the concept of overall insulation, and points out that different parts of the body require different amounts of insulation. The literature on effects of clothing is difficult to coordinate at the present time, as much of it is still in military service reports which are yet to be published and amplified.

For prematurely born infants, the optimum temperature varies from 100 to 75 F, depending upon the stage of development. The optimum relative humidity for these infants is placed at 65 percent.[31] No data are yet available on the optimum air conditions for full term infants and young children up to school age. Satisfactory air conditions for these age groups are assumed to vary from 75 to 68 F with natural indoor humidities. For children (having high metabolism) at school, in winter clothes, 70 F has been considered correct, with 55 F recommended for gymnasiums.

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CHAPTER 7

AIR CONDITIONING IN THE PREVENTION AND TREATMENT OF DISEASE

Sanitary Ventilation, Control of Airborne Infection, Value of Air Cooling Under Tropical Conditions, Treatment of Disease, Operating Rooms, Nurseries for Premature Infants, Fever Therapy, Cold Therapy, Allergic Disorders, Oxygen Therapy, General Hospital Air Conditioning

THE late war caused an increase of interest in the preventive aspects of air conditioning. It re-emphasized the importance of the control of airborne infection and demonstrated the value of air cooling under tropical conditions for the prevention of heat rash, for promoting proper rest and sleep, and in the convalescence of patients.

The problem of air conditioning or air purification in shelters, hospitals, or any buildings, following an atomic explosion has the attention of engineers, military personnel, public health authorities, private physicians, and the general public. The type and structure of shelters are important. In strategic areas likely to be bombed, they should be windowless, underground, strong enough to resist blast and have sufficient cover to protect against initial radiation. Suitable ventilation is a necessity, and could be provided by the use of pressurized installation in which any air taken from the outside is forced through a ventilator. Air conditioning and heating or cooling systems could then be kept in continuous operation for improving inside air conditions and controlling room temperature. Heat transmission through the walls, carbon dioxide accumulation, and the total number of people using the shelter, are factors in determining the necessity for using outside air. Such a shelter should be closed to outside air during the period immediately following a blast.

A surface or subsurface atomic burst would result in wide dispersion of radio-active particles in high concentrations.[1] There is relatively little danger from such particles after a high atomic burst. Windows in ordinary buildings close to the blast would be broken, and consequently, air conditioning systems would provide no protection against this hazard. At a distance of 1200 ft from ground zero, many windows would not be broken; outside leakage would therefore be slight, and air conditioning systems could be used with advantage.

The recent incident in Donora, Pa., has focused the attention of health authorities and engineers on the problem of air contamination by toxic gases. The exact way in which such substances affect the human being is unknown.[2] There has been a great deal of speculation, but in reality there is little specific information concerning this problem. It is believed that in such cases the combination of contaminents, rather than any single substance, produces the toxic effects. It is known that low atmospheric pressure, with accumulation of toxic substances in increasingly higher concentration, produced the situation at Donora. The U. S. Public Health team, in their investigation, found that this same city had experienced several previous incidences of lesser severity during the preceding 30 years. This was shown by much higher mortality rates during certain periods. A high percentage of the population of this city suffered to some extent during the recent *smog*. Older individuals were more seriously affected with cardio-respiratory symptoms.

Study of such a situation is rather difficult, as explained in the aforementioned report. The medical profession has much to learn about the effects of these substances on human beings, and until more precise knowledge is obtained, it is difficult to know when precise controls are really needed. Further research is urgently needed.[3]

SANITARY VENTILATION

During the last 15 years great popular interest has been aroused in the spread of respiratory infection indoors and control by ventilation or its sanitary equivalent by air disinfection. Three important documents have appeared in English, Swedish and French literature recently.[4, 5, 6]

In this country, where the study initiated, the Council of Physical Medicine of the *American Medical Association* approved the radiant disinfection of air in 1943, and two sub-committees of the Committee on Research and Standards of the *American Public Health Association* have reported favorably upon the control of airborne infection by sanitary ventilation and on air sanitation, respectively.[7, 8, 9, 10]

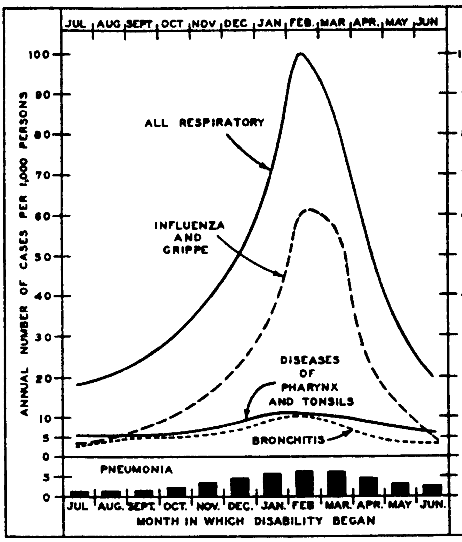
The work of a Technical Advisory Committee on Air Sterilization of the ASHVE has from time to time reported progress in the Journal since 1944, and has approved a set of definitions, formulations and factors which is now being reviewed by the Committee of Research and Standards of the *A.P.H.A.* with the purpose of joint adoption by the Association and the Society.

Since this important new field of sanitary ventilation is yet in its infancy, just emerging from the research and development stage, it has not been possible to prepare a comprehensive treatment in The Guide. Those professionally interested must still refer to the official publications of the organizations listed, and the enormous technical literature on the subject which they review. The following section on Control of Airborne Infection gives some idea of the scope of the subject.

CONTROL OF AIRBORNE INFECTION

The majority of airborne diseases are spread indoors where people gather. Any program of air sanitation is influenced by a number of factors. In the winter months, the closing of doors, windows and other means of access to the outside air to conserve warmth, as well as the crowding of persons indoors, provides conditions conducive to a high incidence of contagion. This seasonal phenomenon, illustrated in Fig. 1 which represents a study made by the *U. S. Public Health Service*, will concern the ventilating engineer insofar as air quality (determined by temperature, humidity, air replenishment and type of air movement and by freedom from contamination) is a major intrinsic factor. Apart from the seasonal picture of airborne contagion, are such extrinsic factors as rate of turnover of personnel, and the marked susceptibility of the recruit in comparison with permanent personnel[11] as shown in Fig. 2 by studies of military personnel housed in barracks. These extraneous variables and the factor of contact infection (direct spray) tend to complicate any evaluation of the effectiveness of air sanitation for elimination of micro-organisms in droplet-nuclei and droplet-dust. Thus, control measures may eliminate consistently 90 percent of airborne organisms in laboratory tests, but cannot effect a decrease in actual incidence of infection exceeding 30 percent. Thirty percent may be the maximal reduction in infection possible by air treatment methods. The distinction should be clearly drawn, therefore, between the effectiveness of a procedure in laboratory tests and its effectiveness and applicability in actually reducing the incidence of airborne disease. On the other hand, recent studies suggest that inhalation of dust-borne bacteria is more important than direct inhalation of infectious droplets or droplet nuclei in the spread of respiratory tract infections.[12]

The following sequence of events has been postulated as occurring in a large proportion of intra-ward infections: (a) ejection of relatively large protected infective particles from patients; (b) rapid venting or settling of these particles so that those remaining airborne are in low con-



Occurrence of diseases causing disability for 8 consecutive days or longer in a group of 100,000 wage earners (10 percent women) in different industries.
[a] Graph obtained from Dean K. Brundage, U. S. Public Health Service.

FIG. 1. STUDY OF AVERAGE MONTHLY FREQUENCY (1921–1926 INCLUSIVE) OF SPECIFIED RESPIRATORY DISEASES[a]

centration; (c) survival of infective particles to permit the accumulation of high concentrations on surfaces; (d) repeated reintroduction of infective particles into the air under the stimulus of ward activities or by air currents of the order of 50 fpm over the floor; and (e) extension of infective areas by air turbulence throughout the ward or hospital. The most important link in this probable infection chain has been demonstrated to be the reintroduction of particles into the air.[13]

Intensive studies on air disinfection have indicated two distinct control measures: (a) suppression of dust and lint, and (b) disinfection of droplet-nuclei. A third measure, control of relative humidity is important. It has been shown that the viability of certain organisms sprayed into the atmosphere from a liquid suspension is dependent on relative humidity. The mortality rate of the organisms is very high at a relative humidity of 50 percent,[14] and decreases at humidities above and below this figure. It has also been reported that the influenza virus loses much of its virulence when the relative humidity is 50 percent.[15]

Well controlled, large scale tests of the various methods of air sterilization conducted in barracks[16, 17] have confirmed the importance of dust control in minimizing the spread of airborne disease, a consideration which has guided the practices of ventilating engineers for a number of years. The importance of the dust factor has been emphasized by many engineers, and has been convincingly demonstrated by subsequent bacteriologic studies aboard ships.

Treatment of floors and bedclothes with oil emulsions has proved effective in reducing bacterial dispersion by as much as 90 percent in Army barracks and station hospitals.[17] The incidence of acute respiratory infections was from 10 to 30 percent lower in barracks with oiled floors and bedclothes than it was in control barracks which received no special treatment. More recent studies, however, have yielded inconsistent results.

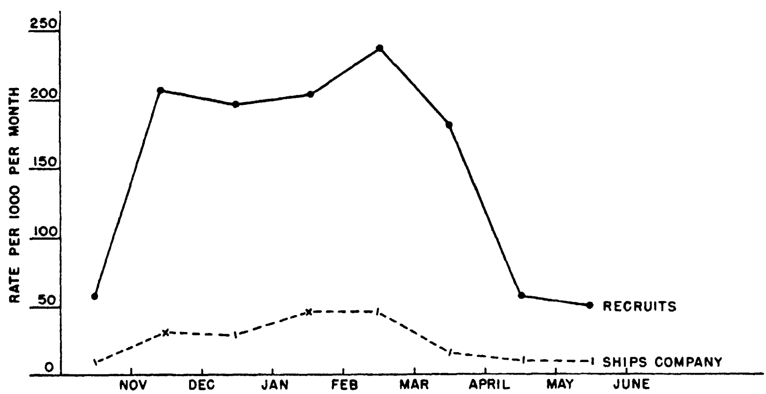


FIG. 2. MONTHLY INCIDENCE OF ACUTE RESPIRATORY ILLNESS AMONG NAVAL RECRUITS AND SHIP'S COMPANY (PERMANENT PERSONNEL)[1]

An emulsifying mixture, *Fixanol C* containing cetyl pyridinium bromide, when incorporated in the oil-in-water emulsion imparted a bactericidal action to the emulsion. Blankets treated with this substance and oil became bactericidal and retained this property for as long as three months. The possibility of hypersensitivity of an occasional individual to bromide drugs should be borne in mind when exposing large groups to treated garments or blankets.[12]

No simple method for disinfecting droplet-nuclei has yet been devised. Under favorable laboratory conditions, propylene glycol in concentrations of 0.07 to 0.14 milligrams per liter, and triethylene glycol in a concentration of 0.0045 milligrams per liter were highly germicidal for most airborne bacteria in clean air when the relative humidity was between 40 and 60 percent.[18, 19, 20] A humidity of 50 percent, without the use of glycol vapor, has been reported as destructive to some bacteria. However, maximum rates of bactericidal action of triethylene glycol vapor will be secured at humidities between 20 to 50 percent.[21]

In a recent report on the effect of triethylene glycol vapor in air disinfection, it is pointed out that the rate of ventilation, as determined by the number of air changes per hour, is important and that continuous vaporization is needed to maintain effective concentrations of glycol vapor.[21] In the absence of an apparatus to measure the concentration of vapor in the room, a slight fog is an indication of adequate concentration. Absence of such a fog indicates a non-bactericidal concentration. The report stated: "However, under experimental conditions we have observed no apparent decomposition of triethylene glycol when temperatures up to 290 F were maintained at the site of vaporization, provided the vaporizing unit was designed so that heat was applied to liquid glycol only at the point of vapor formation. On the other hand, decomposition was frequently observed when the vaporizing temperature was raised above 290 F, or when liquid glycol was kept above 120 F for long periods of time. The precise nature of the decomposition products is as yet unknown, but they may well be irritant or toxic and their formation should be avoided. For this reason, it is felt that a limit of 290 F should be applied to the temperature employed in vaporization, and furthermore, that liquid glycol should not be heated above 120 F for long periods of time."

It was also recommended that a given vaporizer be capable of various outputs as measured by grams of liquid glycol vaporized per hour; for instance, from 0.5 g to at least 2.0 g per 1000 cu ft of volume treated, for small vaporizing units. Such high rates secure bactericidal concentration in less than an hour. Each room should be furnished with a single small vaporizer, and one vaporizer should not be expected to handle a volume greater than 4000 cu ft. Apparatus for the production of glycol vapor, and an independent duct system for carrying this vapor and diluting air for large rooms or spaces, and unit type vaporizers for small spaces, have been described recently.[22] There is also available a device for the automatic regulation of glycol vapor in the air called the glycostat. This instrument has been calibrated to measure the degrees of saturation of the air with glycol vapor by direct reading of the variations in the intensity of light reflected from the glycol condensing surface of the wheel of the instrument.[23]

Under practical conditions, particularly in the presence of dust in the air, glycol effectiveness is much reduced. The use of other chemical aerosols that have been tried is limited by their toxicity, odor, or destructiveness to fabrics and metals. A recently reported controlled experiment in the offices of the Metropolitan Life Insurance Company showed that under ordinary working conditions triethylene glycol vapor failed to reduce the number of airborne bacteria and the incidence rate of minor respiratory infections.[24]

Ultraviolet radiation of floors and upper air has been studied extensively at the Naval Training Center, Sampson, N. Y. In barracks housing naval recruits, hospital admissions for respiratory infections (mostly *catarrhal fever*) were 25 percent lower in a group of men exposed to ultraviolet radiation—2537 Angstrom Units, 1 to 7 ergs per (cm^2) (sec) at bed level—than they were in adjacent control barracks without ultraviolet radiation.[25]

A controlled study over a six-year period on the evaluation of ultraviolet radiation of sleeping quarters as a supplement of accepted methods of respiratory disease control was recently reported.[26] The amount of radiation over the last two years of the period was about five times that recommended commercially. No significant effect in the incidence of disease could be detected in about 400 inmates during the six-year period. The data imply that air layers were not sufficiently mixed. However, more efficient mixing would have been obtained at the expense of increased circulation of dust and lint. Sources of ultraviolet radiation should be so situated as to protect the eyes of the occupants of the room from direct or reflected rays. A combination of ultraviolet radiation and dust control measures is believed to be more effective than either one of the two used alone, but the proof for this has yet to come.

There is no doubt that these methods will reduce the number of bacteria in the air of an enclosed space, but there is still considerable question as to whether they are practical measures for reducing the total number of respiratory infections among personnel, since exposure by contact is an important factor. Bourdillon and his group concluded that there is justification for attempting to reduce the load of air-borne microorganisms by the methods now at our disposal.[4] They have explored the properties of a number of compounds, and their conclusions are in agreement with those of American investigators.

The present status is admirably reviewed by the Committee on Sanitary Engineering of the *National Research Council*,[27] and by a subcommittee of the *American Public Health Association*.[9] Both committees feel that the problem of air disinfection is still in the experimental stage. Knowledge concerning the effectiveness of glycol vapors has not kept pace with the development of vaporizing devices, and there is real danger that commercial exploitation of the various devices may discredit the method and discourage careful research in this important field.[28] More experimentation is needed for arriving at a definite conclusion concerning its use in industry and public buildings.

VALUE OF AIR COOLING UNDER TROPICAL CONDITIONS

The commissioning of a class of naval hospital ships with all wards, laboratories and living spaces air cooled is a notable achievement to provide better treatment of patients, especially those suffering from extensive burns, by control of environmental factors. Although statistics are not at hand to indicate the deaths or retarded recoveries of patients due to lack of air cooling in ships operating in tropical waters, it is generally agreed among competent observers that high temperature and humidity are major factors in prolonging disability and increasing mortality of the sick and injured. Physiologic data obtained on healthy men, moreover, show the large loss of body fluids and the stress on the cardiovascular system in terms of increased pulse rate when these men are continuously subjected to high temperatures. Even at rest about 50 cc of fluid per hour are lost as sweat[29] through intact skin. In burn patients the difficulty, encountered in temperate climates, of maintaining fluid and electrolyte balance is tremendously augmented by the additional evaporative fluid loss in hot environments.

Patients who have such varied conditions as heart disease, thyrotoxicosis, shock from any cause, severe hemorrhage, or those who have had an anesthetic, will invariably store heat when subjected to a hot, humid environment. The gradient between the body surface temperature and the environmental temperature is such that loss by radiation is slight. The heat loss by evaporation in a warm, humid environment is low whether the patient does or does not perspire. The heat regulatory center may be temporarily deranged following an anesthetic, brain injury, or after an overdose of barbiturate. Loss of fluids and electrolytes is anoter influencing factor. Cooling the body is the answer to this problem, and this can best be done in a cool room of low relative humidity where conditions for heat loss are ideal. This measure is also valuable in controlling temperature height of patients with various acute febrile diseases.[30]

Frequently from 50 to 75 percent of personnel aboard naval vessels operating in tropical waters are afflicted with heat rash to a degree that interferes with rest and sleep. In carefully controlled experiments[29] it was possible to produce a fulminating type of rash in all men living continuously at an effective temperature of 85 (90 F dry-bulb and 83 F wet-bulb). In the control group, 12 out of 24 hr were spent in a relatively cool atmosphere of 75 ET (80 F dry-bulb, and 70 F wet-bulb). These men either remained free from heat rash, or occasionally developed a mild form. Thus, intermittent cooling to a degree which prevented sweating in men at rest eliminated a serious handicap to good performance of duty.

In both laboratory tests and aboard hospital ships a relatively cool living environment of 76 to 78 ET provided an atmosphere conducive to rest and sleep without excessive sweating. Berthing spaces tended to have extremely low odor levels. Motivation, initiative and alertness, in contrast to the usual irritability and lack of incentive incident to residence in tropical climate, were maintained.[31]

Little has been done, however, to obtain practical methods for application of air conditioning under heavy heat loads and on the enormous scale that would be needed to modify life in the tropics. It is not improbable that cooled houses in a tropical climate, if used consistently for one generation, would modify the whole character of a population. The obvious advantages of part time cooling on personnel to promote rest and sleep in tropical areas would provide a prophylactic measure of great potential importance.

TREATMENT OF DISEASE

In the past few years considerable progress has been made in using air conditioning as an adjunct in the treatment of various diseases. Among the important applications are those in operating rooms, nurseries for premature infants, maternity and delivery rooms, children's wards, clinics for arthritic patients, heat therapy, cold therapy, oxygen therapy, X-ray rooms, the control of allergic disorders, and for the physiological effects in industry.

Normal individuals may be subjected to considerable strain in adjusting to hot, humid conditions. Heat loss by radiation is reduced, as is loss by evaporation of sweat. Individuals with certain disease processes are at a still greater disadvantage since they may also have difficulty in the transport of heat from the interior to the surface of the body via the circulation.

Patients with thyrotoxicosis tolerate hot, humid conditions or *heat waves* very poorly. Their metabolism is high, and therefore their heat production is excessive. They may be unable to eliminate heat from the body surface as rapidly as it is produced and transported to the skin. They develop hyperthermia or fever, and a tachycardia or rapid heart rate. The demand on the circulation for transport of heat from the interior of the body to the skin surface is increased. The increased body temperature leads to increased cell metabolism, and in turn to still greater heat production. This vicious cycle may threaten life if the cardio-vascular or transport mechanism breaks down. A cool, dry environment favors the loss of heat by radiation and evaporation from the skin, and may save the life of the patient with thyrotoxicosis.

Cardiac patients may be unable to maintain the circulation necessary to insure normal heat loss. Individuals with head injuries, those subjected to brain operations, and those with barbiturate poisoning may have hyperthermia, especially in a hot environment, due to a disturbance in the heat regulatory center of the brain. Obviously, one of the most important factors in recovery is an environment in which the patient can lose heat by radiation and evaporation, namely, a cool room with dehumidified air.

The patient in shock, or the patient who has had a severe hemorrhage, may have an inadequate volume of circulating blood and be unable to maintain an adequate skin circulation. This may result in heat storage or fever. Patients with extensive skin burns may be unable to lose heat adequately from the limited uninvolved skin surface, and thus develop a fever. They need adequate fluid replacement, saline solution, plasma or blood to expand the circulating blood volume and thereby improve peripheral circulation. A cool environment is valuable in aiding heat loss after adequate skin circulation is established.

A hot, dry environment (89.6 F and 35 percent relative humidity) has been used over an extended period for the treatment of patients with rheumatoid arthritis, with reported improvement.[32]

OPERATING ROOMS

The widest application of air conditioning in hospitals is in operating rooms. Complete air conditioning of operating wards is important because winter humidification helps reduce the danger of anesthetic gases; summer cooling with some dehumidification tends to eliminate excessive fatigue and to protect the patient and operating personnel; and finally, filtering aids the removal of allergens from the operating room air.

Reducing Explosion Hazard

Explosion hazards in operating rooms began with the introduction of modern anesthetic gases and apparatus. Ether administered by the old drop method gives rise to an explosive mixture, but in practice this method is still regarded as comparatively safe. When ether is mixed with pure oxygen, or nitrous oxide in certain concentrations, the explosion hazard may be as great as with ethylene-oxygen, or cyclopropane-oxygen mixtures.

Of the anesthetic gases nitrous oxide alone does not explode but supports combustion. Ether, vinyl ether, ethylene, and cyclopropane are as potentially dangerous as gasoline or illuminating gas in the home.[33] Chloroform does not explode violently in contact with flame, but decomposes to liberate phosgene. All of the anesthetic gases and vapors, except ethylene, are heavier than air. Although the incidence of injury or death from explosion is negligible compared with other hazards in the operating room, the dramatic features surrounding an explosion justify continued investigation to eliminate the hazard.

During the course of ethylene anesthesia, the mixture, usually 80 percent ethylene and 20 percent oxygen, is so rich that the danger of explosion is slight in the immediate vicinity of the face mask, but leakage of ethylene into the air may accumulate to any lower concentration, and thus introduce a serious hazard. The most dangerous period is at the end of the operation when the patient's lungs and the anesthesia apparatus are customarily *washed out* with oxygen with or without the addition of carbon dioxide. Even when this procedure is omitted, it is difficult in practice to avoid dilution of the anesthetic gas with air during the normal course of breathing following the administration. In either case the mixture would pass through the explosion range and extraordinary precaution is necessary for the safety of the patient and operating personnel.

In a study[34] of 230 anesthetic explosions and fires, 70 percent of the explosions and 60 percent of the deaths were caused by igniting agents other than static sparks. In 1941 the *National Fire Protection Association*[35] made certain recommendations for safe practice based on available information. Some of these recommendations are:

Windows should be kept closed so that the air conditioning system can prevent pooling of explosive anesthetic gases. Twelve air changes per hour and a humidity of 55 percent are advised. If a higher humidity were compatible with the well being of the patient and personnel, it should be maintained. All electrical installations should comply with the standards set by the *National Electrical Code* for use in explosive situations. Cautery equipment should not be used in hazardous locations. To prevent static sparks, all bodies in an operating room should be conductive or coupled. It is essential that adequate grounding be provided for the floor and every object in the operating room. Conductive rubber should be used on shoes, leg tips, operating table coverings and all rubber parts of the anesthesia equipment. All furniture in contact with the floor should be metal. In the absence of complete grounding facilities, the simple method of intercoupling patient, operating table, anesthetist and gas machine at ground potential may be used.

Experience has shown that neither high humidity nor intercoupling devices have eliminated the danger from static electric discharge. The removal of gas concentrations from the operating table area, by means of specially devised exhaust ventilation, should be thoroughly tested. Portable duct systems as installed aboard ship should be acceptable. Serious explosions can occur in a closed system, but proper precautions will reduce this hazard to a minimum.

It should be realized that when a room and the occupants have been completely grounded, there is always the possibility that the patient or the operator might receive a dangerous shock if a short circuit developed in any of the electrical equipment.

A comprehensive study of the explosion problem and of the general causes and prevention of operating room hazards, by the *University of Pittsburgh*, the A.S.H.V.E. Research Laboratory, and the *U. S. Bureau of Mines* has led to a fruitful attempt to eliminate the explosive range of cyclopropane, one of the best but most difficult gases to handle. The use of helium as a diluent in the total gaseous mixture controls the oxygen concentration by displacement and, because of its flame quenching properties, it is the ideal gas for this purpose. In addition, a gaseous mixture containing helium is more difficult to ignite by electric discharges, and this quality also increases the safety factor of anesthetic administration.

Operating Room Conditions

Little is known about optimum air conditions for maintaining normal body temperatures during anesthesia and the immediate post-operative period. An anesthetized patient displays dilation of blood vessels in the skin resulting in profuse sweating and (it has been believed) inability to regulate body temperature. From this it was concluded that all anesthetized patients suffered considerable heat loss, although there may be little more than 0.8 F variation in the rectal temperature during the course of the operation.[36] The severe physiological effects, such as excessive sweating and rapid pulse, of high operating room temperatures on attendants and patients during the hot months signify the need for proper cooling. Statements of surgeons who operate in both air conditioned and non-air conditioned rooms strongly indicate that the recuperative power of the patient is greater when operated upon in air conditioned rooms.[36]

Although the comfortable air conditions for the operators are not identical with those for the patient, it is usually not difficult to compromise within a range of 55 to 60 percent relative humidity and 72 to 80 F temperature. The work just cited reported that 68 to 70 F effective temperature not only furnished comfort for the operating room workers, but apparently prevented exhaustion of the patient as evidenced by rapid convalescence in the recovery ward. Additional heat may be furnished to the patient locally or by suitable covering, according to body temperature in individual cases.

In the control of airborne infection in the operating room, the prevention of dispersal of infectious materials into the air, control of dust, and proper ventilation supersede attempts to remove or kill pathogenic organisms. The bacterial content of conditioned operating rooms is generally lower than that of non-conditioned rooms.

Bacterial counts aboard an air-conditioned submarine were found to be exceptionally low and not cumulative with time, although all of the air was recirculated for more than 12 hours[37] without replenishment. The removal of bacteria by the process of air cooling and condensation of moisture out of air, merits further study.[38]

The degree of air contamination can be reduced by proper ventilation if velocity of air over the floor does not exceed 50 fpm. Research is in progress on the use of filtered air flowing through a system of mechanical cleaners which protect the patient against infection from attendants, and from bacteria-containing air in the corridor or ward.[39]

Operations are frequently postponed on allergic patients during asthmatic manifestations through fear of complications. The removal of airborne allergens, therefore, is in some cases an important function of the air conditioning system in preparing patients for operation.

Central system air conditioning plants and unit air conditioners prove satisfactory in operating rooms when producing between 8 and 12 air changes per hour of filtered and properly conditioned air, without recirculation, during the course of anesthesia. A separate exhaust fan system is usually necessary to confine and remove the gases and odors. Double windows are desirable and often necessary to prevent condensation and frosting on the glass in cold weather, and to minimize drafts. The air flow of 8 to 12 air changes in operating rooms should: (1) reduce the concentration of the anesthetic to well below the pharmacologic threshold in the vicinity of the operating personnel; (2) remove the great amounts of heat and sometimes moisture, from sterilizing equipment if inside the operating room, from the powerful surgical lights, from solar heat, and from the bodies of the operatives; and (3) provide extra capacity for quickly preparing the room for emergency operations. Much can be gained by thermal insulation of sterilizing equipment, and by thorough exhaust ventilation of sterilizing rooms adjoining the operating rooms. An air conditioned recovery ward in connection with the air conditioned operating room, is of great value in stabilizing peripheral circulation, and in reducing excessive loss of fluids on hot humid days.

NURSERIES FOR PREMATURE INFANTS

One of the most important requirements in the care of premature infants is the stabilization of body temperature. This is necessary because the infant's heat regulatory system is not fully developed, with the resultant tendency for environmental temperature to influence body temperature. The younger the premature infant, the greater is the tendency. As the infant's metabolism is low, heat production is not adequate to maintain a normal body temperature in a cool environment. The resistance to infection is low, and the mortality rate is high. In general, the younger the age of the premature infant, the higher the mortality rate.

Nurseries constructed for metabolic research should be air conditioned so that conditions are reproducible. Results of such studies may be invalid if environmental conditions are not identical, since fluid and electrolyte loss may vary greatly with change in environmental conditions.

Air Conditioning Requirements

The optimum air conditions for growth and development of premature infants were determined by extensive research[40] at the Children's Hospital, Boston, Mass., using four valid criteria, namely, stability of body temperature, gain in weight, incidence of digestive syndromes, and mortality. Individual temperature requirements varied widely (from 72 to 100 F) according to the constitutional state of the infants and body weights. The optimum relative humidity was about 65 percent, and the air movement less than 20 fpm.

A single nursery conditioned to 77 F and 65 percent relative humidity was found to fulfill satisfactorily the requirements of the majority of premature infants. Additional heat for weak (or debilitated) infants may be furnished in the cribs or by means of electric incubators placed inside the conditioned nursery, and the temperature adjusted according to individual requirements. In this way multiplicity of chambers and of air conditioning apparatus is obviated; the infants in the heated beds derive the benefit of breathing cool humid air, and the nurses and doctors need not expose themselves to extreme conditions.

Importance of Humidity: Although external heat is an important factor in the maintenance of normal body temperature, humidity appears to be of equal or greater importance. When the premature nurseries at the Children's Hospital were kept at relative humidity between 25 and 50 percent for two weeks or longer, the body temperature became unstable, gain in weight diminished, the incidence of gastro-intestinal disturbances increased, and the mortality rose. On the other hand, continuous exposure to air conditions with 55 to 65 percent relative humidity gave satisfactory results over a period of years. The initial physiologic loss of body weight (loss occurring within first four days of life) was found to vary inversely with the humidity. In the old nurseries with natural humidity it averaged 12.4 percent of the birth weight; in the conditioned nurseries it was 8.9 percent with 25 to 49 percent relative humidity, and 6.0 percent with 50 to 75 percent relative humidity. The number of days required to regain the birth weight was correspondingly maximum in the old nursery, and minimum in the conditioned nurseries under high humidity.

Maximum gains in body weight occurred in the conditioned nurseries under high humidity (55 to 65 percent) in infants weighing less than 5 lb. The gains were less under low humidity (25 to 50 percent) in the same nurseries, and in the old nurseries prior to the installation of air conditioning apparatus.

The incidence and severity of digestive syndromes, with diarrhea, persistent vomiting, diminishing gain or loss of body weight, and other symptoms, were generally from two to three times as high under low as under high humidity.

Summarizing, the best chances for life in premature infants are created by maintaining a relative humidity of 65 percent in the nursery, and by providing a uniform environmental temperature just sufficiently high to keep the body temperature within normal limits. Medical and nursing care are, of course, factors of equal and sometimes of greater importance.

Air Conditioning Equipment

Many of the installations now in use are of the central system type providing for filtration, for humidification and heating in cold weather, and for cooling and dehumidification in hot weather. A ventilation rate between 8 and 12 air changes per hour is desirable to remove odors and maintain uniformity of temperatures in extremes of weather. Recirculation should not be used in these wards owing to odors and the possibility of infection. There should be a frequent change in spray water.

Control of Airborne Infection

The protection of the premature and older infant against infection is of the utmost importance. It was found in one installation equipped with air conditioning, germicidal lights and mechanical barriers that air conditioning alone did not prevent the spread of respiratory cross-infections.[41] Bacterial ultraviolet barriers, air conditioning and mechanical barriers are efficient. However, infections are brought in by, and often spread by, ward personnel in spite of these measures.

FEVER THERAPY

Artificial production of high fever in man can be considered an imitation of nature's way of overcoming invading pathogenic organisms. The action may be direct and specific by destruction of the invading organism within the safe limit of human temperatures, or indirect in the case of heat resistant organisms, by general mobilization of the defensive mechanisms of the body.

Although the action may be direct and specific by destruction of the invading organisms within the safe human limits, fever therapy exerts much of its benefit through the improvement of the mechanism of bodily defense. A serious challenge to the theory on which fever therapy is based comes from the demonstration that high fever causes a reduction in the concentration of circulating antibodies in experimental animals.

Patients for fever therapy should be carefully selected. The most serious complications which may arise are heat stroke, heat exhaustion and circulatory collapse. The chief minor complications are heat cramps, fever blisters and mild dehydration.

The limits of induced systemic fever are usually between 104 and 107 F (rectal), and the duration from 3 to 8 hours at a time. The total period of fever treatment varies with the type of the organism involved.

The diseases which respond favorably to artificial fever therapy are gonorrhea and its complications (which include arthritis, pelvic infections in women, and involvement of the eye), syphilis and chorea.

The most striking results are seen in gonorrhea and syphilis, since the causative organisms can be destroyed at temperatures compatible with human life. However, the use of fever therapy has decreased since penicillin has been found so effective in the treatment of gonorrhea and syphilis. Mild fever, up to 101 F for one hour, has recently been used in the treatment of rheumatoid arthritis. This degree of fever is not bactericidal, but is believed to stimulate the body defense mechanism.

Equipment for Production of Fever

Artificial fever can be induced by injections of various crystalloid or colloid substances, bacterial products of typhoid and malarial organisms, or by physical methods using hot baths, radiant heat cabinets, hot humidified air cabinets, or by short wave diathermy in combination with a cabinet.

The relative advantages of various methods have been evaluated clinically.[42] Among the devices for the production of fever by physical means, the one most widely used is the hot humid air or *air conditioned* cabinet. This apparatus was developed at the *Kettering Institute for Medical Research* at Miami Valley Hospital in Dayton, Ohio.

In the earlier studies of the Society, temperatures were elevated more easily using saturated atmospheres. A fever therapy apparatus[43] using these same principles has proved efficient as a means of inducing and maintaining fever in a body, with small likelihood of burns because of the comparatively low dry-bulb temperatures.

When heat is necessary in treating legs or arms, such media as short or long wave diathermy, micro-waves, infrared, water baths, etc., have been used extensively. A recent development, a saturated atmosphere heating unit, similar to one previously described has proven satisfactory, because heat may be administered over longer periods which render deep heating possible without fear of burns or shocks. Local heating has been somewhat satisfactory in relieving the painful symptoms of peripheral vascular disease.

This procedure, however, is not without danger. Elevation of tissue temperature increases cell metabolism and the need for oxygen. The inadequate blood supply and oxygen deficiency may lead to tissue death or gangrene. Application of heat to the trunk or abdomen, with consequent reflex dilatation of the vessels of the extremities, eliminates this danger of local heat application.

Short wave diathermy within the cabinet during the induction phase has been used. When the desired body temperature has been reached by electrical induction, the atmosphere of the enclosure is kept at saturation to prevent heat loss, thus maintaining the patient's temperature at the desired point. The two underlying principles in the production of fever by the hot, humid air cabinet are: (1) the transfer of heat by conduction from the circulating hot air to the body, and (2) prevention of heat loss. The latter is more important. In an atmosphere of high humidity, the heat loss by evaporation is markedly decreased.

COLD THERAPY

Cold as an anesthetic agent was advocated by Allen several years ago.[44] Freezing of the tissue must be avoided. For certain patients, in whom amputation of an extremity is indicated, the application of a tourniquet with cooling of the affected extremity down to near freezing (5 C or 40 F) is of value. The patient, following this procedure can be prepared for surgery without the handicap of absorption of septic products and severe pain. This procedure has proven especially valuable in the neglected diabetic patient with an infected gangrenous extremity. Time for treatment of coma and hydration of the patient is gained. However, if amputation of an extremity is not indicated, the application of a tourniquet and packing in ice are dangerous procedures, since loss of the limb usually results. An extremity with inadequate blood supply can be readily cooled without the use of a tourniquet, but such an extremity is also usually eventually lost. Theoretically, cooling is said to reduce the metabolism of the tissue with suspension of the vital processes. It also reduces the blood flow to practically zero, and few extremities with inadequate blood supply remain viable or recover.

Packing in ice, or use of low temperatures, is contra-indicated in the treatment of patients with frostbite, immersion foot or trench foot. The affected extremities should be exposed to the air in a cool room and not rubbed with snow or packed in ice. The lowering of temperature by packing the body in ice for treatment of cancer has not proven successful.

The methods used for refrigeration, depending upon available facilities, are as follows:[44]

(1) Cracked or shaved ice which is simple and has the advantage of not freezing tissues. However, it is cumbersome and sloppy to handle and is unsuited to prolonged treatments.

(2) Use of ice in a pail for immersion of local parts.

(3) Special boxes for holding ice with padded or curtained openings for the limb.

(4) Bare ice bags and cloth bags for iced wet dressings for prolonged treatments and convenience.

(5) A double chambered cabinet using dry ice has been constructed.

(6) Electrical refrigerating apparatus, consisting of a compact noiseless unit that pumps fluid to various types of applications, is available. The applicators may be in the form of blankets containing rubber tubes suitable for covering the entire body, or all or part of a limb. Special applicators are available for insertion into various body cavities, and for inducing dental anesthesia.

(7) An air chamber at regulated temperature for treatments of frostbite and immersion foot, and amputation stumps.

The electrical apparatus is costly, but has the advantages of thermostatic regulation, light weight, freedom of movement, and permits prolonged treatments with heat, as well as cold over the range of temperatures therapeutically desirable.

ALLERGIC DISORDERS

Hay fever, asthma, eczema and contact dermatitis are classified as allergic disorders. The allergic individual responds to contact with a variety of substances, which are innocuous to a non-allergic person, with severe manifestations of hypersensitivity.

These substances are known as allergens and consist of airborne irritants such as dusts, molds, feathers, pollens, animal dander and others; of food protein such as milk, wheat, eggs, etc., or of simple chemicals brought in contact with the skin. They may enter the body by various routes of which inhalation is the most common type. Ingestion of offending food substances is not infrequent.

The offending substance reacts with the sensitized cells of the mucous membranes or skin. During this reaction, histamine or a histamine-like substance is released and causes (a) increased capillary permeability, (b) secretion of mucus and (c) muscular contraction. In the eyes and nose this produces itching, redness and lacrimation or rhinorrhea, in short, the symptoms of hay fever. In the lungs it causes, in addition to the secretory response, a contraction of the smooth muscles of the bronchi resulting in bronchial asthma.

It is commonly known that non-specific environmental factors such as dust, irritating gases, change of temperature and humidity may precipitate asthmatic attacks in allergic subjects, even in the absence of exposure to specific allergens. It is assumed that the presence of frequent allergic bronchial constriction renders the smooth muscles of the bronchi so sensitive to various non-specific stimuli that the threshold of their response to such irritation is considerably lower than that of a non-allergic individual.

Air Conditioning Apparatus

Of all the measures to relieve a specifically sensitive individual, elimination of exposure to the responsible allergen is the most efficient, though not always a practical, form of treatment. In recent years considerable effort has been made to accomplish such elimination by removal of respiratory allergens from enclosures by filtration or other air conditioning processes.

Paper or cloth filters, mounted in inexpensive window or floor units, prove quite satisfactory in many cases, but since dust and smoke frequently cause asthmatic attacks, it is desirable that an air filter, to be of full value in the treatment of asthma, should remove all possible dusts and pollens regardless of size or amount. Electrostatic air cleaners are more efficient than most commonly used types for capturing very fine dust.

Although the chief remedial factor in the treatment by conditioned air is the filtration of pollen, a certain amount of cooling and dehumidification appears to be desirable. A comfortable temperature between 70 and 75 F, and a relative humidity well below 50 percent proved satisfactory.[45] Direct drafts, overcooling or overheating are apt to initiate or aggravate the symptoms.

Limitations of Air Conditioning Methods

The results obtained with air filtration, or other air conditioning processes, in the control of allergic conditions, are fairly comparable to those obtained by desensitization treatment, so long as the patients remain in the pollen-free atmosphere. For all practical purposes filtration gives only temporary relief. In mild cases sleeping in an air conditioned space may make it possible for the individual to pass more comfortable nights. With rare exceptions, the symptoms recur on exposure to pollen-laden air. Moreover, the usefulness of air conditioning methods is limited, because all cases are not caused by airborne substances. Cases of bacterial asthma do not respond to treatment with filtered air.

Despite these limitations, air conditioning methods possess definite advantages in the simplicity of treatment, convenience, and under certain conditions, almost immediate relief.[46] Pollen cases are usually relieved of most of their symptoms within 1 to 3 hr after exposure to properly filtered air. A pollen-free atmosphere is especially valuable when desensitization has given little or no relief, and when desensitization is not advisable owing to intercurrent illness.

OXYGEN THERAPY

Oxygen therapy is used to prevent or relieve anoxia. Some of the more important clinical conditions in which oxygen treatment is beneficial include pneumonia, severe anemia, cardiac decompensation, pulmonary atelectasis, asphyxia and asthma. The effectiveness of oxygen therapy is dependent on the concentration of the oxygen in the inspired air, or the partial pressure of oxygen in the pulmonary alveoli.

Oxygen is usually administered by nasal catheter, face mask or tent.[47] The necessity of air conditioning in oxygen therapy arises from the fact that oxygen is too expensive a gas to waste in the ventilation of oxygen tents and oxygen chambers. Air conditioning is applied to the oxygen tent or chamber through reconditioning of the atmosphere in a closed circuit. Excessive heat, moisture and carbon dioxide are removed.

Oxygen Tents

In oxygen tents, the air enriched with oxygen is usually circulated by means of a small motor blower which sends the air over soda lime to remove carbon dioxide, and then over ice to remove excess heat and moisture. The concentration of oxygen in the tent is regulated by means of a pressure reducing valve and flow meter. In an inadequately cooled tent, high temperatures and humidities are inevitable, increasing the discomfort of the patient and imposing an added strain on an already overburdened heart. Oxygen therapy under such conditions may do more harm than good. An ice melting rate of approximately 10 lb per hour gives satisfactory results in patients with fever in a medium size oxygen tent.

Oxygen tents are confining to the patient. They may terrify the restless and delirious patient. Medical and nursing care is complicated, as the tent must be opened or removed with attendant loss of oxygen. Oxygen concentrations of 50 percent or more are difficult to maintain, and it is a problem to keep the temperature and humidity low enough in hot weather. However, with attention to details, the patient can be made quite comfortable. In fact, during hot, humid weather an oxygen tent may be very valuable in controlling a patient's temperature, since the upper part of the body within the cooled tent loses heat rapidly.

Oxygen Chambers

The conventional oxygen chamber is an air-tight sheet metal enclosure of fire-proof construction, large enough to accommodate one or two patients. Trap doors or curtains are provided for the personnel, food and service, to avoid loss of oxygen. Glass windows in the ceiling and walls admit light from outside the chamber. The air conditioning system may be of the gravity type, or of the fan type using mechanical refrigeration or air drying agents.

The temperature and humidity requirement in oxygen therapy depends primarily upon the physical condition of the patient, and secondarily upon the type of disease. In pneumonias[48] prescribed conditions should be a temperature of 60 to 75 F, humidity 20–50 percent, moderate air movement, oxygen concentration of 50 percent, and carbon dioxide of less than one percent.

Oxygen in Aviation

An important application of the principle of oxygen therapy is in aviation. At the present time all high altitude military airplanes in this country are provided with gaseous oxygen equipment, and military personnel are required to utilize oxygen at all times while in flight above 15,000 ft, or between 12,000 to 15,000 ft for longer than two hours, or between 10,000 to 12,000 ft for longer than six hours. The use of oxygen in commercial aviation will depend on the height and duration of the flights, as well as the state of health of the passengers. The necessity for portable, comfortable equipment, the possible fire hazards due to smoking, and the use of oxygen on sleeper planes are some of the difficulties facing civil airline operators. The pressure cabin airplane is a solution to the problem.

GENERAL HOSPITAL AIR CONDITIONING

Complete conditioning of a large hospital involves a capital investment and running expenses which may not be justified. In clean and quiet districts, the requirements of almost all general and private wards during the cool season of the year can be satisfactorily fulfilled by the use of conventional heating equipment, in conjunction with window air supply and gravity or mechanical exhaust. Insulation against heat and sound is much more important than humidification in winter; it will also help in keeping the building cool in warm weather. Excessive outside noise and dust may require the use of silencers and air filters in the openings.

Cooling and dehumidification in warm weather are important. In new hospitals particularly, the desirability of cooling certain sections of the building should be given serious consideration. Financial reasons may preclude the cooling of the entire building, but the needs of the average hospital can be met by the use of built-in room coolers and a few portable units which can be wheeled from ward to ward when needed.

In the North, and certain sections of the Pacific Coast, cooling is needed but a few days during summer, while in the South, it can be used to advantage from May to October, and in tropical climates almost continuously throughout the year.

F. L. Grocott of the Anglo-Iranian Oil Co. states that in Iran, the medical staff after 10 years' experience with air conditioning, demand a uniform environment of 75 F and 50 percent relative humidity (70 ET) under all summer outside conditions for general wards and treatment rooms, and 70 F with 30 to 50 percent relative humidity (65–66 ET) for winter conditions. In the operating rooms, 70 F with 50 percent relative humidity (66 ET) is demanded all the year 'round, although the annual external range is 40 F to 120 F. No ill effects have been noted in the medical personnel, though they are exposed to changes from external to internal conditions many times daily. Temperature shock in either direction seems to create discomfort for a short interval, but if the individual is in good health, no injury results.[49]

Aside from comfort and recuperative power of the patients, cooling is of great assistance in the treatment of fevers in the new-born and in post-operative cases, in enteric disorders, fevers, heat stroke, heart failure, thyroid crisis, and in a variety of other ailments which often accompany summer heat waves.

Problem of Odors

The evacuation of battle casualties in aircraft and their subsequent hospitalization have stimulated efforts to minimize odors arising from draining wounds, old odorous casts, and gangrenous wounds. For aircraft, chemical sprays and vapors, perfumes, oxidizing gases and simple exhaust methods are unsatisfactory. An ideal deodorant would purify the air by means of odor adsorption so that subsequently the air can be recirculated.

Based upon the effectiveness of activated carbon commercially and industrially to adsorb odors, individual adsorption units have been used successfully. In hospital wards the question of superiority of adsorption methods for elimination of odors over other methods remains to be answered. The present status of the problem is that the commercial aspect is highly controversial.[50]

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CHAPTER 8

AIR CONTAMINANTS

Classification of Air Contaminants; Sizes of Airborne Particles; Air Pollution by Smoke, Ash and Cinders; Smoke Abatement and Air Pollution Control; Odor Nuisance; Maximum Allowable Concentrations of Industrial Air Contaminants; Flammable Gases and Vapors; Combustible Dusts; Atmospheric Pollen; Airborne Bacteria

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THE normal constituents of the earth's atmosphere are oxygen, nitrogen, carbon dioxide, water vapor, argon, small or negligible amounts of other inert gases, hydrogen, variable traces of ozone, and small quantities of microscopic and submicroscopic solid matter, sometimes called *permanent atmospheric impurities*. From the viewpoint of the air conditioning engineer, all other airborne substances may be termed *contaminants*. This term is applied preferably, however, to undesirable or chance impurities, since the occasion may arise for adding to the air controlled amounts of solid or gaseous diluents for the prevention of explosions; germicidal vapors or mists (aerosols) for bacteria control; masking substances for odor control; or a substitute for one of the normal gases, as, for example, when helium is used to replace nitrogen in atmospheres for compressed air workers or divers.

Control of the chemical quality of air is one of the functions of complete air conditioning, and some knowledge of the composition, concentration and properties of air contaminants under various circumstances is therefore essential.

Air contaminants arise from the normal processes of wear, erosion, windstorm, sea-spray evaporation, thermal disintegration, earthquake, volcanic eruption, combustion, manufacturing, transportation, agriculture, and the biochemical or biological processes of life. They are classified at various times as organic and inorganic, visible or invisible, microscopic or macroscopic, particulate or gaseous, toxic or harmless, beneficial or destructive. The following classification is based chiefly upon the origin or method of formation of air contaminants.

CLASSIFICATION OF AIR CONTAMINANTS

Dusts, Fumes, and Smokes are solid particulate air contaminants.

Dusts are solid particles projected into the air by natural forces, such as wind, volcanic eruption or earthquake, and by mechanical processes, such as crushing, grinding, milling, drilling, demolition, shovelling, conveying, screening, bagging and sweeping. Some of these forces *produce* dust from larger masses, while others simply *disperse* materials that are already pulverized. Generally, particles are not called *dust* unless they are smaller than about 100 microns. Dusts may be of *mineral* type, such as rock, ore, metal, sand; *vegetable*, such as grain, flour, wood, cotton, pollen; or *animal*, such as wool, hair, silk, feathers, leather.

Fumes are solid particles commonly formed by the condensation of vapors from normally solid materials such as molten metals. Metallic fumes generally occur as the oxides in air because of the highly reactive nature of finely divided matter. Fumes may also be formed by sublimation, distillation, calcination, or chemical reaction, whenever such processes create airborne particles predominately below the 1 micron size. Fumes permitted to *age* tend to flocculate into clumps or aggregates of larger size, thereby facilitating removal from air.

Smokes are the extremely small solid particles produced by incomplete combustion of organic substances such as tobacco, wood, coal, oil, tar and other carbonaceous materials. The term *smoke* is commonly applied to the mixture of solid, liquid and gaseous products of combustion, although the technical literature prefers to distinguish between such components as soot or carbon particles, fly-ash, cinders, tarry matter, unburned gases, and gaseous combustion products. The finest particulate constituents are much less than 1 micron in size, often in the range of 0.1 to 0.3 micron.

Mists and Fogs are liquid particulate air contaminants.

Mists are very small airborne droplets of materials that are ordinarily liquid at normal temperatures and pressures. They may be formed by atomizing, spraying, splashing, mixing, violent chemical reaction, electrolytic evolution of gas from a liquid, or escape of a dissolved gas upon release of pressure. Very small droplets expelled or atomized into the air by sneezing constitute mists containing microorganisms that become air contaminants.

Fogs are limited by some classifications to airborne droplets formed by condensation from the vapor state. This arbitrary distinction between mist and fog is of minor importance, as both terms are used to indicate the particulate state of airborne liquids (occasionally termed *aerosols*). *Fog nozzles* are so named because of their ability to produce extra fine droplets as compared to the mist from ordinary spray devices. The highly volatile nature of some liquids quickly reduces their airborne droplets from the mist to the fog range, and eventually to the vapor phase until the air becomes saturated with that liquid. Many droplets in fogs or clouds are microscopic and submicroscopic in size, and may be conceived as the transition state between the larger mists and the vapors.

Vapors and Gases are non-particulate air contaminants.

Vapors are the gaseous phase of substances that are either liquid or solid in their commonly known state, examples being gasoline, kerosene, benzene, carbon tetrachloride, mercury, iodine, camphor. Vapors may be changed to the solid or liquid form by increasing the pressure, decreasing the temperature or applying both processes simultaneously. They are removed from the air by condensation with less difficulty than are the gases.

Gases are normally formless fluids which tend to occupy a space or enclosure completely and uniformly at ordinary temperatures and pressures. The following substances qualify as gases: oxygen, nitrogen, carbon dioxide, carbon monoxide, hydrogen, ammonia, sulfur dioxide. Gases, likewise, may be solidified or liquefied by the proper control of temperature and pressure.

The preceding classification is not suitable for the airborne living organisms, which range in size from the submicroscopic viruses to the largest pollen grains, not considering the smallest insect life. Bacteria range from about 0.2 to 5 microns in size, fungus spores from 1 to 10 microns, and pollen from 5 to 150 microns.

SIZES OF AIRBORNE PARTICLES

Fig. 1 is a graphic tabulation of the properties of airborne solids and liquids arranged according to size on the micron scale. There are 25,400 microns in 1 inch.

Particles larger than 10 microns are unlikely to remain suspended in air currents of moderate strength, but settle out by gravity at speeds dependent upon the shape, size and specific gravity of the particle, wind velocity, orientation of the collecting surface, and topography. These larger particles are of major interest to the engineer in the solution of nuisance problems, but it is usually the smaller particles, or those below 10 microns, that remain in the air long enough to be of hygienic as well as economic significance.

Industrial dust particles are predominantly of the order of 1 micron in size. Tremendous numbers are also present in the submicroscopic range below 0.5 micron, but those below 0.1 micron are not believed at present to be of practical importance, possibly due to their exceedingly small mass in comparison with the balance of airborne matter. In fact, particles this small may become the *permanent* atmospheric impurities that have little, if any, opportunity of settling because of the continual motion imparted to them by air currents and the molecular activity of gases (Brownian Movement).

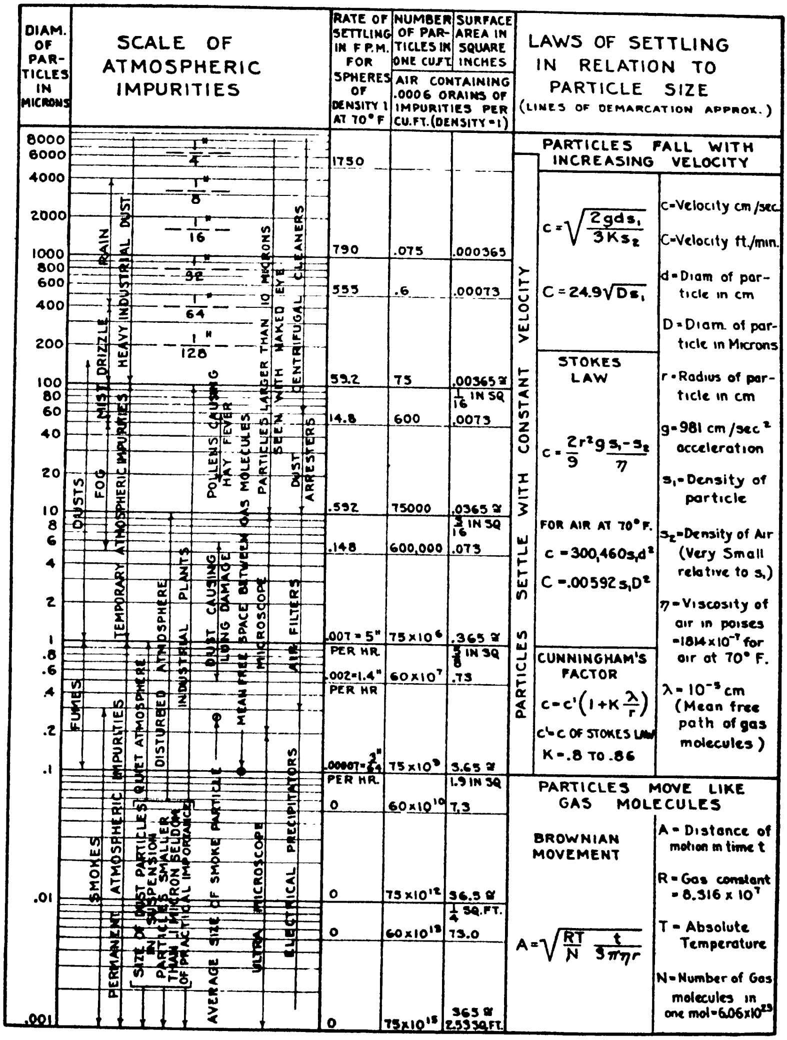

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FIG. 1. SIZES AND CHARACTERISTICS OF AIRBORNE PARTICULATE MATTER

The survey[1] of atmospheric pollution in 14 American cities conducted from 1931 to 1933 indicated the average size of outdoor dust particles to be 0.5 micron, as collected by the Owens jet dust counter and measured under the microscope. Inability of the *light field* microscope to reveal particles in the 0.1 micron vicinity may have influenced the determination of average particle size.

The lower limit of particle size visible to the naked eye cannot be stated definitely. It depends not only upon the individual eye, but also upon the shape and color of the particle, intensity and quality of the light, and nature of the background or opportunity for contrast. Under ideal conditions a particle of 10-micron size might be recognized, while under less favorable conditions it may be impossible to distinguish a particle smaller than 50 microns. The lower limit of visibility probably ranges from 10 to 50 microns.

Dusts, powders and granular materials are frequently classified by reference to the size of screens used for separation. Particles above 40 microns are said to be the *screen sizes* and those below, the *sub-screen* or microscopic sizes. Approximate or theoretical sizes of particles corresponding to the mesh scale of the U. S. Standard Sieve Series are given in Table 1.

Microscopic examination of screened dust indicates that the average diameter of a sample of irregular particles may be substantially larger than the openings of the screen through which it has passed, if the particle shapes deviate considerably from the spherical form.[2] The smallest dimension of many such particles will correspond with the maximum permissible distance between the wires of commercial screens made to *ASTM* Standard Specifications. Screening does not give sharp separation into size groups, and accordingly, such a classification is statistical rather than absolute.

TABLE 1. RELATION OF SCREEN MESH TO PARTICLE SIZE

U. S. Standard Sieve Mesh	400	325	200	140	100	60	35	18
Nominal Sieve Opening in Microns	37	44	74	105	149	250	500	1000

AIR POLLUTION BY SMOKE, ASH AND CINDERS

Total airborne solids settling in urban areas are usually reported as *soot fall* in tons per (square mile) (month). Such data published for the cities in this country range from 20 to 200 tons per (square mile) (month). To the air conditioning engineer this information may indicate the effectiveness of smoke abatement or fuel combustion control methods in his locality, but it does not provide a suitable index of the suspended dust that air cleaners in a ventilating system are expected to capture.[3, 4, 5] Gravimetric or weight data of the type given in Table 2 are preferable. In some cases airborne particle counts may be necessary, as for pollen, bacteria, spores, and insoluble dusts causing illness or lung disease.

Dust concentrations by weight cannot be converted readily to concentrations by particle count because of the variability of particle size, shape and specific gravity, and the inherent characteristics of dust counting and weighing procedures. One milligram of dust per cubic meter of air may represent dust counts from 1 million to 100 million particles per cubic foot of air (lightfield microscope technic) according to the size distribution of the airborne dust sample. Information of this type for a specified application is best obtained by simultaneous sampling for both counting and weighing and noting carefully at the time all factors that might affect the reproducibility of the count-weight ratio.

Smoke Abatement and Air Pollution Control

Successful abatement of atmospheric pollution caused by smoke requires the combined efforts of the combustion engineer, industrial executive, public health officer, city planning commission and the community at large. Electrification of industry and railroads, increase in the use of domestic oil and gas furnaces, and segregation of industrial districts are gradually providing effective aid in the solution of this problem. In the large cities where nuisance from smoke, fly-ash and cinders is more serious, limited areas obtain some relief by the use of district heating. (See Chapters 13, 14 and 15 for further discussion on fuel burning technic.)

TABLE 2. DUST CONCENTRATION RANGES

LOCATION	GRAINS PER 1000 CU FT[a]	MILLIGRAMS PER CUBIC METER
Rural and suburban districts	0.02–0.2	0.05–0.5
Metropolitan district	0.04–0.4	0.1 - 1.0
Industrial districts	0.1 -2.0	0.2 - 5.0
Ordinary factories or workrooms	0.2 -4.0	0.5 -10
Excessively dusty factories or mines	4–400	10–1000
Minimum explosive concentrations	4000–200,000	10,000–500,000

[a] 1 grain per 1000 cu ft = 2.3 milligrams per cubic meter.
1 oz per cubic foot = 1 gram per liter = 1000 grams per cubic meter

Many present ordinances limit the number of minutes in any one hour that smoke of a specified density (determined by comparison with a Ringelmann Chart which is described in Chapter 49) may be discharged.

There is now considerable interest and activity in the control of air pollution factors in addition to smoke. Difficulty in the establishment of acceptable criteria for certain corrosive and irritant gases, such as fluorides and the oxides of sulphur and nitrogen discharged with the gases of combustion, and the frequently complicated technical and economic problems encountered in control, have delayed the drafting and enforcement of legislative measures. Recent reports of an increased incidence of diseases, such as pneumonia and lung cancer, in areas high in certain air contaminants, require further critical investigation before acceptance. The values finally adopted will undoubtedly be lower than the M.A.C. (Maximum Allowable Concentration) limits for use in industry, because the exposure is continuous compared with the 8-hour day, 5 or 6-day week upon which M.A.C. values are based, and because the exposed population contains individuals with greater variation in age and health status.

In foggy weather, or with an *inversion* of atmospheric conditions, accumulation of gaseous contaminants may cause irritation of eyes, nose, and respiratory passages, and possibly cause even more serious physiological effects. The Meuse Valley fog disaster (Belgium 1930) and the Donora *smog* (Pennsylvania 1948) are classic examples in the history of gaseous air pollution. In both instances it is believed that irritant gases, principally from industrial plants, accumulating during periods of exceptionally prolonged meteorological *inversion* and fog, contributed to the illness of many persons, and to the death of some who were especially susceptible.

Absorption of Solar Radiation

Absorption of solar ultraviolet light by smoke and soot is recognized as a health problem in many industrial cities. Measurements of solar radiation in Baltimore[6] by actinic methods demonstrated that ultraviolet light intensity in the country was 50 percent greater than in the city. In New York City[7] a loss as great as 50 percent in visible light was found by photoelectric measurements.

ODOR NUISANCE

A problem companionate with smoke abatement is the control of odor nuisance in the neighborhood of industrial plants discharging noxious or offensive air contaminants. Community planning and zoning will avoid much of the difficulty in the future, but meanwhile many industrial cities must resort to corrective measures by requiring installation of air cleaning devices, alteration of manufacturing processes, or termination of the offensive operation in residential or commercial districts.

Control of outdoor odor nuisance is especially troublesome because of the extremely minute quantities of contaminant that are capable of offending through a wide area. New industrial chemicals with strange or unfamiliar odors tend to receive more attention from the neighborhood than the customary odors generated by well known processes and raw materials. Methods of odor control currently in use include charcoal adsorption, scrubbing towers and air washers, chlorination, condensation, masking, passage of the odorous air through combustion chambers, dispersion through a tall stack, and best of all, substitution of less offensive materials whenever possible.[8, 9, 10, 11]

Control of air quality within buildings ventilated for human occupancy is discussed in Chapter 6. Tobacco smoke odors, cooking odors and body odors are air contaminants of the nuisance type which now command a decisive position in the standards of air quality for indoor comfort. However, the engineer will find, at times, that odors originating *outside* buildings in industrial or business districts may determine the kind and capacity of equipment he must provide for a high quality air supply installation.

INDUSTRIAL AIR CONTAMINANTS

Many industrial processes are sources of contaminants. Their control is an important function of the ventilating or air conditioning engineer, because the atmosphere within buildings is the medium whereby such finely divided matter is dispersed and transported from the source to remote locations where it may cause property damage, nuisance, fire, explosion, disease and even death.

Tables 3, 4 and 5 give the *maximum allowable concentration* values for many industrial air contaminants as adopted by the Sectional Committee Z37 on Allowable Concentrations of Toxic Dusts and Gases of the *American Standards Association*, and as reviewed annually by the Committee on Threshold Limits and adopted by the *American Conference of Governmental Industrial Hygienists* at its 1951 meeting.

It must be emphasized that these limits in the great majority of instances are only *suggested maximum working levels* since they are estimates, based in many cases upon incomplete environmental and medical studies. Where there is agreement between the *ASA* Standard Z37 and *A.C.G.I.H.* values, reliability of the *ASA* Standard is increased. The values are not fixed, but are subject to revision, upward or downward, with the development of new information. There is by no means complete agreement regarding these values among responsible industrial hygienists.

In applying these guide limits, the following factors must be considered:

1. The duration of exposure is 8 hr a day for 5 or 6 days a week.

2. The measurements are indicative of the concentration in the breathing zone of the exposed person.

3. The M.A.C. is usually accepted as the average exposure value when the upper limits do not greatly exceed the M.A.C. value. For example, it cannot be assumed that if 100 ppm is considered *safe* for 8-hour exposure, that 800 ppm for one hour will be permissible.

4. Two substances with similar M.A.C. values may be quite different as to physiological effects at other concentrations. In one instance the selection of a limit may be predicated upon a discomfort factor, with a wide margin of safety before systemic effects would be encountered, while in another case the value represents an appreciable fraction of the concentration which may be associated with severe and irreversible injury.

5. These values are *upper* limits; it is desirable to operate well below the levels if the engineering and economic factors permit. The prudent engineer will incorporate a reasonable margin of safety in his estimates of ventilation capacity.

In Table 3, Column 1 lists the *ASA* Sectional Committee Z37 M.A.C. values; Column 2, those of the *A.C.G.I.H.* Committee on Threshold Limits. Column 3 gives the value in grams per cubic meter or ounces per 1000 cu ft for the corresponding *A.C.G.I.H.* limits; Column 4 shows the liquid ounces of chemical, if liquid at 20 C (68 F), which, if equally dispersed in 1000 cu ft, would give the corresponding M.A.C.

In Table 4, Columns 1 and 2 are respectively the M.A.C. values from the *ASA* and *A.C.G.I.H.* Committees.

In Table 5, the M.A.C. values for a number of industrial dusts as given were obtained from data of the *A.C.G.I.H.* Committee on Threshold Limits.

Information on the properties and effects, with respect to health, of specific industrial air contaminants is available in publications listed at the end of this chapter.

FLAMMABLE GASES AND VAPORS

Adequate ventilation is a primary requirement for minimizing the hazard of fire or explosion due to gases and vapors. The need for good ventilation is not removed by the use of other precautions, such as the elimination of known ignition sources, segregation of hazardous operations, adoption of safe building construction, and installation of automatic alarms. Some safety engineers regard over-ventilation of an operation employing flammable liquids as a legitimate operating charge for the privilege or necessity of using a dangerous process. However, it is not possible to

TABLE 3. MAXIMUM ALLOWABLE CONCENTRATION OF GASES AND VAPORS

SUBSTANCE	ASA STANDARDS M.A.C. ppm by volume	THRESHOLD LIMIT VALUES *A.C.G.I.H.*[a] 1951 ppm	Gm/cu m or Oz/1000 cu ft	Oz/1000 cu ft[b]
Acetaldehyde		200	0.36	0.43
Acetic acids		10	0.02554	0.02
Acetic anhydride		5	0.02085	0.02
Acetone		500	1.2065	1.21
Acrolein		0.5	0.1145	0.013
Acrylonitrile		20	0.04336	0.05
Ammonia		100		
Amyl acetate		200	1.064	1.16
iso-Amyl alcohol		100	0.36	0.41
Aniline		5	0.019	0.02
Arsine		0.05		
Benzene (benzol)	100	35	0.11393	0.12
Bromine		1	0.00653	0.03
1,3-Butadiene		1000	2.210	3.3
n-Butanol		50	0.1515	0.18
2-Butanone		250	0.735	0.88
n-Butyl acetate		200	0.948	1.04
Butyl "cellosolve"		200	0.966	1.02
Carbon dioxide		5000		
Carbon disulfide	20	20	0.0622	0.05
Carbon monoxide	100	100		
Carbon tetrachloride		50	0.313	0.19
"Cellosolve"		200	0.736	0.76
"Cellosolve" acetate		100	0.54	0.53
Chlorine		1		
2-Chlorobutadiene		25	0.0905	0.09
Chloroform		100	0.488	0.31
1-Chloro-1-nitropropane		20	0.101	0.08
Cyclohexane		400	1.375	1.69
Cyclohexanol		100	0.409	0.42
Cyclohexanone		100	0.401	0.41
Cyclohexene		400	1.34	1.58
Cyclopropane (propene)		400	0.6875	0.91
o-Dichlorobenzene		50	0.3005	0.24
Dichlorodifluoromethane		1000	4.94	3.18
1,1-Dichloroethane		100	0.405	0.33
1,2-Dichloroethane (ethylene dichloride)		75	0.3038	0.23
1,2-Dichloroethylene		200	0.794	0.59
Dichloroethyl ether		15	0.0878	0.08
Dichloromethane		500	1.74	1.25
Dichloromonofluormethane		1000	4.20	2.83
1,1-Dichloro-1-nitroethane		10	0.0589	0.4
1,2-Dichloropropane (propylene dichloride)		75	0.3265	0.29
Dichlorotetrafluoroethane		1000	6.98	4.36
Dimethylaniline		5	0.02475	0.02
Dimethylsulfate		1	0.00515	0.003
Dioxane		100	0.36	0.33
Ethyl acetate		400	1.44	1.52
Ethyl alcohol		1000	1.881	2.2
Ethyl benzene		200	0.868	0.96
Ethyl bromide		200	0.892	0.59
Ethyl chloride		1000	2.64	2.75
Ethylene chlorhydrin		5	0.01645	0.015
Ethylene oxide		100	0.18	0.09
Ethyl ether		400	1.212	1.67
Ethyl formate		100	0.303	0.315
Ethyl silicate		100	0.851	0.87
Formaldehyde	10	5	0.006135	
Gasoline		500	2.045	2.89
Heptane		500	2.05	2.87
Hexane		500	1.76	2.54
Hydrogen chloride		5	0.00746	0.004
Hydrogen cyanide		10	0.01104	0.015
Hydrogen fluoride		3	0.00245	0.003

TABLE 3. MAXIMUM ALLOWABLE CONCENTRATION OF GASES AND VAPORS (Concluded)

SUBSTANCE	ASA STANDARDS M.A.C. ppm by volume	THRESHOLD LIMIT VALUES *A.C.G.I.H.*[a] 1951		
		ppm	Gm/cu m or Oz/1000 cu ft	Oz/1000 cu ft[b]
Hydrogen selenide		0.05		
Hydrogen sulfide	20	20		
Iodine		1	0.01039	0.0002
Isophorone		25	0.141	0.15
Mesityl oxide		50	0.2005	0.22
Methanol	200	200	0.2618	0.32
Methyl acetate		200	0.606	0.62
Methyl bromide		20	0.0778	0.04
Methyl butanone		100	0.352	0.41
Methyl "cellosolve"		25	0.7775	0.078
Methyl "cellosolve" acetate		25	0.1208	0.11
Methyl chloride	100	100	0.2086	0.22
Methyl cyclohexane		500	2.005	2.44
Methyl cyclohexanol		100	0.466	0.49
Methyl cyclohexanone		100	0.458	0.48
Methyl formate		100	0.2554	0.25
Methyl iso-butyl ketone		100	0.409	0.49
Monochlorobenzene		75	0.3465	0.3
Monofluorotrichloromethane		1000	5.61	3 59
Mononitrotoluene		5	0.028	0.23
Naphtha (coal tar)		200	0.638	0.71
Naphtha (petroleum)		500	1.945	2.6
Nickel carbonyl		1	0.00697	0.006
Nitrobenzene		1	0.00503	0.004
Nitroethane		100	0.307	0.275
Nitrogen oxides (other than nitrous oxide)	25	25		
Nitroglycerine		0.5	0.00464	0.003
Nitromethane		100	0.2495	0.21
2-Nitropropane		50	0.182	0.17
Octane		500	2.33	3.17
Ozone		1		
Pentane		1000	2.94	4.5
Pentanone (methyl propanone)		200	0.704	0.82
Phosgene		1		
Phosphine		0.05		
Phosphorus trichloride		0.5	0.00281	0.0018
iso-Propanol		400	1.022	1.22
Propyl acetate		200	0.834	0.885
iso-Propyl ether		500	2.085	2.75
Stibine		0.1		
Stoddard solvent		500	2.845	3.4
Styrene monomer	400	200	0.85	0.89
Sulfur chloride		1	0.00552	0.003
Sulfur dioxide		10		
1,1,2,2-Tetrachloroethane		5	0.343	0.02
Tetrachloroethylene		100	0.679	0.405
Toluene	200	200	0.752	0.83
Toluidine		5	0.0219	0.02
Trichloroethylene	200	100	0.536	0.35
Turpentine		100	0.556	0.58
Vinyl chloride (chloroethane)		500	1.28	1.33
Xylene	200	200	0.868	0.96

[a] A.C.G.I.H. = American Conference of Governmental Industrial Hygienists.
[b] Liquid ounces (at 20 C) of chemical in 1000 cu ft.

apply a reasonable safety factor to the ventilation estimate without consideration of the concentrations of gases or vapors that approach the danger point. Safety engineers prefer to limit the concentration to $\frac{1}{4}$ or $\frac{1}{5}$ of the lower explosive limit, and this fact should be given full weight in determining the capacity and design of ventilating equipment. Rarely should consideration be given to *operation above the upper explosive limit* in the open areas of buildings or rooms—even though unoccupied—because the danger of temporary drop of gas concentration to a point within the explosive range is too great.

Ability of a flammable liquid to form explosive mixtures is determined largely by its vapor pressure, volatility, or rate of evaporation. *Flash point* is a convenient method of expressing this property in terms of the temperature scale. It may be defined as the temperature to which a combustible liquid must be heated to produce a flash when a small flame is passed across the surface of the liquid. The higher the flash point, the more safely can the liquid be handled. Liquids with flash points under 70 F should be regarded as highly flammable.

TABLE 4. LIMITS FOR TOXIC DUSTS, FUMES AND MISTS

SUBSTANCE	A.S.A. STANDARDS, M.A.C. mg/cu m	THRESHOLD LIMIT VALUES, *A.C.G.I.H.* 1951 mg/cu m
Antimony		0.5
Arsenic		0.5
Barium		0.5
Cadmium	0.1 (W)	0.1
Chlorodiphenyl		1
Chromic acid & chromates (As CrO_3	0.1	0.1
Cyanide as CN		5
Dinitrotoluene		1.5
Fluorides		2.5
Iron Oxide fume		15
Lead	0.15	0.15
Magnesium oxide fume		15
Manganese	6	6
Mercury	0.1	0.1
Pentachloroanphthalene		0.5
Pentachlorophenol		0.5
Phosphorus (yellow)		0.1
Phosphorus pentachloride		1
Phosphorus pentasulfide		1
Selenium compounds as selenium		0.1
Sulfuric acid		1
Tellurium		0.1
Tetryl		1.5
Trichloronaphthalene		5
Trinitrotoluene		1.5
Zinc oxide fumes		15

Upper and lower limits of flammability of gases and vapors, and the flash points of the corresponding liquids are given in Table 6.

Methods for estimating the flammable limits of mixtures of gases or vapors must be applied with caution; the reader is referred to other publications for this information.[12, 13]

Design of equipment for the control of combustible anesthetics is outlined in Chapter 7. Construction of equipment for handling air containing flammable substances, or operating in atmospheres so contaminated, is discussed in Chapter 45.

It is customary to report the concentrations of flammable gases or vapors in percent by volume, or *volume percent* Comparison with concentrations on the *part per million* scale used in chemical, medical or industrial hygiene literature is readily made by the conversion: 1 percent = 10,000 ppm (parts of contaminant per million parts of air, by volume, or in other words, cubic feet of contaminant per million cubic feet of air). It will be noted in Table 6 that nearly all of the substances listed have lower explosive limits above 1.0 percent, while the *maximum allowable concentrations* for gases and vapors in Table 3 are below 1000 ppm or 0.1 percent in most cases. Therefore, control of toxic or injurious vapors to levels below their maximum allowable concentrations for health usually requires much more effective ventilation than for the prevention of a fire hazard.

COMBUSTIBLE DUSTS

A dust explosion is essentially a sudden pressure rise caused by the very rapid burning of airborne dust. The primary explosion often originates from a small amount of dust in suspension exposed to a source of ignition, and the pressure and vibration it creates may be sufficient to dislodge large accumulations of dust on horizontal ledges or surfaces of the building and equipment, thereby creating a secondary explosion of great force.

TABLE 5. LIMITS FOR MINERAL DUSTS

SUBSTANCE	THRESHOLD LIMIT VALUES *A.C.G.I.H.* 1951 mppcf[a]
Alundum	50
Asbestos	5
Carborundum	50
Dust (nuisance, no free silica)	50
Mica	20
Portland cement	50
Silica—high (above 50% free SiO_2)	5
Silica—medium (5 to 50% free SiO_2)	20
Silica—low (below 5% free SiO_2)	50
Slate (below 5% free SiO_2)	50
Soapstone (below 5% free SiO_2)	20
Talc	20
Total dust (below 5% free SiO_2)	50

[a] mppcf—million particles per cubic foot of air, standard light field count.

Thus the air conditioning engineer is involved for two reasons: (1) to obtain a movement of dust-laden air into exhaust hoods or openings, and through ventilating or pneumatic conveying ducts, in a manner that will prevent accumulation of highly flammable dust at points where it could ignite inside the equipment; and (2) to so design *process ventilation* as to prevent the escape of dust which might settle on horizontal surfaces and become a potential source of disaster at some distance from the dusty operation. (See Chapter 45).

Intensity of a dust explosion depends upon: chemical and thermal properties of the dust; particle size and shape; concentration in air; proportion of inert dust in the air; moisture content and composition of the air; size and temperature of the ignition source; and degree of dispersion of the dust cloud. Investigations on the explosibility of dusts require determination of the maximum pressure developed during explosion of a known air concentration, as well as determination of the rate of pressure rise. Investigators frequently experience difficulty in obtaining dust suspensions of uniform dispersion, and this should be kept in mind when comparing results from several sources.[14]

Minimum explosive concentrations of airborne dusts already tested range from 0.01 to 0.5 oz per cubic foot, or 10 to 500 grams per cubic meter

Table 6. Approximate Limits of Flammability of Single Gases and Vapors In Air at Ordinary Temperatures and Pressures[a]

Gas or Vapor	Lower Limit Percent by Volume	Upper Limit Percent by Volume	*Closed Cup*[b] Flash Point Fahrenheit
Acetyldehyde	4.0	57	−17
Acetone	2.5	12.8	0
Acetylene	2.5	80	....
Allyl alcohol	2.5		70
Ammonia	15.5	26.6	....
Amyl alcohol	1.2		100
Amyl chloride	1.6		...
Amylene	1.6	7.7	....
Benzene (benzol)	1.3	6.8	12
Benzyl chloride	1.1	.	140
Butane	1.8	8.4	....
Butyl acetate	1.4	15.0	84
Butyl alcohol	1.7		....
Butylene	2.0	9.7	....
Carbon disulfide	1.2	50	−22
Carbon monoxide	12.5	74.2	....
Crotonaldehyde	2.1	15.5	55
Cyclohexane	1.3	8.4	1
Cyclopropane	2.4	10.5	....
Decane	0.67	2.6	115
Dichloroethylene (1, 2)	9.7	12.8	57
Diethyl selenide	2.5		...
Dioxane	2.0	22.2	54
Ethane	3.1	12.5	
Ether (diethyl)	1.8	36.5	−49
Ethyl acetate	2.2	11.5	28
Ethyl alcohol	3.3	19.0	54
Ethyl bromide	6.7	11.3	...
Ethyl cellosolve	2.6	15.7	104
Ethyl chloride	4.0	14.8	−58
Ethylene	2.7	28.6	.
Ethylene dichloride	6.2	15.9	56
Ethyl formate	2.7	16.5	−4
Ethyl nitrite	3.0	50	−31
Ethylene oxide	3.0	80	....
Furfural (125 C)	2.1		140
Gasoline (variable)	1.4-1.5	7.4-7.6	−50
Heptane	1.0	6.0	25
Hexane	1.2	6.9	−15
Hydrogen cyanide	5.6	40.0	.
Hydrogen	4.0	74.2	...
Hydrogen sulfide	4.3	45.5	...
Illuminating gas (coal gas)	5.3	33.0	..
Isobutyl alcohol	1.7	..	82
Isopentane	1.3	....	....
Isopropyl acetate	1.8	7.8	43
Isopropyl alcohol	2.0	.	53
Methane	5.0	15.0	.
Methyl acetate	3.1	15.5	14
Methyl alcohol	6.7	36.5	52
Methyl bromide	13.5	14.5	....
Methyl butyl ketone	1.2	8.0	....
Methyl chloride	8.2	18.7	....
Methyl cyclohexane	1.1	.	25
Methyl ethyl ether	2.0	10.1	−35
Methyl ethyl ketone	1.8	9.5	30
Methyl formate	5.0	22.7	−2
Methyl propyl ketone	1.5	8.2	....
Natural gas (variable)	4.3	13.5	....
Naphtha (benzine)	1.1	6.0	20–110
Naphthalene	0.9	..	176
Nonane	0.83	2.9	88
Octane	0.95	3.2	56
Paraldehyde	1.3		....
Pentane	1.4	7.8	....

TABLE 6. APPROXIMATE LIMITS OF FLAMMABILITY OF SINGLE GASES AND VAPORS IN AIR AT ORDINARY TEMPERATURES AND PRESSURES[a] (Concluded)

GAS OR VAPOR	LOWER LIMIT PERCENT BY VOLUME	UPPER LIMIT PERCENT BY VOLUME	*Closed Cup*[b] FLASH POINT FAHRENHEIT
Propane	2.1	10.1	
Propyl acetate	1.8	8.0	58
Propyl alcohol	2.1	13.5	59
Propylene	2.0	11.1	
Propylene dichloride	3.4	14.5	60
Propylene oxide	2.0	22.0	
Pyridine	1.8	12.4	74
Toluene (toluol)	1.3	7.0	40
Turpentine	0.8		95
Vinyl ether	1.7	27.0	
Vinyl chloride	4.0	21.7	
Water gas (variable)	6.0	70	
Xylene (xylol)	1.0	6.0	63

[a] Adapted from: Fire and Explosion Hazards of Combustible Gases and Vapors, by G. W. Jones; Chapter 13, Industrial Hygiene and Toxicology, edited by F. A. Patty (Interscience Publishers, 1948); Properties of Flammable Liquids, Gases and Solids (Associated) Factory Mutual Fire Ins. Cos., January (1940); and National Fire Codes for Flammable Liquids, Gases, Chemicals and Explosives—1945 (*National Fire Protection Association*).

[b] *Closed cup* refers to the equipment used in flash point determinations.

of air. Maximum pressures generated have been reported as high as 500 psi, although they are more likely to be of the order of 50 psi. Investigations on the flammable characteristics of dusts are currently made at 0.1 and 0.5 oz. per cubic foot.[15-21]

ATMOSPHERIC POLLEN

Properties of pollen grains discharged by weeds, grasses and trees and responsible for hay fever, are of special interest to designers of air cleaning equipment (see Allergic Disorders in Chapter 7, and Air Cleaning, Chapter 33). Whole grains and fragments transported by the air range chiefly between 10 and 50 microns in size, but some have been measured as small as 5 microns, and others over 100 microns in diameter. Ragweed pollen grains are fairly uniform in size within the range of 15 to 25 microns. Pollen grains can be removed from the air more readily than the particles of dust prevalent in outdoor air or produced by dusty processes, since the latter predominate in the range of 0.1 to 10 microns in size.

Most grains are quite hygroscopic and therefore vary in weight with the humidity. Illustrations and data on individual pollen grains are available in the botanical literature[22, 23, 24] Geographical distribution of plants known to produce hay fever is also recorded.[25, 26]

The quantity of pollen grains in the air is generally estimated by exposing an adhesive-coated glass plate outdoors for 24 hr, and then counting calibrated areas under the microscope. Methods are available for determining the number of grains in a measured volume of air,[25, 27, 28] but their greater accuracy has not caused them to replace the more simple *gravity slide method* used for most pollen counts. Counting technics vary somewhat, but the daily pollen counts reported in local newspapers during the hay fever season usually represent the number of grains found on 1.8 sq cm of a 24-hr *gravity slide*.

Hay fever sufferers may notice the first symptoms when the pollen count is 10 to 25, and in some localities the maximum figures for the seasonal peak may approach 1000 for a 24-hr period, depending upon the sampling and reporting methods of the laboratory. Translation of gravity counts by special formulas to a volumetric basis, or the number of grains per cubic yard or per cubic foot of air, is unreliable because of the complexity of the modifying factors. When such information is important, it is best obtained directly by a volumetric instrument. The number of pollen grains per cubic yard of air evidently varies from 2 to 20 times the number found on 1 sq cm of a 24-hr gravity slide, depending on grain diameter, shape, specific gravity, wind velocity, humidity and physical placement of the collecting plate.[29, 30, 31]

AIRBORNE BACTERIA

Study of the occurrence and significance of micro-organisms in the atmospheres of the indoor world is absorbing the energies of a substantial number of physicians, bacteriologists, aerobiologists, physicists, public health workers, engineers and hospital personnel. Some data are available on the types and quantities of bacteria found in a variety of spaces, but it is not possible at present to use this information as a conclusive index of the potential health hazard of a given environment. The reported number of airborne organisms may vary from 1 to 1000 per cubic foot of air, influenced somewhat by the method of testing.[32] Many are attached to the dust particles present in the air.

Where it seems advisable or desirable to control the bacterial content of rooms, public conveyances or buildings, highly effective methods are available (see Chapter 7), and their extended use may do much to assist the workers in this field in accumulating the necessary mass of evidence that will decide the practical value of air sterilization for the control of communicable disease. It is now well established that ultraviolet radiation is feasible for the protection or preservation of pharmaceuticals, cosmetics, and food products.

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[26] Hay Fever: A Geographical and Botanical Survey, by E. R. Squibb and Sons, New York, 1937.

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[29] The Volumetric Incidence of Atmospheric Allergens, by O. C. Durham (*Journal of Allergy*, Vol. 14, September, 1943, p. 455–461).

[30] The Volumetric Incidence of Atmospheric Allergens, II: Simultaneous Measurements by Volumetric and Gravity Slide Methods, by O. C. Durham (*Journal of Allergy*, Vol. 15, May, 1944, p. 226–235).

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[32] Sampling Devices, by H. G. DuBuy and A. Hollaender (*American Journal of Medical Science*, Vol. 209, February, 1945, p. 172–177).

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Toxicology and Hygiene of Industrial Solvents, by Lehmann and Flury, translated by Eleanor King and H. F. Smyth, Jr. (Williams and Wilkins, Baltimore, 1943).

CHAPTER 9

HEAT TRANSMISSION COEFFICIENTS OF BUILDING MATERIALS

Heat Transfer Symbols; Calculating Overall Coefficients; Conductivity of Homogeneous Materials; Soil Conductivity; Surface Conductance; Air Space Conductance; Practical Coefficients and Their Use; Computed Coefficients of Walls, Roofs, Ceilings and Floors; Combined Ceiling and Roof Coefficients; Calculating Surface Temperatures; Water Vapor and Condensation; Vapor Transmission; Condensation Control

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THE design of air conditioning or heating systems for buildings requires a knowledge of the thermal properties of the walls enclosing the space. (The term *walls* in this case, includes windows, doors, ceilings, floors, roofs and skylights). The rate of heat flow through the walls under steady-state conditions at design temperatures is usually the basis for calculating the heat required. For a given wall under standard conditions the rate is a specific value designated as U, the *overall coefficient of heat transmission.* It may be determined by test in a guarded hot box apparatus, or it may be computed from known values of the thermal conductance of the various components. Because testing of all combinations of building materials is impracticable, the procedure and necessary data for calculation of the value of U are given in this chapter, together with tables of computed values for the more common constructions.

HEAT TRANSFER SYMBOLS

U = overall coefficient of heat transmission (air to air); the time rate of heat flow expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference between air on the inside and air on the outside of a wall, floor, roof or ceiling). The term is applied to the usual combinations of materials, and also to single materials, such as window glass, and includes the surface conductance on both sides.

k = thermal conductivity; the time rate of heat flow through a homogeneous material under steady conditions through unit area per unit temperature gradient in the direction perpendicular to the area. Its value is expressed in Btu per (hour) (square foot) (Fahrenheit degree per inch of thickness). Materials are considered homogeneous when the value of k is not affected by variation in thickness or size of sample within the range normally used in construction.

C = thermal conductance; the time rate of heat flow through a material from one of its surfaces to the other per unit temperature difference *between the two surfaces.* Its value is expressed in Btu per (hour) (square foot) (Fahrenheit degree). The term is applied to specific materials as used, either homogeneous or heterogeneous.

f = film or surface conductance; the time rate of heat flow between a surface and the surrounding air. Its value is expressed in Btu per (hour) (square foot of surface) (Fahrenheit degree temperature difference). Subscripts i and o are used to differentiate between inside and outside surface conductances, respectively.

a = thermal conductance of an air space; the time rate of heat flow through an air space per unit temperature difference *between the boundary surfaces.* Its value is expressed in Btu per (hour) (square foot of area) (Fahrenheit degree). The conductance of an air space is dependent on the temperature difference, the height, the depth, the position and the character of the boundary surfaces. The relationships are not linear, and accurate values must be obtained by test and not by computation.

R = thermal resistance. Its value is obtained from the reciprocal of heat transfer as expressed by U, k, C, f or a. It is expressed in (hours) (square feet) (Fahrenheit degrees) per (Btu). For example, a wall with a U value of 0.25 would have a

resistance value of $1/0.25 = 4.0$. Therefore, 4 hr would be required for the flow of one Btu for each square foot of area and each degree of temperature differential.

CALCULATING OVERALL COEFFICIENTS

From Chapter 5, Equation 7, the total resistance to heat flow through a wall is equal numerically to the sum of the resistances in series.

$$R_T = R_1 + R_2 + R_3 + R + \ldots + R_n \quad (1)$$

where, $R_1 + R_2$, etc., are the individual resistances of the wall components.

R_T = total resistance.

For a wall of a single homogeneous material of conductivity k and thickness x, with surface coefficients f_i and f_o,

$$R_T = \frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_o} \quad (2)$$

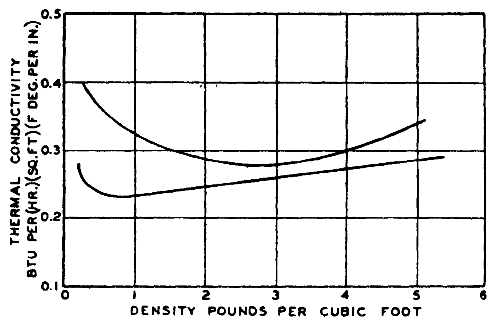


Fig. 1. Typical Variation of Thermal Conductivity with Density—for Fibrous Material

Then by definition, $U = 1/R_T$

For a wall with air space construction and consisting of two homogeneous materials of conductivities k_1 and k_2, thicknesses x_1 and x_2, respectively, and separated by an air space of conductance a,

$$R_T = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{1}{a} + \frac{x_2}{k_2} + \frac{1}{f_o} \quad (3)$$

and $U = 1/R_T$

In the case of types of building materials having non-uniform or irregular sections such as hollow clay tile or concrete blocks, it is necessary to use the conductance C of the section unit as manufactured instead of a conductivity k. The resistance of the section $1/C$ is therefore substituted for x/k in Equations 2 and 3.

CONDUCTIVITIES AND CONDUCTANCES

The method of calculating the overall coefficient of heat transmission for a given construction is comparatively simple, but accurate values of conductivities and conductances must be used to obtain satisfactory results. In addition, there are sometimes parallel heat flow paths of different resistances in the same wall, and these may necessitate modification of the formula. In such cases calculated results should be checked by test.

The determination of the fundamental conductivities and conductances requires considerable skill and experience to obtain accurate results. It is recommended that thermal conductivities of homogeneous materials be determined by means of the Guarded Hot Plate.[1] For determination of conductances, a Guarded Hot Box method[2] is generally used.

Tables 1 and 2 give conductivities and conductances which are quite generally used in calculation, and which have been selected from various sources. Wherever possible, the properties of the material and test conditions are given. In selecting and applying heat transmission values to any construction, caution is necessary, since coefficients for the same material may differ because of variations which occur in test methods, in the materials themselves, or in the temperature of the material when tested.

Conductivity of Homogeneous Materials

Thermal conductivity is a property of a homogeneous material and of types of building materials such as lumber, brick, and stone, which may be considered homogeneous. Most insulating materials, except reflective

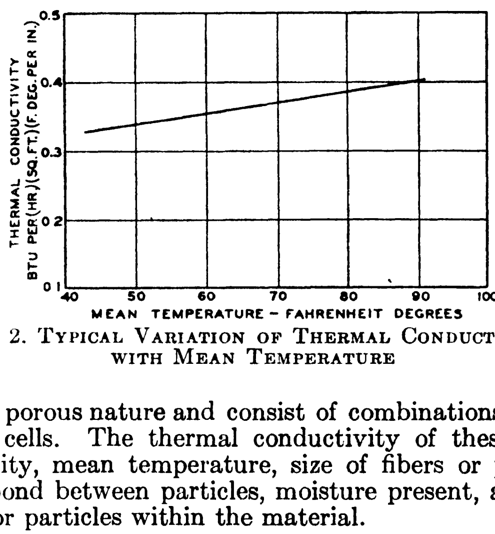


FIG. 2. TYPICAL VARIATION OF THERMAL CONDUCTIVITY WITH MEAN TEMPERATURE

types, are of a porous nature and consist of combinations of solid matter with small air cells. The thermal conductivity of these materials will vary with density, mean temperature, size of fibers or particles, degree and extent of bond between particles, moisture present, and the arrangement of fibers or particles within the material.

The effect of density upon conductivity (at constant mean temperature) is illustrated for two fibrous materials in Fig. 1. It will be noted that for each there is an optimum density for lowest conductivity. Typical variation of conductivity with mean temperature is shown in Fig. 2.

Thermal Conductivity of Soil

The following statements are based largely on results of a study[3] made in the *Engineering Experiment Station, University of Minnesota*, and published in *Bulletin* No. 28. Tests were made on nineteen different soils which represented a wide textural variety, including gravel, sand, sandy loam, silt loam and clay, as well as some crushed rocks and a fibrous peat. Moisture contents in tests varied from air-dried values to those greater than the optimum moisture content; densities varied from a loosely-poured condition to the maximum density obtainable by heavy ramming. The general findings of the investigation are as follows:

Effect of Temperature. Soils were tested at several mean temperatures. The degree of influence of temperature depends upon whether it is above or below freez-

Table 1. Conductances (C) for Surfaces and Air Spaces

All conductance values expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference)

Section A. Surface Conductances for Still Air[a]

Position of Surface	Direction of Heat Flow	Surface Emissivity $e = 0.83$	Surface Emissivity $e = 0.05$
Horizontal	Upward	1.95	1.16
Horizontal	Downward	1.21	0.44
Vertical		1.52*	0.74

Section B. Conductance of Vertical Spaces at Various Mean Temperatures[b]

Mean Temp Fahr Deg	Conductances of Air Spaces for Various Widths in Inches						
	0.128	0.250	0.364	0.493	0.713	1.00	1.500
20	2.300	1.370	1.180	1.100	1.040	1.030	1.022
30	2.385	1.425	1.234	1.148	1.080	1.070	1.065
40	2.470	1.480	1.288	1.193	1.125	1.112	1.105
50	2.560	1.535	1.340	1.242	1.168	1.152	1.149
60	2.650	1.590	1.390	1.295	1.210	1.195	1.188
70	2.730	1.648	1.440	1.340	1.250	1.240	1.228
80	2.819	1.702	1.492	1.390	1.295	1.280	1.270
90	2.908	1.757	1.547	1.433	1.340	1.320	1.310
100	2.990	1.813	1.600	1.486	1.380	1.362	1.350
110	3.078	1.870	1.650	1.534	1.425	1.402	1.392
120	3.167	1.928	1.700	1.580	1.467	1.445	1.435
130	3.250	1.980	1.750	1.630	1.510	1.485	1.475
140	3.340	2.035	1.800	1.680	1.550	1.530	1.519
150	3.425	2.090	1.852	1.728	1.592	1.569	1.559

Section C. Conductances and Resistances of Air Spaces Faced on One Surface with Reflective Insulation[c]

Location and Position of Air Space	Direction of Heat Flow	Temp[d] Diff Fahr Deg		Conductance[e] (C)			Resistance[e] $\left(\frac{1}{C}\right)$		
		Winter	Summer	No. of Air Spaces 1	2	3	No. of Air Spaces 1	2	3
Rafter Space (8 in.) Horizontal	Down	45			0.10	0.07		10.00	14.29
Horizontal	Up	45			0.27	0.17		3.70	5.88
Horizontal	Down		25		0.09	0.06		11.11	16.67
Horizontal	Up		25		0.24	0.16		4.17	6.25
30 deg slope	Down	45			0.15	0.10		6.67	10.00
30 deg slope	Up	45			0.25	0.17		4.00	5.88
30 deg slope	Down		25		0.13	0.09		7.69	11.11
30 deg slope	Up		25		0.23	0.14		4.35	7.14
Stud Space (3⅝ in.) Vertical[f]		30		0.34			2.94		
Vertical		40			0.23	0.13		4.35	7.69
Vertical[f]			15	0.32			3.13		
Vertical			20		0.18	0.11		5.56	9.09
Vertical[g]		30		0.46			2.17		

[a] Radiation and Convection from Surfaces in Various Positions, by G. B. Wilkes and C. M. F. Peterson (A.S.H.V.E. Transactions, Vol. 44, 1938, p. 513.

[b] A.S.H.V.E. Research Report No. 825—Thermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren (A.S.H.V.E. Transactions, Vol. 35, 1929, p. 165).

[c] Thermal Test Coefficients of Aluminum Insulation for Buildings, by G. B. Wilkes, F. G. Hechler and E. R. Queer (A.S.H.V.E. Transactions, Vol. 46, 1940).

[d] Temperature difference is based on total space between plaster base and sheathing, flooring or roofing.

[e] These air space conductance and resistance values are based on one reflective surface (aluminum) having an emissivity of 0.05 facing each space, and are based on total space between plaster base and sheathing, flooring or roofing. The rafter and stud spaces are divided into equal spaces.

[f] Stud space is lined on plaster base side with loose paper with aluminum on surface facing air space The resistance of the small air space between the plaster base and paper was 0.43.

[g] Radiation and Convection Across Air Spaces in Frame Construction, by G. B. Wilkes and C. M. F. Peterson (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 351).

* The recommended surface conductance for calculating heat losses for still air for non-reflective surfaces is 1.65 Btu. For a 15 mph wind velocity, the recommended value is 6.0 Btu. These coefficients were derived from Fig. 4, which was based on tests conducted at the *University of Minnesota*, and apply to vertical surfaces.

Table 2. Conductivities (k) and Conductances (C) of Building and Insulating Materials

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference). Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Fahr Deg)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
BUILDING BOARDS (Non-Insulating)	Compressed cement and asbestos sheets	118	119	4.1	—	0.24	—	(1)
	Corrugated asbestos board	20.4	110	0.48	—	2.08	—	(2)
	Pressed asbestos mill board	60.5	86	0.84	—	1.19	—	(3)
	Gypsum board—gypsum between layers of heavy paper	62.8	70	1.41	—	0.71	—	(3)
	3/8 in. gypsum board	—	—	—	3.73	—	0.27	
	1/2 in. gypsum board	—	—	—	2.82	—	0.35	
	5/8 in. gypsum board	53.5	90	—	2.60	—	0.38	(1)
FRAME CONSTRUCTION COMBINATIONS	1 in. fir sheathing and building paper	—	30	—	0.86	—	1.16	(4)
	1 in. fir sheathing, building paper and yellow pine lap siding	—	20	—	0.50	—	2.00	(4)
	1 in. fir sheathing, building paper and stucco	—	20	—	0.82	—	1.22	(4)
	Pine lap siding and building paper, siding 4 in. wide	—	16	—	0.85	—	1.18	(4)
	Yellow pine lap siding	—	—	—	1.28	—	0.78	(4)
MASONRY MATERIALS Brick	Damp or wet	—	—	5.0[e]	—	0.20	—	(2)
	Common yellow clay brick[a]	—	—	4.8	—	0.21	—	(4)
	One tier yellow common clay brick, one tier face brick, approx. 8 in. thick[a]	—	—	—	0.77	—	1.30	(4)
Clay Tile, Hollow	2 in. Tile, 5/8 in. plaster both sides	120.0	110	—	1.00	—	1.00	(2)
	4 in. Tile, 5/8 in. plaster both sides	127.0	100	—	0.60	—	1.67	(2)
	6 in. Tile, 5/8 in. plaster both sides	124.3	105	—	0.47	—	2.13	(2)
	8 in. Tile, average of 8 types (Walls No. 59, 63, 64, 66, 67, 90, 91, 92[a])	—	—	—	0.52	—	1.92	(4)
	12 in. Clay tile wall: 8 in. x 5 in. x 12 in. and 4 in. x 5 in. x 12 in.[a]	—	—	—	0.26	—	3.84	(4)

Authorities:

[1] *National Bureau of Standards*, tests based on samples submitted by manufacturers.

[2] A. C. Willard, L. C. Lichty and L. A. Harding, tests conducted at the *University of Illinois*.

[3] J. C. Peebles, tests conducted at *Armour Institute of Technology*, based on samples submitted by manufacturers.

[4] F. B. Rowley, et al, tests conducted at the *University of Minnesota*.

[5] A.S.H.V.E. Research Laboratory.

[6] E. A. Allcut, tests conducted at the *University of Toronto*.

[a] See Thermal Conductivity of Building Materials, by F. B. Rowley and A. B. Algren (*University of Minnesota Engineering Experiment Station Bulletin* No. 12).

[b] Heat Transmission Through Insulation as Affected by Orientation of Wall, by F. B. Rowley and C. E. Lund (A.S.H.V.E. Transactions, Vol. 49, 1943, p. 331).

[c] The Effect of Convection in Ceiling Insulation, by G. B. Wilkes and L. R. Vianey (A.S.H.V.E. Transactions, Vol. 49, 1943, p. 196).

[d] See A.S.H.V.E. Research Report No. 915—Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. Transactions, Vol. 38, 1932, p. 47).

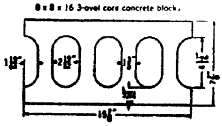
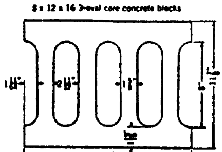
[e] See *Heating, Ventilating and Air Conditioning*, by Harding and Willard, revised edition, 1932.

[f] See BMS13, *U. S. Department of Commerce, National Bureau of Standards*, Washington, D. C.

[g] Roofing, 0.15 in. thick (1.34 lb per square foot), covered with gravel (0.83 lb per square foot), combined thickness assumed 0.25.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING AND INSULATING MATERIALS—Continued

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference). Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness.

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Fahr Deg)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
MASONRY MATERIALS —(Continued)								
CONCRETE	Sand and gravel aggregate, various ages and mixes	—	—	11.35 to 16.36	—	0.09 to 0.06	—	(5)
	Sand and gravel aggregate	142	75	12.6	—	0.08	—	(4)
	Limestone aggregate	132	75	10.8	—	0.09	—	(4)
	Cinder aggregate	97	75	4.9	—	0.22	—	(4)
	Steam treated limestone slag aggregate[a]	74.6	75	2.27	—	0.44	—	(4)
	Pumice (Mined in California) aggregate[a]	65.0	75	2.42	—	0.41	—	(4)
	Expanded burned clay aggregate[a]	59.9	75	2.28	—	0.44	—	(4)
	Burned clay aggregate[a]	67.1	75	2.86	—	0.35	—	(4)
	Blast furnace slag aggregate	76.0	70	1.6	—	0.63	—	(3)
	Expanded vermiculite aggregate	20	90	0.68	—	1.47	—	(3)
	Expanded vermiculite aggregate	26.7	90	0.76	—	1.32	—	(3)
	Expanded vermiculite aggregate	35	90	0.86	—	1.16	—	(3)
	Expanded vermiculite aggregate	50	90	1.10	—	0.91	—	(3)
	Expanded Vermiculite aggregate, 1:9.4 mix	26.3	119	0.97	—	1.03	—	(1)
	Expanded Vermiculite aggregate, 1:2.9 mix	45.6	119	1.60	—	0.63	—	(1)
	Perlite aggregate, 1:10.3 mix	24.4	119	0.75	—	1.33	—	(1)
	Perlite aggregate 1:2.9 mix	47.6	119	1.45	—	0.69	—	(1)
	Concrete plank	76	75	2.5	—	0.40	—	(3)
	Cellular concrete	40.0	75	1.06	—	0.94	—	(3)
	Cellular concrete	50.0	75	1.44	—	0.69	—	(3)
	Cellular concrete	60.0	75	1.80	—	0.56	—	(3)
	Cellular concrete	70.0	75	2.18	—	0.46	—	(3)
	Air-cooled slag aggregate	124.2	119	5.3	—	0.19	—	(1)
	Air-cooled slag aggregate	124.9	119	5.9	—	0.17	—	(1)
8 In. Concrete Blocks 8 x 8 x 16 3-oval core concrete block. 	8 in. three oval core, sand and gravel aggregate[a]	126.4	40	—	0.90	—	1.11	(4)
	8 in. three oval core, crushed limestone aggregate[a]	134.3	40	—	0.86	—	1.16	(4)
	8 in. three oval core, cinder aggregate[a]	86.2	40	—	0.58	—	1.73	(4)
	8 in. three oval core, burned clay aggregate[a]	67.7	40	—	0.50	—	2.00	(4)
	8 in. three oval core, expanded blast furnace slag aggregate[a]	—	40	—	0.49	—	2.04	(4)
	8 in. three oval core, air-cooled slag aggregate	—	40	—	0.68	—	1.47	(4)
12 In. Concrete Blocks 8 x 12 x 16 3-oval core concrete blocks 	12 in. three oval core, sand and gravel aggregate[a]	124.9	40	—	0.78	—	1.28	(4)
	12 in. three oval core, cinder aggregate[a]	86.2	40	—	0.53	—	1.88	(4)
	12 in. three oval core, burned clay aggregate[a]	76.7	40	—	0.47	—	2.13	(4)

See footnotes on first page of Table 2.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING AND INSULATING MATERIALS—Continued

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference.) Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness.

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Fahr Deg)	Conductivity or Conductance		Resistance		Authority
				(k)	(C)	Per Inch Thickness $\left(\frac{1}{k}\right)$	For Thickness Listed $\left(\frac{1}{C}\right)$	
MASONRY MATERIALS—(continued)								
Gypsum	8 in. solid gypsum partition tile[a]	—	—	2.41	—	0.42	—	(4)
	8 in. three cell gypsum partition tile[a]	—	—	—	0.74	—	1.35	(4)
	4 in. three cell gypsum partition tile[a]	—	—	—	0.60	—	1.67	(4)
	87½ percent gypsum, 12½ percent wood chips	51.2	74	1.66	—	0.60	—	(4)
PLASTERING MATERIALS	Gypsum plaster	—	—	3.30	—	0.30	—	
	Gypsum plaster, ⅜ in. thick	—	73	—	8.80	—	0.11	(4)
	Cement plaster	—	—	8.00	—	0.13	—	(2)
	Wood, lath and plaster, total thickness ¾ in.	—	70	—	2.50	—	0.40	(4)
	Gypsum plaster and expanded vermiculite, 4 to 1 mix	39.9	75	0.85	—	1.18	—	(3)
	Gypsum vermiculite plaster							
	Mix. 100 lb: 2 cu ft	46.7	63	1.84	—	0.54	—	(1)
	Mix. 100 lb: 3 cu ft	43.5	64	1.63	—	0.61	—	(1)
	Gypsum perlite plaster							
	Mix. 100 lb: 2 cu ft	49.1	64	1.68	—	0.60	—	(1)
	Mix. 100 lb: 3 cu ft	42.5	64	1.35	—	0.74	—	(1)
	Gypsum sand plaster							
	Mix. 100 lb: 200 lb	104	53	5.55	—	0.18	—	(1)
	Mix. 100 lb: 300 lb	107	52	5.77	—	0.17	—	(1)
	Insulating plaster 0.9 in. thick applied to ⅜ in. gypsum board	54.0	75	—	1.07	—	0.93	(3)
ROOFING	Asbestos shingles	65.0	75	—	6.0	—	0.17	(3)
	Asphalt, composition or prepared	70.0	75	—	6.5	—	0.15	(3)
	Asphalt shingles	70.0	75	—	6.5	—	0.15	(3)
	Built-up roofing, bitumen or felt, gravel or slag surfaced[a]	—	—	1.33	—	0.75	—	(2)
	Slate	—	—	10.00	—	0.10	—	
	Wood shingles	—	—	—	1.28	—	0.78	
WOODS	Balsa	20.0	90	0.58	—	1.72	—	(1)
	Balsa	8.8	90	0.38	—	2.63	—	(1)
	Balsa	7.3	90	0.33	—	3.03	—	(1)
	California redwood, 0 % moisture[a]	28.0	75	0.70	—	1.43	—	(4)
	Cypress	28.7	86	0.67	—	1.49	—	(1)
	Douglas fir, 0 % moisture[a]	34.0	75	0.67	—	1.49	—	(4)
	Eastern hemlock, 0 % moisture[a]	30.0	75	0.76	—	1.32	—	(4)
	Long leaf yellow pine, 0 % moisture[a]	40.0	75	0.86	—	1.16	—	(4)
	Mahogany	34.3	86	0.90	—	1.11	—	(1)
	Hard maple, 0 % moisture[a]	46.0	75	1.05	—	0.95	—	(4)
	Maple	44.3	86	1.10	—	0.91	—	(1)
	Maple, across grain	40.0	75	1.20	—	0.83	—	(3)
	Norway pine, 0 % moisture[a]	32.0	75	0.74	—	1.35	—	(4)
	Red cypress, 0 % moisture[a]	32.0	75	0.79	—	1.27	—	(4)
	Red oak, 0 % moisture[a]	48.0	75	1.18	—	0.85	—	(4)
	Short leaf yellow pine, 0 % moisture[a]	36.0	75	0.91	—	1.10	—	(4)
	Soft elm, 0 % moisture[a]	34.0	75	0.88	—	1.14	—	(4)

See footnotes on first page of Table 2.

Table 2. Conductivities (k) and Conductances (C) of Building and Insulating Materials—Continued

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degrees temperature difference.) Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness.

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Fahr Deg)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
WOODS—(continued)	Soft maple, 0 % moisture[a]	42.0	75	0.95	—	1.05	—	(4)
	Sugar pine, 0 % moisture[a]	28.0	75	0.64	—	1.56	—	(4)
	Virginia pine	34.3	86	0.96	—	1.04	—	(1)
	West coast hemlock, 0 % moisture[a]	30.0	75	0.79	—	1.27	—	(4)
	White pine	31.2	86	0.78	—	1.28	—	(1)
	Yellow pine	—	—	1.00	—	1.00	—	(3)
	Sawdust, various	12.0	90	0.41	—	2.44	—	(1)
	Shavings, various from planer	8.8	90	0.41	—	2.44	—	(1)
	Shavings, from maple beech and birch (coarse)	13.2	90	0.36	—	2.78	—	(1)
INSULATING MATERIALS Blanket and Bat Insulations	Chemically treated wood fibers held between layers of strong paper	3.62	70	0.25	—	4.00	—	(3)
	Flax fibers between strong paper	4.90	90	0.28	—	3.57	—	(1)
	Chemically treated hog hair between kraft paper	5.76	71	0.26	—	3.85	—	(3)
	Chemically treated hog hair between kraft paper and asbestos paper	7.70	71	0.28	—	3.57	—	(3)
	Hair felt between layers of paper	11.00	75	0.25	—	4.00	—	(3)
	Kapok between burlap or paper	1.00	90	0.24	—	4.17	—	(1)
	Stitched and creped expanding fibrous blanket	1.50	70	0.27	—	3.70	—	(3)
	Paper and asbestos fiber with emulsified asphalt binder	4.2	94	0.28	—	3.57	—	(1)
	Cotton insulating bat	0.875	72	0.24	—	4.17	—	(3)
	Cotton fibers Short Staple Linters, Fireproofed	6.25	90	0.25	—	4.00	—	(1)
		4.50	90	0.24	—	4.17	—	(1)
		2.45	90	0.24	—	4.17	—	(1)
		1.60	90	0.26	—	3.85	—	(1)
		0.85	90	0.29	—	3.45	—	(1)
		0.65	90	0.30	—	3.33	—	(1)
	Felted cattle hair	13.00	90	0.26	—	3.84	—	(1)
	Felted cattle hair	11.00	90	0.26	—	3.84	—	(1)
	Felted hair and asbestos	7.80	90	0.28	—	3.57	—	(1)
	Ground paper between two layers, each ⅛ in. thick made up of two layers of kraft paper (sample ⅜ in. thick)	12.1	75	—	0.40	—	2.50	(4)
	Mineral Wool	4.5	—	0.27	—	3.70	—	
Reflective	See Table 1, Section C	—	—	—	—	—	—	
Insulating Board	Made from sugar cane fiber	13.5	70	0.33	—	3.03	—	(3)
	Made from hard wood fibers	15.20	70	0.32	—	3.12	—	(3)
	Made from wood fiber	15.90	72	0.33	—	3.03	—	(3)
	Made from wood fiber	15.00	70	0.33	—	3.03	—	(3)
	Made from wood fiber	—	52	0.33	—	3.03	—	(6)
	Made from wood fiber	15.20	—	0.33	—	3.03	—	(3)
	Made from wood fiber	16.90	90	0.34	—	2.94	—	(1)
	Made from licorice root	16.1	81	0.34	—	2.94	—	(3)
	½ in. insulating boards without special finish[f] (eleven samples)	16.5	90	0.33	—	3.03	—	(1)
		to		to		to		
		21.8	—	0.40	—	2.50	—	
	1 in. insulating board[a]	13.2	—	0.34	—	2.94	—	(4)

See footnotes on first page of Table 2.

Table 2. Conductivities (k) and Conductances (C) of Building and Insulating Materials—Concluded

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference.) Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness.

Material	Description	Density (Lb per Cu Ft)	Mean Temp (Fahr Deg)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
INSULATING MATERIALS—(Continued) Loose Fill Type	Made from ceiba fibers	1.90	75	0.23	—	4.35	—	(3)
	Made from ceiba fibers	1.60	75	0.24	—	4.17	—	(3)
	Chemically treated wood fibers	4.0	75	0.28	—	3.57	—	(4)
	Fibrous material made from dolomite and silica	1.50	75	0.27	—	3.70	—	(3)
	Fibrous material made from slag	9.40	103	0.27	—	3.70	—	(1)
	Redwood bark	3.00	90	0.31	—	3.22	—	(1)
	Redwood bark	5.00	75	0.26	—	3.84	—	(3)
	Glass wool fibers 0.0003 in. to 0.006 in. in diameter	1.50	75	0.27	—	3.70	—	(3)
	Granular insulation made from combined silicate of lime and alumina	4.20	72	0.24	—	4.17	—	(3)
	Expanded vermiculite	7.0	70	0.48	—	2.08	—	(1)
	Regranulated cork about 3/16 in. particles	8.10	90	0.31	—	3.22	—	(1)
	Hand applied granular mineral wool 2 in. to 6 in. thick, horizontal position[b]. No covering	6.05 to 7.13	—	0.30 to 0.33	—	3.33 to 3.03	—	(4)
	4 in. machine blown granular mineral wool, horizontal position[b]. No covering	5.74	—	0.30	—	3.33	—	(4)
	Rock wool	10.0	90	0.27	—	3.70	—	(1)
Slab Insulations	Corkboard, no added binder	14.0	90	0.34	—	2.94	—	(1)
	Corkboard, no added binder	10.6	90	0.30	—	3.33	—	(1)
	Corkboard, no added binder	7.0	90	0.27	—	3.70	—	(1)
	Corkboard, no added binder	5.4	90	0.25	—	4.00	—	(1)
	Corkboard[a]	8.7	—	0.29	—	3.45	—	(4)
	Corkboard, asphaltic binder	14.5	90	0.32	—	3.12	—	(1)
	Chemically treated hog hair with film of asphalt	10.0	75	0.28	—	3.57	—	(3)
	Sugar cane fiber insulation blocks encased in asphalt membrane	13.8	70	0.30	—	3.33	—	(3)
	Made from shredded wood and cement	24.2	72	0.46	—	2.17	—	(3)
	Made from shredded wood and cement[a]	29.8	—	0.77	—	1.30	—	(4)
	Cellular glass	9.0	75	0.42	—	2.38	—	(1)
	Cellular glass	9.0	50	0.40	—	2.50	—	(1)

See footnotes on first page of Table 2.

ing. For increases of moisture content exceeding about 6 to 12 percent, the conductivity of frozen soil becomes progressively greater than that of the unfrozen soil.

Effect of Density. Density affects the thermal conductivity of a soil in about the same manner for all soils, at any moisture content, and for either the frozen or unfrozen condition. On the average, each one pound per cubic foot increase in dry density increases the thermal conductivity by about 3 percent.

Effect of Moisture. An increase in moisture content, up to the point of saturation, causes an increase in thermal conductivity. The rate of increase is indicated by the following values. Average conductivities, in Btu per (square foot) (hour) (Fahrenheit degree per inch), of four sands at a density of 110 lb per cu ft were: 6.8 at 2.5 percent moisture, 8.9 at 5 percent moisture, 11.2 at 10 percent moisture. Five soils of a fine texture at a density of 100 lb per cu ft, gave average conductivities of 6.7 at 10 percent moisture, and 9.5 at 20 percent. Thus, the doubling of moisture content within the ranges cited increases the conductivity by approximately 30 or 40 percent. At higher moisture contents the percentage increase would be less.

Effect of Soil Characteristics. The thermal conductivity of the soil, at a given density and moisture content varies in general with the texture of a soil, being relatively high for coarse-textured soils and relatively low for fine-textured soils. The mineral composition of the soils also affects the conductivity. Quartz tends to give high values, whereas minerals such as plagioclase feldspar and pyroxene, which are constituents of basic rocks tend to give low values of thermal conductivity. These points are illustrated by the values in Table 3 which lists seventeen soils in approximate order of their magnitude of thermal conductivity from greatest to least for seven different density-moisture content conditions. Some of the values in this table have been determined by extrapolation and are consequently approximate. Blank spaces in the table indicate that the density or moisture content, or both, are such that no tests were possible for that condition or that no tests were sufficiently

TABLE 3. THERMAL CONDUCTIVITY (k) VALUES OF SOILS IN APPROXIMATE ORDER OF DECREASING VALUES[a]

Mean Temperature—40 F

Soil No	Soil Designation	Moisture Content—%						
		4	4	4	10	10	20	20
		Dry Density-lb per cu ft						
		100	110	120	90	110	90	100
P4714	Fine Crushed Quartz	12.0	16.0					
P4703	Crushed Quartz	11.5	16.0	22.0				
P4701	Graded Ottawa Sand	10.0	14.0					
P4709	Fairbanks Sand	8.5±	10.5	13.5		15.0		
P4604	Lowell Sand	8.5	11.0			13.5		
P4601	Chena River Gravel		9.0±	13.0				
P4705	Crushed Feldspar	6.0	7.5	9.5				
P4706	Crushed Granite	5.5	7.5	10.0				
P4711	Dakota Sandy Loam		6.5	9.5		13±		
P4704	Crushed Trap Rock	5.0	6.0	7.0				
P4713	Ramsey Sandy Loam	4.5	6.5			10.0		
P4502	Northway Fine Sand	4.5	5.5			8.5		
P4503	Northway Sand	4.5	6.0			7.5±		
P4708	Healy Clay	4.0±			5.5	9.0±	8.0	10.0
P4602	Fairbanks Silt Loam				5.0	9.0±	7.5	10.0
P4710	Fairbanks Silty Clay Loam				5.0	9.0±	7.5	9.5
P4505	Northway Silt Loam				4.0±	7.0±	6 0±	7.0±

[a] k = Btu per (square foot) (hour) (Fahrenheit degree per inch).

close to permit a reasonable extrapolation of the data. Granular soils, particularly those with high quartz contents, head the tabulation or have the greatest conductivity at a given condition. Sandy loam soils are midway in the table and fine grained soils such as clay and silt loam are last.

Estimating Thermal Conductivity. The four diagrams of Fig. 3 are presented to aid in the estimate of the thermal conductivity of any soil. Two of the charts are for sands or sandy soils, and two for silt and clay soils. One of the diagrams for each type of soils is for the frozen, and the other for the unfrozen condition. It is expected that these charts will give conductivity values with a precision of 25 percent. The effect of such factors as density, moisture content, freezing, or texture may be easily approximated by use of these graphs.

Specific Heat of Soils

Tests to determine specific heat were run on twelve soils. On five of the soils, tests were made at three or four mean temperatures varying from about 10 to 140F. The specific heat values of all twelve soils varied by only a small amount (about 0.01), and averaged 0.19 at 140 F. The specific heat values of the soils decreased with a decrease in temperature. The average value at zero F would be about 0.16. Values at temperatures between zero and 140 F can be estimated by considering a straight-line relationship between the two values given.

Surface Conductance

The surface conductance of a wall is the combined heat transfer to or from the wall by radiation, convection and conduction. Each of the three portions making up the total may vary, independently of the others, thus affecting the total conductance. The heat transfer by radiation between two surfaces is controlled by the character of the surfaces (emissivity), the temperature difference between them, and the solid angle through which they *see* each other. The heat transfer by convection and conduction is

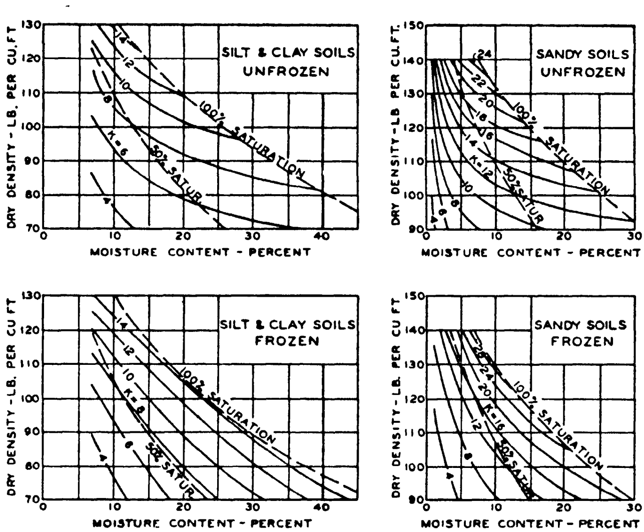


FIG. 3. DETERMINING THERMAL CONDUCTIVITY OF SOILS FROM DENSITY AND MOISTURE CONTENT

controlled by the roughness of the surface, by air movement, and temperature difference between the air and the surface.

The importance of the effect of temperature of *surrounding surfaces* on the surface conductance, due to the effect on radiation, is illustrated in Table 4, which applies to a vertical surface at 80 F, with ambient air at 70 F and effective emissivity equal to 0.83.[4]

In many cases, because the heat resistance of the internal parts of the wall is high compared with the surface resistance, the surface factors are of minor importance. In other cases, *e.g.*, single glass windows, the surface resistances constitute almost the entire resistance and are therefore very important. An analysis of various factors affecting surface conductance and the difference between surface and air temperatures will be found in Reference 5. (See also Chapter 23.)

The convection part of the surface conductance is affected markedly by air movement. This is illustrated by Fig. 4, which shows the results of tests[6] made on 12 in. square samples of different materials at a mean temperature of 20 F, and for wind velocities up to 40 mph. These conductances include the radiation portion of the coefficient which, for the conditions of the tests, was about 0.7 Btu per (hr) (sq ft) (F deg). More recent tests[7] on smooth surfaces show that surface length also affects significantly the convection part of conductance; the average value decreases as the surface length increases. Moreover, observations[8] of the magnitude of low temperature radiant energy received from outdoor surroundings show that only under certain conditions may the out-of-doors be treated as a black body radiating at air temperature.

Because of these factors, the selection of surface conductance coefficients for a practical building, becomes a matter of judgment. *In calculating the overall heat transmission coefficients for the walls, etc., of Tables 6 to 19, 1.65 has been selected as an average inside surface conductance, and 6.0 as an average outside surface conductance for a 15 mph wind.* Both values combine the effects of convection and radiation, and are applicable to ordinary building materials. They should not be used for low emissivity surfaces such as bright metal. Values of U for windows in Table 20 have been computed from somewhat different data, as described in a later section, in order to give proper weight to actual surface conductance.

In special cases, where surface conductances become important factors in the overall rates of heat transfer, more selective coefficients may be required. Data given in Table 1, Section A, and principles and data given in Chapter 5, Heat Transfer, may be applied in such cases.

Table 4. Variation in Surface Conductance Coefficient with Different Temperatures of Surrounding Surface

Surrounding Surface Temperature	75 F	70 F	69 F	60 F	50 F
Convection—Btu per (hr) (sq ft)	6.6	6.6	6.6	6.6	6.6
Radiation— Btu per (hr) (sq ft)	4.4	8.6	9.6	17.0	24.9
Total—Btu per (hr) (sq ft)	11.0	15.2	16.2	23.6	31.5

Air Space Conductance

The transfer of heat across an air space involves the boundary surfaces as well as the intervening air; consequently, the factors influencing surface conductance play an important part in determining the conductance of the air space. The coefficients given for air space conductance represent the total conductance from surface to surface.

The radiation portion of the coefficient is affected by the difference in temperature between the boundary surfaces and by their respective emissivities, and is practically independent of depth. The convection and conduction transfer is controlled by depth and shape of the air space, the roughness of the boundary surfaces, the mean temperature, and the direction of heat flow. For air spaces usually employed in building construction, the radiation and convection factors vary independently of each other.

Table 1, Section B gives experimentally-determined conductances of vertical air spaces bounded by such materials as paper, wood, plaster, etc., having emissivity coefficients of 0.8 or higher, and having extended parallel surfaces perpendicular to the direction of heat flow. The conductances decrease as the depth is increased, but change only slightly for spaces greater than $\frac{3}{4}$ in. Air space tests reported by Wilkes and Peterson, gave conductance values for air spaces of $3\frac{5}{8}$ in. depth having boundary surfaces with emissivity values of 0.83 as follows:[9] vertical, 1.17; horizontal (heat flow upward), 1.32; horizontal (heat flow downward), 0.94. Since, in buildings, the same constructions may be used for conditions where the direction of heat flow may be in one direction or its opposite, and since much of the construction involves vertical air spaces, an average value of

1.10 Btu per (hour) (square foot) (Fahrenheit degree temperature difference) was chosen for use in calculating the overall coefficients in Tables 6 to 19 wherever air spaces ¾ in. or more in depth were involved.

If one or both boundary surfaces of an air space are faced with metals which have low emissivity surfaces, the radiant heat transfer will be greatly reduced in comparison with that occurring from surfaces of ordinary building materials. Table 1, Section C gives conductances and resistances of air spaces *bounded by one reflective surface* with an emissivity of 0.05. These values include heat transferred both by radiation and convection, but the radiation component is relatively small for the test conditions.

When insulating materials are installed with single or multiple air spaces,

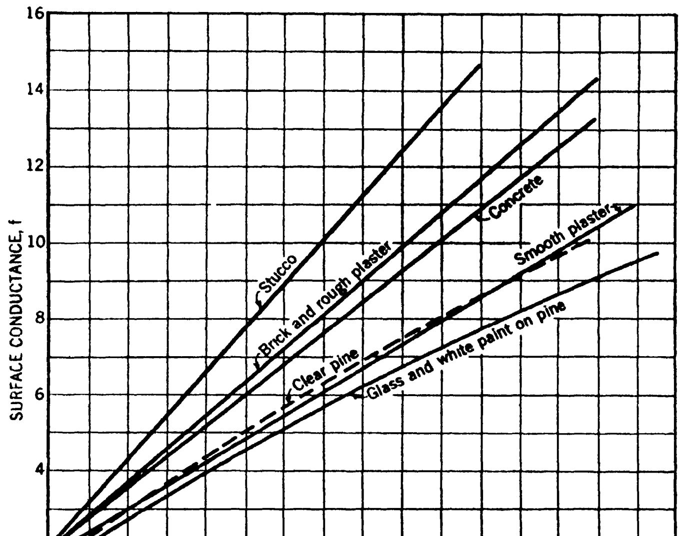


FIG. 4. CURVES SHOWING RELATION BETWEEN SURFACE CONDUCTANCES FOR DIFFERENT SURFACES AT 20 F MEAN TEMPERATURE

the position (vertical, horizontal or inclined) of the material and the direction of heat flow must be taken into consideration. For example, the resistance to *upward* heat flow is about one-third the resistance to *downward* heat flow in a horizontal position (Table 1, Section C). The difference between the conductance through vertical air spaces and that through horizontal and sloping air spaces with upward heat flow is considerably less. For upward heat flow, it is recommended that a value of 0.46 be used for the conductance of horizontal or sloping air spaces bounded on one side by reflective materials having an emissivity of approximately 0.05. The same conductance value is also recommended for similar vertical air spaces.

When considering heat transfer to and from reflective surfaces in building construction, the emissivity should be known. This can be determined directly for the long wave length radiation corresponding to average room

TABLE 5. CONDUCTIVITIES (k) AND CONDUTANCES (C) USED IN CALCULATING HEAT TRANSMISSION COEFFICIENTS (U) IN TABLES 6 TO 19

These constants are expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference). Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness

Material	Description	Conductivity or Conductance		Resistance	
		(k)	(C)	Per Inch Thickness $\left(\frac{1}{k}\right)$	For Thickness Listed $\left(\frac{1}{C}\right)$
AIR SPACES					
Bounded by Ordinary Materials	Vertical[a], ¾ in. or more in width	—	1.10	—	0.91
Bounded by Aluminum Foil	Vertical[a], ¾ in. or more in width	—	0.46	—	2.17
EXTERIOR FINISHES (Frame Walls)					
Brick Veneer	4 in. thick (nominal)	—	2.27	—	0.44
Stucco (1 in.)		12.50	—	0.08	—
Wood Shingles		—	1.28	—	0.78
Yellow Pine Lap Siding		—	1.28	—	0.78
INSULATING MATERIALS					
Aluminum Foil	See Air Spaces	—	—	—	—
Bats and Blankets	Made from mineral or vegetable fiber or animal hair, enclosed or open	0.27	—	3.70	—
Corkboard	Pure, no added binder	0.30	—	3.33	—
Insulating Board	Vegetable fiber	0.33	—	3.03	—
Mineral Wool	Fiber made from rock, slag or glass	0.27	—	3.70	—
Vermiculite	Expanded	0.48	—	2.08	—
INTERIOR FINISHES					
Composition Wallboard	3/16 in. to ⅜ in. thick	0.50	—	2.00	—
Gypsum Plaster		3.30	—	0.30	—
Gypsum Board (⅜ in.)	Plain or decorated	—	3.70	—	0.27
Gypsum Lath (⅜ in.) and Plaster	Plaster thickness assumed ½ in.	—	2.4	—	0.42
Insulating Board (½ in.)	Plain or decorated	—	0.66	—	1.52
Insulating Board Lath (½ in.) and Plaster	Plaster thickness assumed ½ in	—	0.60	—	1.67
Insulating Board Lath (1 in.) and Plaster	Plaster thickness assumed ½ in.	—	0.31	—	3.18
Metal Lath and Plaster	Plaster thickness assumed ¾ in.	—	4.40	—	0.23
Plywood (⅜ in.)	Plain or decorated	—	2.12	—	0.47
Wood Lath and Plaster		—	2.50	—	0.40
MASONRY MATERIALS					
Brick	Adobe, assumed 4 in. thick	—	0.89	—	1.12
Brick	Common, assumed 4 in. thick	—	1.25	—	0.80
Brick	Face, assumed 4 in. thick	—	2.30	—	0.43
Cement Mortar		12.00	—	0.08	—
3 in. Clay Tile (Hollow)		—	1.28	—	0.78
4 in. Clay Tile (Hollow)		—	1.00	—	1.00
6 in. Clay Tile (Hollow)		—	0.64	—	1.57
8 in. Clay Tile (Hollow)		—	0.60	—	1.67
10 in. Clay Tile (Hollow)		—	0.58	—	1.72
12 in. Clay Tile (Hollow)		—	0.40	—	2.50
16 in. Clay Tile (Hollow)		—	0.31	—	3.23
Concrete	Light weight aggregate[b]	2.50	—	0.40	—
Concrete	Sand and gravel aggregate	12.00	—	0.08	—
3 in. Concrete Blocks	Hollow, cinder aggregate	—	1.28	—	0.78
4 in. Concrete Blocks	Hollow, cinder aggregate	—	1.00	—	1.00
8 in. Concrete Blocks	Hollow, gravel aggregate	—	1.00	—	1.00
12 in. Concrete Blocks	Hollow gravel aggregate	—	0.80	—	1.25
8 in. Concrete Blocks	Hollow, cinder aggregate	—	0.60	—	1.66
12 in. Concrete Blocks	Hollow, cinder aggregate	—	0.53	—	1.88
8 in. Concrete Blocks	Hollow, light weight aggregate[b]	—	0.50	—	2.00
12 in. Concrete Blocks	Hollow, light weight aggregate[b]	—	0.47	—	2.13
Gypsum Fiber Concrete	87½ percent gypsum and 12½ percent wood chips	1.66	—	0.60	—
3 in. Gypsum Tile	Hollow	—	0.61	—	1.64
4 in. Gypsum Tile	Hollow	—	0.46	—	2.18
Stucco		12.50	—	0.08	—
Tile and Terrazzo	For flooring	12.00	—	0.08	—
Stone		12.50	—	0.08	—

[a] Conductance values for horizontal air spaces depend on whether the heat flow is upward or downward, but in most cases it is sufficiently accurate to use the same values for horizontal as for vertical air spaces.

[b] Expanded slag, burned clay or pumice.

TABLE 5. CONDUCTIVITIES (k) AND CONDUCTANCES (C) USED IN CALCULATING HEAT TRANSMISSION COEFFICIENTS (U) IN TABLES 6 TO 19

These constants are expressed in Btu per (hour) (square foot) (Farhenheit degree temperature difference.) Conductivities (k) are per inch thickness and conductances (C) are for thickness or construction stated, not per inch thickness

Material	Description	Conductivity or Conductance		Resistance	
		(k)	(C)	Per Inch Thickness $\left(\frac{1}{k}\right)$	For Thickness Listed $\left(\frac{1}{C}\right)$
ROOFING MATERIALS					
Asbestos Shingles		—	6.00	—	0.17
Asphalt Shingles		—	6.50	—	0.15
Built-up Roofing	Assumed thickness ⅜ in.	—	3.53	—	0.28
Heavy Roll Roofing		—	6.50	—	0.15
Slate	Assumed ½ in.	10.00	20.00	0.10	0.05
Wood Shingles		—	1.28	—	0.78
SHEATHING					
Gypsum (½ in.)		—	2.82	—	0.35
Insulating Board (25/32 in.)		—	0.42	—	2.37
Plywood (5/16 in.)		—	2.56	—	0.39
Fir or Yellow Pine (1 in.)	Actual thickness 25/32 in.	—	1.02	—	0.98
Fir, Plus Building Paper	Actual thickness 25/32 in.	—	0.86	—	1.16
SURFACES					
Still air	Ordinary non-reflective materials, vertical	—	1.65	—	0.61
15 mph Wind Velocity	Ordinary non-reflective materials, vertical	—	6.00	—	0.17
WOODS					
Fir Sheathing (1 in.) Building Paper and Yellow Pine Lap Siding		—	0.50	—	2.00
Maple or Oak		1.15	—	0.87	—
Yellow Pine or Fir		0.80	—	1.25	—

and wall temperatures. The possibility of change in emissivity with time of exposure due to surface coatings, chemical action, deposition of dust, etc., must be considered in selecting a material for use.[10]

PRACTICAL COEFFICIENTS AND THEIR USE

For practical purposes it is necessary to have average coefficients that may be applied to various materials and types or construction without the necessity of making actual tests. In Table 2 coefficients are given for a group of materials which have been selected from tests by various authorities. Since there is some variation in the resulting values due to variations in materials and in test conditions, average values for the usual conditions encountered in building practice have been selected and listed in Table 5. These coefficients were used in the calculation of overall coefficients given in Tables 6 to 19. These tables constitute typical examples of combinations frequently used, but any special constructions not given can be computed by the use of the conductivity or conductance values in Table 2 and the fundamental heat transfer formulas.

Caution

The user should realize that the average conductivity and conductance values given in Tables 2 or 5 do not necessarily apply to all products of the same general description. In using these values, judgment should be exercised with regard to the extent to which the product (either as received or as applied) will comply with the tabulated values. Exact conductivities or conductances for specific materials should be obtained from the maker.

Insulating Materials

In order to determine the benefit derived from the addition of insulating materials to a given construction, the overall coefficient of heat transmission U_t of the insulated construction may be compared with the corresponding coefficient U without insulation. Attention is called to the necessity of applying the insulating material in accordance with the manufacturer's specification The engineer must evaluate carefully the economic considerations involved in the selection of an insulating material as adapted to various building constructions. Lack of proper evaluation, or improper installation may lead to unsatisfactory results.

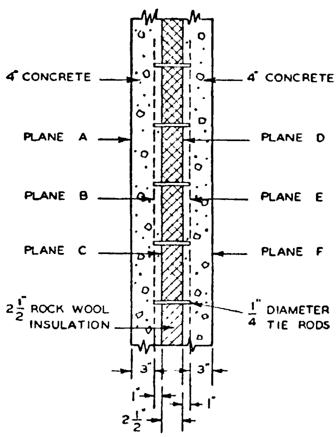


FIG. 5. SECTION OF CONCRETE WALL HAVING STEEL TIE RODS AND INSULATION

Computed Heat Transmission Coefficients

Computed overall heat transmission coefficients of many common types of building construction are given in Tables 6 to 20, inclusive, each coefficient being identified by a serial number, except in Tables 19 and 20. For example, the coefficient U of a brick veneer, frame wall with wood sheathing and ½-in. of plaster on gypsum lath is 0.27 (Wall No. 28-C in Table 6) and with 2 inches of blanket or bat insulation, the coefficient would be 0.097 (No. 49-B in Table 7).

In the analysis of any wall construction for the purpose of calculating the overall coefficient of heat transmission U, it is first necessary to determine the paths of heat flow, that is, whether they are parallel or series, or a combination of both. This is in accordance with the basic laws of heat transfer which state that *in parallel flow the conductances are additive*, while *in series flow the resistances are additive*. Likewise, in order to determine the total resistance for the wall, the conductance must be known.

The importance of this analysis cannot be over-emphasized. This is especially true in wall constructions in which there are parallel paths of heat flow, and one path has a high heat transfer, while others have a low heat transfer. The method of making this calculation can best be shown by *Example 1* and Fig. 5. As this wall was tested by the hot box method

at the *University of Minnesota*, a direct comparison can be made between calculated and tested values.

Example 1: Calculate the coefficient of heat transmission U for wall as shown in Fig. 5. Wall construction consists of two 4-in. concrete walls separated by a 2½-in. space filled with insulation; ¼-in. diameter metal tie rods are imbedded a distance of 1 in. in each 4-in. concrete wall, and spaced 9 in. vertically and 12 in. horizontally. Values of k are: insulation 0.30, concrete 12.00, tie rods 400.00.

Solution: In Fig. 5 the following paths of heat flow from plane A to plane F will be noted:

1. From A to B: One path through 3 in. of concrete.
2. From B to C: Two paths, (a) through 1 in. of tie rod, and (b) through 1 in. of concrete.
3. From C to D: Two paths, (a) through 2½ in. of tie rod, and (*b*) through 2½ in. of insulation.
4. From D to E: Two paths, (a) through 1 in. of tie rod, and (b) through 1 in. of concrete.
5. From E to F: One path through 3 in. of concrete.

It will be noted that items 2 and 4 are paths of similar flow, and could be treated as one. If equilibrium or steady state heat transfer is assumed, there will exist a temperature difference between the metal tie rod and the concrete, and also between the metal tie rod and the insulating material. The rate of heat transfer between these materials is dependent upon their conductivity values and the temperature difference. As the conductivity of the metal tie rods is considerably higher than that of the concrete or insulating material, it cannot be assumed that the same rate of heat transfer takes place for all parallel paths. Likewise, an appreciable error would be made by assuming that no heat transfer takes place between the metal tie rod and the surrounding materials. Although the pattern of the isotherms is unknown, the following method of calculation does partially take into account the heat flow between the metal tie rods and its bounding materials.

Parallel Flow. The conductances through the areas of parallel heat flow may be determined as follows:

1. The area of each ¼-in. diameter tie rod is 0.00036 sq ft, and as the tie rods are spaced 9 in. vertically, and 12 in. horizontally, there will be 0.00036 × $\frac{4}{3}$ = 0.00048 sq ft of tie rod to each square foot of wall area. Then from plane B to plane C, the conductance C_1 is

$$C_1 = \frac{0.00048}{1.0} \times \frac{400}{1.0} + \frac{0.99925}{1.0} \times \frac{12}{1.0} = 0.192 + 11.994 = 12.189$$

2. For tie rod and insulation from plane C to plane D the conductance C_2 is

$$C_2 = \frac{0.00048}{1.0} \times \frac{400}{2.5} + \frac{0.99952}{1.0} \times \frac{0.30}{2.5} = 0.077 + 0.120 = 0.197$$

3. For tie rod and concrete from plane D to plane E the conductance C_3 is

$$C_3 = \frac{0.00048}{1.0} \times \frac{400}{1.0} + \frac{0.99952}{1.0} \times \frac{12}{1.0} = 0.192 + 11.994 = 12.186$$

Series Flow. After the conductance values have been determined, the total resistance and U value can be determined as follows:

$$R_T = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{1}{C_1} + \frac{1}{C_2} + \frac{1}{C_3} + \frac{x_2}{k_2} + \frac{1}{f_o}$$

$$R_T = \frac{1}{1.65} + \frac{3.0}{12.0} + \frac{1}{12.186} + \frac{1}{0.197} + \frac{1}{12.186} + \frac{3.0}{12.0} + \frac{1}{6.0}$$

$$R_T = 0.606 + 0.250 + 0.0821 + 5.076 + 0.0821 + 0.250 + 0.167 = 6.513$$

$$U = \frac{1}{R_T} = \frac{1}{6.513} = 0.153 \text{ Btu per (hr) (sq ft) (F deg).}$$

The Hot Box test value, from *University of Minnesota*, for this wall, corrected for a 15 mph wind velocity, was $U = 0.150$ Btu per (hr) (sq ft) (F deg). The error between the calculated and test values would be

$$\frac{0.153 - 0.150}{0.150} \times 100 = 2 \text{ percent.}$$

If the effect of the tie rods were omitted from the calculations, the overall U value would be 0.103. Although the percentage of area occupied by the tie rods per square foot of wall area is $\frac{0.00048}{1.0} \times 100 = 0.048$ percent, the error between the calculated and test values would be

$$\frac{0.150 - 0.103}{0.150} \times 100 = 31 \text{ percent.}$$

In making the calculations for values of U shown in Tables 6 to 19, the following conditions have been assumed:

Equilibrium or steady-state heat transfer, eliminating effects of heat capacity.

Surrounding surfaces at ambient air temperatures.

Exterior wind velocity of 15 mph.

Surface emissivity of ordinary building materials = 0.83.

No correction for position or direction of heat flow. (Average coefficients used).

Air spaces are $\frac{3}{4}$ in. or more in width.

Variation of conductivity with mean temperature neglected.

Corrections for framing made on basis of parallel heat flow through 2 × 4 in. (nominal) studs, 16 in. on centers, the framing covering 15 percent of wall area.

Actual thicknesses of lumber assumed to be as follows:

Nominal	*Actual*	*Nominal*	*Actual*
1 in. (S-2-S)	$\frac{25}{32}$ in.	3 in. (S-2-S)	$2\frac{5}{8}$ in.
$1\frac{1}{2}$ in. (S-2-S)	$1\frac{5}{16}$ in.	4 in. (S-2-S)	$3\frac{5}{8}$ in.
2 in. (S-2-S)	$1\frac{5}{8}$ in.	Finish flooring, (maple or oak)	$1\frac{3}{16}$ in.
$2\frac{1}{2}$ in. (S-2-S)	$2\frac{1}{8}$ in.		

Coefficients for frame construction are corrected for the effect of framing where such correction would increase the coefficients, but not where the correction would decrease the coefficients.[11]

It should be noted that the effects of poor workmanship in construction and installation have an increasingly greater percentage effect on heat transmission as the coefficient becomes numerically smaller. Failure to meet design estimates may be caused by lack of proper attention to exact compliance with specifications. A factor of safety may be employed as a precaution when it is judged desirable.

Roof Coefficients

Computations for wood shingle roofs applied over wood stripping are based on 1 by 4 in. wood strips, spaced 2 in. apart. Values for roofs containing Spanish and French clay roofing tile are assumed the same as for slate roofs. Values for pitched roofs in Table 17 apply where the roof is over a heated attic or top floor, so that the heat passes directly through the roof structure, including any interior finish material.

Combined Ceiling and Roof Coefficients

If the attic space between the ceiling and roof is unheated, the combined coefficient from room air below the ceiling to exterior air can be calculated from the following formula:

$$R_T = \frac{1}{U_{ce}} + \frac{1}{nU_r} \tag{4}$$

and

$$U = 1/R_T \tag{5}$$

where U = combined coefficient to be used with ceiling area.
R_T = total resistance of ceiling and roof.
U_{ce} = coefficient of transmission of ceiling.
U_r = coefficient of transmission of roof.
n = ratio of roof area to ceiling area.

It should be noted that the overall coefficient U should be multiplied by the ceiling area to determine heat loss, and not by the roof area. Values of U_r and U_{ce} should be calculated using a value of 2.2 (the reciprocal of one-half the air space resistance) rather than 1.65 for the conductances of surfaces facing the attic, since the attic is equivalent to an air space.

If the attic contains windows, dormers and vertical wall spaces, and if their area is small compared to that of the roof, they may be considered part of the roof area. For accuracy, the sum of the coefficients of each individual section, multiplied by its percentage of the total area, should be used as U_r. Where attic wall areas are large, it is preferable to estimate the attic temperature as illustrated in Chapter 11, and calculate the heat loss through the ceiling by multiplying the value of U_{ce} for the ceiling by the difference in temperature above and below the ceiling.

Basement Floor, Basement Wall and Concrete Slab Floor Coefficients

The heat transfer through basement walls and floors to the ground is dependent on the temperature difference between the air within and that of the ground, on the material constituting the wall or floor, and on the conductivity of the surrounding earth. The conductivity of the earth will vary with local conditions, and is usually unknown. Tests[12] at the A.S.H.V.E. Relearch Laboratory indicate a heat flow of approximately 2.0 Btu per (hr) (sq ft) through an uninsulated contrete basement floor, with a temperature difference of 20 F between ground temperature and the air temperature 6 in. above the floor. Based on this result a coefficient of 0.10 Btu per (hr) (sq ft) (Fahrenheit degree difference) is recommended for calculation where it is desirable to allow for the small basement floor heat loss, *e.g.*, for heated basements.

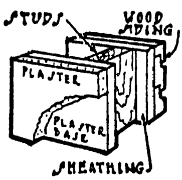
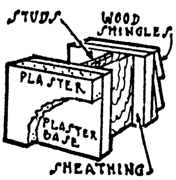
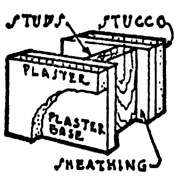
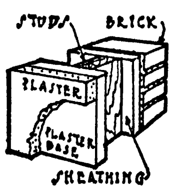
For basement walls the same coefficient may be used, but due to closer proximity to the surface of the ground, the temperature difference for winter design conditions will be greater than for the floor. The test results indicate a unit area heat loss, at mid-height of the basement wall approximately twice that of the same floor area.

For concrete slab floors laid in contact with the ground at grade level, recent tests[13] indicate that for small floor areas (equal to that of a house 25 feet square) the heat loss may be calculated as proportional to the length of exposed edge rather than total area. This amounts to 0.81 Btu per (hr) (lineal foot of exposed edge) (Fahrenheit degree difference between the inside air temperature and the average outside air temperature). It should be noted that this may be appreciably reduced by insulating under the ground slab, and also along the edges between the floor and the abutting walls. See also sections on Basement Temperatures and Heat Loss, and on Floor Heat Loss in Basementless Houses, in Chapter 11.

Table 6. Coefficients of Transmission (U) of Frame Walls

These coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

No Insulation Between Studs[a] (See Table 7)

Exterior Finish	Interior Finish	Type of Sheathing: Gypsum (½ in. thick) A	Plywood (5/16 in. thick) B	Wood[f] (25/32 in. thick) Bldg. Paper C	Insulating Board (25/32 in. thick) D	Wall Number
Wood Siding (Clapboard) 	Metal Lath and Plaster[b]	0.33	0.32	0.26	0.20	1
	Gypsum Board (⅜ in.) Decorated	0.32	0.32	0.26	0.20	2
	Wood Lath and Plaster	0.31	0.31	0.25	0.19	3
	Gypsum Lath (⅜ in.) Plastered[c]	0.31	0.30	0.25	0.19	4
	Plywood (⅜ in.) Plain or Decorated	0.30	0.30	0.24	0.19	5
	Insulating Board (½ in.) Plain or Decorated	0.23	0.23	0.19	0.16	6
	Insulating Board Lath (½ in.) Plastered[c]	0.22	0.22	0.19	0.15	7
	Insulating Board Lath (1 in.) Plastered[c]	0.17	0.17	0.15	0.12	8
Wood[d] Shingles 	Metal Lath and Plaster[b]	0.25	0.25	0.26	0.17	9
	Gypsum Board (⅜ in.) Decorated	0.25	0.25	0.26	0.17	10
	Wood Lath and Plaster	0.24	0.24	0.25	0.16	11
	Gypsum Lath (⅜ in.) Plastered[c]	0.24	0.24	0.25	0.16	12
	Plywood (⅜ in.) Plain or Decorated	0.24	0.24	0.24	0.16	13
	Insulating Board (½ in.) Plain or Decorated	0.19	0.19	0.19	0.14	14
	Insulating Board Lath (½ in.) Plastered[c]	0.19	0.18	0.19	0.13	15
	Insulating Board Lath (1 in.) Plastered[c]	0.14	0.14	0.15	0.11	16
Stucco 	Metal Lath and Plaster[b]	0.43	0.42	0.32	0.23	17
	Gypsum Board (⅜ in.) Decorated	0.42	0.41	0.31	0.23	18
	Wood Lath and Plaster	0.40	0.39	0.30	0.22	19
	Gypsum Lath (⅜ in.) Plastered[c]	0.39	0.39	0.30	0 22	20
	Plywood (⅜ in.) Plain or Decorated	0.39	0.38	0.29	0.22	21
	Insulating Board (½ in.) Plain or Decorated	0.27	0.27	0.22	0.18	22
	Insulating Board Lath (½ in.) Plastered[c]	0.26	0.26	0.22	0.17	23
	Insulating Board Lath (1 in.) Plastered[c]	0.19	0.19	0.16	0.14	24
Brick Veneer[e] 	Metal Lath and Plaster[b]	0.37	0.36	0.28	0.21	25
	Gypsum Board (⅜ in.) Decorated	0.36	0.36	0.28	0.21	26
	Wood Lath and Plaster	0.35	0.34	0.27	0.20	27
	Gypsum Lath (⅜ in.) Plastered[c]	0.34	0.34	0.27	0.20	28
	Plywood (⅜ in.) Plain or Decorated	0.34	0.33	0.27	0.20	29
	Insulating Board (½ in.) Plain or Decorated	0.25	0.25	0.21	0.17	30
	Insulating Board Lath (½ in.) Plastered[c]	0.24	0.24	0.20	0.16	31
	Insulating Board Lath (1 in.) Plastered[c]	0.18	0.18	0.15	0.13	32

(See text p. 182.

[a] Coefficients not weighted; effect of studding neglected.
[b] Plaster assumed ¾ in. thick.
[c] Plaster assumed ½ in. thick.
[d] Furring strips (1 in. nominal thickness) between wood shingles and all sheathings except wood.
[e] Small air space and mortar between building paper and brick veneer neglected.
[f] Nominal thickness, 1 in.

TABLE 7. COEFFICIENTS OF TRANSMISSION (U) OF FRAME WALLS WITH INSULATION BETWEEN FRAMING[a, b]

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

COEFFICIENT WITH *NO* INSULATION BETWEEN FRAMING	COEFFICIENT WITH INSULATION BETWEEN FRAMING				NUMBER
	MINERAL WOOL OR VEGETABLE FIBERS IN BLANKET OR BAT FORM[c] (Thickness below)			3⅝ IN. MINERAL WOOL BETWEEN FRAMING[d]	
	1 IN.	2 IN.	3 IN.		
	A	B	C	D	
0.11	0.078	0.063	0.054	0.051	33
0.13	0.088	0.070	0.058	0.055	35
0.15	0.097	0.075	0.062	0.059	37
0.17	0.10	0.080	0.066	0.062	39
0.19	0.11	0.084	0.069	0.065	41
0.21	0.12	0.088	0.072	0.067	43
0.23	0.12	0.091	0.074	0.069	45
0.25	0.13	0.094	0.076	0.071	47
0.27	0.14	0.097	0.078	0.073	49
0.29	0.14	0.10	0.080	0.075	51
0.31	0.14	0.10	0.081	0.076	53
0.33	0.15	0.10	0.083	0.077	55
0.35	0.15	0.11	0.084	0.078	57
0.37	0.16	0.11	0.085	0.080	59
0.39	0.16	0.11	0.086	0.081	61
0.41	0.16	0.11	0.087	0.082	63
0.43	0.17	0.11	0.088	0.082	65

(See text p. 187.)

[a] This table may be used for determining the coefficients of transmission of frame constructions with the types and thicknesses of insulation indicated in Columns A to D inclusive between framing. Columns A, B and C may be used for walls, ceilings or roofs with only one air space between framing but are not applicable to ceilings with *no* flooring above. (See Table 12.) Column D is applicable to walls only. *Example:* Find the coefficient of transmission of a frame wall consisting of wood siding, 25/32 in. insulating board sheathing studs, gypsum lath and plaster, with 2 in. blanket insulation between studs. According to Table 6, a wall of this construction with *no insulation* between studs has a coefficient of 0.19 (Wall No. 4D). Referring to Column B above, it will be found that a wall of this value with 2 in. blanket insulation between the studs has a coefficient of 0.084

[b] Coefficients corrected for 2 x 4 framing, 16 in. on centers—15 percent of surface area.

[c] Based on one air space between framing.

[d] No air space.

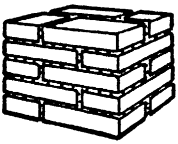
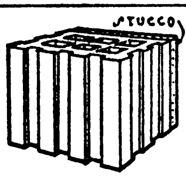
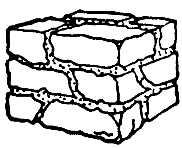
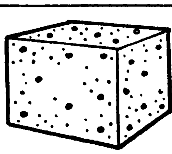
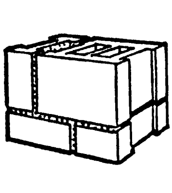
Glass Coefficients

The U values for glass sheets and hollow glass block, given in Sections A, B and C of Table 20, have been computed by methods and data given in an A.S.H.V.E. Research Paper.[14] It is assumed that the surface conductance for convection loss to the air is 4.0 Btu per (hr) (sq ft) (F deg). It is also assumed that the glass loses heat by radiation to the ground and to the clear sky, which together have an effective radiating temperature *below* the air temperature. It is therefore necessary to determine, by trial and error, the temperature of the outdoor glass surface such that the sum of the radiation and convection losses equals the heat conducted through the glass section, and equals the heat delivered to the glass from the heated space. This heat flow, divided by the air-to-air temperature difference, results in a U value which is used in the usual manner. The equivalent surface conductance for radiation and convection combined, based on *air-to-surface* temperature difference, therefore varies from about 5.5 for single glass to about 6.6 for double glass for exactly the same environmental design conditions.

It is assumed that the room air temperature equals the average tem-

Table 8. Coefficients of Transmission (U) of Masonry Walls

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

Type of Masonry		Thickness of Masonry Inches	Interior Finish (Plus Insulation Where Indicated)									Wall Number
			Plain Walls—No Interior Finish	Plaster (½ in.) on Walls	Metal Lath and Plaster[f] Furred[h]	Gypsum Board (⅜ in.) Decorated—Furred[h]	Gypsum Lath (⅜ in.) Plastered[g]—Furred[h]	Insulating Board (½ in.) Plain or Decorated—Furred[h]	Insulating Board Lath (½ in.) Plastered[g]—Furred[h]	Insulating Board Lath (1 in.) Plastered[g]—Furred[h]	Gypsum Lath[g] Plastered Plus 1 in. Blanket Insulation—Furred[h]	
			A	B	C	D	E	F	G	H	I	
Solid[a] Brick		8	0.50	0.46	0.32	0.31	0.30	0.22	0.22	0.16	0.14	67
		12	0.36	0.34	0.25	0.25	0.24	0.19	0.19	0.14	0.13	68
		16	0.28	0.27	0.21	0.21	0.20	0.17	0.16	0.13	0.12	69
Hollow[b] Tile (Stucco Exterior Finish)		8	0.40	0.37	0.27	0.27	0.26	0.20	0.20	0.15	0.13	70
		10	0.39	0.37	0.27	0.27	0.26	0.20	0.19	0.15	0.13	71
		12	0.30	0.28	0.22	0.22	0.21	0.17	0.17	0.13	0.12	72
		16	0.24	0.24	0.19	0.19	0.18	0.15	0.15	0.12	0.11	73
Stone[c]		8	0.70	0.64	0.39	0.38	0.36	0.26	0.25	0.18	0.16	74
		12	0.57	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	75
		16	0.49	0.45	0.31	0.31	0.29	0.22	0.22	0.16	0.14	76
		24	0.37	0.35	0.26	0.26	0.25	0.19	0.19	0.15	0.13	77
Poured Concrete[d]		6	0.79	0.71	0.42	0.41	0.39	0.27	0.26	0.19	0.16	78
		8	0.70	0.64	0.39	0.38	0.36	0.26	0.25	0.18	0.16	79
		10	0.63	0.58	0.37	0.36	0.34	0.25	0.24	0.18	0.15	80
		12	0.57	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	81
Hollow Concrete Blocks		Gravel Aggregate										
		8	0.56	0.52	0.34	0.34	0.32	0.24	0.23	0.17	0.15	82
		12	0.49	0.46	0.32	0.31	0.30	0.22	0.22	0.16	0.14	83
		Cinder Aggregate										
		8	0.41	0.39	0.28	0.28	0.27	0.21	0.20	0.15	0.13	84
		12	0.38	0.36	0.26	0.26	0.25	0.20	0.19	0.15	0.13	85
		Light Weight Aggregate[e]										
		8	0.36	0.34	0.26	0.25	0.24	0.19	0.19	0.15	0.13	86
		12	0.34	0.33	0.25	0.24	0.24	0.19	0.18	0.14	0.13	87

(See text p. 182.)

[a] Based on 4 in. hard brick and remainder common brick.

[b] The 8 in. and 10 in. tile figures are based on two cells in the direction of heat flow. The 12 in. tile is based on three cells in the direction of heat flow. The 16 in. tile consists of one 10 in. and one 6 in. tile, each having two cells in the direction of heat flow.

[c] Limestone or sandstone.

[d] These figures may be used with sufficient accuracy for concrete walls with stucco exterior finish.

[e] Expanded slag, burned clay or pumice.

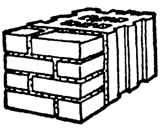
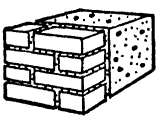
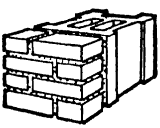
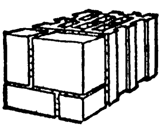
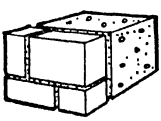
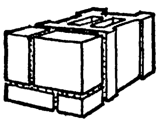
[f] Thickness of plaster assumed ¾ in.

[g] Thickness of plaster assumed ½ in.

[h] Based on 2 in. furring strips; one air space.

TABLE 9. COEFFICIENTS OF TRANSMISSION (U) OF BRICK AND STONE VENEER MASONRY WALLS

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

TYPICAL CONSTRUCTION	FACING	BACKING	INTERIOR FINISH (PLUS INSULATION WHERE INDICATED)									WALL NUMBER
			Plain Walls—no Interior Finish	Plaster (½ in.) on Walls	Metal Lath and Plaster[e]—Furred[g]	Gypsum Board (⅜ in). Decorated—Furred[g]	Gypsum Lath (⅜ in.) Plastered[f]—Furred[g]	Insulating Board (½ in.) Plain or Decorated—Furred[g]	Insulating Board Lath (½ in.) Plastered[f]—Furred[g]	Insulating Board Lath (1 in.) Plastered[f]—Furred[g]	Gypsum Lath Plastered[f] Plus 1 in. Blanket Insulation—Furred[g]	
			A	B	C	D	E	F	G	H	I	
	4 In. Brick Veneer[a]	6 In. Hollow Tile[b]	0.35	0.34	0.25	0.25	0 24	0.19	0.18	0.14	0.13	88
		8 In. Hollow Tile[b]	0.34	0.32	0.25	0.24	0.23	0.19	0.18	0.14	0.13	89
		6 In. Concrete	0.59	0.54	0.35	0.35	0.33	0.24	0.23	0.17	0.15	90
		8 In. Concrete	0.54	0.50	0.33	0.33	0.31	0.23	0.23	0.17	0.15	91
		8 In. Concrete Blocks[c] (Gravel Aggregate)	0.44	0.41	0.29	0.29	0.28	0.21	0.21	0.16	0.14	92
		8 In. Concrete Blocks[c] (Cinder Aggregate)	0.34	0.33	0.25	0.24	0.24	0.19	0.18	0.14	0.13	93
		8 In. Concrete Blocks[c] (Light Weight Aggregate)[d]	0.31	0.29	0.23	0.23	0.22	0.18	0.17	0.14	0.12	94
	4 In. Cut Stone Veneer[a]	6 In. Hollow Tile[b]	0.37	0.35	0.26	0.26	0.25	0.19	0.19	0.15	0.13	95
		8 In. Hollow Tile[b]	0.36	0.34	0.25	0.25	0.24	0.19	0.19	0.14	0.13	96
		6 In. Concrete	0.63	0.58	0.37	0.36	0.34	0 25	0.24	0.18	0.15	97
		8 In. Concrete	0.57	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	98
		8 In. Concrete Blocks[c] (Gravel Aggregate)	0.47	0.44	0.30	0.30	0.29	0.22	0.21	0.16	0 14	99
		8 In. Concrete Blocks[c] (Cinder Aggregate)	0.36	0.34	0.25	0.25	0.24	0.19	0.19	0.15	0.13	100
		8 In. Concrete Blocks[c] (Light Weight Aggregate)[d]	0.32	0.30	0.23	0.23	0.22	0.18	0.17	0.14	0.12	101

(See text p. 182.)

[a] Calculation based on ½ in. cement mortar between backing and facing, except in the case of the concrete backing which is assumed to be poured in place.

[b] The hollow tile figures are based on two air cells in the direction of heat flow.

[c] Hollow concrete blocks.

[d] Expanded slag, burned clay or pumice.

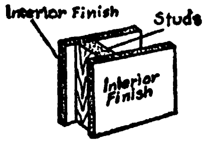
[e] Thickness of plaster assumed ¾ in.

[f] Thickness of plaster assumed ½ in.

[g] Based on 2 in. furring strips; one air space.

Table 10. Coefficients of Transmission (U) of Frame Partitions or Interior Walls[a]

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on still air (no wind) conditions on both sides.



Interior Finish	Single Partition (Finish on one side only of studs)	Double Partition (Finish on both sides of studs)		Partition Number
		No insulation between studs	1 in. Blanket[d] between studs. One air space.	
	A	B	C	
Metal Lath and Plaster[b]	0.69	0.39	0.16	1
Gypsum Board (⅜ in.) Decorated	0.67	0.37	0.16	2
Wood Lath and Plaster	0.62	0.34	0.15	3
Gypsum Lath (⅜ in.) Plastered[c]	0.61	0.34	0.15	4
Plywood (⅜ in.) Plain or Decorated	0.59	0.33	0.15	5
Insulating Board (½ in.) Plain or Decorated	0.36	0.19	0.11	6
Insulating Board Lath (½ in.) Plastered[c]	0.35	0.18	0.11	7
Insulating Board Lath (1 in.) Plastered[c]	0.23	0.12	0.082	8

(See text p. 182.)

[a] Coefficients not weighted; effect of studding neglected.

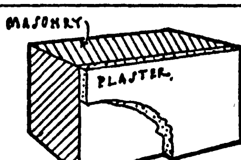
[b] Plaster assumed ¾ in. thick.

[c] Plaster assumed ½ in. thick.

[d] For partitions with other insulations between studs refer to Table 7, using values in Column B of above table, in left-hand column of Table 7. *Example:* What is the coefficient of transmission (U) of a partition consisting of gypsum lath and plaster on both sides of studs with 2 in. blanket between studs? *Solution:* According to above table, this partition with no insulation between studs (No. 4B) has a coefficient of 0.34. Referring to Table 7, it will be found that a wall having a coefficient of 0.34 with no insulation between studs, will have a coefficient of 0.10 with 2 in. of blanket insulation between studs (No. 56B).

Table 11. Coefficients of Transmission (U) of Masonry Partitions

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on still air (no wind) conditions on both sides.



Type of Partition		Thickness of Masonry (Inches)	Type of Finish			Partition Number
			No Finish (Plain walls)	Plaster One Side	Plaster Both Sides[a]	
			A	B	C	
Hollow Clay Tile		3	0.50	0.47	0.43	9
		4	0.45	0.42	0.40	10
Hollow Gypsum Tile		3	0.35	0.33	0.32	11
		4	0.29	0.28	0.27	12
Hollow Concrete Tile or Blocks	Cinder Aggregate	3	0.50	0.47	0.43	13
		4	0.45	0.42	0.40	14
	Light Weight Aggregate[b]	3	0.41	0.39	0.37	15
		4	0.35	0.34	0.32	16
Common Brick		4	0.50	0.46	0.43	17

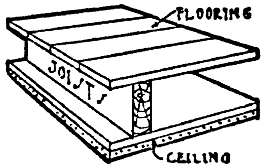
(See text p. 182.)

[a] 2 in. solid plaster partition, U = 0.53.

[b] Expanded slag, burned clay or pumice.

TABLE 12. COEFFICIENTS OF TRANSMISSION (U) OF FRAME CONSTRUCTION CEILINGS AND FLOORS

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference between the air on the two sides) and are based on still air (no wind) conditions on both sides.



TYPE OF CEILING	INSULATION BETWEEN, OR ON TOP OF, JOISTS (No Flooring Above)												WITH FLOORING[g] (On Top of Ceiling Joists)		Number
	None	Insulating Board on Top of Joists		Blanket or Bat Insulation[f] Between Joists[a]			Vermiculite Insulation Between Joists[a]			Mineral Wool Insulation Between Joists[a]			Single Wood Floor[b]	Double Wood Floor[c]	
		½ In.	1 In.	1 In.	2 In.	3 In.	2 In.	3 In.	4 In.	2 In.	3 In.	4 In.			
	A	B	C	D	E	F	G	H	I	J	K	L	M	N	
No Ceiling		0.37	0.24										0.45	0.34	1
Metal Lath and Plaster[d]	0.69	0.26	0.19	0.19	0.12	0.093	0.18	0.14	0.11	0.12	0.093	0.077	0.30	0.25	2
Gypsum Board (3/8 in.) Plain or Decorated	0.67	0.26	0.18	0.19	0.12	0.092	0.18	0.13	0.10	0.12	0.092	0.077	0.30	0.24	3
Wood Lath and Plaster	0.62	0.25	0.18	0.19	0.12	0.091	0.17	0.13	0.10	0.12	0.091	0.076	0.28	0.24	4
Gypsum Lath (3/8 in.) Plastered[e]	0.61	0.25	0.18	0.19	0.12	0.091	0.17	0.13	0.10	0.12	0.091	0.076	0.28	0.24	5
Plywood (3/8 in.) Plain or Decorated	0.59	0.24	0.18	0.19	0.12	0.091	0.17	0.13	0.10	0.12	0.091	0.076	0.28	0.23	6
Insulating Board (½ in.) Plain or Decorated	0.36	0.19	0.15	0.16	0.10	0.082	0.14	0.12	0.097	0.10	0.082	0.069	0.22[h]	0.19[h]	7
Insulating Board Lath (½ in.) Plastered[e]	0.35	0.19	0.15	0.15	0.10	0.081	0.14	0.11	0.096	0.10	0.081	0.068	0.21	0.18	8
Insulating Board Lath (1 in.) Plastered[e]	0.23	0.15	0.12	0.12	0.089	0.072	0.12	0.097	0.084	0.089	0.072	0.061	0.16	0.14	9

(See text p. 184)

[a] Coefficients corrected for framing on basis of 15 percent area, 2 in. x 4 in. (nominal) framing, 16 in on centers.

[b] 25/32 in. yellow pine or fir.

[c] 25/32 in. pine or fir sub-flooring plus 13/16 in. hardwood finish flooring.

[d] Plaster assumed ¾ in. thick.

[e] Plaster assumed ½ in. thick.

[f] Based on insulation in contact with ceiling, and consequently no air space between.

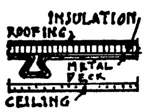
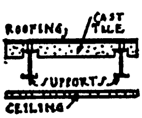
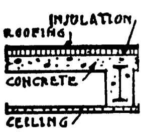
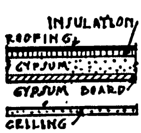
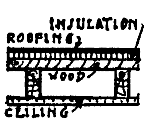
[g] For coefficients for constructions in Columns M and N (except No. 1) with insulation between joists, refer to Table 7. *Example:* The coefficient for No. 3-N of Table 12 is 0.24. With 2 in. blanket insulation between joists, the coefficient will be 0.093. (See Table 7.) (Column D of Table 7 applicable only for 3⅝ in. joists.)

[h] For 25/32 in. insulating board sheathing applied to the under side of the joists, the coefficient for *single* wood floor (Column M) is 0.18 and for *double* wood floor (Column N) is 0.16. For coefficients with insulation between joists, see Table 7.

TABLE 16. COEFFICIENTS OF TRANSMISSION (U) OF FLAT ROOFS COVERED WITH BUILT-UP ROOFING. WITH LATH AND PLASTER CEILINGS[a]

(See Table 15 for Flat Roofs with No Ceilings)

These coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

Type of Roof Deck	Thickness of Roof Deck (Inches)	No Insulation	Insulation on Top of Deck (Covered with Built-Up Roofing)							Number
			Insulating Board (Thickness Below)				Corkboard (Thickness Below)			
			½ In.	1 In.	1½ In.	2 In	1 In.	1½ In.	2 In.	
		A	B	C	D	E	F	G	H	
Flat Metal Roof Deck 		0.46	0.27	0.19	0.15	0.12	0.18	0.14	0.11	12
Precast Cement Tile 	1⅝ in	0.43	0.26	0.19	0.15	0.12	0.18	0.14	0.11	13
Concrete 	2 in.	0.42	0.26	0.19	0.14	0.12	0.18	0.14	0.11	14
	4 in.	0.40	0 25	0.18	0 14	0.12	0.17	0.13	0.11	15
	6 in	0.37	0 24	0.18	0.14	0.11	0.17	0.13	0.11	16
Gypsum Fiber Concrete[b] on ½ in. Gypsum Board 	2½ in.	0.27	0.19	0.15	0.12	0.10	0.14	0.12	0.097	17
	3½ in	0.23	0.17	0.14	0.11	0.097	0.13	0.11	0.091	18
Wood[c] 	1 in.	0.31	0.21	0.16	0.13	0.11	0.15	0.12	0.10	19
	1½ in.	0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.095	20
	2 in.	0.24	0.17	0.14	0.11	0.097	0.13	0.11	0.092	21
	3 in.	0.18	0.14	0.12	0.10	0.087	0.11	0.095	0.082	22

[a] Calculations based on metal lath and plaster ceilings, but coefficients may be used with sufficient accuracy for gypsum lath or wood lath and plaster ceilings. It is assumed that there is an air space between the under side of the roof deck and the upper side of the ceiling.

[b] 87½ percent gypsum, 12½ percent wood fiber. Thickness indicated includes ½ in. gypsum board.

[c] Nominal thicknesses specified—actual thicknesses used in calculations.

TABLE 17. COEFFICIENTS OF TRANSMISSION (U) OF PITCHED ROOFS

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

TYPE OF CEILING (APPLIED DIRECTLY TO ROOF RAFTERS)	WOOD SHINGLES (ON 1 x 4 WOOD STRIPS[a] SPACED 2 IN. APART)				ASPHALT SHINGLES OR ROLL ROOFING (ON SOLID WOOD SHEATHING)[a]				SLATE OR TILE[b] (ON SOLID WOOD SHEATHING)[a]				NUMBER
	INSULATION BETWEEN RAFTERS				INSULATION BETWEEN RAFTERS				INSULATION BETWEEN RAFTERS				
	None	Blanket or Bat (Thickness Below)			None	Blanket or Bat (Thickness Below)			None	Blanket or Bat (Thickness Below)			
		1 In.	2 In.	3 In.		1 In.	2 In.	3 In.		1 In.	2 In.	3 In.	
	A	B	C[c]	D[c]	E	F	G[c]	H[c]	I	J	K[c]	L[c]	
No Ceiling Applied to Rafters	0.48[f]	0.15	0.10	0.081	0.52[f]	0.15	0.11	0.084	0.55[f]	0.16	0.11	0.085	1
Metal Lath and Plaster[d]	0.31	0.14	0.10	0.081	0.33	0.15	0.10	0.083	0.34	0.15	0.10	0.083	2
Gypsum Board (3/8 in.) Decorated	0.30	0.14	0.10	0.080	0.32	0.15	0.10	0.082	0.33	0.15	0.10	0.083	3
Wood Lath and Plaster	0.29	0.14	0.10	0.080	0.31	0.14	0.10	0.081	0.32	0.15	0.10	0.082	4
Gypsum Lath (3/8 in.) Plastered[e]	0.29	0.14	0.10	0.079	0.31	0.14	0.10	0.081	0.32	0.15	0.10	0.082	5
Plywood (3/8 in.) Plain or Decorated	0.29	0.14	0.099	0.079	0.30	0.14	0.10	0.081	0.31	0.15	0.10	0.081	6
Insulating Board (1/2 in.) Plain or Decorated	0.22	0.12	0.090	0.072	0.23	0.12	0.091	0.074	0.24	0.13	0.092	0.074	7
Insulating Board Lath (1/2 in.) Plastered[e]	0.22	0.12	0.088	0.072	0.22	0.12	0.090	0.073	0.23	0.12	0.091	0.074	8
Insulating Board Lath (1 in.) Plastered[e]	0.16	0.10	0.078	0.064	0.17	0.10	0.079	0.065	0.17	0.10	0.080	0.066	9

(See text on p. 184.)

[a] Coefficients corrected for framing on basis of 15 percent area, 2 in. x 4 in. (nominal), 16 in. on centers.

[b] Figures in Columns I, J, K and L may be used with sufficient accuracy for rigid asbestos shingles on wood sheathing. Layer of slater's felt neglected.

[c] Sheathing and wood strips assumed 25/32 in. thick.

[d] Plaster assumed 3/4 in. thick.

[e] Plaster assumed 1/2 in. thick.

[f] No air space included in 1-A, 1-E or 1-I; all other coefficients based on one air space.

Table 18. Combined Coefficients of Transmission (U) of Pitched Roofs[a] and Horizontal Ceilings—Based on Ceiling Area[b]

Coefficients are expressed in Btu per (hour) (square foot of ceiling area) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

Ceiling Coefficient[f] (From Table 12)	Type of Roofing and Roof Sheathing						Number
	Wood Shingles on Wood Strips[d]			Asphalt Shingles[c] or Roll Roofing on Wood Sheathing[e]			
	No Roof Insulation (Rafters Exposed) (U_r = 0.48)	½ In. Insulating Board on Under Side of Rafters (U_r = 0.22)	1 In. Insulating Board on Under Side of Rafters (U_r = 0.16)	No Roof Insulation (Rafters Exposed) (U_r = 0.53)	½ In. Insulating Board on Under Side of Rafters (U_r = 0.23)	1 In. Insulating Board on Under Side of Rafters (U_r = 0.17)	
	A	B	C	D	E	F	
0.10	0.085	0.073	0.066	0.087	0.074	0.067	19
0.11	0.092	0.078	0.07	0.094	0.079	0.071	20
0.12	0.090	0.082	0.074	0.10	0.083	0.075	21
0.13	0.11	0.087	0.078	0.11	0.088	0.079	22
0.14	0.11	0.091	0.081	0.11	0.093	0.083	23
0.15	0.12	0.096	0.084	0.12	0.097	0.086	24
0.16	0.13	0.10	0.087	0.13	0.10	0.089	25
0.17	0.13	0.10	0.090	0.13	0.10	0.092	26
0.18	0.14	0.11	0.093	0.14	0.11	0.095	27
0.19	0.14	0.11	0.095	0.15	0.11	0.098	28
0.20	0.15	0.11	0.098	0.15	0.12	0.10	29
0.21	0.15	0.12	0.10	0.16	0.12	0.10	30
0.22	0.16	0.12	0.10	0.17	0.12	0.11	31
0.23	0.16	0.12	0.10	0.17	0.12	0.11	32
0.24	0.17	0.13	0.11	0.18	0.12	0.11	33
0.25	0.17	0.13	0.11	0.18	0.13	0.11	34
0.26	0.18	0.13	0.11	0.19	0.13	0.11	35
0.27	0.18	0.13	0.11	0.19	0.13	0.12	36
0.28	0.19	0.14	0.12	0.19	0.14	0.12	37
0.29	0.19	0.14	0.12	0.20	0.14	0.12	38
0.30	0.20	0.14	0.12	0.20	0.14	0.12	39
0.34	0.21	0.15	0.12	0.22	0.15	0.13	40
0.35	0.22	0.15	0.13	0.22	0.15	0.13	41
0.36	0.22	0.15	0.13	0.23	0.15	0.13	42
0.37	0.23	0.15	0.13	0.23	0.16	0.13	43
0.45	0.25	0.17	0.13	0.26	0.17	0.14	44
0.59	0.29	0.18	0.14	0.30	0.19	0.15	45
0.61	0.29	0.18	0.15	0.31	0.19	0.15	46
0.63	0.30	0.19	0.15	0.31	0.19	0.15	47
0.67	0.31	0.19	0.15	0.33	0.20	0.16	48
0.69	0.31	0.19	0.15	0.33	0.20	0.16	49

(See text on p. 184.)

[a] Calculations based on ⅓ pitch roof (n = 1.2) using the following formula:

$$U = \frac{U_r \times U_{ce}}{U_r + \frac{U_{ce}}{n}}$$

U = combined coefficient to be used with ceiling area.
U_r = coefficient of transmission of the roof.
U_{ce} = coefficient of transmission of the ceiling.
n = the ratio of the area of the roof to the area of the ceiling.

[b] Use ceiling area (not roof area) with these coefficients.

[c] Coefficients in Columns D, E and F may be used with sufficient accuracy for tile, slate and rigid asbestos shingles on wood sheating.

[d] Based on 1 x 4 in. strips spaced 2 in. apart.

[e] Sheathing assumed 25/32 in. thick.

[f] Values of U_{ce} to be used in this column may be selected from Table 12.

TABLE 19. COEFFICIENTS OF TRANSMISSION (U) OF SOLID WOOD DOORS

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based upon an outside wind velocity of 15 mph.

NOMINAL THICKNESS INCHES	ACTUAL THICKNESS INCHES	$U^{a,b}$ EXPOSED DOOR	$U^{a,b}$ WITH GLASS STORM DOOR[c]
1	25/32	0.69	0.35
1¼	1 1/16	0.59	0.32
1½	1 5/16	0.52	0.30
1¾	1⅜	0.51	0.30
2	1⅝	0.46	0.28
2½	2⅛	0.38	0.25
3	2⅝	0.33	0.23

[a] Computed using $k = 1.15$ for wood; $f_i = 1.65$, $f_o = 6.0$ and 1.10 for air space.
[b] A U value of 0.85 may be used for single exposed doors containing thin wood panels or single panes of glass, and 0.39 for the same with glass storm doors
[c] 50 percent glass and thin wood panels.

TABLE 20. COEFFICIENTS OF TRANSMISSION (U) OF WINDOWS, SKYLIGHTS AND GLASS BLOCK WALLS

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based upon the following outdoor conditions: 0 F air temperature, clear skies, no solar radiation

SECTION A—VERTICAL GLASS SHEETS

NUMBER OF SHEETS	ONE	TWO			THREE		
Air Space, inches	None	¼	½	1[a]	¼	½	1[a]
U	1.13	0.61	0.55	0.53	0.41	0.36	0.34

SECTION B—HORIZONTAL GLASS SHEETS (HEAT FLOW UP)

NUMBER OF SHEETS	ONE	TWO		
Air Space, inches	None	¼	½	1[a]
U	1.40	0.70	0.66	0.63

SECTION C—WALLS OF HOLLOW GLASS BLOCK

DESCRIPTION	U
5¾ x 5¾ x 3⅞ in. thick	0.60
7¾ x 7¾ x 3⅞ in. thick	0.56
7¾ x 7¾ x 3⅞ in. thick with glass fiber screen dividing the cavity	0.48

SECTION D—APPROXIMATE APPLICATION FACTORS FOR WINDOWS
MULTIPLY FLAT GLASS U VALUES BY THESE FACTORS

WINDOW DESCRIPTION	SINGLE GLASS		DOUBLE GLASS[b]		WINDOWS WITH STORM SASH[c]	
	Percent[d] Glass	Factor	Percent[d] Glass	Factor	Percent[d] Glass	Factor
Sheets	100	1.00	100	1.00		
Wood Sash	80	0.90	80	0.95	80	0.90
Wood Sash	60	0.80	60	0.85	60	0.80
Metal Sash	80	1.00	80	1.20	80	1.00[e]

(See text p. 187.)
[a] For 1 in. or greater.
[b] Unit type double glazing (two lights or panes in same opening).
[c] Use with U values for two sheets with 1 in. air space.
[d] Based on area of exposed portion of sash; does not include frame or portions of sash concealed by frame.
[e] For metal storm sash or metal sash with attached storm pane.

value of 1.65 used in computing U values given in other tables in this chapter. These values should therefore be used in estimating the temperature at which condensation on glass surfaces will occur.

The application factors given in Section D of Table 20 are based upon hot box tests summarized in a research bulletin[16], and are approximate only. In practice, some variation in heat flow through windows having the same ratio of glass to sash area, may be expected because of difference in construction details and in air space edge effects.

CALCULATING SURFACE TEMPERATURES

In many heating and cooling load calculations it is necessary to determine the inside surface temperature or the temperature of the surfaces within the structure. As the resistance of any path of heat flow is expressed in Fahrenheit degrees per (Btu) (hour) (square foot), the resistances through any two paths of heat flow would be proportional to the temperature drop through these paths, and can be expressed as follows:

$$\frac{R_1}{R_2} = \frac{(t_i - t_x)}{(t_i - t_o)} \tag{6}$$

where

R_1 = the resistance from the inside air to any point in the structure at which the temperature is to be determined.

R_2 = the overall resistance of the wall from inside air to outside air.

t_i = inside air temperature.

t_x = temperature to be determined.

t_o = outside air temperature.

Example 2: Determine the inside surface temperature for a wall having an overall coefficient of heat transmission U = 0.25, inside air temperature 70 F, outside air temperature −20 F.

Solution:

$$R_1 = 1/f_i = 1/1.65 = 0.606$$
$$R_2 = 1/U = 1/0.25 = 4.00$$

Then, by Equation 6

$$\frac{0.606}{4.00} = \frac{70 - t_x}{70 - (-20)}$$

$$t_x = 56.4 \text{ F}$$

The same procedure can be used for determining the temperature at any point within the structure.

A chart for determining inside wall surface temperature is given in Fig. 12 of Chapter 23, Panel Heating.

WATER VAPOR AND CONDENSATION IN CONSTRUCTION

Water as a vapor is present in all air and as adsorbed moisture in building materials such as wood. Even dense materials like glass hold considerable adsorbed moisture on their surfaces. In each of these places water may be harmless or even desirable, if its quantity is not excessive. Excessive moisture in building materials may cause mould, rot, and rust. Water blistering may damage seriously the exterior paint on wood siding when the siding moisture content rises above a safe level. While excessive moisture in building materials may be caused by rain leakage, it frequently is due to water vapor migration, a phenomenon likely to be associated with a temperature difference. Thus it may occur in the walls and roofs of heated buildings in winter, and in the enclosure of refrigerated spaces at all seasons. Water vapor released within a building, either incidentally or intentionally, may result in excessive moisture in the structure.

The behavior of water vapor is too often overlooked or given scant consideration in the design and construction of buildings and in the layout of air conditioning processes. It is an important factor to consider in the construction of residences and public buildings in cold climates and to a lesser extent in warm climates. It is extremely important to consider the moisture problem in the construction of cold storage and low temperature rooms. Manufacturing processes which demand a high humidity, require buildings designed to reduce the effect of moisture on the structure.

Moisture problems in residences occur in winter and become increasingly important as homes are built smaller and tighter. Water vapor originates from such necessary living requirements as cooking, laundering, bathing and the breathing and perspiration of people. In a typical family of four, the average daily production of water vapor from these sources may be as much as 25 lb, and may be much greater where such appliances as humidifiers, automatic washers and dryers are used.[16] Another large source of water vapor is sometimes the bare earth in a crawl space or basement. All this water vapor must escape from the dwelling.

Visible Condensation

Just as moisture collects on the outer surface of a glass of cold water, so does it also condense on other cold materials. In winter, visible condensation may collect on cold closet walls and attic roofs and is commonly observed on frosted window panes. Although condensation, if liquid, may enter an unpainted surface as fast as it forms and thus be unseen, any condensation on a visible surface will for convenience here be called *visible condensation* to distinguish it from *concealed condensation*. Within residences and public buildings, visible condensation occurs in winter and may damage decorative finishes and window sash.

Interior visible condensation occurs when any surface is colder than the dewpoint of the near-by air. The temperature of any such surface—wall, roof, or glass—is dependent upon the air temperature inside and outside the building and the heat transfer coefficient U of the surface structure. Based on a value of 1.65 for the inside surface conductance, Fig. 6 shows the relative humidity in a room at 70 F when visible condensation will appear at various U-values. The curves for single and double glass at their usual U-values are included. It should be noted that U-values as commonly used are an average for a large area within which there may be spots, such as the studs in an insulated wall, where the transmittance is higher. The inside surface temperature of a wall will, in general, be lower at the bottom due to such things as stratification of inside air and the effects of air leakage and of convection in walls with air spaces. Since condensation seeks the coldest spot, values from Fig. 6 can be applied only with caution. As a result, the limit of relative humidity for a non-homogeneous wall is lower than might be inferred from its average U-value.

The avoidance of interior visible condensation is partly a construction and partly an operating problem. It is accomplished by reducing the interior dew-point temperature or by raising the surface temperatures that are below the dew-point, or both. The dew-point temperature may be lowered by giving attention to the sources of the moisture, and in winter, may be controlled by ventilation, or possibly by some moisture absorption process. The temperatures of the inside room surfaces in winter may be increased by adding insulation to outside walls, by double glazing of windows, by circulating warm air over the surface, or perhaps by direct heating of the surface. The most expedient method of overcoming a surface condensation difficulty will depend upon special conditions surrounding the problem.

Vapor Transmission through Materials

The condensation of moisture within buildings is not limited to visible surfaces. Vapor permeates through certain materials very readily and may penetrate exterior or cold walls and contact material therein having a temperature below the dew point of the vapor. At these places the vapor will condense to form liquid water or frost. Such concealed condensation may, if excessive, cause serious damage which is particularly insidious when

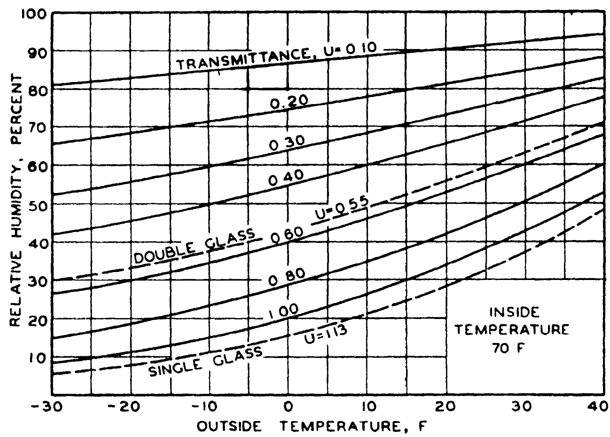


FIG. 6. RELATIVE HUMIDITY AT WHICH VISIBLE CONDENSATION WILL APPEAR ON INSIDE SURFACE

it continues without detection. An accumulation of hidden condensation often causes great difficulty in long-range processes.

The principal mechanism by which water vapor passes through solid materials is a process of diffusion, the net transfer requiring a difference of vapor pressure. Various writers have suggested the possibility that adsorbed moisture (which is neither vapor nor liquid) moves from a region of high concentration to one of lower concentration without the benefit of a vapor pressure difference, but this action has not been conclusively demonstrated and probably is negligible in the problems here considered. The property of a material which enables it to transmit vapor is known as its vapor *permeability*. Other forces which play an important part are capillarity and gravity (when the vapor changes to liquid at any point in its path), and the hygroscopic adsorption of moisture (which, for many materials, is nearly proportional to relative humidity).

The term *permeability* has frequently been applied to the rate of vapor transmission for the thickness of the material considered or tested, but this use is not consistent with the use of *conductivity* (thermal) which relates to a property of the material based on unit thickness. It has been suggested[17] that the term *permeance* (similar to *conductance* in heat transfer) be used when referring to any specimen of definite thickness, or an assembly of such pieces. This recommendation is followed in this chapter. The term *permeability*, as used herein, defines a property of the material and is numerically equal to the permeance of a unit thickness.

The theory covering water vapor transmission through materials leads to the following formula,

$$W = MAT\,\Delta p \tag{7}$$

where,

W = total weight of vapor transmitted through the specimen, grains.
A = area of the specimen, square feet.
T = time during which the transmission occurred, hours.
Δp = the difference of the vapor pressure across the specimen, inches of mercury.
M = the permeance of the specimen, in *perms*, or grains per (square foot) (hour) (inch of mercury vapor pressure difference).

The basic units in Equation 7 are favored by the building industry. The designation *perm* for the unit of permeance has been proposed[17] as a convenient substitute for the unit, 1 grain per (square foot) (hour) (inch of mercury vapor pressure difference), and this recommendation is followed herein.

The weight of vapor transmitted is unquestionably proportional to area and time, but is not always proportional to the vapor pressure difference. Proportionality is a useful relation when applied with caution in a limited range, but the expression *per inch of mercury* does not sanction an unrestricted extension of this relation. In other words, the permeance of a specimen is not a constant under every condition. This fact must be considered but is generally not an obstacle in the solution of many practical problems.

Vapor resistance is the reciprocal of permeance, and theory indicates that the vapor resistance of a homogeneous specimen is proportional to its thickness. Permeance, therefore, is inversely proportional to thickness, and:

$$M = \frac{\mu}{t}$$

$$\text{or, } \mu = \text{Mt} \tag{8}$$

where,

M = the permeance of the specimen, perms.
t = the thickness of the specimen, inches.

Evidently μ is the permeance of a unit thickness of the material, which is its permeability, as above defined. Using consistent units, permeability is expressed in *perm-inches*, a perm-inch being equal to one grain per (square foot) (hour) (inch of mercury per inch of thickness.)

Equations 7 and 8 may be combined to give:

$$W = \mu AT\frac{\Delta p}{t} \tag{9}$$

where,

μ = the average permeability of the material. (The spot permeability in thin elements may be progressively different throughout the thickness.)

The overall vapor resistance of an assembly (like a wall) of materials in series is the sum of the resistances of its component parts provided condensation does not take place within the assembly. Expressed in the more usual terms, the permeances (M_1, M_2, M_3, etc.) of the individual pieces may be combined by use of the formula

$$M = \frac{1}{\frac{1}{M_1} + \frac{1}{M_2} + \frac{1}{M_3} + \cdots \frac{1}{M_n}} \tag{10}$$

Equation 10 holds for materials that are reasonably homogeneous and in a condition of *steady state* where the transmission at all points is a vapor diffusion process as, for example, in a vapor transmission test. Actually, the conditions of moisture movement through a building wall are generally

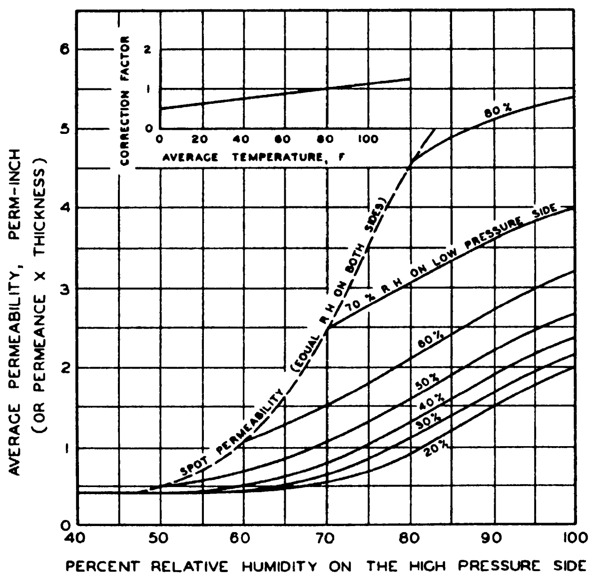


FIG. 7. PERMEABILITY OF WOOD (SUGAR PINE)

different. A steady state, where the entering and leaving moisture are equal, rarely exists, and frequently, the moisture in some portion of the path is liquid, in which case forces of capillarity and gravity are usually more important. It is therefore evident that the formula can be used only for certain portions of a building structure. Another caution is that the permeances of the several pieces must apply at the existing conditions.

The permeability of a material has been defined as one of its properties but it is not a fixed property for all conditions of exposure. Some materials like wood, because of their structure and hygroscopicity, are much more permeable to water vapor when the relative humidity is high. Since the equilibrium moisture content of permeable materials is increased to a greater or less degree by exposure to high relative humidity, it is likely that this sorbed moisture contributes to the mechanism of transfer.

The variations in the permeability of sugar pine wood are shown in Fig. 7.[18] It is notable that high relative humidity on either side of a specimen increases its permeance and the average permeability of the piece.

The *spot permeability* is shown, but the average is more readily used in practical calculation. Temperature also affects permeability, but for most materials is considered a minor factor, although data are few. These variations are to be expected in most materials and therefore, due care is required in choosing for each the proper value at its exposure conditions. Exact calculation by any of the preceding formulas, therefore, requires a knowledge of such variations as shown in Fig. 7 for each material, but approximate calculations are readily made and are adequate for most requirements.

Permeance Data and Testing

The simplest method of finding the vapor permeance of a specimen is to seal it over the top of a cup containing desiccant or water, placing it in a

TABLE 21. CONVERSION FACTORS FOR VAPOR TRANSFER UNITS

To Obtain ↓	Multiply Number of → By	WVT Units: $\frac{\text{grams}}{(24\ \text{hrs})\ (\text{sq m})}$	WVT Units: $\frac{\text{grains}}{(\text{hr})\ (\text{sq ft})}$
$\frac{\text{grams}}{(24\ \text{hrs})\ (\text{sq m})}$	by same method*	1	16.7
$\frac{\text{grains}}{(\text{hr})\ (\text{sq ft})}$	by same method*	0.0597	1
Perms by same method*	method A & B, 73.4 F method C & D, 90 F method E, 100 F	0.144 0.0840 0.0344	2.41 1.41 0.575

Nominal Test Conditions: Method	Temperature-F	% Relative Humidity on the Two Sides of Specimen: In cup	Outside cup
A	73.4	0	50
B	73.4	100	50
C	90	0	50
D	90	100	50
E	100	0	90

* Data obtained by one method cannot be reliably converted to another method.

controlled atmosphere, and weighing it periodically. The steady rate of weight gain or loss is normally the water vapor transfer. When the cup contains a desiccant the procedure is called the *dry-cup method* and when the cup contains water, the *wet-cup method*. Usually the outside atmosphere is held at 50 percent relative humidity, thus providing in either method substantially the same difference of vapor pressure, but the results obtained by the two methods on the same specimen are likely to be much different, the wet method producing the higher values.

It is obvious that any statement of permeance of a specimen should include the conditions of test. The permeance of a piece of material in a given service is best known if tested under conditions duplicating the service. Its permeance may be adequately judged, however, if it is tested by both dry and wet methods thus providing relative humidity conditions that usually include those to be encountered in service.

Unfortunately, the conditions of testing have not been standardized, and test data have frequently been presented in terms of weight transmitted per (unit area) (unit time). Such data may be called *water vapor transmission data* or WVT data and values are either high or low depending on the difference of vapor pressure chosen for the test. When this difference is known, WVT data can be converted to permeance, care being taken if conversion of the basic units (weight, area and time) is also required. The following formula applies:

$$\text{Permeance} = \frac{\text{WVT rating}}{\Delta p} \qquad (11)$$

where,

WVT rating = weight of vapor transmitted, grains per (sq ft) (hour).
Δp = vapor pressure difference in the test, in inches of mercury.
Permeance is expressed in perms.

Table 21 presents the conversion factors applicable to the commonly used units and test methods.

Table 22 presents some data on typical building materials showing in each case the source and method and, where applicable, the thickness tested.

Water-proofed building papers are listed in *Federal Specifications* UU-P-147, May 24, 1948, according to water vapor resistance required as:

Class A. For uses where a high degree of water vapor resistance is required.
Class B. For uses where a lower degree of water vapor resistance and of water resistance is required.
Class C. For uses where a moderate degree of water resistance is required.
Class D. For uses where low resistance to water vapor is required.

It may be noted that a paper may be *water-proof* i.e. possess water resistance, and still have low water vapor resistance.

Detail requirements in these specifications are given as follows, the specified WVT Test being a dry method at 73F:

Class A paper shall have a minimum tensile strength in each direction of either 35 lb per inch width, or 20 lb per inch width, as specified in the invitation for bids. Paper of both strengths shall have a minimum water resistance of 24 hr, and a maximum water vapor permeability (WVT) of 4 grams per square meter per 24 hr, (i.e. 0.576 perm).

Class B paper shall have a minimum tensile strength in each direction of either 35 lb per inch width, or 20 lb per inch width, as specified in the invitation for bids. Paper of both strengths shall have a minimum water resistance of 16 hr, and a maximum water vapor permeability (WVT) of 6 grams per square meter per 24 hr, (i.e. 0.864 perm).

Class C paper shall have a minimum tensile strength in each direction of either 35 lb per inch width, or 20 lb per inch width, as specified in the invitation for bids. Paper of both strengths shall have a minimum water resistance of 8 hr.

Class D paper shall have a minimum tensile strength in each direction of 20 lb per inch width. The paper shall have a minimum water resistance of 10 min., a minimum water vapor permeability (WVT) of 35 grams per square meter per 24 hr, (i.e 5.04 perms).

Concealed Condensation in Heated Buildings

Water vapor produced in a building necessarily raises the vapor pressure above that outside thus providing the force that causes its diffusion into exterior walls. The amount of vapor pressure rise in the building depends on the amount of vapor produced and inversely on its chance to escape. The resulting balance may be expressed in terms of relative humidity if the inside temperature is 70 F. The relative humidity in heated buildings

TABLE 22. PERMEANCE AND PERMEABILITY OF MATERIALS TO WATER VAPOR

MATERIAL	PERMEANCE PERM	PERMEABILITY PERM-INCH	% RH_1-RH_2	METHOD†	REF.‡
AIR (still)		120.	92-73	b	18
INSULATION					
Cellular glass		0.0		d	
Corkboard		2.1-2.6	75-0	d·	21
Corkboard		9.5	100-45	w	24
Structural Insulating Board (vegetable, uncoated)		20-50	40-x	t	22
Mineral Wool (unprotected)		116.	100-30	w	19
WOOD					
Sugar Pine (see Fig. 7)		0.4-5.4	various	tv	18
Plywood (Exterior type 3 ply D.F.), ¼ in.	0.72		50-	4	26
Plywood (Interior type 3 ply D.F.), ¼ in.	1.86		50-	4	26
MASONRY					
Concrete (1:2:4 Mix)		3.2	100-45	w	24
Concrete (8" cored block wall, limestone agrgt.)	2.4		79-68	t	18
Brick wall—with mortar—4 in.	0.8		50-x	t	23
Tile wall—with mortar—4 in.	0.12		50-x	t	23
INTERIOR FINISH					
Plaster on wood lath	11.		100-30	w	19
Plaster on metal lath—¾"	15.		40-x	t	22
Plaster on plain gypsum lath (with studs)	20.		40-85	t	18
Gypsum wall board—plain—⅜ in.	50.		50-20	v	28
Insulating wall board (uncoated)—½ in.	50-90		40-x	t	22
* PAINT—2 coats					
Asphaltic paint on plywood	0.4		100-30	w	19
Aluminum in varnish on wood	0.3-0.5		95-0	d	25
Enamels, brushed on smooth plaster	0.5-1.5		92-0	b	18
Primers or *Sealers* on insulating wall board	0.9-2.1		40-x	t	22
Various *Primers* + 1 coat flat paint on plaster	1.6-3.		40-x	t	22
Flat paint (alone) on insulating wall board	4.		40-x	t	22
Water Emulsions on insulating wall board	30.-85.		40-x	t	22
* PAINT—Exterior, 3 coats					
White *lead & oil* prepared paint on wood siding	0.3-1.0		50-0	d	28
White lead-zinc oxide & linseed oil on wood	0.9		95-0	d	25

	Lb. per 500 sq ft	PERMEANCE-PERMS dry cup	wet cup	
* BUILDING PAPERS AND FELTS				
Duplex sheet, asphalt laminae, *aluminum foil* one side	43	0.002	0.176	27
Saturated and *coated* felt heavy roll roofing	326	0.05	0.24	27
Kraft and *asphalt laminae*, Reinforced 30-120-30	34	0.3	1.8	27
Insulation back up, asphalt-sat., one side glossy	21	0.4	0.6-4.2	27
Asphalt-saturated and coated sheathing paper	43	0.3	0.6	27
Asphalt-saturated sheathing paper	22	3.3	20.2	27
15-pound asphalt felt	70	1.0	5.6	27
15-pound tar felt	70	4.0	18.2	27
Single sheet Kraft, double infused	16	30.8	41.9	27

* Description is a guide only, and does not insure permeance.

† Methods: d—dry cup; w—wet cup; t—two temperatures; b—special cell; v—air velocity both sides; 4—average of four methods.

‡ References. No. 22 also includes *Bulletins* 22 and 25 of the *Engineering Experiment Station, University of Minnesota*. No. 28 includes data to be published by the *Engineering Experiment Station, The Pennsylvania State College.*

covers nearly all of the possible range. In zero weather it may be only 10 percent in an office, and 85 percent in an industrial plant where humidification is required for a process, or where vapor release is incidental to a process. In residences the relative humidity in cold winter weather ranges from 10 percent to 60 percent, the latter figure applying to a very small, crowded and unventilated dwelling. A 40 percent level is considered representative of a substantial number of modern tightly constructed small houses although the average house relative humidity is probably below 25 percent. Surveys in residences show that the relative humidity increases as would be expected in warmer weather. Fig. 8 represents the results of one such survey.[19]

When water vapor is allowed to enter a wall and condensation occurs on its outer cold elements, it appears as frost or liquid. If the weather temperature rises frequently, frost melts and becoming liquid, is likely to penetrate capillary materials like wood, or run down when the surface is non-absorbing or is already saturated with water. In weather that is continuously cold for a long period, the frost may build back into a cavity or fibrous insulation and, when it reaches a warmer plane, will run to lower, cooler levels where it forms a mass of ice. Water seepage to the weather side may occur harmlessly in masonry walls when the weather is above freezing but water seepage into the building must obviously be avoided. In typical frame construction with wood sheathing which has large water absorbing capacity, seepage is rare and occurs only after a long period of steady cold weather. More generally, moisture accumulates in wood sheathing and siding through the colder months and reaches a peak in late

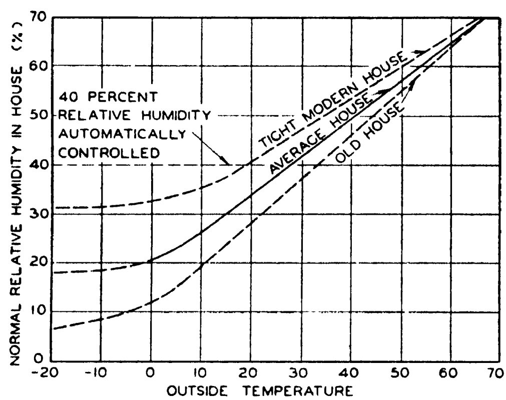


FIG. 8. RELATIVE HUMIDITY IN DWELLINGS

winter, after which the drying of spring and summer completes the annual cycle. The average winter temperature and its duration are factors in the condensation problem. In Fig. 9 the United States is divided into three condensation zones based on winter weather conditions.[20] Zone I roughly includes those areas where the design temperature is −20 F or colder; zone II those for which the design temperature is zero to −20 F, and zone III those at zero and warmer. Within each zone, similar degrees of condensation trouble are to be expected, and similar corrective measures apply.

In roofs, the condensation problem is much the same as in walls. The roof covering may be even more resistant to the escape of vapor than wall coverings such as paint; and while paint is likely to be ruptured by excessive moisture, no such relief occurs in roofs. Thus roofs furnish conspicuous examples of rapid decay in lumber.

In crawl spaces over uncovered damp ground, a large water evaporation sometimes occurs and causes condensation on the outer ends of floor joists and other members that are below the floor line and near to the outside. Water vapor from the crawl space may also enter walls, be transported by rising air in a stack effect, and even reach the attic by this route when the wall structure permits. Ventilation, as discussed later in this chapter, is an important correction factor in these cases.

Insulation in a wall or roof reduces heat loss and lowers the temperature of the outer elements of the structure, thus increasing the possibility of condensation if the vapor path to the cold surface is not blocked. Since low vapor resistance is a characteristic of fibrous insulation, the needed vapor resistance must be provided by other means. It is to be noted that, in typical residential conditions, condensation does not occur in fibrous insulation itself, except when frost has formed on sheathing and gradually

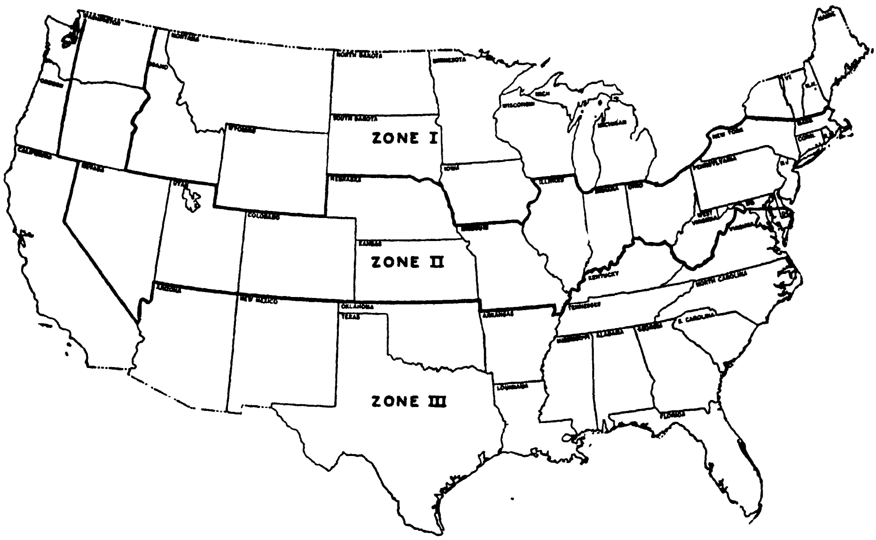


Fig. 9. Condensation Zones in the United States
(Zones Include Areas with Design Temperatures about as follows: Zone I, −20 F and lower; Zone II, 0 F to − 20 F; and Zone III, above 0 F)

built backward among the fibers. Wet insulation may result from this condition or from liquid condensation seeping down from a higher level.

Control of Concealed Condensation

An excessive accumulation of moisture in walls (or roofs) can be prevented by one or more of the following measures: (1) provide a vapor barrier to limit vapor entrance into the wall, (2) ventilate the building to reduce vapor pressure therein, (3) ventilate the wall cavity to remove vapor that has entered.

1. *Vapor Barrier.* A vapor barrier is the principal and most obvious correction, but each measure is more effective if aided by the effect of another. In habitations, some ventilation of the living space, either incidental or planned, is necessary. Also, a small amont of cavity ventilation is essential in cases where the vapor inflow is not completely stopped and the moisture storing capacity of the outer wall elements is slight. This applies to some prefabricated designs using metal siding.

Vapor barrier sheets are often built into the wall near the warm surface. In wood frame walls they may be applied to the inside surface of the studs. They are sometimes attached to the warm side of the insulating materials, or they may be applied on the cold side of plaster base materials. Special designs may be attached like wall paper to the inside of the wall, when satisfactory from the decorative view point. Sheet barriers often contain asphalt as the vapor resisting ingredient; metal foils, so placed that they are not too cold, may also be used.

The interior wall board or finish material may itself be vapor resistant, or a barrier coating may be applied to its concealed side when that side will not be too cold. The interior finished surface may be coated with a suitable paint having the required vapor resistance and also serving as the decorative finish, or it may be covered by another coat.

A paint coat on the interior finish, though of adequate resistance, is not likely to be so effective as a sheet barrier properly applied during the wall construction. This applies especially to houses of more than one story having cavities in ceilings which open into the outside walls. Such cavities allow vapor entering the ceiling to diffuse or be transported to the cold areas. Stoppage of this path is difficult, requiring normally the painting of the ceiling as well as the walls. Similar treatment may be required on internal partitions, or at least the first stud space adjacent to the cold wall.

The necessary barrier resistance depends on a number of factors. When the vapor flow occurs in annual cycles as in heated buildings, the requirement is not so exacting as it is for a cold storage room where there is no chance for drying out an accumulation of moisture. In a heated structure covered on the outside with materials highly resistant to water vapor such as paint or roll roofing, the winter season is a time of moisture accumulation in the cold outer elements and their safe moisture holding capacity is an important factor in determining the barrier requirement. A house without sheathing requires a better barrier; and a prefabricated design with only a sheet of metal outside of insulation requires very high barrier resistance. The interior vapor pressure and the length and severity of the winter are also important.

For typical frame dwellings with wood sheathing and siding in the northern United States, a barrier permeance of one perm or less has been found satisfactory. There are cases, however, in residential construction where a one perm barrier would not be adequate and there are also many industrial applications in which a very much higher vapor resistance is required. In any event, the choice of an adequate barrier implies that its permeance be definitely established. The usually accepted test procedure for this purpose is a dry method at a temperature of 70 F to 80 F. If obtainable at reasonable cost (including good application), a barrier better than required should be chosen for any construction. Despite the theoretical possibility of safely discharging some vapor through a wall, a higher than minimum permeance is not preferred.

An exact statement showing which buildings require a vapor barrier is not readily formulated. However, in view of the distressing results its omission may bring, it is tentatively recommended that the walls of every well constructed modern dwelling include a vapor barrier when the construction includes any material that would be damaged by moisture or its freezing. This applies to all condensation zones when the U value for the wall is lower than 0.25 Btu per (sq ft) (hr) (F deg), and it applies in zone I and zone II to walls of higher transmittance.

In applying vapor resistance to a wall, there are certain fundamental principles which should be followed. First, the vapor barrier should be placed as near to the warm surface of the wall as practicable. Second, it should be continuous with no direct openings through the barrier. Good workmanship and application are very important. Workmanship that leaves two openings through the barrier, or around its margin, at different levels, connecting air spaces at only slightly different temperatures, leaves a path for thermosyphon air rotation which will transport large amounts of water vapor from the warmer space to the colder. If a membrane barrier is used back of the plaster or interior finish, its joints should be made over some solid framing member, and not between the studs or in similar places. Usually a two-inch lap over a framing member will make a sufficiently tight joint when the interior finish is applied. Such a lap, however, without backing would not be adequate. Barriers attached to the warm side of insulation should form a continuous unbroken membrane over the entire insulated area. Edges should be lapped over framing members; ends of strips should be fastened by lapping over plates or headers. All openings

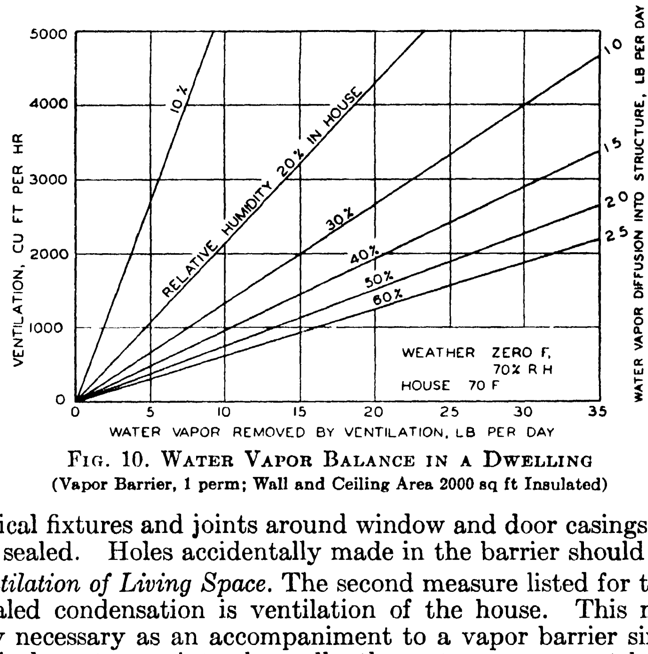


FIG. 10. WATER VAPOR BALANCE IN A DWELLING
(Vapor Barrier, 1 perm; Wall and Ceiling Area 2000 sq ft Insulated)

for electrical fixtures and joints around window and door casings should be carefully sealed. Holes accidentally made in the barrier should be sealed.

2. *Ventilation of Living Space.* The second measure listed for the control of concealed condensation is ventilation of the house. This measure is obviously necessary as an accompaniment to a vapor barrier since, if the barrier blocks entrance into the walls, the water vapor must be removed by other means. No great volume of air change is necessary, however, and normal infiltration alone is frequently all that is required in winter weather.

The effectiveness of ventilation is shown in Fig. 10, which also shows the small amount of water vapor escaping into the barrier equipped, well insulated walls and ceilings (2000 sq ft) of a typical small dwelling, the floor being neglected. Evidently, ventilation of 2000 cu ft per hr will remove 21 lbs of vapor per day with the relative humidity at 40 percent, while at the same time 1.5 lbs escapes into the structure. The total vapor production (22.5 lb) is a typical amount. Double glass will be barely safe from visible condensation as shown in Fig. 6. By reference to Chapter 6, 2000 cu ft per hr appears to be near the minimum for odor control and ventilation would have to be higher when cooking is done. By reference to Chapter 10, it appears that usual infiltration will normally supply the necessary air change, but that supplementary ventilation may be necessary in kitchen and laundry for proper vapor control and for the reduction of peaks in relative humidity which would otherwise occur in those areas.

3. *Ventilation of Structure.* The third measure listed for the control of concealed condensation, ventilation of the structure itself, is effective in certain cases especially as a supplement to warm side vapor resistance which is considerable but not of itself fully adequate. Air from outside is used. The vents must be shielded from the entrance of rain and insects.

Attics and crawl speces may be considered as parts of the structure, and for these portions ventilation is practically a necessity. Attic ventilation has long been an established practice but its effectiveness is likely to be diminished by the newer practice of adding insulation to ceilings. Insulation requires added ventilation which in turn necessitates adequate insulation. The recommended ventilation shown in Table 23 for dwellings[20] is based on such insulation. The *net area* refers to the total of all openings free from obstructions. The use of louvers and 8-mesh screen (usually recommended) requires a gross area 2.25 times that listed. In zone I, a ceiling vapor barrier is recommended for all constructions. It is also necessary that stray openings from walls into the attic, or around a loose fitting attic door be avoided. The *stack effect* allows a large inflow of warm air from the dwelling, transporting much vapor to a danger area. More desirable ventilation of the house can be arranged.

Crawl spaces under dwellings where the earth is damp and uncovered require a high rate of ventilation. At least four openings, one at each corner, as high as possible, should be provided.[20] Their total net area may be calculated by the formula:

$$a = \frac{2L}{100} + \frac{A}{300} \qquad (12)$$

where:

L = the perimeter of the crawl space, linear ft.
A = the area of the crawl space, square feet.
a = the total net area of all vents, (or the gross area if a 4-mesh screen is used), square feet.

This ventilation is usually sufficient but cools the first floor so much that insulation is needed. A better treatment is a cover on the damp ground. This cover may be a concrete slab, or merely heavy roll roofing laid on a graded surface with its edges lapped 2 in. (but not necessarily cemented). With this barrier, the vent area may be reduced to 10 percent of that calculated by Equation 12.

In building walls, cavity ventilation can be applied in a moderate climate as the sole vapor control system. In general, however, air passages in walls designed to remove an unrestricted vapor supply are unduly large and may waste considerable heat. On the other hand, a barrier as the only control measure would, in some cases, require so high a resistance as to be impractical. Ventilation of the structure in conjunction with a vapor barrier, is a procedure with important applications, but its general utility has not been fully investigated. Ventilation is most effective when each structural space has a clearly defined air passage with an inlet and outlet. In walls a small thermosyphon effect may be utilized by locating one vent at the bottom and one at the top of each space.

The best time to vapor-proof a building is during its construction. After a building is completed, ventilation of the occupied space is the most easily applied of the three basic control measures. Paint that is chosen for its low

TABLE 23. RECOMMENDED GOOD PRACTICE[20]-LOFT AND ATTIC VENTILATION[a]

FLAT ROOF—SLOPE LESS THAN 3 INCHES IN 12 INCHES
Condensation Zone I: Total net area of ventilation should be 1/300th[b] distributed uniformly at the eaves *plus* a vapor barrier in the top story ceiling. Free circulation must be provided through all spaces. Condensation Zone II and III: Same as for Zone I.
GABLE ROOF—SLOPE OVER 3 INCHES IN 12 INCHES
Condensation Zone I: Total net area of at least 2 louvers on opposite sides located near the ridge to be 1/300th[b] *plus* a vapor barrier in the top story ceiling. Condensation Zone II: Same ventilation as for Zone I. A vapor barrier is not considered necessary. Condensation Zone III: Same as for Zone II.
HIP ROOF
Condensation Zone I: Total net area of ventilation should be 1/300th[b] with 1/600th[b] distributed uniformly at the eaves *and* 1/600th[b] located at the ridge with all spaces interconnected. A vapor barrier should also be used in the top story ceiling. Condensation Zone II: Same ventilation as for Zone I. A vapor barrier is not considered necessary. Condensation Zone III: Same as for Zone II.
GABLE OR HIP ROOF—WITH OCCUPANCY CONTEMPLATED
Condensation Zone I: Total net area of ventilation shoud be 1/300th[b] with 1/600th[b] distributed uniformly at the eaves *and* 1/600th[b] located at the ridge with all spaces interconnected. A vapor barrier should also be used on the warm side of the top full story ceiling, the dwarf walls, the sloping part of the roof, and the attic story ceiling. Condensation Zone II: Same as for Zone I. Condensation Zone III: Same as for Zone I except that a vapor barrier is not considered necessary if insulation is omitted.

[a] It is recognized that in many areas increased ventilation may be desirable for summer comfort. For winter comfort, insulation is recommended between a living space and a loft or attic ventilated at these rates.
[b] Refers to area enclosed within building lines at eave level.

vapor permeance can be applied as a barrier on the interior with good results, care being taken that all areas, including parts of partitions and ceilings which offer an indirect vapor path to the cold wall, are covered. Ventilation of the wall cavity is effective in certain cases especially to supplement the foregoing measures. When such venting is required, each cavity space isolated by framing should be separately vented with an inlet and outlet judiciously placed, to accomplish proper air change.

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CHAPTER 10

INFILTRATION AND VENTILATION

Causes of Infiltration, Infiltration Due to Wind Pressure, Infiltration Due to Temperature Difference, Sealing of Vertical Openings, Natural Ventilation, Wind Forces, Temperature Difference Forces, Heat Removal, Effect of Unequal Openings, Combined Wind and Temperature Forces, Types of Openings, General Ventilation Rules, Ventilation of Animal Shelters, Garage Ventilation

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THE air leakage which takes place through various apertures in buildings must be considered in heating and cooling calculations, and properly evaluated. This *infiltration*, as it is sometimes designated, takes place through cracks around doors and windows, through solid walls, and through fireplaces and chimneys. Although the latter sources of leakage may be considerable, they are often neglected on the assumption that dampers would be closed during periods of extreme cold weather, or that the fireplace will be in use at such times, and will therefore contribute to the heat supplied and lessen the heating load.

CAUSES OF INFILTRATION

The displacement of heated air in buildings by unheated outside air is due to two causes, namely, (1) the pressure exerted by the wind, and (2) the difference in density of outside and inside air because of differences in temperature. The former is generally referred to as *infiltration* and the latter as *stack* or *chimney effect.*

In either case an exact estimate of the amount of infiltration under design conditions is difficult to make. The complicating factors include (1) variations in building construction, particularly as to width of crack or size of openings through which air leakage takes place; (2) the variations in wind velocity and direction; (3) the exposure of the building with respect to air leakage openings, and with respect to adjoining buildings; (4) the variations in outside temperatures which influence the chimney effect; (5) the relative area and resistance of openings on the windward and leeward sides, and on the lower floors and on the upper floors; and (6) the influence of a planned air supply and the related outlet vents. Tight construction is essential for preventing large heat loss due to infiltration.

INFILTRATION DUE TO WIND PRESSURE

The wind causes a pressure to be exerted on one or two sides of a building. As a result, air comes into the building on the windward side through cracks or porous construction, and a similar quantity of air leaves on the leeward side through like openings. In general, the resistance to air movement is similar on the windward to that on the leeward side. This causes a building up of pressure within the building, and a lesser air leakage than that experienced in single wall tests as determined in the laboratory. It is assumed that actual building leakages, owing to this building up of pressure, will be 80 per cent of laboratory test values. While there are cases where this is not true, tests in actual buildings substantiate the factor for the general case. Mechanical ventilating systems are frequently designed to produce positive or negative pressures in an enclosure, which are greater or lower than prevalent wind pressures. In such designs, if the specified rate at which air is to be supplied to, or removed from, the enclosure by positive means, exceeds the infiltration rate, it is common practice to use the greater value in determining the heating capacity to warm the outside air.

Infiltration Through Walls

Data on infiltration through brick and frame walls are given in Table 1.[1] The brick walls listed in this table are walls which show poor workmanship, and which are constructed of porous brick and lime mortar. For good workmanship, the leakage through hard brick walls with cement-lime mortar does not exceed one-third the values given. These tests indicate that plastering reduces the leakage by about 96 percent; a heavy

TABLE 1. INFILTRATION THROUGH WALLS[a]

Expressed in cubic feet per square foot per hour

Type of Wall		Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
8½ in. Brick Wall[b]	Plain	2	4	8	12	19	23
	Plastered[c]	0.02	0.04	0.07	0.11	0.16	0.24
13 in. Brick Wall[b]	Plain	1	4	7	12	16	21
	Plastered[c]	0.01	0.01	0.03	0.04	0.07	0.10
	Plastered[d]	0.03	0.10	0.21	0.36	0.53	0.72
Frame Wall, with lath and plaster[e]		0.03	0.07	0.13	0.18	0.23	0.26

[a] The values given in this table are 20 percent less than test values to allow for building up of pressure in rooms, and are based on test data reported in the papers listed in chapter footnotes.

[b] Constructed of porous brick and lime mortar—workmanship poor.

[c] Two coats prepared gypsum plaster on brick.

[d] Furring, lath, and two coats prepared gypsum plaster on brick.

[e] Wall construction: Bevel siding painted or cedar shingles, sheathing, building paper, wood lath and three coats gypsum plaster.

coat of cold water paint, 50 percent; and three coats of oil paint carefully applied, 28 percent. The infiltration through walls ranges from 6 to 25 percent of that through windows and doors in a 10-story office building, with imperfect sealing of plaster at the baseboards of the rooms. With perfect sealing the range is from 0.5 to 2.7 percent; or a practically negligible quantity, which indicates the importance of good workmanship in proper sealing at the baseboard. It will be noted from Table 1 that the infiltration through properly plastered walls can be neglected.

The value of building paper, when applied between sheathing and shingles, is indicated by Fig. 1, which represents the effect on outside construction only, without lath and plaster. The effectiveness of plaster properly applied is no justification for the use of low grade building paper, or of the poor construction of the wall containing it. Not only is it difficult to secure and maintain the full effectiveness of the plaster, but also it is highly desirable to have two points of high resistance to air flow with an

air space between them. The infiltration indicated in Fig. 1 is that determined in the laboratory, and should be multiplied by the factor 0.80 to give proper working values.

Window and Door Leakage

There are two methods of estimating air leakage through window and door cracks, namely, (1) the crack method, and (2) the air change method. The crack method is generally regarded as being more accurate than the air change method, provided the variables, such as crack width and clearance, can be properly evaluated.

Crack Method

The crack method is based on known air leakage factors for various types of windows, and widths of crack and clearance. The wind velocity

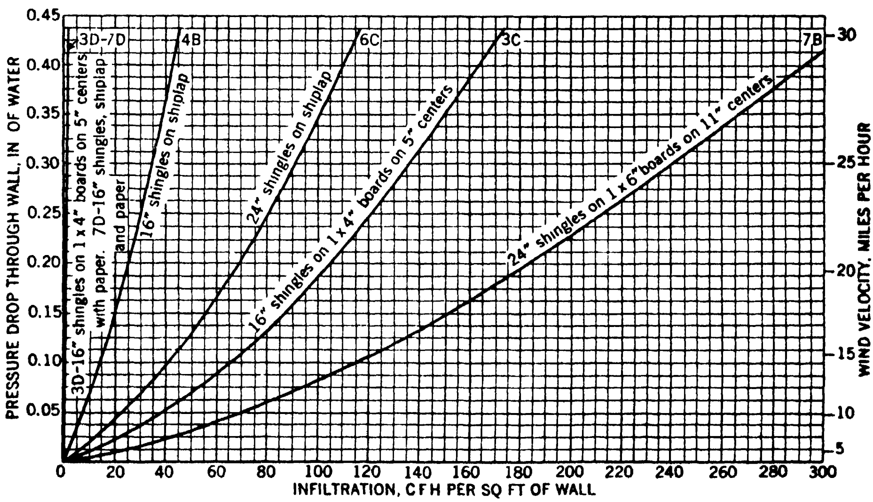

Fig. 1. Infiltration Through Various Types of Shingle Construction

and length of crack are also considered when the crack method is employed. The amount of infiltration for various types of windows is given in Table 2.[2] The fit of double-hung wood windows is determined by crack and clearance. Crack thickness is equivalent to one-half the difference between the inside window frame dimension and the outside sash width. The difference between the width of the window frame guide and the sash thickness is considered as the clearance. The length of the perimeter opening or crack for a double-hung window is equal to three times the width, plus two times the height, or in other words, it is the outer sash perimeter length, plus the meeting rail length. All of the window crack in any given room is not necessarily used in estimating the infiltration heat loss by the crack method. The length of crack to be selected in any given case depends on the number of exposed sides, as explained in Chapter 11.

Values of leakage shown in Table 2 for the average double-hung wood window were determined by using, on nine windows tested in the laboratory, the average measured crack and clearance of a large number of windows found in a field survey. In addition, the table gives figures for a

Table 2. Infiltration Through Windows

Expressed in Cubic Feet per Foot of Crack per Hour[a]

Type of Window	Remarks	Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
Double-Hung Wood Sash Windows (Unlocked)	Around frame in masonry wall—not calked[b]	3	8	14	20	27	35
	Around frame in masonry wall—calked[b]	1	2	3	4	5	6
	Around frame in wood frame construction[b]	2	6	11	17	23	30
	Total for average window, non-weather-stripped, 1/16-in. crack and 3/64-in. clearance.[c] Includes wood frame leakage[d]	7	21	39	59	80	104
	Ditto, weatherstripped[d]	4	13	24	36	49	63
	Total for poorly fitted window, non-weather-stripped, 3/32-in. crack and 3/32-in. clearance.[e] Includes wood frame leakage[d]	27	69	111	154	199	249
	Ditto, weatherstripped[d]	6	19	34	51	71	92
Double-Hung Metal Windows[f]	Non-weatherstripped, locked	20	45	70	96	125	154
	Non-weatherstripped, unlocked	20	47	74	104	137	170
	Weatherstripped, unlocked	6	19	32	46	60	76
Rolled Section Steel Sash Windows[k]	Industrial pivoted, 1/16-in. crack[g]	52	108	176	244	304	372
	Architectural projected, 1/32-in. crack[h]	15	36	62	86	112	139
	Architectural projected, 1/64-in. crack[h]	20	52	88	116	152	182
	Residential casement, 1/64-in. crack[i]	6	18	33	47	60	74
	Residential casement, 1/32-in. crack[i]	14	32	52	76	100	128
	Heavy casement section, projected, 1/64-in. crack[j]	3	10	18	26	36	48
	Heavy casement section, projected, 1/32-in. crack[j]	8	24	38	54	72	92
Hollow Metal, vertically pivoted window[f]		30	88	145	186	221	242

[a] The values given in this table, with the exception of those for double-hung and hollow metal windows are 20 percent less than test values to allow for building up of pressure in rooms, and are based on test data, reported in the papers listed in chapter footnotes.

[b] The values given for frame leakage are per foot of sash perimeter, as determined for double-hung wood windows. Some of the frame leakage in masonry walls originates in the brick wall itself, and cannot be prevented by calking. For the additional reason that calking is not done perfectly and deteriorates with time, it is considered advisable to choose the masonry frame leakage values for calked frames as the average determined by the calked and non-calked tests.

[c] The fit of the average double-hung wood window was determined as 1/16-in. crack and 3/64-in. clearance by measurements on approximately 600 windows under heating season conditions.

[d] The values given are the totals for the window opening per foot of sash perimeter, and include frame leakage and so-called *elsewhere leakage*. The frame leakage values included are for wood frame construction, but apply as well to masonry construction assuming a 50 percent efficiency of frame calking.

[e] A 3/32-in. crack and clearance represent a poorly fitted window, much poorer than average.

[f] Windows tested in place in building, so that no reduction from test values is necessary, as mentioned in footnote a.

[g] Industrial pivoted window generally used in industrial buildings. Ventilators horizontally pivoted at center or slightly above, lower part swinging out.

[h] Architecturally projected made of same sections as industrial pivoted, except that outside framing member is heavier, and it has refinements in weathering and hardware. Used in semi-monumental buildings such as schools. Ventilators swing in or out and are balanced on side arms. 1/32-in. crack is obtainable in the best practice of manufacture and installation, 1/64-in. crack considered to represent average practice.

[i] Of same design and section shapes as so-called *heavy section casement*, but of lighter weight. 1/64-in. crack is obtainable in the best practice of manufacture and installation, 1/32-in. crack considered to represent average practice.

[j] Made of heavy sections. Ventilators swing in or out and stay set at any degree of opening. 1/64-in. crack is obtainable in the best practice of manufacture and installation, 1/32-in. crack considered to represent average practice.

[k] With reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions, is negligible. With 3/64-in. crack, representing poor installation, leakage at contact with steel framework is about one third, and at mullions, about one-sixth of that given for industrial pivoted windows in the table.

poorly fitted window. All of the figures for double-hung wood windows are for the *unlocked* condition. Just how a window is closed, or fits when it is closed, has considerable influence on the leakage. The leakage will be high if the sash are short, if the meeting rail members are warped, or if the frame and sash are not fitted squarely to each other. It is possible to have a window with approximately the average crack and clearance that will have a leakage at least double that of the figures shown. Values for the average double-hung wood window in Table 2 are considered to be easily obtainable figures, provided the workmanship on the window is good. Should it be known that the windows under consideration are poorly fitted, the larger leakage values should be used. Locking a window generally decreases its leakage, but in some cases may push the meeting rail members apart and increase the leakage. On windows with large clearances, locking will usually reduce the leakage.

Wood casement windows may be assumed to have the same unit leakage as for the average double-hung wood window when properly fitted. Locking, a normal operation in the closing of this type of window, maintains the crack at a low value.

For metal pivoted sash, the length of crack is the total perimeter of the movable or ventilating sections. Frame leakage on steel windows may be neglected when they are properly grouted with cement mortar into brick work or concrete. When they are not properly sealed, the linear feet of

TABLE 3. INFILTRATION THROUGH 72-INCH REVOLVING DOOR AND 36-INCH SWINGING DOOR[a,b]

(Cubic Feet per Person per Passage)

Usage	Freely-Revolving Door	Door Equipped with Brake
Infrequent	75	60
Average	60	50
Heavy	40	40
39-Inch Swinging Door	100	

[a] These figures are based on the assumption that there is no wind pressure and that swinging doors are in use in one wall only. Any swinging doors in other walls should be kept closed to insure air conditioning in accordance with these recommended standards.

[b] From Application Engineering Standards for Air Conditioning for Comfort 1947, *Air Conditioning & Refrigerating Machinery Association, Inc.*, Washington, D. C. Used by permission.

sash section in contact with steel work at mullions should be figured at 25 percent of the values given in Table 2 for industrial pivoted windows.

When storm sash are applied to well fitted windows, some reduction in infiltration is secured; the application of the sash provides an air space which reduces the heat transmission and helps prevent the frosting of the windows.[3] By applying storm sash to poorly fitted windows, a reduction in leakage of 50 percent may be obtained, the effect so far as air leakage is concerned, being roughly equivalent to that obtained by the installation of weatherstrips.

Door Leakage

Doors vary greatly in fit because of their large size and tendency to warp. For a well fitted door, the leakage values for a poorly fitted double-hung wood window may be used. If poorly fitted, twice this figure should be used. If weatherstripped, the values may be reduced one-half. A single door which is frequently opened, as might be the case in a store, should have a value applied which is three times that for a well fitted door. This extra allowance is for opening and closing losses, and is kept from being greater by the fact that doors are not used as much in the coldest and windiest weather.

The infiltration rate through swinging and revolving doors is generally a matter of judgment by the engineer making *cooling* load determinations, and in the absence of adequate research data, the values given in Table 3 represent current engineering practice. Some tests of infiltration through swinging and revolving doors have been reported.[4] The data in Table 3 are indicative of what might be expected in this connection, but it should be noted that Table 3 is based on a no-wind condition, and therefore not directly applicable to *heating* design.

Air Change Method

The amount of air leakage may be estimated by assuming a certain number of air changes per hour for each room, the number of changes assumed being dependent upon the type, use, and location of the room, as indicated in Table 4. Where it is not possible to determine or pre-determine with accuracy the width of crack or clearance of windows, or where other sources of air leakage cannot readily be evaluated, as is often the case, the use of the air change method may be justified.[5]

The values in Table 4 may be used with reasonable accuracy for residences, and are the requirements for each room. The total infiltration allowance for the entire building should be one-half the sum of the infiltration allowances of the individual rooms, since whatever air enters on the windward side, generally leaves the building on the leeward side, and the infiltration requirements therefore do not exist simultaneously on all sides or in all rooms. An allowance of one air change per hour for all sources of air leakage for the entire volume may be considered average for a well constructed residence.

Table 4. Air Changes Taking Place under Average Conditions in Residences, Exclusive of Air Provided for Ventilation[a]

Kind of Room or Building	Number of Air Changes taking Place per Hour	Kind of Room or Building	Number of Air Changes taking Place per Hour
Rooms, 1 side exposed	1	Rooms with no windows or outside doors	½ to ¾
Rooms, 2 sides exposed	1½	Entrance Halls	2 to 3
Rooms, 3 sides exposed	2	Reception Halls	2
Rooms, 4 sides exposed	2	Bath Rooms	2

[a] For rooms with weatherstripped windows or storm sash, use ½ these values, where applicable, but never less than ½ air change.

The air leakage, due to opening and closing of doors in vestibules, is sometimes based on the air change method, even though the air leakage estimates for other rooms are based on the crack method. Except for vestibules and reception halls, it is not advisable to attempt to apply the air change method to factories and industrial and commercial buildings, because of wide variations in the type and percentage of fenestration which is the principal source of air leakage in such buildings.

INFILTRATION DUE TO TEMPERATURE DIFFERENCE

The air exchange due to temperature difference, inside to outside, is a chimney effect, causing air to enter through openings at lower levels, and to leave at higher levels.[6] Although it is not appreciable in low buildings, this loss should be considered in tall, single story buildings with openings near the ground level and near the ceiling. Also in tall, multi-story buildings it may be a considerable item, unless the sealing between various floors and rooms is quite perfect.

In tall buildings, temperature difference or chimney effect will produce

a head that will add to the effect of the wind at lower levels, and subtract from it at higher levels. On the other hand, the wind velocity at lower levels may be somewhat abated by surrounding obstructions. Furthermore, the chimney effect is reduced in multi-story buildings by the partial isolation of floors, thereby preventing free upward movement, so that wind and temperature difference may seldom cooperate to the fullest extent. Making the assumption that the *neutral zone*[7] is located at mid-height of a building, and that the temperature difference is 70 deg, Equations 1 and 2 may be used to determine an equivalent wind velocity to be used in connection with Tables 1 and 2, that will allow for both wind velocity and temperature difference:

$$V_e = \sqrt{V^2 - 1.75a} \tag{1}$$

$$V_e = \sqrt{V^2 + 1.75b} \tag{2}$$

where

V_e = equivalent wind velocity to be used in conjunction with Tables 1 and 2, miles per hour.
V = wind velocity upon which infiltration would be determined if temperature difference were disregarded, miles per hour.
a = distance of windows under consideration from mid-height of building if *above* mid-height, feet.
b = distance if *below* mid-height, feet.

The coefficient 1.75 allows for about one-half the temperature difference head.

For buildings of unusual height, Equation 1 would indicate negative infiltration at the highest stories, which condition may, at times, actually exist.

Sealing of Vertical Openings

In tall, multi-story buildings, every effort should be made to seal off vertical openings, such as stair-wells and elevator shafts, from the remainder of the building. Stair-wells should be equipped with self-closing doors, and, in exceptionally high buildings, should be closed off into sections of not over 10 floors each. Plaster cracks should be filled. Elevator enclosures should be tight, and solid doors should be used.

If the sealing of the vertical openings is made effective, no allowance need be made for the chimney effect. Instead, the greater wind movement at the greater heights makes it advisable to install additional heating surface on the upper floors above the level of neighboring buildings, this additional surface being increased as the height is increased. One arbitrary rule is to increase the heating surface on floors above neighboring buildings by an amount ranging from 5 per cent to 20 per cent. This extra heating surface is required only on the windward side and on windy days, and hence, automatic temperature control is especially desirable with such installations.

In stair-wells that are open through many floor levels, although closed off from the remainder of each floor by doors and partitions, the stratification of air makes it advisable to increase the amount of heating surface at the lower levels, and to decrease the amount at higher levels. One rule is to calculate the heating surface of the entire stair-well in the usual way, and to place 50 per cent of this in the bottom third, the normal amount in the middle third, and the balance in the top third.

Infiltration and Air for Combustion

Infiltration in buildings normally supplies the air required for combustion by fuel-burning appliances, but in some cases weatherstripping, sealing and calking may reduce infiltration to the point that special openings must be provided to supply adequate air to the heating appliances.

NATURAL VENTILATION

Ventilation by natural forces finds application in industrial plants, public buildings, schools, dwellings, garages, and in farm buildings.

The natural forces available for moving air into, through, and out of buildings are: (*a*) wind forces, and (*b*) the difference in temperature between the air inside and outside a building. The air movement may be caused by either of these forces acting alone, or by a combination of the two, depending upon atmospheric conditions, building design, and location. The ventilating results obtained will vary, from time to time, due to variation in the velocity and direction of the wind, and the temperature difference. The arrangement, location, and control of the ventilating openings should be such that the two forces act cooperatively rather than in opposition.

WIND FORCES

In considering the use of natural wind forces for producing ventilation, account must be taken of: (1) average wind velocity; (2) prevailing wind direction; (3) seasonal and daily variations in velocity and direction; and (4) local wind interference by nearby buildings, hills or other obstructions of similar nature.

Values are given in Table 3, Chapter 12 for the average wind velocities for the months June to September in various localities throughout the United States, while Table 1, Chapter 11, lists similar values for the winter. In almost all localities, the summer wind velocities are lower than those in the winter, and in about two-thirds of the localities the prevailing direction is different during the summer and winter. While the tables give no average velocities below 5 mph, there will be times when the velocity is lower, even in localities where the seasonal average is considerably above 5 mph. There are relatively few places where the velocity falls below one-half of the average for many hours per month. Consequently, if the natural ventilating system is designed for wind velocities of one-half of the average seasonal velocity, it should prove satisfactory in almost every case.

Equation 3 may be used for calculating the quantity of air forced through ventilation openings by the wind, or for determining the proper size of such openings to produce given results:

$$Q = EAV \tag{3}$$

where

Q = air flow, cubic feet per minute.
A = free area of inlet openings, square feet.
V = wind velocity, feet per minute, = miles per hour × 88.
E = effectiveness of openings. (E should be taken at 0.50 to 0.60 for perpendicular winds, and 0.25 to 0.35 for diagonal winds.[8])

The precision of results obtained by the use of Equation 3, depends upon the placing of the openings, as the formula assumes that ventilating openings have a flow coefficient slightly greater than that of a square-edged orifice. If the openings are not advantageously placed with respect to the wind, the flow per unit area of the openings will be less and, if unusually well placed, the flow will be slightly more than that given by the formula. Inlets should be placed to face directly into the prevailing wind, while outlets should be placed in one of the five places listed:

1. On the side of the building directly opposite the direction of the prevailing wind.
2. On the roof in the low pressure area caused by the jump of the wind (see Fig. 2).
3. On the sides adjacent to the windward face where low pressure areas occur.

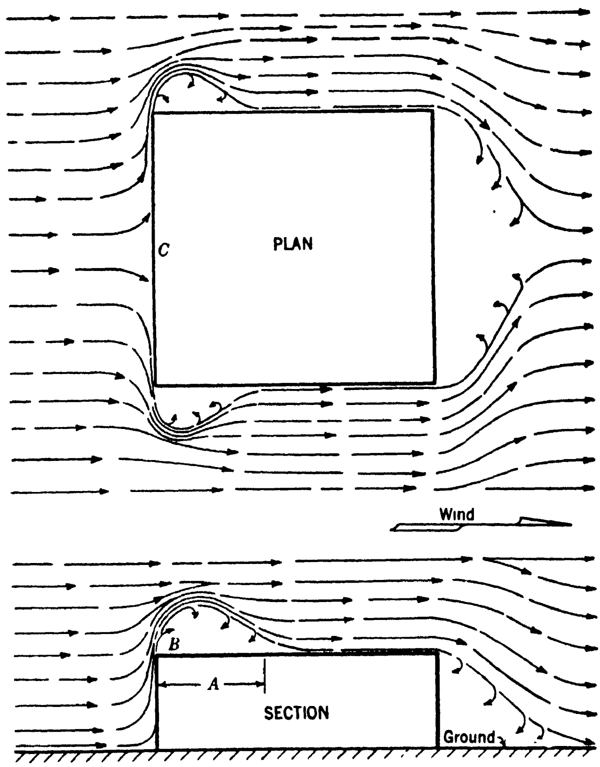

FIG. 2. THE JUMP OF WIND FROM WINDWARD FACE OF BUILDING. (A—LENGTH OF SUCTION AREA; B—POINT OF MAXIMUM INTENSITY OF SUCTION; C—POINT OF MAXIMUM PRESSURE)

4. In a monitor on the side opposite from the wind.
5. In roof ventilators or stacks.

TEMPERATURE DIFFERENCE FORCES[7]

The stack effect produced within a building, when the outdoor temperature is lower than the indoor temperature, is due to the difference in weight of the warm column of air within the building and cooler air outside. The flow due to stack effect is proportional to the square root of the draft head, or approximately

$$Q = 9.4A\sqrt{h(t - t_o)} \tag{4}$$

where

Q = air flow, cubic feet per minute.
A = free area of inlets or outlets (assumed equal), square feet.
h = height from inlets to outlets, feet.
t = *average* temperature of indoor air in height h, Fahrenheit degrees.
t_o = temperature of outdoor air, Fahrenheit degrees.
9.4 = constant of proportionality, including a value of 65 percent for effectiveness of openings. This should be reduced to 50 percent (constant = 7.2) if conditions are not favorable.

HEAT REMOVAL

In problems of heat removal, knowing the amount of heat to be removed and having selected a desirable temperature difference, the amount of

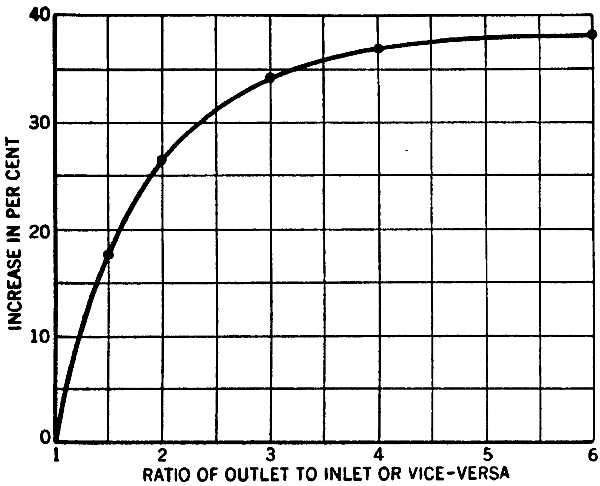

FIG. 3. INCREASE IN FLOW CAUSED BY EXCESS OF ONE OPENING OVER ANOTHER

air to be passed through the building per minute, to maintain this temperature difference, can be determined by means of Equation 5.

$$Q = \frac{H}{0.0175\,(t - t_o)} \qquad (5)$$

where

Q = air flow, cubic feet per minute.
H = heat removed, Btu per minute.
$t - t_o$ = inside-outside temperature difference, Fahrenheit degrees.

EFFECT OF UNEQUAL OPENINGS

The largest flow per unit area of openings is obtained when inlets and outlets are equal, and the preceding equations are based on this condition. Increasing outlets over inlets, or vice-versa, will increase the air flow, but not in proportion to the added area. When solving problems having an unequal distribution of openings, use the smaller area, either inlet or outlet, in the equations, and add the increase as determined from Fig. 3.

COMBINED FORCES OF WIND AND TEMPERATURE

Equations have already been given for determining the air flow due to temperature difference and wind. It must be remembered that when both forces are acting together, even without interference, the resulting air flow is not equal to the sum of the two estimated quantities. The flow through any opening is proportional to the square root of the sum of the heads acting on that opening.

When the two heads are about equal in value, and the ventilating openings are operated so as to coordinate them, the total air flow through the building is about 10 percent greater than that produced by either head acting independently under conditions ideal to it. This percentage de-

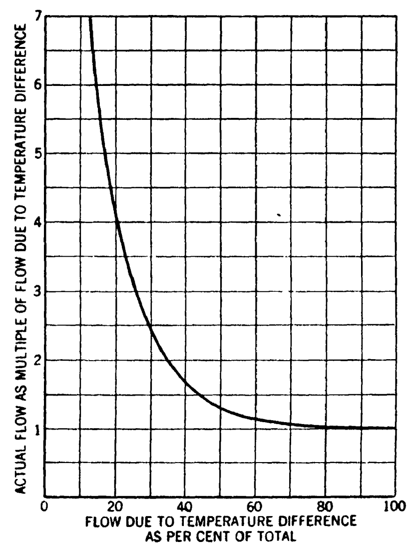

FIG. 4. DETERMINATION OF FLOW CAUSED BY COMBINED FORCES OF WIND AND TEMPERATURE DIFFERENCE

creases rapidly as one head increases over the other. The effect of the larger head will predominate.

The wind velocity and direction, the outdoor temperature, or the indoor distribution, cannot be predicted with certainty, and refinement in calculations is not justified; consequently, a simplified method can be used. This may be done by using the equations and calculating the flows produced by each force separately, under conditions of openings best suited for coordination of the forces. Then, by determining, as a percentage, the ratio of the flow produced by temperature difference to the sum of the two flows, the actual flow due to the combined forces can be approximated from Fig. 4.

Example 1: Assume a drop forge shop, 200 ft long, 100 ft wide, and 30 ft high. The cubical content is 600,000 cu ft, and the height of the air outlet over that of the inlet is 30 ft. Oil fuel of 18,000 Btu per lb is used in this shop at the rate of 15 gph (7.75 lb per gal). Desired summer temperature difference is 10 deg, and the prevailing wind is 8 mph perpendicular to the long dimension. What is the necessary area for the inlets and outlets, and what is the rate of air flow through the building?

Solution for Temperature Difference Only: The heat $H = \frac{15 \times 7.75 \times 18{,}000}{60} =$ 34,875 Btu per min.

By Equation 5, the air flow required to remove this heat with an average temperature difference of 10 deg is

$$Q = \frac{H}{0.0175(t - t_o)} = \frac{34{,}875}{0.0175 \times 10} = 199{,}286 \text{ cfm.}$$

This is equal to about 20 air changes per hour. From Equation 4, the inlet (or outlet) opening area should be

$$A = \frac{Q}{9.4\sqrt{h(t - t_o)}} = \frac{199{,}286}{9.4\sqrt{30 \times 10}} = 1224 \text{ sq ft.}$$

The flow per square foot of inlet or outlet would be 199,286 ÷ 1224 = 163 cfm, with all windows open.

Solution for Wind Only: With 1,224 sq ft of inlet openings distributed around the sidewalls, there will be about 410 sq ft in each long side and 202 sq ft in each end. The outlet area will be equally distributed on the two sides of the monitor, or 612 sq ft on each side. With the wind perpendicular to the long side, there will be 410 sq ft of opening in its path for inflow, and 612 in the lee side of the monitor for outflow, with the windward side closed. The air flow, as calculated by Equation 3, will be:

$$Q = 0.60 \times 410 \times 704 = 173{,}200 \text{ cfm.}$$

This gives 17.3 air changes per hour, which should be more than ample when there is no heat to be removed.

Solution to Combined Heads: Since the windward side of the monitor is closed when the wind is blowing, the flow due to temperature difference must be calculated for this condition, using Fig. 3. This chart shows that, when inlets are twice the size of the outlets, in this case 1,224 sq ft in the sidewalls and 612 sq ft in the monitor, the flow will be increased 26.5 percent over that produced by equal openings. Using the smaller opening and the flow per square foot obtained previously, the calculated amount for this condition will be

$$612 \times 163 \times 1.265 = 126{,}200 \text{ cfm.}$$

Adding the two computed flows:

Temperature Difference	= 126,200 =	42 percent.
Wind	= 173,200 =	58 percent.
Total	299,400 =	100 percent.

From Fig. 4, it is determined that, when the flow, due to temperature difference, is 42 percent of the total, the actual flow, due to the combined forces, will be about 1.6 times that calculated for temperature difference alone, or 201,920 cfm.

The original flow, due to temperature difference alone, was 199,286 cfm with all openings in use. The effect of the wind is to increase this to 201,920 cfm, even though half of the outlets are closed.

A factor of judgment is necessary in the location of the openings in a building, especially those in the roof, where heat, smoke, and fumes are to be removed. Usually, windward monitor openings should be closed, but if the wind is low enough for the temperature head to overcome it, all windows may be opened.

TYPES OF OPENINGS

Types of openings may be classified as: (1) windows, doors, monitor openings and skylights; (2) roof ventilators; (3) stacks connecting to registers; and (4) specially designed inlet or outlet openings. The various types and principles of operation are discussed in following paragraphs.

Windows, Doors and Skylights

Windows have the advantage of transmitting light, as well as providing ventilating area, when open. Their movable parts are arranged to open in various ways; they may open by sliding either vertically or horizontally, by tilting on horizontal pivots at or near the center, or by swinging on pivots at the top, bottom or side. Regardless of their design, the air flow per square foot of opening may be considered to be the same under the same conditions. The type of pivoting should receive consideration from the standpoint of weather protection, and certain types may be advantageous in controlling the distribution of incoming air. Deflectors are sometimes used for the same purpose, and these devices should be considered a part of the ventilation system.

Roof Ventilators

The function of a roof ventilator is to provide a storm and weatherproof air outlet. These are actuated by the same forces of wind and temperature head which create flow through other types of openings. The capacity of a ventilator depends upon four things: (1) its location on the roof; (2) the resistance it and the duct work offer to air flow; (3) the height of draft; and (4) the efficiency of the ventilator in utilizing the kinetic energy of the wind for inducing flow by centrifugal or ejector action.

For maximum flow induction, a ventilator should be located on that part of the roof where it will receive the full wind without interference. If ventilators are installed within the suction region created by the wind passing over the building, or in a light court, or on a low building between two high buildings, their performance will be seriously influenced. Their normal ejector action, if any, may be completely lost.

The base of the ventilator should be of a taper-cone design to produce the effect of a bell-mouth nozzle whose coefficient of flow is considerably higher than that of a square-entrance orifice. If a grille is provided at the base, or if the base or structural members present obstructions, additional resistance is introduced, and the base opening should be increased in size accordingly.

Air inlet openings located at lower levels in the building, should be at least equal to, and preferably larger than, the combined throat areas of all roof ventilators. The air discharged by a roof ventilator depends on wind velocity and temperature difference, and, in general, its performance will be the same as any monitor opening located in the same place but, due to the four capacity factors already mentioned, no simple formula can be devised for expressing ventilator capacity.

Roof ventilators may be classified as stationary, pivoting or oscillating, and rotating. Generally, these have a round throat, but the continuous-ridge ventilator would fall in the stationary classification. When selecting roof ventilators, some attention should be given to ruggedness of construction, storm-proofing features, dampers and damper operating mechanisms, possibility of noise, original cost, and maintenance.

Natural ventilation units may be used to supplement power-driven supply fans, and under favorable weather conditions it may be possible to stop the power-driven units. Units are not subject to code tests for ratings. Generally, they must be selected from manufacturers' tables. It is, therefore, very important to consider the reliability of the ratings used.

Controls

Gravity ventilators may have dampers controlled by hand, thermostat, or wind velocity, in combination with a fan. The thermostat station may be located anywhere in the building, or it may be located within the ventilator itself. The purpose of wind velocity control is to obtain a definite volume of exhaust regardless of the natural forces, the fan motor being energized when the natural exhaust capacity falls below a certain minimum, and again shut off when the wind velocity rises to the point where this minimum volume can be supplied by natural forces.

Stacks

Stacks or vertical flues are really chimneys which function through the effects of the wind and temperature difference. Like the roof ventilator, the stack outlet should be located so that the wind may act upon it from any direction. With little or no wind, the chimney effect depends entirely on temperature difference to produce a removal of air from the rooms where the inlet openings are located.

GENERAL VENTILATION RULES

A few of the important considerations, in addition to those already outlined, are:

1. Inlet openings in the building should be well distributed, and should be located on the windward side near the bottom, while outlet openings are located on the leeward side near the top. Outside air will then be supplied to the zone to be ventilated.
2. Inlet openings should not be obstructed by buildings, trees, sign boards, etc., outside, nor by partitions inside.
3. Greatest flow per square foot of total opening is obtained by using inlet and outlet openings of nearly equal areas.
4. In the design of window ventilated buildings, where the direction of the wind is quite constant and dependable, the orientation of the building, together with amount and grouping of ventilation openings, can be readily arranged to take full advantage of the force of the wind. Where the wind's direction is quite variable, the openings should be arranged in sidewalls and monitors so that, as far as possible, there will be approximately equal areas on all sides. Thus, no matter what the wind's direction, there will always be some openings directly exposed to the pressure force, and others to a suction force, and effective movement through the building will be assured.
5. Direct short circuits between openings on two sides at a high level may clear the air at that level without producing any appreciable ventilation at the level of occupancy.
6. In order that temperature difference may produce a motive force, there must be vertical distance between openings. That is, if there are a number of openings available in a building, but all are at the same level, there will be no motive head produced by temperature difference, no matter how great that difference might be.
7. In order that the force of temperature difference may operate to maximum advantage, the vertical distance between inlet and outlet openings should be as great as possible. Openings in the vicinity of the neutral zone are less effective for ventilation.
8. In the use of monitors, windows on the windward side should usually be kept closed, since, if they are open, the inflow tendency of the wind counteracts the outflow tendency of temperature difference. Openings on the leeward side of the monitor result in cooperation of wind and temperature difference.
9. In an industrial building where furnaces that give off heat and fumes are to be installed, it is better to locate them in the end of the building exposed to the prevailing wind. The strong suction effect of the wind at the roof near the windward end will then cooperate with temperature difference, to provide for the most active and satisfactory removal of the heat and gas-laden air.

10. In case it is impossible to locate furnaces in the windward end, that part of the building in which they are to be located should be built higher than the rest, so that the wind, in splashing therefrom, will create a suction. The additional height also increases the effect of temperature difference to cooperate with the wind.

11. The intensity of suction, or the vacuum produced by the jump of the wind, is greatest just back of the building face. The area of suction does not vary with the wind velocity, but the flow due to suction is directly proportional to wind velocity.

12. Openings much larger than the calculated areas are sometimes desirable, especially when an increase in occupancy may occur, or when extremely hot days may be anticipated. In the former case, free openings should be located at the level of occupancy for psychological reasons.

13. In single story industrial buildings, particularly those covering large areas, natural ventilation must be accomplished by taking air in and out of the roof openings. Openings in the pressure zones can be used for inflow, and openings in the suction zone, or openings in zones of less pressure, can be used for outflow. The ventilation is accomplished by the manipulation of openings to get air flow through the zones to be ventilated.

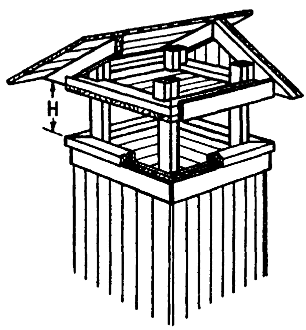

FIG. 5. RECOMMENDED TYPE OF COVER FOR WOODEN OUTLET FLUE

The opening *H* should equal one-half the least dimension of the flue. Heavy insulation of the level deck is essential.

VENTILATION OF ANIMAL SHELTERS[9]

Animal shelters require ventilation to remove moisture, odors and, in the case of dairy stables, excess heat.

Outlets. Outlet flues for natural draft systems should be round or approximately square. A thermal resistance ($1/U$) of not less than two is required in their side walls. They should extend at least two feet above the highest part of the roof. Only one outlet is recommended for each room or pen. The use of several outlet flues may result in excessive up-drafts in some flues, and down-drafts in others.

If flues have roofs or covers, these should be high enough to provide unobstructed openings on all sides, equal in height to one-half the least dimension of the flue, Fig. 5. A level, heavily insulated ceiling under the flue roof, and over the entire area of the flue, is important.

Inlets. Inlets should direct the incoming air vertically upward so as to avoid drafts on the animals, and to insure immediate mixing of incoming air with the room air. A reasonably uniform distribution around the stable or pen is desirable. Inlet flues, that deliver air close to the side walls, stimulate convection currents, which is desirable. When so placed, they also tend to bathe the side walls with cool air, thus reducing temperature difference between the inside and outside of the wall.

From the standpoint of air movement, insulation of inlet flues is not important. Condensation is, however, likely to occur on them, unless the thermal resistance of their walls is at least two.

Controls or throttling devices in inlet flues are seldom required. If used, they are best applied to the inlets and limited to the side of the building facing prevailing winter winds. They should be so made that an opening, at least one inch wide by the width of the flue, will always remain open.

Amounts of heat and water produced by livestock vary not only with the different kinds of animals, but also with age, weight, feed consumption and production. These facts, and the vagaries of the weather, make exact calculations impossible. The following practical recommendations are based on numerous, carefully checked observations.

It is desirable to keep the relative humidity of livestock shelters below 85 percent. Temperatures may be as indicated in the discussion for each kind of animal.

Dairy Stables

The most usually accepted temperatures for dairy stables, where cows are confined in stanchions or tie stalls, are from 45 to 55 F. These temperatures are readily maintained in winter weather by the body heat of the herd in well constructed, well stocked and well ventilated stables. Stable volume in excess of 600 cu ft, and exposed wall area in excess of 130 sq ft per 1000 lb animal weight are, in general, undesirable.

Side walls should have an overall thermal resistance of from 2 to 5, depending on the temperature zone and wind exposure. Thermal resistance of the ceiling should be 50 percent greater than that of the side walls. In stables of this size and so insulated, a ventilation rate of 3200 to 3800 cu ft per (hr) (1000 lb of animal) usually insures good conditions.

The following recommendations do not apply to so-called pen stables or loafing barns in which there is a thick manure and bedding pack on the floor, and in which doors are normally kept open.

Outlets. One flue will serve a stable 200 ft long. In stables over 120 ft long, the flue should be about midway between the ends or, if the stable is L-shaped, near the angle. In shorter stables, it may be at any convenient location.

The exhaust point in the stable should be not more than 18 in. above the floor. This permits removal of only the coolest air, and prevents rapid fluctuations in stable temperature.

A basic rule for finding the cross-section of the outlet flue is

$$A_o = \frac{176\,N}{\sqrt{h}} \qquad (6)$$

where

A_o = area of the outlet flue, square inches.
N = weight of animal population, thousands of pounds.
h = vertical distance from top of inlet flues to top of outlet flue, feet.

For large flues the flue area obtained from the basic formula may well be reduced according to the chart, Fig. 6, because of a decrease in friction.

Example 2: Assume a stable in which the vertical height from the top of the inlets to the top of the outlet flue is 32.5 ft, and in which 38 cows, averaging 1300 lb, will be housed. . Determine required size of outlet flue.

Solution: From Equation 6

$$A_o = \frac{176 \times (38 \times 1.300)}{\sqrt{32.5}} = 1525 \text{ sq in.}$$

From Fig. 6 the factor to be applied to this area is 94.5 percent. Therefore, the area of flue required is 1525 $\times$ 0.945 = 1441 sq in.

Inlets. Inlet flues, each approximately 60 sq in. in area, have given good results. One such flue should be provided for each 3500 lb animal weight. They should deliver air from points 12 to 18 in. below the ceiling.

Sheep Barns

Shelters used for breeding and feeding stock usually have enough openings, so that no special provision for ventilation is required. Barns for winter lambing flocks, however, require ventilation systems. Fermentation

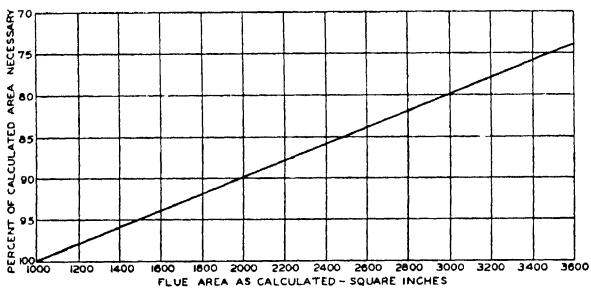

FIG. 6. MODIFICATION OF FLUE AREA FOR OUTLETS EXCEEDING 1000 SQ IN.

in the floor pack of manure produces heat, vapor and odor. These must be added to the ventilation load regularly produced by the animals.

Outlets. In practice, results obtained by the basic formula, Equation 7, when modified by the use of the chart, Fig. 6 have given good results:

$$A_o = \frac{4A_f}{\sqrt{h}} \tag{7}$$

where

A_o = area of the outlet, square inches.
A_f = floor area, square feet.
h = vertical height from top of inlet openings to top of outlet flue, feet.

The bottom of the outlet flue should be 15 to 24 in. above the surface of of the manure pack.

Inlets. Provide one inlet, 60 sq in. in area, for each 150 sq ft of floor area. Inlets should be well distributed around the side walls, and designed to deliver air near the ceiling (see section on Dairy Stables).

Swine Barns

Community swine barns, because of the extent of slop feeding and the absence of daily cleaning of the pens, are the most difficult farm buildings to ventilate satisfactorily. Farrowing pens, to which supplemental heat is supplied, present less of a problem. In all cases, good floor drainage to remove urine and excess spilled water is important.

Temperatures of from 50 to 55 F are usually recommended for farrowing pens. It is desirable to maintain temperatures above freezing in all other pens in community houses. For barns that are well stocked and adequately ventilated, this requires walls with an overall thermal resistance of from 3 to 6, and celings with 40 to 50 percent greater resistance.

Outlets. The outlet flue should draw air from a level of 15 to 18 in. above the floor. Equation 8 is the basic formula for flue area, and gives reasonably good results when modified according to the chart, Fig. 6.

$$A_o = \frac{5 \times A_f}{\sqrt{h}} \tag{8}$$

Inlets. At least one inlet flue should be used for each pen. Swine are given or select definite nesting places, and care must be exercised to avoid having inlets located over them.

Total inlet area should be approximately 70 percent of outlet area. The area of individual inlets is best determined from the total area required, and the number that can be so installed as to meet previous specifications. Inlets should deliver air from points 12 to 15 in. below the ceiling, or from a level deflector on a sloping ceiling.

Poultry Laying Houses

From the standpoint of ventilation, poultry laying houses may be divided into cold houses and warm houses. The former are uninsulated, except in the ceiling. The inside-outside temperature difference is seldom more than 5 F deg. In the warm house, because of insulation or supplemental heat, the temperature seldom falls below 32 F, and is usually above 45 F.

Outlets. The outlet flue is best placed near the middle of the pen. It is advisable to limit the length of pens to 80 ft.

The area of the outlet flue may be determined from Equation 9, and modified according to the chart, Fig. 6.

$$A_o = \frac{2.5 \times A_f}{\sqrt{h}} \tag{9}$$

For a cold house, the bottom of the flue should be at the level of the insulated ceiling. In warm houses, the bottom of the flue should be 12 to 15 in. above the level of the floor litter.

Inlets. Inlets may be approximately 60 sq in. in area. The total inlet area may be equal to 70 percent of the outlet area. Inlets in cold and in warm houses should deliver air from points 12 to 24 in. above the floor. In cold houses which normally do not have storm sash, windows may be raised enough to give the desired area, and be fitted with baffle boards to direct the air straight and upward.

In houses less than 20 ft in width, all inlets may be on one side. In wider houses, a rather uniform distribution of inlets is essential.

GARAGE VENTILATION

Because of hazards resulting from carbon monoxide and other physiologically harmful or combustible gases or vapors in garages, the importance of proper ventilation of these buildings cannot be over-emphasized. During the warm months of the year, garages are usually ventilated adequately because the doors and windows are kept open. As cold weather sets in, more and more of the ventilation openings are closed, and consequently on extremely cold days, the carbon monoxide concentration runs high.

Many garages can be satisfactorily ventilated by natural means, particularly during the mild weather, when doors and windows can be kept open. However, the A.S.H.V.E. Code of Minimum Requirements for Heating and Ventilating Garages, adopted in 1935, states that natural ventilation may be employed for the ventilation of storage sections where it is practicable to maintain open windows or other openings at all times. The code specifies that such openings shall be distributed as uniformly as possible in at least two outside walls, and that the total area of such openings shall be equivalent to at least 5 percent of the floor area. The code further states that where it is impracticable to operate such a system of natural ventilation, a mechanical system shall be used, and shall provide for either the supply of 1 cfm of outdoor air for each square foot of floor area, or for removal of the same amount and its discharge to the outside as a means of flushing the garage.[10]

Research

Cooperative research on garage ventilation, undertaken by the A.S.H.V.E. Committee on Research at Washington University, St. Louis, Mo., and at the University of Kansas, Lawrence, Kans., and tests conducted at the A.S.H.V.E. Research Laboratory, have resulted in authoritative papers on the subject.

Some of the conclusions based on work at the Laboratory are:

1. Upward ventilation results in a lower concentration of carbon monoxide at the breathing line and a lower temperature above the breathing line, than does downward ventilation for the same rate of carbon monoxide production, air change, and the same temperature at the 30-in. level.
2. A lower rate of air change and a smaller heating load are required with upward than with downward ventilation.
3. In the average case, upward ventilation results in a lower concentration of carbon monoxide in the occupied portion of a garage, than that obtained with mixing of the exhaust gases and the air supplied. However, the variations in concentration from point to point, together with the possible failure of the advantages of upward ventilation to accrue, suggest the basing of garage ventilation on complete mixing, and an air change sufficient to dilute the exhaust gases to the allowable concentration of carbon monoxide.
4. The rate of carbon monoxide production by an idling car is shown to vary from 25 to 50 cfh, with an average rate of 35 cfh.
5. An air change of 350,000 cfh per idling car is required to keep the carbon monoxide concentration down to one part in 10,000 parts of air

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CHAPTER 11

HEATING LOAD

General Procedure, Design Outdoor Weather Conditions, Inside Temperatures, Attic Temperatures, Temperatures in Unheated Spaces, Ground Temperatures, Basement Temperatures and Heat Loss, Floor Heat Loss in Basementless Houses, Transmission Heat Loss, Infiltration Heat Loss, Selection of Wind Velocities, Auxiliary Heat Sources, Intermittently Heated Buildings, Residence Heat Loss Problems

PRIOR to designing a heating system, an estimate must be made of the maximum probable heat loss of each room or space to be heated, based on maintaining a selected inside air temperature during periods of design outdoor weather conditions. The heat losses may be divided into two groups, namely: (1) the transmission losses or heat transmitted through the confining walls, floor, ceiling, glass or other surfaces; and (2) the infiltration losses or heat required to warm outside air which leaks in through cracks and crevices, around doors and windows, opening of doors and windows, or heat required to warm outside air used for ventilation.

GENERAL PROCEDURE

The general procedure for calculating heat losses of a structure is:

1. Select the design outdoor weather conditions: temperature, wind direction and wind velocity. The data on climatic conditions given in Table 1 and the isotherms of average design temperature in Fig. 1 will be helpful, but should be used with judgment as suggested in the section Design Outdoor Weather Conditions.

2. Select the inside air temperature, which is to be maintained in each room during the coldest weather. (See Table 2).

3. Estimate temperatures in adjacent unheated spaces and the attic. The attic temperature need not be estimated if the combined roof and ceiling coefficient is used.

4. Select or compute the heat transmission coefficients for outside walls and glass; also for inside walls, floors, or top-floor ceilings, if these are next to unheated space; include roof if next to heated space. (See Chapter 9).

5. Measure net area of outside wall, glass and roof next to heated spaces, as well as any cold walls, floors or ceilings next to unheated space. Such measurements are made from building plans, or from the actual building, using inside dimensions.

6. Compute the heat transmission losses for each kind of wall, glass, floor, ceiling and roof in the building by multiplying the heat transmission coefficient in each case by the area of the surface in square feet, and the temperature difference between the inside and outside air. (See Items 1, 2, and 3).

7. Select unit values and compute the heat equivalent of the infiltration of cold air taking place around outside doors and windows. These unit values depend on the kind or width of crack, wind velocity, and the temperature difference between the inside and outside air; the result expresses the heat required to warm up the cold air leaking into the building per hour. (See Chapter 10).

8. When positive ventilation using outdoor air is provided by an air heating or an air conditioning unit, the heat required to warm the outside air to room temperature must be provided by the unit; if mechanical exhaust from the room is provided, in amount equal to the outside air drawn in by the unit, the natural infiltration losses must also be provided for by the unit. If no mechanical exhaust is used, and the outdoor air supply equals or exceeds the amount of natural infiltration which would occur without ventilation, the natural infiltration may be neglected.

9. The sum of the heat losses by transmisions (Item 6) through the outside walls and glass, as well as through any cold floors, ceilings or roof, plus the heat equivalent

(Item 7) of the cold air entering by infiltration, or required to replace mechanical exhaust, represents the total heat loss equivalent for any building.

DESIGN OUTDOOR WEATHER CONDITIONS

There are no hard and fast rules for selecting the design outdoor weather conditions to be used for a given locality or type of building or heating system, and the selection is to some extent a matter of judgment and experience. The outside design temperature seldom is taken as the lowest temperature, or even the lowest daily mean temperature ever recorded in a given locality. Such temperatures rarely recur in successive years. Likewise the wind direction and velocity prevailing at the time of design outside conditions frequently are entirely different from those prevailing during the winter.

The A.S.H.V.E. Technical Advisory Committee on Weather Design Conditions has recommended the adoption for heating load calculations of an *outside design temperature which is equalled or exceeded during 97½ percent of the hours in December, January, February and March.*

Complete data of this nature are not available, but Column 8, Table 1, lists this recommended design temperature based on airport station readings for the period indicated, generally the five years, 1935–1939. In most cases these stations are outside of the city and these data apply primarily to rural areas. In general the use of the airport data for buildings within an adjacent city will not make an appreciable difference in design load.

Because of the limited data available, design temperatures in common use are listed in Column 10. Many of these values were furnished by A.S.H.V.E. members—the balance were taken from an ACRMA Bulletin,[1] manufacturers' publications and other sources,[2] and a few were estimated. The map, Fig. 1, shows isotherms approximated for these design temperatures. They may be used as a guide for localities not listed in the table. Interpolation between these lines is suggested, and due consideration must be given to elevations and other local conditions. Large differences in climate occur within relatively small distances of Weather Bureau stations in hilly and mountainous regions. Experience and judgment are necessary to deal properly with this factor.

Column 8 of Table 1 gives the maximum wind velocity which occurred with temperatures the same as, and lower than, those shown in Column 8. Winter average velocities for all temperatures are given in Column 11.

Column 6 lists the average annual minimum temperature which is the average of readings of the one lowest temperature occurring for each year the station has been in existence. It is of interest as a guide to the lowest temperature to be expected, except for an occasional extreme of short duration.

INSIDE TEMPERATURES

The inside air temperature which must be maintained within a building is understood to be the dry-bulb temperature at the breathing line, 5 ft above the floor, or at the seating level, 30 in. above the floor, and not less than 3 ft from the outside walls. Inside air temperatures usually specified, vary in accordance with the intended use of the building. Table 2 presents values which conform to good practice.

The proper dry-bulb temperature to be maintained depends upon the relative humidity and air motion, as explained in Chapter 6. In other

Table 1. Winter Climatic Conditions[a]

Col. 1 State	Col. 2 Station[b]	Col. 3 Elevation[c] ft	Col. 4 Period of Record[d]	Col. 5 Lowest Temp. on Record[d] °F	Col. 6 Average Annual Min. Temp.[e] °F	Col. 7 Avg. Winter Temp.[f] °F	Col. 8 Design Dry-Bulb Temp. on TAC 97½% Basis[g] °F	Col. 9 Wind Vel. at Design Temp.[h] Mph	Col. 10 Design Dry-Bulb Temp. in Common Use[i] °F	Col. 11 Avg. Wind Vel. Dec. Jan. Feb.[j] Mph
Ala.	Anniston . . C	733	1893–1947	−10		52.3			5	
	Birmingham C	711	1893–1945	−10	12	53.8			10	8.0
	Birmingham A	615	1939–1947	−10		51.8	21	7.4		
	Mobile C	143	1872–1947	−1	22	58.9			15	9.9
	Mobile A	219	1940–1947	13			30	8.5		
	Montgomery C	293	1872–1947	−5	18	56.4			10	7.5
	Montgomery A	226	1938–1944	9	19					
Ariz..	Flagstaff C	6957	1899–1947	−25	−15	35.9			−10	7.7
	Kingman A	3473	1935–1939[d]	8			22	8.6		
	Phoenix C	1122	1895–1947	16	26	59.5			25	5.4
	Phoenix . . A	1112	1937–1947	21		57.7	31	5.6		
	Tucson . C		Up to 1946	6		57.2			25	5.2
	Tucson A	2561	1935–1939[d]	19		57.2	30	6.3		
	Winslow C	4853	Up to 1946	−19	−1	43.6			−10	
	Winslow A	4899	1932–1947	−18		43.5	6	7.5		
	Yuma C	146	1876–1946[d]	22	27	62.5			30	6.7
Ark.	Fort Smith C	545	1882–1945	−15	6	50.4			10	8.3
	Fort Smith A	463	1945–1947	7						
	Little Rock . . C	451	1879–1942	−12	10	51.6			5	8.3
	Little Rock A	282	1942–1947	7			21	6.6		
Cal..	Bakersfield . . A	499	1937–1946[d]	19	25	55.5	33	5.6	25	
	Burbank A	740	1931–1947	21		57.5	35	4.9		
	Daggett A	1925	1935–1939[d]				27	8.5		
	Eureka C	115	1886–1947	20	29	49.3			30	7.3
	Fresno C	387	1887–1939	17	26	53.8			25	5.4
	Fresno A	281	1939–1947	23			32	4.5		
	Los Angeles C	534	1877–1947	28	37	59.3			35	6.4
	Oakland A	21	1929–1947	23		51.9	36	7.5	30	
	Red Bluff C	341[c]	1877–1934	17		53.9				6.1
	Red Bluff A	346	1944–1947	25		52.9				
	Redding A	579	1935–1939[d]	17			32	7.1		
	Sacramento . C	116	1877–1947	17	28	53.0			30	7.2
	Sacramento A	22	1938–1947	22						
	San Diego C	90	1871–1940	25	37	58.2			35	6.3
	San Diego . A	34	1939–1947	34			43	5.8		
	San Francisco C	164	1875–1947	27	37	54.2			35	7.5
	San Jose C	100[c]	Up to 1946	22		53.5			25	
	Williams . A	124	1935–1939[d]				29	8.3		
Col.	Denver C	5398	1871–1947	−29	−11	39.0			−10	7.5
	Denver A	5379	1934–1947	−30		37.0	0	8.8		
	Grand Junction C	4587[c]	1889–1945[d]	−21	−2	39.9			−15	4.4
	Pueblo . . C	4770	1889–1938[d]	−27		41.1			−20	7.9
	Pueblo . . A	4810	1939–1947	−26			2	6.4		
Conn.	Hartford C	229	1905–1940	−18		36.4			0	8.7
	Hartford A	20	1940–1947	−24			4	8.3		
	New Haven C	180	1872–1946[d]	−15	−1	38.4			0	9.4
	New Haven A	17	1943–1947	−4			11	8.1		
D.C.	Washington C	128	1871–1947	−15	−1	43.4			0	7.8
	Washington . A	20	1935–1939[d]				14	7.4		
Fla.	Apalachicola C	23	1922–1947	18		60.9			25	8.4
	Jacksonville C	104	1871–1947	10	29	62.0			25	9.0
	Jacksonville A	29	1938–1947	16		60.6	31	7.0		
	Key West C	23	1871–1947	41		73.1			45	10.6
	Key West A	48	1939–1947	42		72.5				
	Miami C	253	1896–1947	27	38	71.4			35	10.1
	Miami . . A	13	1940–1947	28	35		45	7.3		
	Pensacola C	67	1879–1947	7	23	59.7			20	10.9
	Pensacola . . A	113	1943–1947	20						
	Tampa . . C	111	1890–1940[d]	19	32	66.0			30	8.6
	Tampa .... A	12	1941–1946	31						
	Titusville . . A	52	1935–1939[d]				38	6.5		
Ga.	Atlanta .... . A	1020	1878–1945[d]	−8	12	51.3	22	10.2	10	11.7
	Augusta .. . C	195	1871–1946[d]	3		54.2			10	6.5
	Augusta .. . A	424	1939–1947	10						
	Macon C	408	1899–1947	7		54.7			15	6.7
	Macon. .... A	432	1939–1947	8						
	Savannah ... C	115	1871–1945	8	22	58.5			20	9.5
	Savannah .. . A	56	1939–1947	16		56.7	29	7.7		
Idaho..	Boise C	2818	1864–1939[d]	−28		40.3			−10	9.1
	Boise .... A	2849	1939–1947	−13	−1	39.8	5	4.5		
	Burley .. .. A	4150	1935–1939[d]	−35		36.9	2	8.8		
	Idaho Falls .... A	4744	1935–1939[d]	−18			−7	7.6		
	Lewiston .. C	763	1900–1944	−23	1	42.7			5	4.1
	Pocatello .. . C	4522	1899–1947	−28	−12	37.1			−5	8.9
	Pocatello .. . A	4467	1938–1947	−23		35.0	6	7.2		
Ill.....	Cairo C	319	1872–1947	−16		46.4			0	9.8
	Chicago C	601[c]	Up to 1946	−23	−8	37.6			−10	12.0

Blank spaces indicate data not yet completed.

TABLE 1. WINTER CLIMATIC CONDITIONS[a]—(CONTINUED)

COL. 1 STATE	COL. 2 STATION[b]		COL. 3 ELEVATION[c] FT	COL. 4 PERIOD OF RECORD[d]	COL. 5 LOWEST TEMP. ON RECORD[d] °F	COL. 6 AVERAGE ANNUAL MIN. TEMP.[e] °F	COL. 7 AVG. WINTER TEMP[f] °F	COL. 8 DESIGN DRY-BULB TEMP. ON TAC 97½% BASIS[g] °F	COL. 9 WIND VEL. AT DESIGN TEMP.[h] MPH	COL. 10 DESIGN DRY-BULB TEMP. IN COMMON USE[i] °F	COL. 11 AVG. WIND VEL. DEC. JAN. FEB.[j] MPH
Ill.	Chicago	A	615	1935–1939[d]			35.1	−3	11.7		
	Moline	A	594	1935–1939[d]			34.6	−6	10.8	−10	
	Peoria	A	660	1935–1939[d]			37.3	6	10.7	−10	8.3
	Springfield	C	603	1879–1947	−24	−7	39 8			−10	11.9
	Springfield	A	608	1930–1947	−19		37.7	−2	11.6		
Ind.	Evansville	C	464	1897–1940	−16	1	45.1				9.6
	Fort Wayne	C	885	1911–1941	−24		37.6			−10	10.4
	Helmer	A	970	1935–1939[d]				−1	11.5		
	Indianapolis	C	816	1871–1946[d]	−25	−6	39.6			−10	11.3
	Indianapolis	A	800	1932–1946[d]	−18		39.0	2	11.4		
	Terre Haute	C	1146	1893–1946[d]	−18	−5	41.7				10.2
	Terre Haute	A	589		−11						
Iowa..	Charles City	C	1023	1891–1945[d]	−34	−22	31.2				7.9
	Davenport	C	648	1872–1947	−27	−13	37.0			−15	10.5
	Des Moines	C	805	1878–1945[d]	−30		36.4			−15	10.1
	Des Moines	A	979	1935–1939[d]			35.4	−8	14.5		
	Dubuque	C	740	1874–1947	−32	−17	34.6			−20	7.1
	Keokuk	C	637	1871–1945[d]	−27	−12	39.3				8.3
	Sioux City	C	1093[c]	1889–1944	−35	−20	32.6			−20	11.5
	Sioux City	A	1098	1940–1946[d]	−24						
Kan.	Concordia	C	1425	1885–1947	−25	−13	40.7			−10	7 7
	Dodge City	C	2515	1942–1947	−26	−10	41.4			−10	10 6
	Dodge City	A	2599	1874–1942	−26						
	Topeka	C	991	1887–1947	−25		42.1			−10	9 2
	Topeka	A	883	1946–1947	−21						
	Wichita	C	1497	1888–1939	−22	−4	43.6			−10	12 4
	Wichita	A	1423	1939–1947	−10			6	14.7		
Ky.	Louisville	C	563	1871–1947	−20	−5	45.1			0	9.8
	Louisville	A	544	1937–1947	−15		43.9	9	8.8		
La.	New Orleans	C	85	1874–1947	7	26	61.6			20	8.6
	New Orleans	A	8	1937–1947	19		60.6	36	12.8		
	Shreveport	A	179	1935–1939[d]		16		27	8.9	20	8.8
Maine	Eastport	C	100	1873–1947	−23	−15	31.5			−10	12.6
	Portland	C	70	1885–1940	−21	−6	33.8			−5	10 4
	Portland	A	66	1940–1947	−39		33.0				
Md.	Baltimore	C	114	1871–1947	−7	8	44.3			0	8.2
	Baltimore	A	43	1935–1939[d]			44.1	13	8.9		
Mass.	Boston	C	356	1870–1935[d]	−18	−3	38.1			0	12 4
	Boston	A	45	1936–1947	−14			8	12 3		
	Nantucket	C	45	1886–1947	−6		39.3			0	14 8
	Nantucket	A	48	1946–1947	12						
Mich.	Alpena	C	615	Up to 1946	−28	−12	29.6			−10	11.0
	Detroit	C	1000	1873–1933	−24	−11	36.5			−10	12.0
	Detroit	A	632	1935–1939[d]				4[k]	11.0		
	Escanaba	C	645	1878–1945[d]	−32		27.3			−15	9 5
	Grand Rapids	C	706	1891–1946[d]	−24		36.0			−10	12 1
	Lansing	C	861	1910–1947	−25		34.0			−10	9 8
	Lansing	A	863	1940–1947	−10						
	Marquette	C	721	1874–1947	−27	−13	28.3			−10	10.6
	Sault St. Marie	C	724[c]	1888–1942[d]	−37	−22	26.0			−20	8 9
Minn...	Duluth	C	1133	1874–1947	−41	−28	24.3			−25	13 4
	Duluth	A	1413	1941–1947	−33						
	Minneapolis	C	945	1890–1947	−34	−23	29.4			−20	11.3
	Minneapolis	A	873	1938–1947	−31						
	St. Paul	C	951	1871–1933	−41	−25	29.0			−20	9.5
	St. Paul	A	708	1937–1947	−26	−18		−15	9.9		
Miss.	Meridian	C	410	1889–1947	−6	15	54.8			10	6.3
	Meridian	A	298	1939–1947	−7						
	Vicksburg	C	316	1874–1947	−1	18	56.8			10	8.3
	Vicksburg	A	266	1941–1947	10		53.6				
Mo.	Columbia	C	739	1889–1947	−26		42.3			−10	8.9
	Columbia	A	787	1939–1947	−18		41.1				
	Kansas City	C	741[c]	1889–1940	−22	−6	41.5			−10	10.3
	Kansas City	A	780	1935–1939[d]				2	10.6		
	St. Louis	C	646	1871–1947	−22	−2	43.6			0	11.8
	St. Louis	A	597	1930–1947	−19		42.3	3	10.8		
	Springfield	A	1270	1935–1939[d]		−5		8	11.0		11.0
Mont...	Billings	A	3584	1935–1939[d]		−30	34.9	−17	9.1	−25	12.4
	Butte	C	5700[c]	1894–1931	−34		30.7			−20	
	Butte	A	5538	1931–1947	−52		27.0	−18	4.8		
	Havre	C	2498	1880–1947	−57	−36	28.4			−30	9.4
	Helena	C	4175[c]	1880–1940[d]	−42	−24	31.6			−20	7.4
	Kalispell	C	3004	1897–1947	−34	−17	31.6			−20	5.2
	Miles City	C	2400	1892–1942[d]	−49	−30	29.0			−35	5.6
	Miles City	A	2629[c]	1935–1939[d]			27.6	−18	8.4		
Neb...	Lincoln	C	1189	1887–1947	−29	−13	37.0			−10	10.6

* Approximate value.

Table 1. Winter Climatic Conditions[a]—(Continued)

Col. 1 State	Col. 2 Station[b]		Col. 3 Elevation[c] ft	Col. 4 Period of Record[d]	Col. 5 Lowest Temp. on Record[d] °F	Col. 6 Average Annual Min. Temp[e] °F	Col. 7 Avg. Winter Temp.[f] °F	Col. 8 Design Dry-Bulb Temp. on TAC 97½% Basis[g] °F	Col. 9 Wind. Vel. at Design Temp.[h] Mph	Col. 10 Design Dry-Bulb Temp. in Common Use[i] °F	Col. 11 Avg. Wind. Vel. Dec. Jan. Feb.[j] Mph
Neb.	Lincoln	A	1185	1933–1947	−26		35.6	−2	12.7		
	North Platte	C	2815	1874–1947	−35	−17	35.4			−20	7.9
	North Platte	A	2788	1935–1939[d]				−9	10.7		
	Omaha	C	1219	1873–1935	−32	−14	36.4			−10	9.7
	Omaha	A	1009	1935–Pres.	−21			−8	11.5		
	Valentine	C	2627	1889–1947	−38	−22	33.6			−25	9.2
Nev...	Elko	A	5079	1935–1939*				−4	4.0		
	Las Vegas	A	1882	1937–1947	8	16	53.8	23	5.3		
	Reno	C	4588	1905–1942	−19		41.7	0		−5	6.0
	Reno	A	4417	1940–1947	−16			7	3.6		
	Winnemucca	C	4293	1871–1947	−36	−10	38.0			−15	8.1
N. H...	Concord	C	343	1871–1941	−35	−15	33.2			−15	6.2
	Concord	A	359	1941–1947	−37						
N. J.	Atlantic City	C	45	1874–1947	−9	6	42.3			5	15.8
	Camden	A	20	1935–1939[d]				12	9.5		
	Newark	A	15	1931–1947	−14			10	11.6	0	
	Sandy Hook	C	19	1914–1938[d]	−11		41.2			0	16.1
	Trenton	C	144	1866–1946	−14	2	42.0			0	10.9
N. M...	Albuquerque	C	5022	1931–1933	5		44.3			0	7.3
	Albuquerque	A	5319	1933–1947	−6			16	7.1		
	El Morro	A	7120	1940–1947	−25	−19		−6	4.6		
	Rodeo	A	4116	1935–1939[d]			51.4	25	8.4		
	Roswell	C	3643	1905–1947	−29		49.1			−10	7.1
	Tucumcari	A	4054	1935–1939[d]				13	9.2		
N. Y...	Albany	C	114	1874–1947	−24	−11	35.2			−10	10.5
	Albany	A	280	1938–1947	−22			0	9.6		
	Binghamton	C	915	1891–1946	−28	−11	34.7			−10	6.8
	Binghamton	A	836	1942–1947	−17						
	Buffalo	C	693[c]	1873–1945[d]	−20	−4	34.8			−5	17.1
	Buffalo	A	726	1935–1939[d]				3	14.0		
	Canton	C	458	1889–1947	−43	−26	29.5			−25	10.5
	Elmira	A	948	1935–1939[d]				5	8.0		
	Ithaca	C	888	1879–1937	−24	−10	34.9			−15	11.3
	New York	C	425	1871–1947	−14	−3	41.1			0	16.8
	Oswego	C	363	1871–1943	−23	−9	34.4			−10	12.1
	Rochester	C	609	1872–1947	−22	−4	35.1			−5	9.6
	Rochester	A	560	1935–1939[d]	−16			4	11.9		
	Syracuse	C	465	1928–1940	−24		34.8			−10	11.2
	Syracuse	A	404	1940–1947	−26			−1	8.9		
N. C..	Asheville	C	2280	1902–1947	−6	6	46.1			0	9.5
	Charlotte	C	809	1878–1947	−5	12	50.6			10	7.3
	Charlotte	A	757	1939–1947	−3			22	7.5		
	Greensboro	A	896	1928–1947	−7		46.4	17	7.8	10	
	Raleigh	C	405	1944–1947	−2	13	50.0			10	7.9
	Raleigh	A	446	1935–1939[d]				20	8.5		
	Wilmington	C	78	1871–1947	5	18	54.6			15	9.4
N. D...	Bismark	C	1675	1875–1940	−45	−31	25.3			−30	9.1
	Bismarck	A	1655	1940–1947	−38		22.9	−24	7.1		
	Devils Lake	C	1481	1904–1947	−46	−33	21.7			−30	10.1
	Dickinson	A	2599	1935–1939[d]			25.6	−20	12.4		
	Fargo	A	900	1935–1939[d]				−25	10.9	−25	
	Pembina	A	830	1935–1939[d]				−30	11.9		
	Williston	C	1919	1879–1947	−50		24.5			−35	8.6
Ohio	Akron	C		1887–1931[d]	−20		37.3			−5	
	Akron	A	104	1935–1939[d]				9	10.6		
	Cincinnati	C	772	1870–1947	−17	−2	43.0			0	8.5
	Cincinnati	A	488	1931–1947	−14			7	8.0		
	Cleveland	C	669	1871–1946[d]	−17	−2	37.2			0	14.7
	Cleveland	A	813	1930–1946[d]	−5			6	13.8		
	Columbus	C	812	1878–1946[d]	−20		40.4			−10	11.6
	Columbus	A	820	1939–1947	−15	−3	38.2	4	10.5		
	Dayton	C	1086	1883–1943	−28		40.4			0	11.1
	Dayton	A	1002	1940–1947	−11						
	Sandusky	C	608	1878–1946[d]	−16		38.0			0	11.0
	Toledo	C	668	1871–1947	−16	−5	37.2			−10	12.1
	Toledo	A	626	1940–1947	−13		35.7	4	12.1		
Okla.	Ardmore	A	762	1935–1939[d]				18	9.7		
	Oklahoma City	C	1264	1890–1947	−17	2	47.9			0	11.5
	Oklahoma City	A	1311	1939–1947	−10			14	14.7		
	Tulsa	A	686	1932–1947	−5		49.0	13	11.3	0	
	Waynoka	A	1529	1935–1939[d]				10	11.3		
Ore.	Arlington	A	881	1935–1939[d]				7	7.8		
	Baker	C	3501	1889–1947	−25	−17	35.2			−5	5.6
	Baker	A	3374	1939–1947	−19		33.8	3	6.4		
	Eugene	C	366	1890–1942[d]	−4		45.9			15	
	Eugene	A	368	1942–1947	9		45.4	23	5.3		
	Medford	C	1428[c]	1911–1929	−10		44.7			5	4.3

Table 1. Winter Climatic Conditions[a]—(Continued)

Col. 1 State	Col. 2 Station[b]		Col. 3 Elevation[c] Ft	Col. 4 Period of Record[d]	Col. 5 Lowest Temp. on Record[d] °F	Col. 6 Average Annual Min. Temp.[e] °F	Col. 7 Avg. Winter Temp.[f] °F	Col. 8 Design Dry-Bulb Temp. on TAC 97½% Basis[g] °F	Col. 9 Wind. Vel. at Design Temp.[h] Mph	Col. 10 Design Dry-Bulb Temp in Common Use[i] °F	Col. 11 Avg. Wind Vel. Dec. Jan Feb.[j] Mph
Ore	Medford	A	1343	1929–1947	−3			23	4.3		
	Portland	C	98	1874–1947	−2	18	46.1			10	7.3
	Portland	A	25	1940–1947	3		44 3	22	8 0		
	Roseburg	C	523	1877–1947	−6	19	46.7			10	3.9
Pa.	Curwensville	A	2219	1943–1947	−10		33.7	0	13.5		
	Erie	C	771	1873–1946	−16	−3	37.3			−5	13.6
	Erie	A	736	1935–1939[d]				6	12.1		
	Harrisburg	C	335[c]	1888–1938[d]	−14	3	39.9			0	7.6
	Harrisburg	A	339	1935–1939[d]				7	9.0		
Pa.	Philadelphia	C	200	1871–1947	−11	6	42.7			0	11 0
	Philadelphia	A	18	1940–1947	1		41.4				
	Pittsburgh	C	929	1875–1947	−20	−2	40.9			0	11 6
	Pittsburgh	A	1284	1935–1947	−16		38.7	6	12.1		
	Reading	C	311	1913–1947	−14		41.2	9[m]		0	9.0
	Scranton	C	877	1901–1947	−19		37.7			−5	7.6
	Sunbury	A	448	1935 1939[d]				7	7.1		
R. I...	Block Island	C	46	1881–1947	−10		40.1			0	20 6
	Providence	C	77	1904–1947	−17	1	37.5			0	12 1
S. C.	Charleston	C	59	1871–1947	7	22	57.4			15	10 5
	Charleston	A	51	1940–1947	14		55.0	26	6.9		
	Columbia	C	401	1887–1947	−2	19	54.4			10	8 0
	Columbia	A	227	1939–1947	9						
	Greenville	C	1006[c]	Up to 1946	−5		49.2			10	8.4
S. D.	Huron	C	1342	1881–1938	−43	−26	28.2			−20	10.7
	Huron	A	1287	1938–1947	−30						
	Rapid City	C	3309	1887–1947	−34	−21	33.4			−20	8.0
	Rapid City	A	3220	1939–1947	−27						
Tenn...	Chattanooga	C	952	1879–1947	−10	9	50.3			10	7.7
	Chattanooga	A	675	1940–1947	6		47 8	19	6.2		
	Knoxville	C	1024	1871–1942	−16	2	47.9			0	7 2
	Knoxville	A	1007	1942–1947	1		46.8				
	Memphis	C	348	1872–1941	−9	9	51.1			0	9 3
	Memphis	A	267	1941–1947	1		50 0	19	8 9		
	Nashville	C	714	1871–1947	−13	5	48 5			0	9 8
	Nashville	A	610	1939–1947	−15			14	7 3		
Texas.	Abilene	C	1748	1885–1944	−6		53 9			15	10 1
	Abilene	A	1756	1940–1947	−9			20	10 4	−5	
	Amarillo	C	3686	1892–1941	−16	0	45.2			−10	12.1
	Amarillo	A	3595	1941–1947	−7			11	12.9		
	Austin	C	625	1897–1942	−1		58.4			20	8 3
	Austin	A	625	1942–1947	13						
	Brownsville	C	140	1922–1943	12	29	66.8			30	10 4
	Brownsville	A	25	1943–1947	30						
	Corpus Christi	C	21	1887 1942[d]	11		63 7			20	11.0
	Corpus Christi	A	45	1943–1946[d]	23						
	Dallas	C	732	1913–1940	−3	13	55.6			0	10 6
	Dallas	A	520	1940–1947	5			23	8 8		
	Del Rio	C	1020	1905–1947	12		60.6			15	8.0
	El Paso	C	3792	1880–1942	−5	16	53.5			10	9.0
	El Paso	A	3956	1939–1947	11			26	8.6		
	Fort Worth	C	708	1898–1939[d]	−8	12	55.1			10	10.5
	Fort Worth	A	728	1940–1947	4						
	Galveston	C	128	1871–1947	8		61.7			20	11 2
	Galveston	A	9	1939–1947	14						
	Houston	C	198	1888–1947	5	22	61 0			20	10 5
	Houston	A	73	1932–1947	5		60.1	33	9 2		
	Palestine	C	555	1881–1947	−6		57 1			15	8 0
	Port Arthur	C	64	1917–1947	11		61.2			20	10 7
	Port Arthur	A	21	1944–1947	24						
	San Antonio	C	770[c]	1885–1941	4	21	60.6			20	8.3
	San Antonio	A	800	1942–1947	17			32	7.6		
	Waco	A	513	1931–1947	7			26	11.6		
	Wink	A	2811	1935–1939[d]				23	8.5		
Utah	Milford	A	5095	1935–1939[d]	−17			−2	7.7		
	Modena	C	5472	1901–1947	−32	−15	36.3			−15	9.0
	Salt Lake City	C	4346	1874–1947	−20	−2	40.0			−10	7.8
	Salt Lake City	A	4254	1928–1947	−30		38.3	7	7.4		
Vt.	Burlington	C	409	1884–1943	−29	−17	31.5			−10	11.6
	Burlington	A	335	1943–1944	−23						
Va. .	Cape Henry	C	24	1874–1947	5		49.2			10	14.0
	Lynchburg	C	644	1874–1944	−7	8	46.8			5	8.1
	Lynchburg	A	951	1944–1947	7						
	Norfolk	C	91	1871–1947	2	15	49.3			15	12.1
	Richmond	C	180	1897–1947	−3	10	47.0			15	8 1
	Richmond	A	172	1929–1947	−12			15	7.1		
	Roanoke	A	1194	1935–1939[d]				21	8.2		
Wash...	Ellensburg	A	1731	1935–1939[d]			36.2	1	3.8		

Table 1. Winter Climatic Conditions[a]—(Continued)

Col. 1 State	Col. 2 Station[b]	Col. 3 Elevation[c] ft	Col. 4 Period of Record[d]	Col. 5 Lowest Temp. on Record[d] °F	Col. 6 Average Annual Min. Temp.[e] °F	Col. 7 Avg. Winter Temp.[f] °F	Col. 8 Design Dry-Bulb Temp. on TAC 97½% Basis[g] °F	Col. 9 Wind Vel. at Design Temp[h] Mph	Col. 10 Design Dry-Bulb Temp. in Common Use[i] °F	Col. 1 Avg. Wind. Vel. Dec., Jan., Feb.[j] Mph
Wash..	North Head C	199	1884–1947	11	24	46.4			20	16.1
	Seattle C	104	1890–1947	3	20	46.3			15	9.8
	Seattle A	47	1928–1947	3		45.1	24	6.3		
	Spokane C	2030	1881–1941	−30	−5	37.7			−15	6.2
	Spokane A	1974	1941–1947	−7			4	5.1		
	Tacoma C	279	1897–1947	7		44.9			15	8.0
	Tatoosh Is. C	110	1883–1947	7		45.4			15	18.9
	Yakima C	1160	1938–1946	−24		39.8			−5	4.1
	Yakima A	1066	1944–1947	−4						
W. Va..	Elkins C	1969[c]	1898–1944	−28	−8	39.4			−10	6.2
	Parkersburg C	685	1888–1947	−27	−1	42.9			−10	7.2
Wisc.	Green Bay C	598	1886–1947	−36	−18	29.8			−20	10.5
	La Crosse C	725	1872–1947	−43	−21	31.7			−25	9.3
	La Crosse A	677	1943–1947	−28		30.5	−17	6.9		
	Madison C	1008	1858–1947	−29		31.4			−15	10.1
	Madison A	884	1935–1939[d]				−8	9.1		
	Milwaukee C	744	1870–1947	−25	−12	33.4			−15	12.1
	Milwaukee A	707	1927–1947	−29		29.0	−6	11.9		
Wyo.	Cheyenne C	6144	1873–1935	−38	−18	33.6			−15	13.3
	Cheyenne A	6161	1935–1947	−34			−3	11.1		
	Lander C	5448	1891–1946	−40	−12	30.0			−18	3.9
	Lander A	5568	1946–1947	−14						
	Rock Springs A	6746	1932–1942	−33		30 1	−7	9.1		
Alta	Edmonton	2219[c]	Up to 1943	−57	−41	22.8			−40	7.5
B C.	Vancouver	22[c]	Up to 1943	2	13	42.6			10	4.5
	Victoria	228[c]	Up to 1943	−2	19	44.0			5	12 6
Man.	Winnipeg	786[c]	Up to 1943	−54	−38	17.2			−35	10.1
N. B..	Fredericton	164[c]	Up to 1943	−35	−25	27.5			−20	9 1
N. S.	Yarmouth	136[c]	Up to 1943	−12	0	34.8			−5	14 3
Ont..	London	912[c]	Up to 1943	−27	−14	32.6			− 5	10.3
	Ottawa	294[c]	Up to 1943	−35	−24	26 4			−20	8 4
	Port Arthur	644[c]	Up to 1943	−40	−29	22.0			−30	8.0
	Toronto	379[c]	Up to 1943	−26	−11	32.7			−10	13.6
P.E.I.	Charlottetown	186[c]	Up to 1943	−27	−13	29.4			−10	9.8
Que.	Montreal	187[c]	Up to 1943	−29	−18	27.9			−15	11.3
	Quebec	296[c]	Up to 1943	−34	−23	24.5			−20	13.3
Sask.	Prince Albert	1414[c]	Up to 1943	−70	−47	16.0			−45	5 1
Y. T.	Dawson	1062[c]	Up to 1943	−68	−54	1.9			−45	3.7
Newf.	St. Johns	428[c]	Up to 1943	−21	−5	31.5			−10	12

[a] United States Data compiled from U. S. Weather Bureau Records for years indicated, and Canadian data from Meteorological Service of Canada corrected to 1946.

[b] Col. 2. The stations followed by letter A are airport stations, all others are city office stations and are followed by letter C.

[c] Col. 3. The elevations marked c are ground elevations of the station. All other elevations given are the actual elevations of the thermometer bulb above mean sea level.

[d] Col. 4. The periods of record indicated apply only to the lowest temperature ever recorded shown in Col 5, and generally extend from a summer month of the first year indicated through the spring months of the last year indicated. The periods marked by d terminated in December of the year indicated.

[e] Average of readings of one lowest temperature obtained for each year.

[f] For period October to April, inclusive.

[g] It should be noted that Col. 8 applies only to airports, as these data for city stations are not available at this time. The temperature shown is the minimum hourly out-door temperature which has been equalled or exceeded 97½ percent of the total hours in December, January and February for the period of record. It is pointed out that in most cases the airport stations are outside of the city and these data would apply primarily to rural areas.

[h] Col. 9 indicates the average wind velocity which occurred at temperatures the same as, and lower than, the temperatures shown in Col. 8.

[i] Col. 10 records design temperatures in use by A S.H.V.E. Members as reported by Chapter Secretaries for the various stations. Where these were not available the design temperatures from an ACRMA publication and various other sources have been inserted.

[j] The wind velocities indicated in Col. 11 were furnished by the U. S. Weather Bureau and corrected through Feb. 1948.

[k] A bulletin prepared by A.S.H.V.E. and U. S. Weather Bureau for annual weather data of city of Detroit indicates 6 as design temperature based on Dec. to March, inclusive.

[m] Computed for Reading by Karl Shelley and O. F. Smith.

words, a person may feel warm or cool at the same dry-bulb temperature, depending on the relative humidity and air motion. The optimum winter

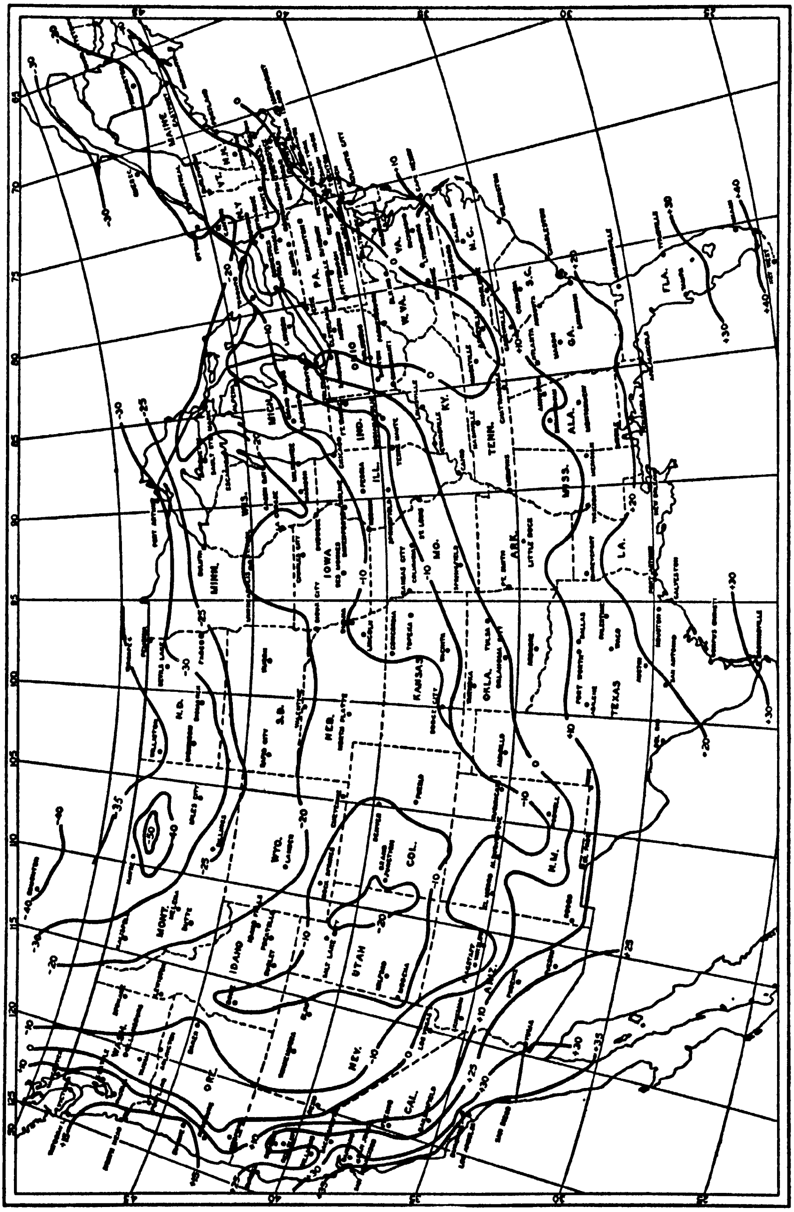

FIG. 1. ISOTHERMS OF WINTER OUTDOOR DESIGN TEMPERATURE

effective temperature for sedentary persons, as determined at the A.S.H.V.E. Research Laboratory, is 67–68 ET.

As explained in Chapter 6 for so-called still air conditions, a relative humidity of approximately 50 percent is required to produce an effective temperature of 68 ET when the dry-bulb temperature is 72.5 F. However, even where provision is made for artificial humidification, the relative humidity is seldom maintained higher than 40 percent during the extremely cold weather, and where no provision is made for humidification, the relative humidity may be 20 percent or less. Consequently, in using the figures listed in Table 2, consideration should be given to the actual relative humidity to be maintained, if provision is to be made for humidification. If no humidification is to be provided, the higher temperatures may not even produce comfort on cold days; if humidity is to be maintained at 50 percent, the lower temperatures will apply.

In rooms having large glass areas, when sun is not shining, or in rooms with walls having a high transmission coefficient, the lowered surface temperature will cause a feeling of coolness even though the air temperature in the room is at or above the temperatures indicated in the table. In rooms of this character, it is desirable to design for even higher temperatures than those listed, unless a compensating higher temperature surface is installed to offset the low temperature surfaces.

TABLE 2. WINTER INSIDE DRY-BULB TEMPERATURES USUALLY SPECIFIED[a]

Type of Building	Deg F	Type of Building	Deg F
Schools—		Theaters—	
Class rooms	70–72	Seating space	68–72
Assembly rooms	68–72	Lounge rooms	68–72
Gymnasiums	55–65	Toilets	68
Toilets and baths	70		
Wardrobe and locker rooms	65–68	Hotels—	
Kitchens	66	Bedrooms and baths	70
Dining and lunch rooms	65–70	Dining rooms	70
Playrooms	60–65	Kitchens and laundries	66
Natatoriums	75	Ballrooms	65–68
		Toilets and service rooms	68
Hospitals—			
Private rooms	70–72	Homes	70–72
Private rooms (surgical)	70–80	Stores	65–68
Operating rooms	70–95	Public buildings	68–72
Wards	68	Warm air baths	120
Kitchens and laundries	66	Steam baths	110
Toilets	68	Factories and machine shops	60–65
Bathrooms	70–80	Foundries and boiler shops	50–60
		Paint Shops	80

[a] The most comfortable dry-bulb temperature to be maintained depends on the relative humidity and air motion. These three factors considered together constitute what is termed the *effective temperature*. (See Chapter 6.) When relative humidity is not controlled separately, optimum dry-bulb temperature for comfort will be slightly higher than shown in Table 2

The inside temperatures specified in Table 2 may be used for panel heated spaces as well as for spaces heated by warm air, radiators or convectors. It is true that warm panel surfaces tend to produce a comfortable environment at a lower room air temperature than when warm panels are not present, but field experience in the United States has indicated that actual reductions in air temperature are slight in operation.

Temperature at Proper Level. In making the actual heat loss computations, however, for the various rooms in a building it is often necessary to modify the temperatures given in Table 2 so that the air temperature at the proper level will be used. By *air temperature at the proper level* is meant, in the case of walls, the air temperature at the mean height between floor and ceiling; in the case of glass, the air temperature at the mean height of the glass; in the case of roof or ceiling, the air temperature at the mean height of the roof or ceiling above the floor of the heated room; and in the case of floors, the air temperature at the floor level.

Temperature at Ceiling. The air temperature at the ceiling is generally higher than at the breathing level due to stratification of air resulting from the tendency of the warmer or less dense air to rise. An allowance for this fact should be made in calculating ceiling heat losses, particularly in the case of high ceilings. However, the exact allowance to be made may be somewhat difficult to determine as it depends on many factors, including (1) the type of heating system, (2) ceiling height, and (3) the inside-outside temperature differential. The type of heating system is particularly important, as the temperature gradient from floor to breathing-level to ceiling may depend to a large extent on whether direct radiation, unit heaters or warm air is used, and in the latter case, whether the air is moved mechanically or by gravity. The temperature of the heating medium is also a factor.

It is impracticable to establish rigid rules for determining the temperature difference to use in all cases. However, for residences and structures having ceiling heights under 10 ft, the comparatively small temperature differential between the breathing level and ceiling generally may be

TABLE 3. APPROXIMATE TEMPERATURE DIFFERENTIALS BETWEEN BREATHING LEVEL AND CEILING, APPLICABLE TO CERTAIN TYPES OF HEATING SYSTEMS[a]

Ceiling Height (Ft)	Breathing Level Temperature (5 ft Above Floor)									
	60	65	70	72	74	76	78	80	85	90
10	3.0	3.3	3.5	3.6	3.7	3.8	3.9	4.0	4.3	4.5
11	3.6	3.9	4.2	4.3	4.4	4.6	3.7	4.8	5.1	5.4
12	4.2	4.6	4.9	5.0	5.2	5.3	5.5	5.6	6.0	6.3
13	4.8	5.2	5.6	5.8	5.9	6.1	6.2	6.4	6.8	7.2
14	5.4	5.9	6.3	6.5	6.7	6.8	7.0	7.2	7.7	8.1
15	6.0	6.5	7.0	7.2	7.4	7.6	7.8	8.0	8.5	9.0
16	6.1	6.6	7.1	7.3	7.5	7.7	7.9	8.1	8.6	9.1
17	6.2	6.7	7.2	7.4	7.6	7.8	8.0	8.2	8.7	9.2
18	6.3	6.8	7.3	7.5	7.7	7.9	8.1	8.3	8.8	9.3
19	6.4	6.9	7.4	7.6	7.8	8.0	8.2	8.4	8.9	9.4
20	6.5	7.0	7.5	7.7	7.9	8.1	8.3	8.5	9.0	9.5
25	7.0	7.5	8.0	8.2	8.4	8.6	8.8	9.0	9.5	10.0
30	7.5	8.0	8.5	8.7	8.9	9.1	9.3	9.5	10.0	10.5
35	8.0	8.5	9.0	9.2	9.4	9.6	9.8	10.0	10.5	11.0
40	8.5	9.0	9.5	9.7	9.9	10.1	10.3	10.5	11.0	11.5
45	9.0	9.5	10.0	10.2	10.4	10.6	10.8	11.0	11.5	12.0
50	9.5	10.0	10.5	10.7	10.9	11.1	11.3	11.5	12.0	12.5

[a] The figures in this table are based on an increase of 1 percent per foot of height above the breathing level (5 ft) up to 15 ft and 1/10 of one degree for each foot above 5 ft. This table is generally applicable to forced air types of heating systems. For direct radiation or gravitywarm air, increase values 50 percent to 100 percent.

neglected without serious error. For higher ceilings, an allowance of approximately 1 percent per foot of height above the breathing level may be made for ceiling heights up to 15 ft and approximately $\frac{1}{10}$ of 1 deg per foot of height above this level. The values in Table 3 are calculated on this basis. For direct radiation and gravity warm air systems, the allowance should be increased from 50 percent to 100 percent over those given in Table 3. These rules should, however, be used with considerable discretion, and they do not apply to some types of heating systems such as those using panel and baseboard radiation, where very low temperature differences between the floor and the ceiling may exist.

Temperature at Floor Level. According to tests at the University of Illinois,[3, 4, 5, 6] the temperature at the floor level ranged from about 2 to 6 deg below that at the breathing level, or somewhat greater than the difference between the breathing level and ceiling temperatures. Tests at the University of Wisconsin[7] indicated a somewhat smaller differential between the floor and breathing level temperatures. As a general rule, if the breathing level to ceiling temperature differential is neglected (as with ceiling heights under 10 ft), the breathing level to floor differential may also be neglected, as the two are somewhat compensating, especially where both floor and ceiling losses are calculated for the same space. In other cases, the 10 ft temperature differentials in Table 3 may be used in arriving at the floor heat loss, these differentials to be subtracted from the breathing level temperature.

ATTIC TEMPERATURES

Frequently, it is necessary to estimate the attic temperature, and in such cases Equation 1 can be used for this purpose:

$$t_a = \frac{A_c U_c T_1 + t_o(A_r U_r + A_w U_w + A_g U_g)}{A_c U_c + A_r U_r + A_w U_w + A_g U_g} \quad (1)$$

where

t_a = attic temperature, Fahrenheit degrees.

t_1 = inside temperature near top floor ceiling, Fahrenheit degrees.

t_o = outside temperature, Fahrenheit degrees.

A_c = area of ceiling, square feet.

A_r = area of roof, square feet.

A_w = area of net vertical attic wall surface, square feet.

A_g = area of attic glass, square feet.

U_c = coefficient of transmission of ceiling, based on surface conductance of 2.20 (upper surface, see Chapter 9). 2.20 = reciprocal of one-half the air space resistance.

U_r = coefficient of transmission of roof, based on surface conductance of 2.20 (lower surface, see Chapter 9).

U_w = coefficient of transmission of vertical wall surface.

U_g = coefficient of transmission of glass.

Example 1. Calculate the temperature in an unheated attic, assuming the following conditions: t_1 = 70; t_o = 10; A_c = 1000; A_r = 1200; A_w = 100; A_g = 10; U_r = 0.50; U_c = 0.40; U_w = 0.30; U_g = 1.13.

Solution: Substituting these values in Equation 1:

$$t_a = \frac{(1000 \times 0.40 \times 70) + 10[(1200 \times 0.50) + (100 \times 0.30) + (10 \times 1.13)]}{(1000 \times 0.40) + (1200 \times 0.50) + (100 \times 0.30) + (10 \times 1.13)}$$

$$t_a = \frac{34{,}413}{1041} = 33.1 \text{ F.}$$

Equation 1 neglects the effect of any interchange of air such as would take place through attic vents or louvers intended to preclude attic condensation. However, according to tests,[8] such venting of attics by means of small louvers or other small openings does not appreciably reduce the attic temperature and may be neglected without serious error.

Neither does this equation take into consideration such factors as heat exchange between chimney and attic or solar radiation to and from the roof. Because of these latter effects, actual attic temperatures are frequently higher than the calculated values using Equation 1. The attic temperature may be calculated in the usual manner by means of Equation 1, allowing the full value of the roof. The error resulting from this assumption will generally be considerably less than if the roof were neglected (as is sometimes the practice) and the attic temperature assumed to be the same as the outside temperature. When relatively large louvers are installed, as is customary in the southern states, the attic temperature is often assumed as the average between inside and outside.

For a shorter, approximate method of calculating heat losses through attics, the combined ceiling and roof coefficient may be used, as described in Chapter 9.

TEMPERATURES IN UNHEATED SPACES

The heat loss from heated rooms into unheated rooms or spaces must be based on the estimated or assumed temperature in such unheated spaces. This temperature will lie in the range between the inside and outside temperatures, depending on the relative areas of the surfaces adjacent to the heated room and those exposed to the outside. If the respective surface areas adjacent to the heated room and exposed to the outside are approximately the same, and if the coefficients of transmission are approximately equal, the temperature in the unheated space may be assumed to be the mean of the inside and outside design temperatures. If, however, the surface areas and coefficients are unequal, the temperature in the unheated space should be estimated by means of Equation 2.

$$t_u = \frac{t(A_1U_1 + A_2U_2 + A_3U_3 + \text{etc.}) + t_o(A_aU_a + A_bU_b + A_cU_c + \text{etc.})}{A_1U_1 + A_2U_2 + A_3U_3 + \text{etc.} + A_aU_a + A_bU_b + A_cU_c + \text{etc.}} \quad (2)$$

where

t_u = temperature in unheated space, Fahrenheit degrees.

t = inside design temperature of heated room, Fahrenheit degrees.

t_o = outside design temperature, Fahrenheit degrees.

A_1, A_2, A_3, etc. = areas of surface of unheated space adjacent to heated space, square feet.

A_a, A_b, A_c, etc. = areas of surface of unheated space exposed to outside, square feet.

U_1, U_2, U_3, etc. = coefficients of transmission of surfaces of A_1, A_2, A_3, etc.

U_a, U_b, U_c, etc. = coefficients of transmission of surfaces A_a, A_b, A_c, etc.

Example 2: Calculate the temperature in an unheated space adjacent to a heated room having surface areas (A_1, A_2, and A_3) in contact therewith of 100, 120, and 140 sq ft and coefficients (U_1, U_2, and U_3) of 0.15, 0.20, and 0.25, respectively. The surface areas of the unheated space exposed to the outside (A_a and A_b) are respectively 100 and 140 sq ft, and the corresponding coefficients are 0.10 and 0.30. The sixth surface is on the ground and is neglected in this example. Assume t = 70 and t_o = − 10.

Solution: Substituting in Equation 2:

$$t_u = \frac{70[(100 \times 0.15) + (120 \times 0.20) + (140 \times 0.25)] + -10[(100 \times 0.10) + (140 \times 0.30)]}{(100 \times 0.15) + (120 \times 0.20) + (140 \times 0.25) + (100 \times 0.10) + (140 \times 0.30)}$$

$$t_u = \frac{4660}{126} = 37 \text{ F.}$$

The temperatures in unheated spaces having large glass areas and having two or more surfaces exposed to the outside (such as sleeping porches and sun parlors), generally are assumed to be the same as outside.

GROUND TEMPERATURES

Ground temperatures to be assumed for estimating basement heat losses usually will differ in the case of basement walls and floors, the temperatures under the floors generally being higher than those adjacent to walls. Factors affecting these temperatures will be discussed.

Temperatures Under Basement Floors

The temperature of the ground under basement floors[9] is affected by heat sources within the basement and is not influenced by atmospheric conditions. In computing losses through basement floors, the ground temperatures may be assumed to be the same as water temperatures at depths of 30 to 60 ft given in Fig. 3, Chapter 34. Test observations indicate that heat losses through basement floors frequently are over-estimated.[10]

Temperatures Adjacent to Basement Walls

Ground temperatures near the surface and under open spaces vary with the climate, the season of the year and the depth below the surface. The nearer the surface (during the cold weather) the lower will be the ground temperature. Frost will penetrate to a depth of over 4 ft in some localities if not protected by snow. A thick blanket of snow will result in a higher ground temperature near the surface. Therefore, in localities where the ground is covered with a heavy blanket of snow throughout the winter, the ground temperatures near the surface will be higher than when little or no snow is present.

Table 4. Below Grade Heat Losses for Basement Walls and Floors

Ground Water Temperature[a]	Basement Floor Loss[b] Btu/Sq Ft	Below Grade Wall Loss[b] Btu/Sq Ft
40	3.0	6.0
50	2.0	4.0
60	1.0	2.0

[a] See Fig. 3, Chapter 34.
[b] Based on basement temperature of 70 F and U of 0.10.

Complete data on ground temperatures adjacent to buildings are not available, but since the recommended transmission coefficient for basement walls in contact with the soil is only 0.10, any reasonable, assumed ground temperature will not materially affect the calculated heat loss.

BASEMENT TEMPERATURES AND HEAT LOSS

The allowance to be made for basement heat loss depends on whether the basement is to be heated or not.

If the basement is *heated* to a specified temperature, the heat loss should be calculated in the usual manner, based on the proper wall and floor coefficients (see Chapter 9) and the outside air and ground temperatures. Heat loss through windows and walls above grade should be based on outside temperatures and the proper air-to-air coefficients. Heat loss through basement walls below grade should be based on the floor and wall coefficients for surfaces in contact with the soil, and on the proper ground temperature.

The heat loss values for below grade basement walls and floors given in Table 4 are sufficiently precise for general practice.

If a basement is completely below grade and is *not heated*, the temperature in the basement normally will range between that in the rooms above and the ground temperature. Basement windows will, of course, lower the basement temperature when it is cold outside, and heat given off by the heating plant will increase the basement temperature. In any case, the exact basement temperature is indeterminate if the basement is not heated. In general, it is found that the transient heat from the heating plant warms the air near the basement ceiling sufficiently to make it unnecessary to make an allowance for floor heat loss from rooms located over the basement.

The temperature in crawl spaces below floors will vary widely depending on the number and size of wall vents, the amount of warm piping present and type of piping insulation. It is necessary, therefore, to evaluate the conditions and to select an appropriate temperature by judgment.

FLOOR HEAT LOSS IN BASEMENTLESS HOUSES

Two types of concrete floors are in common use in basementless houses: (a) the floor not heated but relying for warmth on radiation received from walls, ceiling, etc.; and (b) the floor containing heating pipes or ducts and constituting a *radiant* slab for heating or partially heating the house.

For type (a) the floor heat loss, economically considered, is of minor importance since it comprises generally about 10 percent of the total heat loss of the house. From the comfort standpoint, however, it may be most important, since houses with cold floors are not successfully heated. In this connection, it should be remembered that a well insulated floor does not assure comfort if down-drafts from windows or exposed walls create pools of chilly air over considerable areas of the floor. For this reason a floor of type (a) should not be used in a severe climate, except with a ceiling panel heating system or other system capable of warming the floor uniformly by radiation.

Data are meager, but the results of some experiments[10, 11] indicate that the heat loss from a concrete slab floor on grade is more nearly proportional to the perimeter than to the area of the floor, and that the heat loss can be estimated by means of the formula:

$$H_F = FP\ (t - t_o) \tag{3}$$

where

H_F = heat loss of the floor, Btu per hour.

P = perimeter or exposed edge of the floor, linear feet.

F = heat loss coefficient, Btu per (hour) (linear foot of exposed edge) (degree difference in temperature between the inside air and the outside air). (F ranges between 0.81 for a floor with no edge insulation to 0.55 for a floor with edge insulation).

t = inside air temperature, Fahrenheit.

t_o = outside air temperature, Fahrenheit.

In most instances the values given in Fig. 2 for edge loss are of sufficient precision for use.[12] The insulation shown extending under the floor (about 2 ft) can also be located along the foundation wall with equal effectiveness if the insulation extends 18 in. to 24 in. below the floor level.

Example 3: Calculate the heat loss from the floor of a 12 ft x 15 ft room with two exposures. The floor is insulated at the edge with 1 in. of insulation, and the house is located in zone B (outside design temperature = −10 F).

Solution: From Fig. 2 the heat loss per foot of exposed edge is 55 Btu per hr. The length of exposed edge is 12 ft + 15 ft = 27 ft, and the total edge loss is 27 x 55 = 1485 Btu per hr.

Floors of type (b), consisting of concrete slabs placed on the ground, and containing heating pipes or ducts, are now in use in many small dwelling houses. The heat loss downward or through the ground from such floors is called the *reverse loss.* Authoritative data for computing or estimating the magnitude of reverse losses are lacking. It is customary for designers to allow a percentage (often in the range from 10 to 20 percent) of the house heat loss to cover the reverse heat loss from such a floor. There is some indication that it may be possible to use Equation 3 to estimate the *reverse loss* by substituting for t the temperature of the heating medium.

The desirability of edge insulation is apparent, but standards of practice have not been established. An inch of waterproof material is the minimum thickness of edge insulation that should be used, and greater thicknesses are recommended.[12] Such a floor usually is placed above a cinder or gravel fill four or more inches thick, both to insulate the floor from the earth and to retard the rise of ground water by capillarity. Obviously, it is important that such floors be laid several inches above grade, and that effective subsoil drainage be provided to avoid slabs soaked by rain or melting snow, and consequent excessive heat loss.

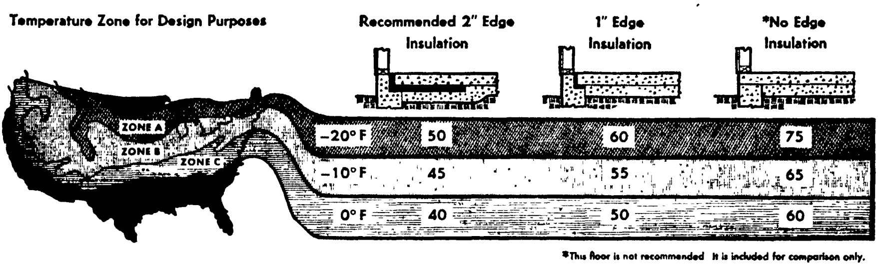

FIG. 2. HEAT LOSS PER FOOT OF EXPOSED EDGE FOR CONCRETE FLOORS AT OR NEAR GRADE LEVEL.[12]

TRANSMISSION HEAT LOSS

The basic formula for the loss of heat by transmission through any surface is given in Equation 4:

$$H_t = AU\,(t - t_o) \tag{4}$$

where

H_t = heat loss transmitted through the wall, roof, ceiling, floor, or glass, Btu per hour.

A = area of wall, glass, roof, ceiling, floor, or other exposed surface, square feet.

U = coefficient of transmission, air to air, Btu per (hour) (square foot) (Fahrenheit degree temperature difference) (Chapter 9).

t = inside temperature near surface involved (this may not necessarily be the so-called breathing line temperature), Fahrenheit degrees.

t_o = outside temperature, or temperature of adjacent unheated space or of the ground, Fahrenheit degrees.

Example 4: Calculate the transmission loss through an 8 in. brick wall having an area of 150 sq ft, if the inside temperature t is 70 F and the outside temperature t_o is − 10 F.

Solution: The coefficient of transmission (U) of a plain 8 in. brick wall is 0.50 (Chapter 9, Table 8). The area (A) is 150 sq ft. Substituting in Equation 4:

$$H_t = 150 \times 0.50 \times [70 - (-10)] = 6000 \text{ Btu per hour.}$$

Transmission Loss Through Ceilings and Roofs

The transmission heat loss through top floor ceilings, attics, and roofs may be estimated by either of two methods:

1. By substituting in Equation 4 the ceiling area A, the inside-outside temperature difference $(t - t_o)$ and the proper value of U:

   a. *Flat roofs.* Select the coefficient of transmission of the ceiling and roof from Tables 15 or 16, Chapter 9, or use appropriate coefficients in Equation 1 if side walls extend appreciably above the ceiling of the floor below.

   b. *Pitched roofs.* Select the combined roof and ceiling coefficient from Table 18, Chapter 9 or calculate the combined roof and ceiling coefficient by means of Equations 4 and 5, Chapter 9, where these formulas are applicable as explained in Chapter 9.

2. By estimating the attic temperature (based on the inside and outside design temperatures) by means of Equation 1, and substituting for t_o in Equation 4, the value of t_a thus obtained, together with the ceiling area A and the ceiling coefficient U. This applies to *pitched roofs*. In the case of *flat roofs* it is not necessary to calculate the attic temperatures, as the ceiling-roof heat loss can be determined as suggested in paragraph 1a.

INFILTRATION HEAT LOSS

The infiltration heat loss includes (1) the *sensible heat loss* or the heat required to warm the outside air entering by infiltration, and (2) the *latent heat loss* or the heat equivalent of any moisture which must be added.

Sensible Heat Loss

The formula for the heat required to warm the outside air which enters a room by infiltration to the temperature of the room, is given in Equation 5:

$$H_s = 0.240\ Qd\ (t - t_o) \tag{5}$$

where

H_s = heat required to raise temperature of air leaking into building from t_o to t, Btu per hour.

0.240 = specific heat of air.

Q = volume of outside air entering building, cubic feet per hour (see Chapter 10).

d = density of air at temperature t_o, pounds per cubic foot.

It is sufficiently accurate to use $d = 0.075$ in which case Equation 5 reduces to

$$H_s = 0.018\ Q\ (t - t_o) \tag{5a}$$

The volume Q of outside air entering per hour depends on the wind velocity and direction, the width of crack or size of openings, the type of openings and other factors, as explained in Chapter 10. Where the crack method is used for estimating leakage, it is more convenient to express the air leakage heat loss in terms of the crack length:

$$H_s = B\ L\ (t - t_o) \tag{5b}$$

where

B = air leakage per (hour) (foot of crack) (Chapter 10) for the wind velocity and type of windows or door crack involved, multiplied by 0.018.

L = length of window or door crack to be taken into consideration, feet.

Example 5: What is the infiltration heat loss per hour through the crack of a 3 x 5 ft average, double-hung, non-weatherstripped, wood window, based on a wind velocity of 15 mph? Assume inside and outside temperatures to be 70 F and zero, respectively.

Solution: According to Table 2, Chapter 10, the air leakage through a window of this type (based on $\frac{1}{16}$ in. crack and $\frac{3}{64}$ in. clearance) is 39 cu ft per (ft of crack) (hour). Therefore, $B = 39 \times 0.018 = 0.70$. The length of crack L is $(2 \times 5) + (3 \times 3)$, or 19 ft; $t = 70$ and $t_o = 0$. Substituting in Equation 5b,

$$H_s = 0.70 \times 19 \times (70 - 0) = 931 \text{ Btu per hour.}$$

Crack Length to be Used for Computations

For designers who prefer to use the crack method, the basis of calculation is as follows: The amount of crack used for computing the infiltration heat loss should be not less than half of the total length of crack in the outside walls of the room. For a building having no partitions, air entering through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, take one-half the total crack for computing each side and end of the building. In a room with one exposed wall, take all the crack; with two exposed walls, take the wall having the most crack; and with three or four exposed walls, take the wall having the most crack; but in no case take less than half the total crack.

In small residences the total infiltration loss of the house is generally considered to be equal to the sum of the infiltration losses of the various rooms. However, this is not necessarily accurate as at any given time infiltration will take place only on the windward side or sides and not on the leeward side. Therefore, for determining the total heat requirements of larger buildings it is more accurate to base the total infiltration loss on the wall having the most total crack, but in no case on less than half of the total crack in the building.

Number of Air Changes to be Used for Computations

Since a certain amount of judgment is required regarding quality of construction, weather conditions, use of room and other factors in estimating infiltration by any method, some designers base infiltration upon an estimated number of air changes rather than upon the length of window cracks. Table 4 of Chapter 10 indicates air changes commonly used, but should be taken only as a guide.

When calculating infiltration losses by the air change method, Equation 5a may be used by substituting for Q the volume of the room multiplied by the number of air changes obtained from Table 4, Chapter 10. For further discussion of the method see section on Air Change Method in Chapter 10.

Latent Heat Loss

When it is intended to add moisture to air leaking into a room in order to maintain proper winter comfort conditions, it is necessary to determine the heat required to evaporate the water vapor added. This heat may be calculated by the equation

$$H_l = Qd\,(W_i - W_o)\,h_{fg} \tag{6}$$

where

H_l = heat required to increase moisture content of air leaking into building from m_o to m_i, Btu per hour.

Q = volume of outside air entering building, cubic feet per hour.

d = density of air at temperature t_i, pounds per cubic foot.
W_i = vapor density of inside air, pounds per pound of dry air.
W_o = vapor density of outside air, pounds per pound of dry air.
h_{fg} = latent heat of vapor at m_i, Btu per pound.

If the latent heat of vapor h_{fg} is assumed to be 1060 Btu per lb, Equation 6 reduces to:

$$H_l = 79.5\, Q\, (W_i - W_o) \tag{6a}$$

Equations 5a, 5b and 6a may also be used for determining the sensible and latent heat gains due to infiltration in cooling load computations.

SELECTION OF WIND VELOCITIES

The effect of wind on the heating requirements of any building should be given consideration for two reasons:

1. Wind movement increases the heat transmission of walls, glass, and roof, affecting poor walls to a much greater extent than good walls.
2. Wind increases materially the infiltration of cold air through the cracks around doors and windows, and even though the building materials themselves (see Tables 1 and 2, Chapter 10).

Theoretically, as a basis for design, the most unfavorable combination of temperature and wind velocity should be chosen. It is entirely possible that a building might require more heat on a windy day with a moderately low outside temperature, than on a quiet day with a much lower outside temperature. However, the combination of wind and temperature, which is the worst, would differ with different buildings, because wind velocity has a greater effect on buildings which have relatively high infiltration losses. It would be possible to compute the heating load for a building for several different combinations of temperature and wind velocity which records show to have occurred, and to select the worst combination, but designers generally do not feel that such a degree of refinement is justified.

Therefore, since Table 1 lists the average velocity of winds occurring at temperatures equalled or exceeded 97½ percent of the winter period for each locality, this value should be the basis for estimating infiltration losses. When using the air change method it will not be necessary to consider the wind velocities. Designers employing the crack method generally use values corresponding to a 15-mile wind. Due to the small effect of the wind velocity on the transmission coefficient, the values in Chapter 9, based on a 15-mile wind may be used with sufficient accuracy for all ordinary conditions.

Exposure Factors

Many designers use empirical *exposure* factors to increase the calculated heat loss of rooms or spaces on the side or sides of the building exposed to the prevailing winds. However, the use of exposure factors is unnecessary when the GUIDE method of calculating heat losses is used. Therefore, exposure factors may be regarded as *factors of safety* for the rooms or spaces exposed to the prevailing winds, to allow for additional capacity for these rooms or spaces, or to *balance the radiation*, particularly in the case of multi-story buildings. Tall buildings may have severe infiltration heat losses, induced by their stack effect (see Chapter 10), which will require special consideration. Although the exposure allowance frequently is assumed to be 15 percent, the actual allowance to be made, if any, must to a large extent be a matter of experience and judgment of the designer, since there are at present no authentic test data available from which rules could be developed for the many conditions encountered in practice.

AUXILIARY HEAT SOURCES

The heat supplied by persons, lights, motors and machinery always should be ascertained in the case of theaters, assembly halls, and industrial plants, but allowances for such heat sources must be made only after careful consideration of all local conditions. In many cases, these heat sources should not affect the size of the heating plant at all, although they may have a marked effect on the operation and control of the system. In general, where audiences are present, the heating system must have sufficient capacity to bring the building to the stipulated inside temperature before the audience arrives. In industrial plants, quite a different condition exists, and heat sources, if always available during occupancy, may be substituted for a portion of the heating installation. In no case should the

Table 5. Heat Equivalents of Various Sources[a]

Machinery (Motor in room = Motor Hp/efficiency x 2544	Btu/hr.
Machinery (Motor outside room) = Motor Hp x 2544	Btu/hr.
Electric Lights = Kilowatts x 3413	Btu/hr.
Gas (Producer = 150) (Manufactured = 535) (Natural = 1000)	Btu/cu ft.

[a] Additional values are given in Chapter 12, Table 29.

actual heating installation (exclusive of heat sources) be reduced below that required to maintain at least 40 F in the building.

Electric Motors and Machinery

Motors and the machinery which they drive, if both are located in the room, convert all of the electrical energy supplied into heat. This heat is retained in the room if the product manufactured is not removed until its temperature is the same as the room temperature.

If power is transmitted to the machinery from the outside, then only the heat equivalent of the brake horsepower supplied is used. In some mills this is the chief source of heating, and it is frequently sufficient to overheat the building even in zero weather, thus requiring cooling by ventilation the year 'round. Table 5 shows the heat output equivalent of various sources of heat in a factory. For information concerning the heat supplied by persons, refer to data given in Chapter 6, and also Table 28, Chapter 12. For appliances see Table 29, Chapter 12.

INTERMITTENTLY HEATED BUILDINGS

In the case of intermittently heated buildings additional heat is required for raising the temperature of the air, the building materials and the material contents of the building to the specified inside temperature. The rate at which this additional heat must be supplied depends upon the heat capacity of the structure and its material contents, and upon the time in which these are to be heated.[13]

This additional heat may be computed and allowed for as conditions require, but inasmuch as the heating system proportioned for taking care of the heat losses will usually have a capacity about 100 percent greater than that required for average winter weather, and inasmuch as most

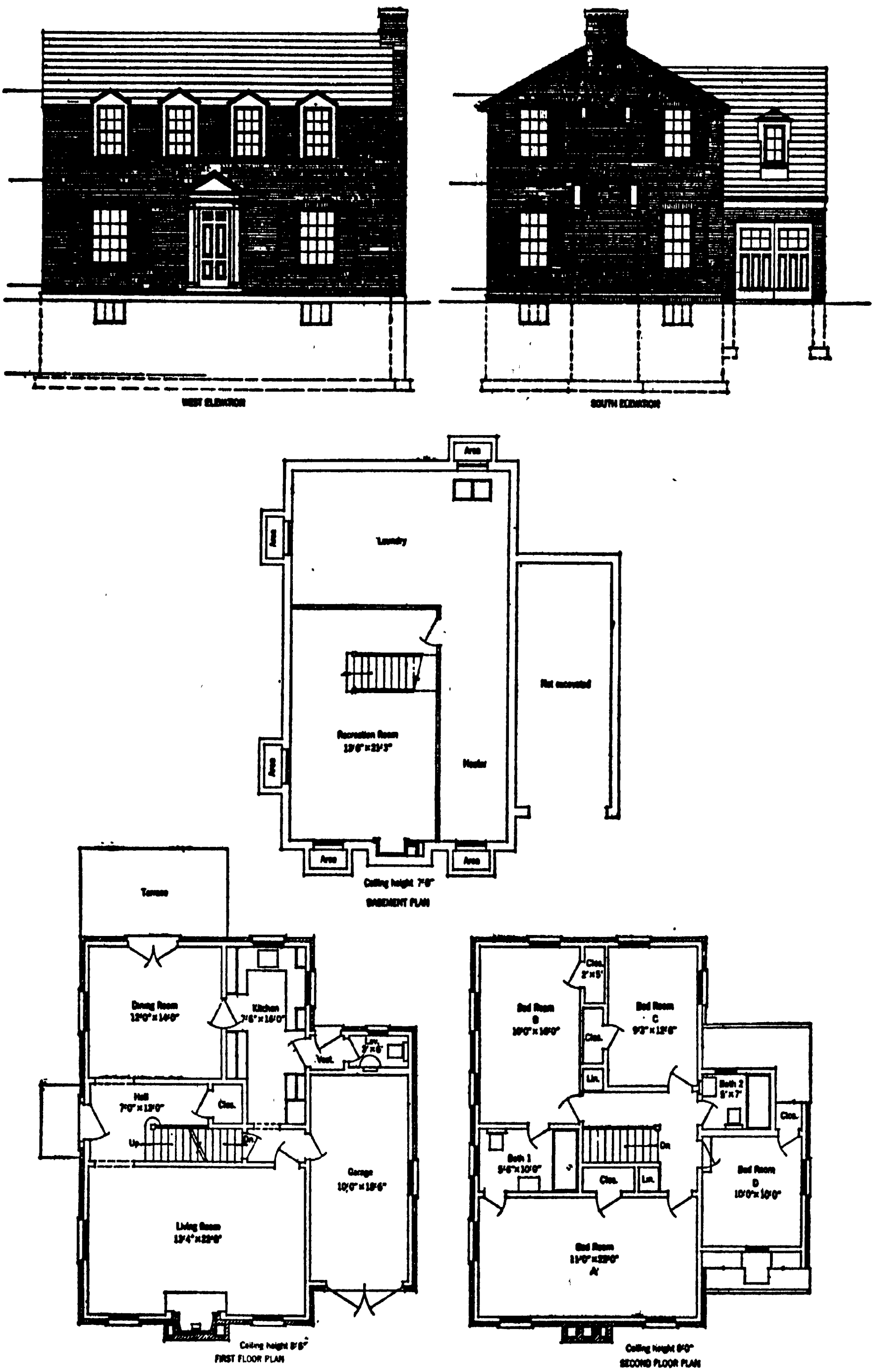

FIG. 3. ELEVATIONS AND FLOOR PLANS OF RESIDENCE

TABLE 6. HEAT LOSS CALCULATION SHEET FOR UNINSULATED RESIDENCE (FIG. 3)

A	B	C	D	E	F	G
ROOM OR SPACE	PART OF STRUCTURE OR INFILTRATION AIR CHANGES	NET AREA OR AIR VOLUME	COEFFICIENT	TEMP. DIFF.[a]	HEAT LOSS (Btu per hour)	TOTALS (Btu per hour)
Bedroom A and Closet	Walls	238 sq ft	0.28	80	5330	
	Glass	40 sq ft	0.45	80	1440	
	Ceiling	252 sq ft	0.69	39.8[d]	6910	
	Infiltration (¾)[g]	1510 cfh[b]	0.018[c]	80	2180	15,860
Bedroom B and Closet	Walls	156 sq ft	0.28	80	3490	
	Glass	40 sq ft	0.45	80	1440	
	Ceiling	170 sq ft	0.69	39.8[d]	4660	
	Infiltration (¾)[g]	1020 cfh[b]	0.018[c]	80	1470	11,060
Bedroom C and Closet	Walls	114 sq ft	0.28	80	2560	
	Glass	27 sq ft	0.45	80	970	
	Ceiling	129 sq ft	0.69	39.8[d]	3540	
	Infiltration (¾)[g]	874 cfh[b]	0.018[c]	80	1260	8,330
Bedroom D and Closet	Walls	118 sq ft	0.28	80	2650	
	Glass	20 sq ft	0.45	80	720	
	Ceiling	110 sq ft	0.69	39.8[d]	3020	
	Floor over garage	110 sq ft	0.25	35[e]	960[p]	
	Infiltration (¾)[g]	660 cfh[b]	0.018[c]	80	950	8,300
Bathroom 1	Walls	30 sq ft	0.28	80	670	
	Glass	14 sq ft	0.45	80	500	
	Ceiling	55 sq ft	0.69	39.8[d]	1510	
	Infiltration (1)[g]	440 cfh[b]	0.018[c]	80	630	3,310
Bathroom 2	Walls	79 sq ft	0.26	80	1640	
	Glass	9 sq ft	0.45	80	320	
	Ceiling	35 sq ft	0.69	39.8[d]	960	
	Floor over garage	35 sq ft	0.25	35[e]	310[p]	
	Infiltration (1)[g]	280 cfh[b]	0.018[c]	80	400	3,630
Living Room	Walls	267 sq ft	0.28	80	5980	
	Walls (adjoining garage)	94 sq ft	0.39[f]	35[e]	1280[p]	
	Glass	50 sq ft	0.45	80	1800	
	Floor	294 sq ft				
	Infiltration (1½)[h]	3745 cfh[b]	0.018[c]	80	5400	14,460
Dining Room	Walls	166 sq ft	0.28	80	3720	
	Glass (doors)	35 sq ft	0.85	80	2380	
	Glass (windows)	20 sq ft	0.45	80	720	
	Floor	168 sq ft				
	Infiltration (1½)[i]	2140 cfh[b]	0.018[c]	80	3080	9,900
Kitchen and Entrance to Garage	Walls	96 sq ft	0.28	80	2150	
	Walls (adjoining garage)	51 sq ft	0.39[f]	35[e]	700[p]	
	Glass	18 sq ft	0.45	80	650	
	Door	17 sq ft	0.51	35	300	
	Floor	125 sq ft				
	Infiltration (1½)[j]	1595 cfh[b]	0.018[c]	80	2300	6,100
Lavette and Vestibule	Walls	82 sq ft	0.28	80	1840	
	Walls (adjoining garage)	85 sq ft	0.39[f]	35[e]	1160[p]	
	Glass	9 sq ft	0.45	80	320	
	Door	19 sq ft	0.51	80	780	
	Floor	30 sq ft				
	Infiltration (1½)[k]	383 cfh[b]	0.018[c]	80	550	4,650
Entrance Hall	Walls	39 sq ft	0.28	80	870	
	Door	21 sq ft	0.38	80	640	
	Ceiling[r]	87 sq ft	0.69	39.8[d]	2390	
	Infiltration (2)[l]	1110 cfh[b]	0.018[c]	80	1600	5,500
Garage	Walls	167 sq ft	0.28	45[e]	2110	
	Glass	53 sq ft	1.13	45	2700	
	Doors	44 sq ft	0.51	45	1010	
	Infiltration (1½)[m]	2360 cfh[b]	0.018[c]	45	1910	
	Floor	29 ft[o]	0.81	45	1060	
	Gain adjoining rooms				−4410[p]	4,380
Recreation Room[q]	Walls	220 sq ft	0.10	38	840	
	Glass	8 sq ft	1.13	80	720	
	Floor	287 sq ft	0.10	20	570	
	Infiltration (1)[n]	2010 cfh[b]	0.018[c]	80	2890	5,020
					TOTAL	100,500

NOTES FOR TABLE 6.

[a] The inside-outside temperature difference is 70− (−10) or 80 F except where otherwise noted.

[b] Volume of infiltration, cfh = (no. air changes) x (floor or ceiling area) x (ceiling height).

[c] From Equation 5a.

[d] The ceiling heat losses are calculated by estimating the attic temperature and then calculating the loss through the ceiling using the proper temperature difference. This unheated attic is not ventilated during winter months. The attic temperature is estimated from Equation 1 to be 30.2 F when the outside temperature is −10 F and room temperature is 70 F. The temperature difference is then 70−30 2 or 39.8 deg. For the insulated residence, attic temperature becomes 4.6 F and temperature difference 70−4.6 = 65.4 deg

[e] Temperature in garage assumed to be 35 F.

[f] Coefficient for wall adjoining garage calculated on basis of metal lath and plaster on both sides of studs (U = 0.39).

[g] One half of value from Table 4, Chapter 10, for storm windows or weatherstripping.

[h] Exposed on two sides, weatherstripped windows offset by fire-place. Use 1½.

[i] Window on one side weatherstripped but double-doors are hard to close tightly. Hence, conservative value of 1¼.

[j] Assuming kitchen vent, door to vestibule usually open, allow full table value of 1½.

[k] One-half value in Table 4, Chapter 10, increased to 1½ by nearby outside door in vestibule.

[l] Full value in Table 4, Chapter 10, to allow for frequent opening of outside door.

[m] Two sides exposed, large doors but large volume. Use value 1½ as given in Table 4, Chapter 10.

[n] Two small unweatherstripped windows in protected location, but fireplace, indicate 1 change.

[p] Heat losses from these rooms into garage are heat gains for garage.

[q] Neglect heat loss to basement, as losses from boiler, piping, etc., will probably keep basement near, f not above, 70 F.

Upstairs hall ceiling figured with downstairs. Heat should be provided downstairs for both.

[s] Linear feet of exposed edge.

TABLE 7. SUMMARY OF HEAT LOSSES OF UNINSULATED RESIDENCE (*Btu Per Hour*)

Room or Space	Walls	Ceiling and Roof	Floor	Glass and Door	Infiltration	Totals
Bedroom A	5330	6910	...	1440	2180	15,860
Bedroom B	3490	4660	...	1440	1470	11,060
Bedroom C	2560	3540	...	970	1260	8,330
Bedroom D	2650	3020	960	720	950	8,300
Bathroom 1	670	1510	...	500	630	3,310
Bathroom 2	1640	960	310	320	400	3,630
Living Room	7260	...	...	1800	5400	14,460
Dining Room	3720	...	...	3100	3080	9,900
Kitchen	2850	...	...	950	2300	6,100
Lavette	3000	...	...	1100	550	4,650
Entrance Hall	870	2390	...	640	1600	5,500
Garage	−1030[a]	−1270[b]	1060	3710	1910	4,380
Recreation	840	...	570	720	2890	5,020
Design Totals	33,850	21,720	2,900	17,410	24,620	100,500
Operating Totals[c]	33,850	21,720	2,900	17,410	12,310	88,190
Percentages[d]	38.4	24.6	3.3	19.7	14.0	100.0

[a] Wall heat loss of 2110 Btuh minus wall heat gains of 1280,700 and 1160 Btuh. [b] Heat gains of 960 and 310 Btuh. [c] Based on ½ computed infiltration. [d] Based on operating totals.

TABLE 8. SUMMARY OF HEAT LOSSES OF INSULATED RESIDENCE (*Btu Per Hour*)

Room or Space	Walls	Ceiling and Roof	Floor	Glass and Door	Infiltration	Totals
Bedroom A	2480	2460	...	1440	2180	8,560
Bedroom B	1620	1660	...	1440	1470	6,190
Bedroom C	1190	1260	...	970	1260	4,680
Bedroom D	1230	1080	690	720	950	4,670
Bathroom 1	310	540	...	500	630	1,980
Bathroom 2	760	250	220	320	400	1,950
Living Room	3370	...	...	1800	5400	10,570
Dining Room	1730	...	...	3100	3080	7,910
Kitchen	1320	...	...	950	2300	4,570
Lavette	1390	...	...	1100	550	3,040
Entrance Hall	410	850	...	640	1600	3,500
Garage	−470[a]	−910[b]	1060	3710	1910	5,300
Recreation	840	...	570	720	2890	5,020
Design Totals	16,180	7,190	2,540	17,410	24,620	67,940
Operating Totals[c]	16,180	7,190	2,540	17,410	12,310	55,630
Percentages[d]	29.1	12.9	4.6	31.3	22.1	100.0

[a] Wall heat loss of 980 Btuh minus wall heat gains of 590, 320 and 540 Btuh. [b] Heat gains 690 and 220 Btuh [c] Based on ½ computed infiltration. [d] Based on operating totals.

buildings may either be continuously heated or have more time allowed for heating up during the few minimum temperature days, no allowance usually is made, except in the size of boilers or furnaces. For churches, auditoriums and other intermittently heated buildings, additional capacity should be provided.

RESIDENCE HEAT LOSS PROBLEMS

Example 6: Calculate the heat loss of the residence shown in Fig. 3 located in the vicinity of Chicago. From Table 1, design outdoor conditions are −10 F and 12 mph wind velocity. Inside temperature from Table 2 is assumed to be 70 F. The attic is unheated. Assume ground temperature to be 50 F (see Fig. 3, Chapter 34) under basement and garage floors and 32 F adjoining basement walls. Estimate infiltration losses by the air change method. No wall, ceiling or roof insulation is to be considered in this problem, but all first and second floor windows, except in the garage, are to have storm sash. The building is constructed as follows (heat transmission coefficients U are parentheses):

Walls: Brick veneer, building paper, wood sheathing, studding, metal lath and plaster (0.28). Walls of dormer over garage, same except wood siding in place of brick veneer (0.26).

Attic Walls: Brick veneer, building paper, wood sheathing on studding (0.42).

Basement Walls: 10 in. concrete (0.10).

Roof: Asphalt shingles on wood sheathing on rafters (0.53).

Ceiling (Second floor): Metal lath and plaster (0.69).

Windows: Double-hung wood windows averaging 70 percent glass (0.45; from Chapter 9, Table 19, Section D, the U value for wood windows with storm sash is 0.53 x application factor; by interpolation this factor is 0.85). Steel casement sash in garage and basement (1.13; from Chapter 9, Table 19, U is 1.13 for all glass and the application factor is 1.00). French doors in dining room 50 percent glass, no storm doors (0.85; from Chapter 9, Table 19, U is 1.13 for all glass; by interpolation the application factor is 0 75).

Floor (Bedroom D): Maple finish flooring on yellow pine sub-flooring; metal lath and plaster ceiling below (0.25).

Floor (Basement and Garage): 4 in. stone concrete on 3 in. cinder concrete (0.10).

Solution: The calculations for this problem are given in Table 6, and a summary of the results in Table 7. The values in column F of Table 6 were obtained by multiplying together the figures in columns C, D, and E. The heat losses are calculated to the nearest 10 Btu. See reference notes for Table 6 for further explanation of data.

Attention is called to the summary of heat losses (Table 7) for the uninsulated residence. As storm windows are used in this instance the *glass* and *door* transmission heat losses of 19.8 percent are relatively small. The infiltration losses of 14.0 percent are also comparatively small because the storm windows are equivalent to weatherstripping. In this problem, the wall, ceiling and floor transmission losses comprise 66.2 percent of the total.

Example 7: Calculate the heat loss of residence shown in Fig. 3 based on the same conditions as in *Example 6* but having construction improved or insulated to obtain coefficients as follows:

Walls, 0 13; *Walls of Dormer over Garage*, 0.12; *Attic Walls*, 0.28; *Walls Adjoining Garage*, 0.18; *Basement Walls (Recreation Room)*, 0.10.

Roof, 0.53.

Ceiling (Second Floor), 0.15.

Windows (Same as in *Example 6*).

Floor (Bedroom D), 0.18.

Solution: The procedure for calculating the heat losses is similar to that for *Example 6*. A summary of the results is given in Table 8.

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CHAPTER 12

COOLING LOAD

Cooling Load Calculations; Design Conditions; Instantaneous Heat Load; Solar Radiation; Periodic Heat Flow; Tables for Calculating Solar Heat Gain Through Walls, Roofs and Glass; Instantaneous Heat Gain vs. Cooling Loads; Load from Interior Partitions, Ceiling and Floors; Load from Outside Air, Ventilation and Infiltration; Effect of Outside Air on Load; Heat Sources Within Conditioned Space; Moisture Transfer Heat Load; Miscellaneous Heat Loads; Required Air Quantity Through Conditioning Equipment; Minimum Entering Air Temperature; Example Cooling Load Calculation

THE variables affecting cooling-load calculations are numerous, often difficult to define precisely, and always intricately inter-related. Most of the components of the cooling load vary in magnitude over a wide range during a 24-hour period, and as the cyclic changes in load components are not usually in phase with each other, careful analysis is required to establish the resultant maximum cooling load for a building or zone. A zoned system must often handle peak loads in different zones at different hours.

Economic considerations must be of particular influence in the selection of equipment for cooling season operation in comfort air conditioning, and this fact, coupled with present inadequacies in available data and knowledge of the air-conditioning art, places a premium on the experienced judgment essential to successful design or practice. Variations in the weather, building occupancy, and other factors affecting load, necessitate carefully coordinated controls to regulate simultaneously the components and the equipment in order to maintain the desired room conditions.

The calculation procedures presented in this chapter deal with the various instantaneous rates of heat gain, both sensible and latent, in a conditioned space. There may be an appreciable difference between the *net instantaneous rate of heat gain* and the *total cooling load* at any instant. This difference is caused by the storage and subsequent release of heat by the structure and its contents. This thermal-storage effect may be quite important in determining an economical cooling equipment capacity. The lack of any adequate means of treating this storage quantitatively in its entirety for a complete structure, must be recognized in judging the procedures and data presented for calculating individual components of the net rate of instantaneous heat gain.

Solar heating calculations involve the same principles as cooling load calculations. Many of the data on solar radiation given in this chapter can be used in calculations for solar heating.

COOLING LOAD CALCULATIONS

Summer cooling load calculations, whether for industrial or comfort applications, require consideration of the following factors:

A. *Design Conditions:* (1) indoor conditions; (2) outdoor conditions; (3) ventilation rate.

B. *Instantaneous Heat Load, Sensible and Latent:* (1) load from solar radiation, sky radiation and from outdoor-indoor temperature differential for glass areas and exterior, walls and roofs, modified by periodic heat flow or lag factors depending on the type of structure; (2) load due to heat gain through interior partitions, ceilings and floors; (3) load due to ventilation, either natural or mechanical; (4) load due to heat sources within the conditioned space such as people, lights, power equipment, and appliances; (5) load due to moisture transfer through permeable building materials; (6) miscellaneous heat sources.

C. *Determination of Air Quantity and Apparatus Dew-Point.*

These factors will be discussed in turn. The material presented leads to an illustrative procedure for a cooling-load calculation, and a numerical example is given to demonstrate the calculations involved.

DESIGN CONDITIONS

Indoor Conditions

Indoor air conditions for human health and comfort have been and continue to be the subject of much discussion and research.

The effective temperature index, explained in Chapter 6 is probably the best available source of design criteria for comfort air conditioning systems for buildings or enclosures in which the air and inside surface

TABLE 1. TYPICAL COMMERCIAL DESIGN ROOM CONDITIONS FOR SUMMER AVERAGE PEAK LOAD IN COMFORT AIR CONDITIONING[a]

Type of Installation	Dry-Bulb Temp	Wet-Bulb Temp[b]	Relative Humidity Per Cent	Grains Per Lb[b]	Effective Temp[c]
Deluxe Application	78	65	50	72.7	72.2
Normal Application	80	67	51	78.5	74.0
15 to 40 min Occupancy	82	68	49	80.0	75.3

[a] Values in Table 1 are for *peak load* conditions. It is general practice to operate a system at approximately 76 F and 50 percent relative humidity at other than peak load.

[b] Psychrometric data for standard barometric pressure.

[c] Fig. 10, Chapter 6, air movement 15 to 25 fpm.

temperatures remain substantially equal; a condition that can safely be assumed for most ordinary comfort air conditioning installations. Other sources of design specifications are to be found in the requirements of codes and ordinances, and in the varied long-term experiences of manufacturers, contractors, and engineering specialists.

Past experience, cumulative over many years, indicates that indoor design conditions for which summer air-conditioning equipment is selected, should not exceed a temperature of 80 F or a relative humidity of 50 percent for the average job in the United States. If these conditions are exceeded, complaints of discomfort may be expected, especially with continuous occupancy. For very brief occupancy only, a slightly higher peak-load design temperature may be employed. In regard to the lower limit of humidity, complaints are not encountered for store installations operated down to 35 percent relative humidity or, for office jobs, somewhat lower. These observations apply to normal commercial practice in this country only; for extremes, such as tropical or very hot regions, it is regarded as more practicable to design for a peak-load outdoor-indoor temperature difference of about 15 to 20 F.

Table 1 offers typical design conditions for average requirements encountered. The *deluxe* figures would also apply in general for localities having a summer outdoor design temperature of 90 F or less; and the *15 to 40 min* occupancy values, or even somewhat higher dry-bulb temperatures, would indicate acceptable conditions for very hot localities. Table 1 *is to be used with good judgment*, for there is no universal rule which may be applied to indoor design conditions.

Guarantees of conditions to be maintained for summer operation are based upon a definite set of load conditions. At other than the guarantee load, the conditions produced by a system are determined by the balance of imposed load and equipment capacity, and by the method adopted for regulating the system operation. Complete specifications of indoor design conditions would include part-load and overload operation, particularly from the viewpoint of economy.

In the field of *industrial air conditioning*, indoor design conditions are established by the requirements of goods and processes, in addition to the

TABLE 2. ILLUSTRATIVE TEMPERATURES AND RELATIVE HUMIDITIES APPLICABLE TO INDUSTRIAL AIR CONDITIONING[a]

CLASSIFICATION	MATERIALS, LOCATION OR PROCESS	TEMPERATURE F	RELATIVE HUMIDITY %
Employee Efficiency	General Machine Shop Work	78–80	50
	Drafting Rooms	78–80	50
	Offices	78–80	50
Storage Prior to Manufacturing	Rough Castings	80	50
	Ceramic Materials	60–80	50
	Pharmaceutical Powders	70–80	15–35
	Sugar	80	35
	Paper	75–80	35
	Electrical Goods	60–80	35–50
	Flour	60–75	55–65
	Rubber	60–75	40–50
	Grains	60	30–45
	Hardened Aluminum Alloys	0 to −30	
Manufacturing Process	Machine Tool Oil Cooling	70–90	
	Precision Parts Honing Machinery	75–80	40–55
	Ceramic Molding	80	60
	Manufacturing of Electrical Wiring	60–80	35–50
	Assembly Line	65–80	40–50
	Gage Rooms	78	50
	Instrument Calibration	68	50–55
	Match Manufacturing	72–74	50
Research and Development	Paper Testing Laboratory	60–80	55–65
	Textile Testing Laboratory	70	65
	Special Process Temperature Boxes	−100 to +170	
	Chemical Laboratories	78	50
	Fibres and Plastics	70–75	50–65
	Drafting	78–80	45–50
	Temperature Shock Tests	−80 to +150	

[a] Taken from the article, "Indoor Climate and Refrigeration for Post-War Industry," by E. K. Heglin, *Cleveland Engineering*, Vol. 40, No 27, July 3, 1947, p. 5.

comfort and efficiency of the workers. No generally-applicable specifications are possible, as each job has its own special requirements. Table 2 offers illustrative general information.

The indoor design conditions suggested have had reference to conditions to be maintained at the level of occupancy. For extremely high ceilings in public or industrial buildings, only the zone from 10 to 15 ft above the floor may be cooled to the full extent. The air temperature at the ceiling would be much higher, and this should be kept in mind when calculating the convective portion of the roof heat gain. A reduction of outdoor-to-indoor air temperature differential may be assumed in such instances; radiation from the inner roof surface is not diminished.

Outdoor Conditions

Summer climatic conditions and suggested design wet-bulb and dry-bulb temperatures are given in Table 3 for various locations in the United

States. The highest temperature ever recorded is for the period of record shown. In some cases it should be noted that this period of record is comparatively short, and higher temperatures may be expected. In making comparisons for other localities than those shown in Table 3, due consideration must be given to elevation.

Column 6 of Table 3 indicates the design dry-bulb temperature suggested by the A.S.H.V.E. Technical Advisory Committee on Weather Design Conditions. This temperature is the maximum hourly outdoor temperature which has been equalled or exceeded 2½ percent of the total hours of June, July, August and September for the period of record, in this case the 5-year period 1935–1939, and should not be confused with the period of record given in Column 4 which applies only to highest temperature ever recorded. Since all of these data (Column 4) are based on airport records, they are not necessarily applicable to cities.

The data given in Columns 7 and 8 were obtained from local A.S.H.V.E. Chapter Secretaries, and represent the design temperatures in local use. Where such information was not available, it was taken from a publication of the A.C.R.M.A.[1] and from various other sources.

The Technical Advisory Committee on Weather Design Conditions has suggested that wet-bulb design temperature be taken as that wet-bulb temperature which has been equalled or exceeded 5 percent of the hours during months of the period of record. While not available for the 1951 edition of The Guide, due to the tremendous task of compiling these data, it is hoped that they will be available for some stations for future editions.

The wind velocity to be used in design should be that wind velocity which accompanies the design temperatures in each instance, but since these data are not available, the average summer wind velocities for the period of record were taken from U. S. Weather Bureau records revised to 1948. It is pointed out that this is not necessarily the velocity which coincides with the design temperatures, but it may serve as a guide to the designer.

Other weather data, such as *daily range of temperature*, are useful particularly when making cooling load calculations for an early morning peak on an east exposure. Daily range of temperature is the difference between the average of the daily maximum dry-bulb temperatures and the average of the daily minimum temperatures. This range is highest in semiarid or desert regions and at high elevations, and lowest near the oceans or very large lakes. The daily range of temperature (Fahrenheit degrees) in July for the principal areas of the United States can be approximated from the following tabulation:

East Sea Shore	12 to 18	East of Mississippi River	19 to 24
Gulf Sea Shore	12 to 18	Mississippi River to Rocky Mountains	24 to 33
Great Lakes Shore	18 to 21	Rocky Mountain Area	33 to 42
West Sea Shore	15 to 20	West Coastal States	20 to 36

Ventilation Rate

The introduction of outside air is necessary for the ventilation of conditioned spaces. Chapter 6 suggests *minimum* outdoor-air requirements for representative applications; but it is to be emphasized that minimum requirements are not necessarily *adequate* requirements for all psychological attitudes and physiological responses.

Local codes and ordinances frequently specify ventilation requirements for public places and for industrial installations.

Recommended and minimum ventilation rates for the most common

TABLE 3. SUMMER CLIMATIC CONDITIONS[a]

Suggested Design Wet-Bulb and Dry-Bulb Temperatures

COL. 1 STATE	COL. 2 STATION[b]		COL. 3 ELEVATION[c] FT	COL. 4 PERIOD OF RECORD[d]	COL. 5 HIGHEST TEMP. EVER RECORDED[d] °F	COL. 6 DESIGN DRY-BULB TEMP. ON T.A.C. 2½% BASIS[e] °F	COL. 7 DESIGN DRY-BULB TEMP. IN COMMON USE[f] °F	COL. 8 DESIGN WET-BULB TEMP. IN COMMON USE[f] °F	COL. 9 AVERAGE SUMMER WIND VELOCITY[g] MPH
Ala.	Anninston	CO	733	1893–1947	105	..	95	78	
	Birmingham	CO	711	1893–1945	107		95	78	5.4
	Birmingham	AP	615	1939–1947	103	94			
	Mobile	CO	143	1872–1947	103		95	80	8 0
	Mobile	AP	219	1940–1947	104	92			
	Montgomery	CO	293	1872–1947	107		95	78	
	Montgomery	AP	226	1938–1944[d]	103				
Ariz.	Flagstaff	CO	6957	1899–1947	93		90	65	
	Kingman	AP	3473	1935–1939	107	99			
	Phoenix	CO	1122	1895–1947	118		105	76	6.0
	Phoenix	AP	1112	1933–1947	117	107			
	Tucson	AP	2561	1935–1939	112	102	105	72	
	Winslow	CO	4853	Up to 1946	107		100	70	
	Winslow	AP	4899	1937–1947	103	95			
	Yuma	CO	146	1876–1946	120		110	78	
Ark.	Fort Smith	CO	545	1882–1945	113		95	76	6 1
	Fort Smith	AP	463	1945–1947	107				
	Little Rock	CO	451	1879–1942[d]	110		95	78	6.2
	Little Rock	AP	282	1942–1947	107	97			
Calif.	Bakersfield	AP	499	1937–1946	113	104	105	70	
	Burbank	AP	740	1931–1947[d]	111	94		.	
	Daggett	AP	1925	1935–1939	113	104			
	Eureka	CO	132	1886–1947	85		90	65	
	Fresno	CO	387	1887–1939	115		105	74	7.9
	Fresno	AP	281	1939–1947	111	103			
	Los Angeles	CO	534	1877–1947	109		90	70	5.8
	Oakland	AP	21	1929–1947	102	80	85	65	
	Red Bluff	CO	305[c]	1877–1934	115		100	70	
	Red Bluff	AP	346	1944–1947	112				
	Redding	AP	579	1935–1939	112	101			
	Sacramento	CO	116	1877–1947	114		100	72	7.9
	Sacramento	AP	22	1938–1947	108				
	San Diego	CO	90	1871–1940	110		85	68	
	San Diego	AP	34	1939–1947	106	79			
	San Francisco	CO	164	1875–1947	101		85	65	10.7
	San Jose	CO	100	Up to 1946	106		91	70	
	Williams	AP	124	1935–1939	116	103			
Colo..	Denver	CO	5398	1871–1947	105		95	64	6.9
	Denver.	AP	5379	1934–1947	104	93			
	Durango	CO	6558	Up to 1946	99		95	65	
	Grand Junction	CO	4587[c]	Up to 1946	105		95	65	6.3
	Pueblo	CO	4770	1889–1938	104		95	65	
	Pueblo	AP	4810	1939–1947	104	95			.
Conn..	Hartford	CO	229	1905–1940[d]	101		93	75	.
	Hartford	AP	20	1940–1947	98	88			
	New Haven	CO	180	1872–1947	101		95	75	7.4
	New Haven	AP	17	1943–1947	94	84			
D. C.	Washington	CO	128	1871–1947	106		95	78	5.9
	Washington	AP	20	1935–1939	106	92			
Fla.	Apalachicola	CO	23	1922–1947	102		95	80	
	Jacksonville	CO	104	1871–1947	104		95	78	8.4
	Jacksonville	AP	29	1938–1947	105	94			
	Key West	CO	23	1871–1947	100		98	78	
	Key West	AP	48	1939–1947	95				
	Miami	CO	253	1896–1947	96		91	79	8.7
	Miami	AP	13	1940–1947	100	89			
	Pensacola	CO	67	1879–1947	103		95	78	
	Pensacola	AP	113	1943–1947	105				
	Tampa	CO	111	1890–1940	98		95	78	7.4
	Tampa	AP	12	1941–1946[d]	98				
	Titusville	AP	52	1935–1939	98	90			
Ga.	Atlanta	AP	1020	1935–1939	102	93	95	76	7.9
	Augusta	CO	195	1871–1946	106		98	76	
	Augusta	AP	424	1939–1947	105				
	Macon	CO	408	1899–1947	105		95	78	
	Macon	AP	432	1939–1947	102				
	Savannah	CO	115	1871–1945[d]	105		95	78	8 0
	Savannah	AP	56	1939–1947	105	93			
Idaho	Boise	CO	2818	1864–1939	112		95	65	5.8
	Boise	AP	2849	1939–1947	109	95			
	Burley	AP	4150	1935–1939	104	94			
	Idaho Falls	AP	4744	1935–1939	100	88			
	Lewiston	CO	763	1900–1944[d]	117		95	65	
	Pocatello	CO	4522	1899–1947	105		95	65	
	Pocatello	AP	4467	1938–1947	103	92			
Ill.	Cairo	CO	319	1872–1947	106		98	78	
	Chicago	CO	601	Up to 1946	105		95	75	9.5
	Chicago	AP	615	1935–1939	107	92			
	Moline	AP	594	1932–1947	106	94	96	76	

Table 3. Summer Climatic Conditions[a] (Continued)
Suggested Design Wet-Bulb and Dry-Bulb Temperatures

Col. 1 State	Col. 2 Station[b]	Col. 3 Elevation[c] Ft	Col. 4 Period of Record[d]	Col. 5 Highest Temp. Ever Recorded[d] °F	Col. 6 Design Dry-Bulb Temp. on T.A.C. 2½% Basis[e] °F	Col. 7 Design Dry-Bulb Temp. in Common Use[f] °F	Col. 8 Design Wet-Bulb Temp. in Common Use[f] °F	Col. 9 Average Summer Wind Velocity[g] Mph
Ill.	Peoria AP	660	1935–1939	111	94	96	76	8.2
	Springfield CO	603	1879–1947	110		98	77	
	Springfield AP	608	1930–1947	109	96			
Ind.	Evansville CO	464	1897–1940	108		95	78	7.0
	Fort Wayne CO	885	1911–1941[d]	106	...	95	75	
	Helmer AP	970	1935–1939	107	89			
	Indianapolis CO	816	1871–1946	106		95	76	8.9
	Indianapolis AP	800	1932–1946	107	91			
	Terre Haute CO	1146	1893–1946	110		95	78	
	Terre Haute AP	589	1941–1946	103				
Iowa	Davenport CO	648	1872–1947	111	...	95	78	
	Des Moines AP	979	1935–1939	111	95	95	78	8.6
	Dubuque CO	740	1874–1947	110	...	95	78	
	Keokuk CO	637	1872–1946	113	...	95	78	
	Sioux City CO	1093[c]	1889–1944[d]	111	...	95	78	
	Sioux City AP	1098	1940–1946	108	...			
Kans.	Concordia CO	1425	1885–1947	116	...	95	78	
	Dodge City CO	2515	1874–1942[d]	109	...	95	78	...
	Dodge City AP	2599	1942–1947	109	...	...		...
	Topeka CO	991	1887–1947	114	...	100	78	...
	Topeka AP	883	1946–1947	108				
	Wichita CO	1497	1888–1939	114		100	75	11.8
	Wichita AP	1423	1939–1947	109	100			
Ky.	Louisville CO	563	1871–1947	107		95	78	7.2
	Louisville AP	544	1937–1947	103	93			
La.	New Orleans CO	85	1874–1947	102		95	80	6.9
	New Orleans AP	8	1937–1947	100	93			
	Shreveport AP	179	1935–1939	109	98	100	78	7.0
Maine.	Eastport CO	100	1873–1947	93		90	70	
	Portland CO	185	1885–1940	103	...	90	73	8.7
	Portland AP	65	1940–1947	99				
Md.	Baltimore CO	114	1871–1947	107		95	78	7.4
	Baltimore AP	43	1935–1939	105	91			
Mass.	Boston CO	356	1870–1935	104		92	75	12.5
	Boston AP	45	1936–1947	101	87			
	Nantucket CO	45	1886–1947	92	...	95	75	
	Nantucket AP	48	1946–1947	82		...		
Mich.	Alpena CO	615	1874–1946	104	...	95	75	
	Detroit CO	1000	1873–1933	104		95	75	9.5
	Detroit AP	632	1934–1947	105	89		73[h]	
	Lansing CO	861	1910–1947	102	...	95	75	
	Lansing AP	863	1940–1947	98	...			...
	Marquette CO	721	1874–1947	108	...	93	73	
Minn.	Duluth CO	1133	1874–1947	106	...	93	73	
	Duluth AP	1413	1941–1947	95	...			
	Minneapolis CO	945	1890–1947	108	...	95	75	10.2
	Minneapolis AP	873	1938–1947	104	...	...		...
	St. Paul CO	951	1871–1933	104		95	75	
	St. Paul AP	708	1937–1947	104	91			
Miss.	Meridian CO	410	1889–1947	105	...	95	79	4.6
	Meridian AP	298	1939–1947	105				
	Vicksburg CO	316	1874–1947	104		95	78	6.4
	Vicksburg AP	266	1941–1947	104	...			
Mo.	Columbia CO	739	1889–1947	111		100	78	
	Columbia AP	787	1939–1947	102				
	Kansas City AP	780	1935–1939	112	101	100	76	9.1
	St. Louis CO	646	1871–1947	110		95	78	9.5
	St. Louis AP	597	1930–1947	111	97			
	Springfield AP	1270	1935–1939	105	96			8.7
Mont.	Billings AP	3584	1935–1947	106	92	90	66	
	Butte AP	5538	1931–1947	100	85			
	Havre CO	2498	1880–1947	108		95	70	
	Helena CO	4175	1880–1940	103		95	67	8.1
	Kalispell CO	3004	1897–1947	101		95	65	
	Miles City AP	2629	1935–1939	108	97			
Nebr.	Lincoln CO	1189	1887–1947	115		95	78	9.7
	Lincoln AP	1185	1933–1947	115	101			
	North Platte CO	2815	1874–1947	109		95	78	8.1
	North Platte AP	2788	1935–1939	109	98		...	
	Omaha CO	1219	1873–1935[d]	111		95	78	...
	Omaha AP	1009	1935–1947	114	98		...	...
	Valentine CO	2627	1889–1947	110		95	78	...
Nev.	Elko AP	5079	1935–1939	102	92		...	...
	Las Vegas AP	1882	1937–1947	117	108	...		
	Reno CO	4588	1905–1942	106		95	65	7.2
	Reno AP	4417	1940–1947	105	93		...	...
	Winnemucca CO	4293	1871–1947	108	...	95	65	...
N. H.	Concord CO	343	1871–1941[d]	102	...	90	73	4.9
	Concord AP	359	1941–1947	99	...	...	...	...

TABLE 3. SUMMER CLIMATIC CONDITIONS[a] (CONTINUED)
Suggested Design Wet-Bulb and Dry-Bulb Temperatures

COL. 1 STATE	COL. 2 STATION[b]	COL. 3 ELEVATION[c] FT	COL. 4 PERIOD OF RECORD[d]	COL. 5 HIGHEST TEMP. EVER RECORDED[d] °F	COL. 6 DESIGN DRY-BULB TEMP. ON T.A.C. 2½% BASIS[e] °F	COL. 7 DESIGN DRY-BULB TEMP. IN COMMON USE[f] °F	COL. 8 DESIGN WET-BULB TEMP. IN COMMON USE[f] °F	COL. 9 AVERAGE SUMMER WIND VELOCITY[g] MPH
N. J..	Atlantic City CO	45	1874–1947	104		95	78	
	Camden AP	20	1935–1939	105	91			
	Newark AP	15	1931–1947	104	89	95	75	
	Trenton CO	144	1866–1946	106		95	78	8.8
N. M..	Albuquerque. CO	5022	1931–1933[d]	99		95	70	7.8
	Albuquerque. AP	5319	1933–1947	101	93			
	El Morro AP	7120	1935–1939	92	84			
	Rodeo AP	4116	1935–1939	104	97			
	Roswell CO	3643	1905–1947[d]	107		95	70	
	Tucumcari AP	4054	1935–1939	107	97			
N. Y.	Albany CO	114	1874–1947	104		93	75	7.5
	Albany AP	280	1938–1947	99	88			
	Binghamton CO	915	1891–1946	103		95	75	
	Binghamton AP	836	1942–1947	97				
	Buffalo AP	726	1935–1939	95	86	93	73	12 1
	Canton CO	458	1906–1947	99		90	73	8 2
	Elmira AP	948	1935–1939	96	88			
	New York CO	425	1871–1947	102		95	75	12.5
	Oswego CO	363	1871–1947	100		93	73	
	Rochester CO	609	1872–1947	102		95	75	
	Rochester AP	560	1935–1939	98	89			
	Syracuse CO	465	1902–1940	102		93	75	
	Syracuse AP	404	1940–1947	97	88			
N. C.	Asheville CO	2280	1902–1947	99		93	75	5.6
	Charlotte CO	809	1878–1947	103		95	78	
	Charlotte AP	757	1939–1947	103	93			
	Greensboro AP	896	1928–1947	101	91	95	78	
	Raleigh CO	405	1887–1947	104		95	78	6.3
	Raleigh AP	446	1944–1947	102	93			
	Wilmington CO	78	1871–1947	103		95	78	8.4
N. D..	Bismarck CO	1675	1875–1940	114		95	73	9.5
	Bismarck AP	1655	1940–1947	109	96			
	Devils Lake.. CO	1481	1904–1947	112		95	70	
	Dickinson AP	2599	1935–1939	112	94			
	Fargo AP	900	1935–1939	115	93	95	75	
	Pembina AP	830	1935–1939	109	92			
	Williston CO	1919	1879–1947	110		95	73	
Ohio	Akron AP	104	1935–1939	101	88	95	75	
	Cincinnati CO	772	1870–1947	108		95	78	5.6
	Cincinnati AP	488	1931–1947	108	94			
	Cleveland CO	669	1871–1946	100		95	75	11.1
	Cleveland AP	813	1930–1946	107	90			
	Columbus CO	812	1878–1946	106		95	76	
	Columbus AP	820	1939–1947	100	90			
	Dayton CO	1086	1883–1943[d]	108		95	78	
	Dayton AP	1002	1940–1947	99				
	Sandusky CO	608	1878–1946	105		95	75	
	Toledo CO	668	1871–1947	105		95	75	
	Toledo AP	626	1940–1947	100	91			
Okla	Ardmore AP	762	1935 1939	110	99			
	Oklahoma City CO	1264	1890–1947	113		101	77	9.8
	Oklahoma City AP	1311	1939–1947	109	99			
	Tulsa AP	686	1932–1947	109	100	101	77	
	Waynoka AP	1529	1935–1939	115	103			
Ore.	Arlington AP	881	1935–1939	111	95			
	Baker CO	3501	1889–1947	104		90	66	
	Baker AP	3374	1939–1947	103	90			
	Eugene CO	366	1890–1942	104		90	68	
	Eugene AP	368	1942–1947	105	88			
	Medford CO	1428	1911–1929	110		95	70	
	Medford AP	1343	1929–1947	115	95			
	Portland CO	98	1874–1947	107		90	68	6 5
	Portland AP	25	1940–1947	105	87			
	Roseburg CO	523	1877–1947	109		90	66	
Pa.	Curwensville AP	2219	1943–1947	90	82			
	Erie CO	771	1873–1946	98		93	75	
	Erie AP	736	1935–1939	96	85			
	Harrisburg AP	339	1935–1939	103	91	95		
	Philadelphia CO	200	1871–1947	106		95	78	9.7
	Philadelphia AP	18	1940–1947	100				
	Pittsburgh CO	929	1875–1947	103		95	75	8.9
	Pittsburgh AP	1284	1935–1947	102	88			
	Reading CO	311	1913–1947	105		95	75	
	Scranton CO	877	1901–1947	103		95	75	
	Sunbury AP	448	1935–1939	101	89			
R. I.	Block Island CO	46	1881–1947	93		95	75	
	Providence CO	77	1904–1947	100		93	75	9.5
S. C.	Charleston CO	59	1871–1947	104		95	78	9 8
	Charleston AP	51	1940–1947	103	91			

TABLE 3. SUMMER CLIMATIC CONDITIONS[a] (CONCLUDED)

Suggested Design Wet-Bulb and Dry-Bulb Temperatures

COL. 1 STATE	COL. 2 STATION[b]		COL. 3 ELEVATION[c] FT	COL. 4 PERIOD OF RECORD[d]	COL. 5 HIGHEST TEMP. EVER RECORDED[d] °F	COL. 6 DESIGN DRY-BULB TEMP. ON T.A.C. 2½% BASIS[e] °F	COL. 7 DESIGN DRY-BULB TEMP. IN COMMON USE[f] °F	COL. 8 DESIGN WET-BULB TEMP. IN COMMON USE[f] °F	COL. 9 AVERAGE SUMMER WIND VELOCITY[g] MPH
S. C..	Columbia	CO	401	1887–1947	106		95	75	
	Columbia	AP	227	1939–1947[d]	104				
S. D. ..	Huron	CO	1342	1881–1938[d]	111		95	75	10.3
	Huron	AP	1287	1938–1947	110				
	Rapid City	CO	3309	1888–1947	106		95	70	7.9
	Rapid City	AP	3220	1939–1947	108				
Tenn.	Chattanooga	CO	952	1879–1947	103		95	76	5.6
	Chattanooga	AP	675	1940–1947	105	94			
	Knoxville	CO	1024	1871–1942[d]	104		95	75	5 7
	Knoxville	AP	1007	1942–1947	102	94			
	Memphis	CO	348	1872–1941[d]	106		95	78	7.3
	Memphis	AP	267	1941–1947	105	96			
	Nashville	CO	714	1871–1947	106		95	78	
	Nashville	AP	610	1939–1947	104				
Texas	Abilene	CO	1748	1885–1944[d]	111		100	74	
	Abilene	AP	1756	1940–1947	109	97			
	Amarillo	CO	3686	1892–1941	107		100	72	11.8
	Amarillo	AP	3595	1941–1947	106	96			
	Austin	CO	625	1897–1942	109		100	78	
	Austin	AP	625	1942–1947	104				
	Brownsville	CO	140	1922–1943[d]	102		100	80	7 0
	Brownsville	AP	25	1943–1947	100				
	Corpus Christi	CO	21	1887–1942	105		95	80	
	Corpus Christi	AP	45	1943–1946	101				
	Dallas	CO	732	1913–1940	110		100	78	9.3
	Dallas	AP	520	1940–1947	109	99			
	Del Rio	CO	1020	1905–1947	111		100	78	
	El Paso	CO	3792	1887–1942	106		100	69	8 4
	El Paso	AP	3956	1939–1947	104	97			
	Fort Worth	CO	708	1898–1939	112		100	78	9.5
	Fort Worth	AP	728	1940–1947	110				
	Galveston	CO	128	1871–1947	101		95	80	9 7
	Galveston	AP	9	1939–1947	101				
	Houston	CO	198	1888–1947	108		95	78	8 8
	Houston	AP	73	1932–1947	105	93			
	Palestine	CO	555	1881–1947	108		100	78	
	Port Arthur	CO	64	1917–1947	102		95	79	
	Port Arthur.	AP	21	1944–1947	98				
	San Antonio	CO	770	1885–1941[d]	107		100	78	7.8
	San Antonio	AP	800	1942–1947	104	98			
	Waco	AP	513	1931–1947	111	98			
	Wink.	AP	2811	1935–1939	110	99			
Utah.	Milford	AP	5095	1935–1939	103	94			
	Modena	CO	5472	1901–1947	101		95	65	
	Salt Lake City.	CO	4346	1874–1947	105		95	65	9 8
	Salt Lake City.	AP	4254	1928–1947	106	95			
Vt.	Burlington	CO	409	1884–1943[d]	100		90	73	8 5
	Burlington	AP	335	1943–1947	101				
Va.	Cape Henry	CO	24	1874–1947	104		95	78	
	Lynchburg	CO	644	1874–1944	106		95	75	
	Lynchburg	AP	951	1944–1947	100				
	Norfolk	CO	91	1871–1947	105		95	78	10 1
	Richmond	CO	180	1897–1947	107		95	78	6.4
	Richmond	AP	172	1929–1947	104	92			
	Roanoke	AP	1194	1935–1939	103	90	95	76	
Wash.	Ellensburg	AP	1731	1935–1939	105	90			
	North Head	CO	199	1884–1947	97		85	65	
	Seattle	CO	104	1890–1947	100		85	65	7.7
	Seattle	AP	47	1928–1947	99	81			
	Spokane	CO	2030	1881–1941[d]	108		93	65	6 5
	Spokane	AP	1974	1941–1947	104	92			
	Tacoma	CO	279	1897–1947	98		85	64	
	Tatoosh Island	CO	110	1883–1947	88				
	Yakima	CO	1160	1928–1946	110		95	65	
	Yakima	AP	1066	1944–1947	103				
W. Va.	Parkersburg	CO	685	1888–1947	106		95	75	5.2
Wisc.	Green Bay	CO	598	1886–1947	104		95	75	9.2
	La Crosse	CO	725	1872–1947	108		95	75	6.4
	La Crosse	AP	677	1943–1947	96	91			
	Madison	CO	1008	1858–1947	107		95	75	7 9
	Madison	AP	884	1935–1939	106	89			
	Milwaukee	CO	744	1870–1947	105		95	75	9.8
	Milwaukee	AP	707	1927–1947	106	87			
Wyo.	Cheyenne	CO	6144	1873–1935	100		95	65	9.2
	Cheyenne	AP	6161	1935–1947	100	89			
	Lander	CO	5448	1891–1946	102		95	65	
	Lander	AP	5568	1936–1947	97				
	Rock Springs	AP	6746	1932–1942	98	87	..	.	...

NOTES FOR TABLE 3

[a] Data compiled from U. S. Weather Bureau Data and various other sources.

[b] Column 2. The station designation AP or CO indicates airport or city office station, respectively.

[c] Column 3. The elevations marked c are ground elevations of the station. All other elevations given are the actual elevations of the thermometer bulb above mean sea level, corrected to 1948.

[d] The periods of record indicated apply only to the highest temperature ever recorded shown in Column 5, and do not necessarily include all of the summer months of the first year indicated. The last year indicated includes July or August, except those marked d which terminate prior to July of that year.

[e] It should be noted that Column 6 applies only to airports, as these data for city stations are not available at this time. The temperature shown is the maximum hourly outdoor temperature which has been equalled or exceeded 2½ percent of the total hours of June, July, August and September for the 5-year period 1935–1939, inclusive. It is pointed out that in most cases the airport stations are outside of the city, and that these data would apply primarily to rural areas.

[f] Columns 7 and 8 record wet and dry-bulb temperatures in use by A.S.H.V.E. Members as reported by Chapter Secretaries for various stations. Where such values were not available, the design temperatures published by A.C.R.M.A., or obtained from various other sources, have been inserted.

[g] The average wind velocities indicated in Column 9 were furnished by the U. S. Weather Bureau, corrected to 1947. In general these velocities are averages for the months of June through September.

[h] The bulletin published by A.S.H.V.E. for the annual weather data of Detroit indicates 73 F as the design wet-bulb temperature which has been equalled or exceeded 5 percent of the hours for period 1935–1939.

[i] Blank spaces indicate data not available.

TABLE 4. VENTILATION STANDARDS[a]

Application	Smoking	Cfm per Person: Recommended	Cfm per Person: Minimum[b]	Cfm per Sq Ft of Floor: Minimum[b]
Apartment Average	Some	20	10	
Deluxe	Some	30	25	0.33
Banking Space	Occasional	10	7½	
Barber Shops	Considerable	15	10	
Beauty Parlors	Occasional	10	7½	
Brokers' Board Rooms	Very Heavy	50	20	
Cocktail Bars	Heavy	40	25	
Corridors (Supply or Exhaust)				0.25
Department Stores	None	7½	5	0 05
Directors' Rooms	Extreme	50	30	
Drug Stores[d]	Considerable	10	7½	
Factories[c, f]	None	10	7½	0.10
Five and Ten Cent Stores	None	7½	5	
Funeral Parlors	None	10	7½	
Garages[c]				1.0
Hospitals Operating Rooms[e, f]	None			2.0
Private Rooms	None	30	25	0.33
Wards	None	20	10	
Hotel Rooms	Heavy	30	25	0.33
Kitchens Restaurant				4.0
Residence				2.0
Laboratories[d]	Some	20	15	
Meeting Rooms	Very Heavy	50	30	1.25
Offices General	Some	15	10	
Private	None	25	15	0.25
Private	Considerable	30	25	0.25
Restaurant Cafeteria[d]	Considerable	12	10	
Dining Room[d]	Considerable	15	12	
School Rooms[c]	None			
Shop, Retail	None	10	7½	
Theater[c]	None	7½	5	
Theater	Some	15	10	
Toilets[c] (Exhaust)				2.0

[a] Taken from present-day practice or large air conditioning companies. [b] When minimum is used, take the larger of the two. [c] See local codes which may govern. [d] May be governed by exhaust. [e] May be governed by special sources of contamination or local codes. [f] All outside air recommended to overcome explosion hazard of anesthetics.

applications are summarized in Table 4. For further general applications, a basis of estimating the cfm per person may be taken as:

1. People not smoking . . . . 7½ *Recommended* *5 Minimum*
2. People smoking . . . . 40 *Recommended* *25 Minimum*

The cooling load due to the introduction of outside air for ventilation is determined once the indoor and outdoor design conditions are fixed. Calculations will be discussed subsequently.

INSTANTANEOUS HEAT LOAD

The total cooling load is frequently divided for convenience into two components, *sensible heat* and *latent heat*. While this subdivision is not imperative, past practice has found it convenient.

A gain of *sensible heat* is considered to occur when there is a direct addition of heat to the enclosure by any one or all of the mechanisms of conduction, convection, and radiation. A gain of *latent heat* is considered to occur when there is an addition of water vapor to the air of the enclosure. For example, when the humidity in an enclosure is increased by water vapor emitted by human occupants, or by water vapor resulting from a process such as cooking, the heat required to vaporize the water does *not* come from the air. Maintenance of a constant humidity ratio in a sealed enclosure requires the condensation of water vapor in the cooling apparatus at a rate equal to its rate of addition within the enclosure. The rate of heat removal from this condensing vapor would be substantially equal to the product of

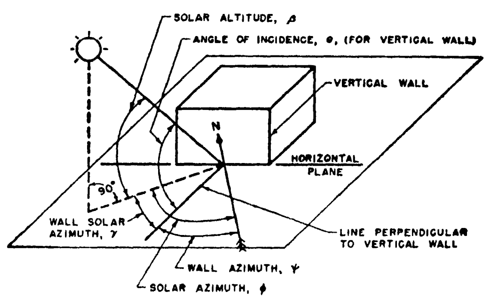


FIG. 1. DEFINITION OF SOLAR ANGLES

the rate of condensation and the latent heat of condensation; this product, expressed in Btu per hour, would be called a *latent heat* load.

As a further example, the infiltration of outdoor air with a high dry-bulb temperature and a high humidity ratio, and the corresponding escape of room air at a lower dry-buld temperature and a lower humidity ratio, would increase both the *sensible heat* load and the *latent heat* load.

SOLAR RADIATION

Magnitude of Solar Radiation

If a plane surface were set perpendicular to the sun's rays (*i.e.*, for *normal incidence*) outside the earth's atmosphere, it would receive solar radiation of about 420 Btu per (hr) (sq ft). A similarly oriented surface, at the surface of the earth, would receive considerably less solar energy because a large part of the radiation is scattered in passing through the air, moisture, smoke, and dust which comprise the earth's atmosphere, and also, because some of the atmospheric constituents, notably water vapor, ozone, and carbon dioxide, absorb solar radiation. The intensity of solar radiation varies with wave length, reaching a peak at about 0.5 microns (a micron equals 1/1000 of a millimeter) and, for practical purposes, is confined to the radiation spectrum between 0.3 and 2.3 microns. The effects of scattering and absorption vary with the wave length, but to make an exact analysis of these phenomena is impracticable in air-conditioning estimates. The important principle to remember is that the total radiation I_t, received by a surface at the earth, is the sum of I_D and I_d, where

$I_D = K\, I_{Dn}$ = the *direct* or *beamed* solar radiation, Btu per (hour)(square foot of receiving surface).

I_{Dn} = the direct solar radiation normal to the sun's rays, Btu per (hour) (square foot of receiving surface).

TABLE 5. Values of I_{Dn}, Direct Solar Radiation Received at Normal Incidence at the Earth's Surface, and Values of I_d, Diffuse or Sky Solar Radiation, Received by Variously Oriented Surfaces

Solar Altitude β, Degrees	Btu per (hour) (square foot): For Clear Atmospheres: Direct[a] Normal Radiation	Diffuse or Sky Radiation[b,c]					For Industrial Atmospheres: Direct[d] Normal Radiation	Diffuse or Sky Radiation[b,e]				
AM → ↓		N	E	S	W	Horiz.		N	E	S	W	Horiz.
5	67	6	11	4	4	7	34	4	11	5	3	9
10	123	11	20	8	7	14	58	8	22	9	7	18
15	166	14	27	11	10	19	80	11	28	13	9	24
20	197	15	32	13	12	23	103	13	36	17	12	31
25	218	16	35	15	13	26	121	16	43	21	16	38
30	235	17	36	17	15	28	136	18	47	24	18	44
35	248	17	36	19	16	30	148	19	50	27	21	48
40	258	18	36	21	17	31	158	20	50	30	23	52
45	266	19	35	23	18	32	165	21	49	31	25	55
50	273	19	33	25	19	33	172	22	47	34	27	58
60	283	21	28	27	21	34	181	22	41	37	30	63
70	289	22	23	29	23	35	188	22	34	41	34	69
80	292	—	—	—	—	—	195	—	—	—	—	—
90	294	—	—	—	—	—	200	—	—	—	—	—
↑ PM →		N	W	S	E	Horiz.		N	W	S	E	Horiz.

[a] Moon's[2] proposed standard for sea level, 20 mm precipitable water vapor, 300 dust particles per cu cm, 2.8 mm Hg partial pressure of ozone.

[b] For 40 deg north latitude on about August 1.

[c] Based on observations by ASHVE Laboratory at Cleveland on cloudless days during which the observed normal incidence values closely approximated the normal incidence values tabulated.

[d] Derived from recommended design sol-air temperatures[4] for New York City for a horizontal surface with absorptivity of 1.0.

I_d = the *sky* or *diffuse* solar radiation, Btu per (hour)(square foot of receiving surface). This comes principally from the atmosphere itself as a consequence of scattering. Vertical surfaces also receive solar radiation by reflection of direct and diffuse radiation from the ground and other objects. Such radiation is usually diffuse. The diffuse radiation strikes at *all* angles.

I_t = total incident solar radiation, Btu per (hour)(square foot of receiving surface).

K = cosine of the angle of incidence, θ. For a vertical surface, θ is defined in Fig. 1.

Standardized, practical-purpose values of the *direct* solar radiation I_{Dn} incident upon a plane *perpendicular* to the sun's rays at the earth's surface, have been proposed by Moon.[2] Table 5 gives these values. They are representative of a *clear* summer day at sea-level elevation, and are nearly identical with values derived from suggested design sol-air temperatures for Lincoln, Nebraska.[3] Values typical of a humid industrial area derived from sol-air data for New York City[4] are also given in Table 5. Day-to-day changes in the amount of dust and water vapor in the atmosphere cause large differences in solar intensity values observed on cloudless days at a given locality. For example, it has been observed in Cleveland that values of the order of those given for industrial atmospheres are usually associated with dry-bulb and wet-bulb temperatures near the design values of 95 F and 75 F (67 F dew-point). On the other hand, values approaching or exceeding those for a clear atmosphere are often encountered during Cleveland summers, but with dew-point and maximum dry-bulb temperatures 10 to 15 deg lower. Considerable judgment, therefore, is required in selecting solar intensity values for design purposes.

Data regarding the irradiation of vertical and horizontal surfaces by diffuse or sky radiation are few. Suggested design values for a 40-deg latitude on August 1 (18 deg declination, north) are given in Table 5 for the two types of atmospheres. These are based upon observations made on cloudless days in Cleveland over a period of several summers. Since less extensive data were available for industrial atmospheres, there is more uncertainty regarding these values. In both instances, the values include an unknown amount of ground reflection, which may be expected to vary with location. It should be noted that clouds which do not obscure the sun tend to *increase* diffuse radiation values. Nearby buildings may *reduce* diffuse irradiation by partial shading.

Calculation Tables

The irradiation of a surface by the sun is the product of I_{Dn}, the direct normal radiation (see Table 5), and the cosine K of the incident angle, θ. For horizontal surfaces, the cosine K equals the sine of the solar altitude. For vertical walls, K is a function of the solar altitude β and the wall solar azimuth γ, thus

$$K = \cos\theta = \cos\beta \cos\gamma \qquad (1)$$

These three angles are defined in Fig. 1. Values of K are given in Table 6 and values of β and γ are given in Table 7 for 18 deg north declination (August 1).

To compute K values for orientations other than those given in Table 6, third angle ϕ, the solar azimuth, is required. In this discussion, ϕ will be measured east from south in the morning, and west from south in the afternoon. Hence, ϕ values are equal to 90 deg minus the γ values for an east or west facing wall, except when Table 7 shows the south walls to be in the shade. In this case ϕ equals $90 + \gamma$, that is, ϕ is greater than 90 deg.

The wall azimuth ψ is the angle, measured east from south to the perpendicular to the wall for walls which have an easterly component, and west from south for those having a westerly component. For example, ψ for a wall facing northeast is 135 deg.

The wall solar azimuth γ may be found according to the following schedule:

For walls facing east of south:	For walls facing west of south:
$\gamma = \phi - \psi$ a.m.	$\gamma = \phi + \psi$ a.m.
$\gamma = \phi + \psi$ p.m.	$\gamma = \phi - \psi$ p.m.

TABLE 6. VALUES OF K, THE COSINE OF THE INCIDENT ANGLE, FOR VARIOUSLY ORIENTED WALLS AND A HORIZONTAL SURFACE

Computed for 18 Deg Declination, North (August 1)

30 Deg North Latitude

Sun Time AM ↓		Cosine K of the Incident Angle						
AM →		N	NE	E	SE	S	SW	Horiz.
6 a.m.	6 p.m.	0.267	0.862	0.952	0.484			0.156
7	5	0.144	0.752	0.919	0.548			0.367
8	4	0.030	0.604	0.824	0.561			0.566
9	3		0.427	0.672	0.524	0.068		0.737
10	2		0.234	0.476	0.438	0.144		0.866
11	1		0.039	0.246	0.310	0.192		0.951
12				0.000	0.147	0.208	0.147	0.978
	↑ PM →	N	NW	W	SW	S	SE	Horiz.

40 Deg North Latitude

Sun Time AM ↓		Cosine K of the Incident Angle						
AM →		N	NE	E	SE	S	SW	Horiz.
5 a.m.	7 p.m.	0.406	0.934	0.914	0.358			0.009
6	6	0.237	0.840	0.951	0.505			0.199
7	5	0.079	0.705	0.919	0.594			0.391
8	4		0.533	0.824	0.631	0.069		0.566
9	3		0.337	0.673	0.614	0.196		0.713
10	2		0.129	0.475	0.542	0.292		0.829
11	1			0.246	0.424	0.354	0.076	0.903
12				0.000	0.265	0.375	0.265	0.927
	↑ PM →	N	NW	W	SW	S	SE	Horiz.

50 Deg North Latitude

Sun Time AM ↓		Cosine K of the Incident Angle						
AM →		N	NE	E	SE	S	SW	Horiz.
5 a.m.	7 p.m.	0.385	0.922	0.920	0.378			0.078
6	6	0.199	0.813	0.951	0.532			0.233
7	5	0.010	0.656	0.918	0.643			0.399
8	4		0.465	0.824	0.700	0.166		0.545
9	3		0.252	0.673	0.609	0.316		0.669
10	2		0.030	0.475	0.642	0.433		0.766
11	1			0.247	0.532	0.505	0.183	0.829
12				0.000	0.375	0.530	0.375	0.848
	↑ PM →	N	NW	W	SW	S	SE	Horiz.

Treat negative values of γ as if they were positive. If γ is greater than 90 deg, the wall is in the shade.

Values of K for other seasons and latitudes may be found in the literature,[5] or may be computed from data given in *Hydrographic Office Bulletin* No. 214, Tables of Computed Altitude and Azimuth[6] and the Ephemeris of the Sun.[7] Table 8 shows the variation of solar declination during the months ordinarily requiring cooling.

Example 1: Find the solar azimuth ϕ at 6:30 p.m. at 40 deg north latitude on August 1st.

Solution: From Table 7 in the column of γ for a wall facing west ϕ for 6:00 p.m.

Table 7. Values of the Wall Solar Azimuth, γ, for Variously Oriented Walls and Solar Altitude

Computed for 18 Deg Declination, North (August 1)

30 Deg North Latitude

Sun Time		Solar Altitude β Degrees	Azimuth Angle γ, Degrees					
AM → ↓			N	NE	E	SE	S	SW
6 a.m.	6 p.m.	9.0	74	29	16	61		
7	5	21.5	81	36	9	54		
8	4	34.5	88	43	2	47	shade	
9	3	47.5	shade	51	6	39	84	
10	2	60.0		62	17	28	73	
11	1	72.0		83	38	7	52	shade
12		78.0		shade	90	45	0	45
	↑ PM →		N	NW	W	SW	S	SE

40 Deg North Latitude

Sun Time		Solar Altitude β Degrees	Azimuth Angle γ, Degrees					
AM → ↓			N	NE	E	SE	S	SW
5 a.m.	7 p.m.	0.5	66	21	24	69		
6	6	11.5	76	31	14	59		
7	5	23.0	85	40	5	50	shade	
8	4	34.5	shade	50	5	40	85	
9	3	45.5		61	16	29	74	
10	2	56.0		76	31	14	59	shade
11	1	64.5		shade	55	10	35	80
12		68.0			90	45	0	45
	↑ PM →		N	NW	W	SW	S	SE

50 Deg North Latitude

Sun Time		Solar Altitude β Degrees	Azimuth Angle γ, Degrees					
AM → ↓			N	NE	E	SE	S	SW
5 a.m.	7 p.m.	4.5	67	22	23	68		
6	6	13.5	78	33	12	57		
7	5	23.5	90	45	0	45	90	
8	4	33 0	shade	57	12	33	78	
9	3	42.0		70	25	20	65	
10	2	50.0		87	42	3	48	shade
11	1	56.0		shade	64	19	26	71
12		58.0			90	45	0	45
	↑ PM →		N	NW	W	SW	S	SE

Table 8. Approximate Solar Declinations in Degrees

Date	Declination	Date	Declination	Date	Declination
April 1	4.5	June 1	22.0	Aug. 1	18.0
April 15	10.0	June 15	23.5	Aug. 15	14.0
May 1	15.0	July 1	23.0	Sept. 1	8.5
May 15	19.0	July 15	21.5	Sept. 15	3.0

is 90 + 14 = 104 deg, and at 7:00 p.m. is 90 + 24 = 114 deg. By interpolation, ϕ for 6:30 p.m. is 109 deg west of south (at 5:30 a.m. ϕ would be 109 deg east of south.)

Example 2: Find K for a wall facing 18 deg east of south at 10:00 a.m. on August 1 at 50 deg north latitude.

Solution: The wall azimuth is 18 deg. The solar azimuth is 48 deg east (Table 7). The wall solar azimuth is 48 − 18 or 30 deg. From Table 7, β is 50 deg. Then

$$K = \cos\beta \cos\gamma = \cos 50 \times \cos 30 = 0.643 \times 0.866 = 0.557.$$

Example 3: Find K for the wall in *Example 2* at 3:00 p.m.

Solution: The solar azimuth is 65 deg west. The wall solar azimuth is therefore 65 + 18 = 83 deg. The angle β is 42 deg.

$$K = \cos 42 \times \cos 83 = 0.743 \times 0.122 = 0.091.$$

Example 4: Find the total solar irradiation for the wall for the conditions of *Example 2*.

Solution: Use clear atmosphere solar intensities. At 50 deg altitude, the direct normal radiation is 273 Btu per (hr) (sq ft). Then,

$$I_D = K \times I_{Dn} = 0.557 \times 273 = 152.0 \text{ Btu per (hr) (sq ft)}.$$

By linear interpolation, the diffuse irradiation is

$$I_d = 25 + \tfrac{18}{90}(33 - 25) = 26.6 \text{ Btu per (hr) (sq ft)}.$$

The total solar irradiation is

$$I_t = 152.0 + 26.6 = 178.6 \text{ Btu per (hr)(sq ft)}.$$

PERIODIC HEAT FLOW THROUGH WALLS AND ROOFS

The calculation of heat flow, through a structural section of a building exposed to the weather, requires consideration of the diurnal cycles of solar irradiation and air temperature. These cycles and other factors lead to a periodic variation in the instantaneous rate of heat flow into the weather surface, and a related periodic variation in the rate of heat flow into the air conditioned space. Because of heat capacity and other factors, these heat flow cycles are, in general, out of time phase and unequal in amplitude.

In order to calculate the rate of heat entry into the weather surface of a building, it is necessary to know:

1. The intensity of direct solar radiation striking the surface.
2. The absorptivity (or reflectivity) of the surface for direct solar radiation.
3. The intensity of diffuse or sky solar radiation striking the surface.
4. The absorptivity (or reflectivity) of the surface for diffuse or sky solar radiation.
5. The rate at which the surface emits radiation to the sky and other surroundings.
6. The rate at which the surface absorbs the low temperature radiation emitted by the sky and other surroundings by virtue of their temperatures and radiating characteristics.
7. The temperature of the surrounding air.
8. The temperature of the outer building surface.
9. The unit convective conductance for heat transfer between the air and the building surface.

The Sol-Air Temperature

The complex interrelationship of the above factors can be considerably simplified through the use of the sol-air temperature concept. The sol-air temperature t_e is the temperature of the outdoor air, which, in the absence of all radiation exchanges, would give the same rate of heat entry into the surface as would exist with the actual combination of incident solar radiation, radiant energy exchange with the sky and other outdoor surroundings, and convective heat exchange with the outdoor air.

The sol-air temperature is developed as follows:

1. The basic heat balance equation which includes the factors listed, gives the rate of heat entry $\left(\frac{q}{A}\right)_L$ into the weather side of a sunlit building surface, and is written:

$$\left(\frac{q}{A}\right)_L = \alpha_D I_D + \alpha_d I_d + f_{oo}(t_o - t_L) + \epsilon_L R_s - \epsilon_L R_L, \text{ Btu per (hr)(sq ft)} \quad (2)$$

where

α = absorptivity (dimensionless) of weather side of wall or roof for incident solar radiation. Subscripts D and d refer respectively to direct and diffuse.

I = incident solar radiation, Btu per (hour)(square foot). Subscripts D and d refer respectively to direct and diffuse.

t_o = outside air temperature, Fahrenheit degrees.

t_L = temperature of weather surface of wall or roof, Fahrenheit degrees.

f_{oo} = unit convective conductance of weather surface, Btu per (hour)(square foot) (Fahrenheit degree).

R_s = low temperature radiant energy falling on surface from outdoor surroundings, Btu per (hour)(square foot of receiving surface).

ϵ_L = emissivity of surface at temperature t_L (also equals absorptivity for R_s), dimensionless.

R_L = low temperature radiant energy emitted by a black body at temperature t_L, Btu per (hour)(square foot).

2. The sol-air temperature is defined as:

$$t_e = t_o + \frac{\alpha_D I_D + \alpha_d I_d + \epsilon_L(R_s - R_L)}{f_{oo}} \quad (3)$$

3. The instantaneous rate of heat entry into the weather surface of the wall or roof becomes:

$$\left(\frac{q}{A}\right)_L = f_{oo}(t_e - t_L) \quad (4)$$

The term $\epsilon_L(R_s - R_L)$ is difficult to evaluate, since t_L, on which R_L depends, cannot be found until t_e and the thermal properties of the structure are known. However, t_L may be estimated from experimental observations of surface temperatures of walls and roofs which appear in the literature[8, 9] on periodic heat flow. R_L can then be found from Chapter 5, Table 5, and ϵ_L can be found from Table 3 of the same chapter. The term R_s represents the low temperature radiant energy from outdoor surroundings which falls on the surface in question (ϵ_L is the fraction absorbed). In the case of horizontal surfaces, all of this energy comes from the atmosphere of which water vapor is the principal radiating component. For horizontal surfaces, Brunt's[10] correlation of a long series of observations shows R_s to be dependent upon the dew-point temperature. For the commonly used design dew-point 67 F, his empirical equation

TABLE 9. DESIGN SOL-AIR TEMPERATURES FOR 40 NORTH LATITUDE AND 18 D DECLINATION, NORTH (AUGUST 1) FOR CLEAR AND INDUSTRIAL ATMOSPHERES

	Sol-Air Temperature, t_e, Fahrenheit Degrees										
Sun Time	Any[b] Surface	Clear Atmospheres					Industrial Atmospheres				
		Hor.	N	E	S	W	Hor.	N	E	S	W
$\frac{\alpha^a}{f_{cro}}$	0.0	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25
12	77	77	77	77	77	77	77	77	77	77	77
1 a.m.	76	76	76	76	76	76	76	76	76	76	76
2	76	76	76	76	76	76	76	76	76	76	76
3	75	75	75	75	75	75	75	75	75	75	75
4	74	74	74	74	74	74	74	74	74	74	74
5	74	74	75	76	74	74	74	75	75	74	74
6	74	85	85	112	76	76	82	80	95	77	76
7	75	102	83	132	79	78	95	81	113	80	79
8	77	119	81	137	86	81	110	82	120	86	82
9	80	136	85	134	99	85	123	85	120	96	86
10	83	149	88	124	110	88	135	88	115	105	90
11	87	160	92	111	119	92	145	92	108	113	95
12	90	165	96	96	124	96	150	95	98	117	98
1 p.m.	93	166	98	98	125	117	151	98	101	119	114
2	94	160	99	99	121	135	146	99	101	116	126
3	95	151	100	100	114	149	138	100	101	111	135
4	94	136	98	98	103	154	127	99	99	103	137
5	93	120	101	96	97	150	113	99	97	98	131
6	91	102	102	93	93	129	99	97	93	96	112
7	87	87	88	87	87	89	87	88	87	92	88
8	85	85	85	85	85	85	85	85	85	85	85
9	83	83	83	83	83	83	83	83	83	83	83
10	81	81	81	81	81	81	81	81	81	81	81
11	79	79	79	79	79	79	79	79	79	79	79
24 hr avg t_m	83.1	109.1	86.5	95.8	92.2	95.8	103.4	86.0	92.9	91.2	92.9

[a] α = surface absorptivity, dimensionless.
f_{cro} = unit surface conductance, radiation and convection combined, Btu per (hr) (sq ft) (F deg).
[b] Values in this column are magnitudes of t_o, the outdoor dry-bulb temperature.

gives R_s as 82 percent of the hemispherical radiation emitted by a *black* surface radiating at a temperature equal to that of the outdoor dry-bulb temperature. For vertical surfaces, R_s varies, since part of the energy is received from the ground, the temperature of which is influenced by solar radiation. Few data for vertical surfaces are available.

Because of the lack of data regarding R_s and the difficulty of evaluating $\epsilon_L(R_s - R_L)$, present practice in calculating sol-air temperatures is to compensate for the term by increasing f_∞. This is a rough approximation, since, during a considerable part of the night, outdoor surface temperatures are at or close to the ambient air temperatures, yet the radiation loss is of appreciable magnitude.

Example 5: If t_o = 90 F, α_t = 0.7, I_t = 200 Btu per (hr)(sq ft), f_∞ = 3.0, find the sol-air temperature for a roof when the dew-point temperature is 67 F.

Solution: Previous experience indicates that the roof temperature under these conditions will be about 120 F. ϵ_L equals 0.9. Using the tabular data of Table 4, Chapter 5, to determine R_s and R_L:

$$t_e = 90 + \frac{0.7\ (200) + 0.9\ (0.82 \times 159 - 196)}{3} = 117.0 \text{ F.}$$

Example 6: Find t_e if compensation for $\epsilon_L(R_s - R_L)$ is made by replacing f_∞ with f_{cro}, the surface conductance for radiation and convection combined, equal to 4.0.

Solution: $t_e = 90 + \dfrac{0.7(200)}{4} = 125.0$ F.

Based upon the solar data of Table 5, sol-air data have been computed for locations in clear and in humid industrial areas where the design dry-bulb temperature is 95 F. As given in Table 9, these are for 40 deg north latitude, 18 deg north declination (August 1). Radiation exchange has been included with the convection transfer; a value of 0.25 was used for α_t/f_{cro}. The data for industrial areas are practically identical with sol-air data for New York City[4] which were derived from an analysis of *U. S. Weather Bureau* records for a 10-year period. A similar analysis for Lincoln, Nebraska[5] leads to somewhat higher sol-air temperatures than the clear atmosphere values given in Table 9 because of the higher dry-bulb temperatures.

Corrections for other conditions indicated by ('), are made as follows:

1. To adjust the data in Table 9 for variations in t_o:
   a. Establish the value of t_o', the dry-bulb temperature, for the locality in question.
   b. Determine $t_o' - t_o$.
   c. Add (algebraically) $t_o' - t_o$ to the data tabulated.
2. To adjust the data of Table 9 for other values of α_t/f_{cro}: interpolate or extrapolate the tabulated data by direct proportion, using the column for $\alpha_t/f_{cro} = 0.0$.

Sol-air temperatures are especially helpful in the calculation of periodic heat transfer, as will be illustrated in the material which follows.

PRINCIPLES OF PERIODIC HEAT FLOW

Calculation principles for periodic heat flow are dealt with briefly in this section; in the section which follows, *practical tables are given to facilitate rapid design estimates.* In addition to the rate of heat entry into the outside building surface, these tables take into account the following factors:

1. The thermal conductivity of material.
2. The density, specific heat and character of material.
3. Thickness of material.
4. Room air temperature.
5. Unit convective conductance between the inside surface and room air; and radiant heat transfer between the inside surfaces and other surfaces in the room.

Time Lag

The fundamental analysis of periodic heat flow is complicated when compared with steady-state calculations on account of the time-variable storage of heat from point to point through a wall or roof. The cyclic variation of outdoor conditions produces a related cyclic variation of temperature and heat flow throughout each structural section exposed to the weather. The cyclic variations undergo a progressive *shift in phase* and *decrease in amplitude* in going through a wall with constant conditions maintained in the indoor space.

By a *shift in phase* is meant that as the cyclic temperature wave passes through the wall, the time of occurrence of the maximum temperature at any point shifts farther and farther behind the time of the outer-surface maximum for successive positions through the wall. The resultant *time lag* between the outer-surface and inner-surface maximum temperatures is important, for it may be the determining factor in fixing the time of the maximum cooling load.

By a *decrease in amplitude* is meant that as the cyclic temperature wave passes through the wall, the difference between the maximum temperature of a cycle and the mean temperature of the cycle, which is the amplitude of the wave by definition, decreases progressively as the wave passes through the wall. The magnitude of the *temperature amplitude* at the inner wall surface is necessary for the determination of the instantaneous rate of heat transfer to the indoor space.

Practical design data for periodic heat flow comprise a means of determining the time lag and amplitude decrement for different wall constructions, and any given outdoor cycle of sol-air temperature. Both analytical and experimental studies have been made on this problem.[11] While the analytical solution has been written, it is far too detailed for direct use in rapid practical work; and the extensive numerical work required to establish a basis for simplified calculations has been only partially completed. The method reported by Mackey and Wright[11] will be adopted as the basis for the design procedure recommended here.

Homogeneous Walls or Roofs, Constant Indoor Temperature

For walls or roofs of a single, homogeneous material, the *instantaneous* rate of heat gain within an enclosure where the indoor air temperature is held constant is, approximately,

$$\frac{q}{A} = U\,(t_m - t_i) + \lambda U\,(t_e^* - t_m) \text{ Btu per (hour) (square foot)} \tag{5}$$

where t_m = 24-hr average sol-air temperature for the particular value of $\frac{\alpha}{f_{cro}}$, Fahrenheit degrees.

λ = amplitude decrement factor, a variable that depends upon the thickness material, and orientation of the wall or roof; see Table 10 for values. The amplitude decrement factor λ as used in this chapter is equivalent to $\left(\frac{1.65 \times \lambda}{U}\right)$ as defined by Mackey and Wright,[11] and also used by Stewart.[12]

t_e^* = sol-air temperature at a time earlier than the time for which the heat gain is being found by an amount that is equal to the time lag of the wall or roof, Fahrenheit degrees; see Table 10 for values of time lag.

U = overall coefficient of heat transfer of the wall or roof, Btu per (hour) (square foot) (Fahrenheit degree),

$$U = \frac{1}{\frac{1}{f_{cri}} + \frac{1}{f_{cro}} + \frac{L}{k}} = \frac{1}{\frac{1}{1.65} + \frac{1}{4} + \frac{L}{k}} = \frac{1}{0.856 + \frac{L}{k}}$$

L = thickness of building material, inches.

k = thermal conductivity of building material, Btu per (hour) (square foot) (Fahrenheit degree per inch).

f_{cri} = unit indoor surface conductance (radiation and convection combined), Btu per (hour) (square foot) (Fahrenheit degree).

f_{cro} = unit outdoor surface conductance (radiation and convection combined), Btu per (hour) (square foot) (Fahrenheit degree).

The time at which the *maximum occurs in the rate of heat entry into the outside surface of walls or roofs* is taken as the time at which the peak point occurs in the sol-air temperature cycle (mean *sun time* is used in the sol-air cycles). The corresponding maximum rate of heat entry follows from Equation 5 with t_e^* being the maximum temperature of the sol-air cycle.

The time of *maximum heat gain to the room* is obtained by adding the time lag to the time of maximum sol-air temperature (from Table 9) for the particular wall or roof.

The magnitude of the second term in Equation 5 relative to the first term indicates the relative portion of the structural heat in-flow assignable to periodic heat flow. The periodic term is continually passing through a *cyclic variation* from zero to a positive maximum, to zero, to a negative maximum, to zero again and so on over each 24-hour cycle. Surfaces with

TABLE 10. PERIODIC HEAT FLOW DATA FOR HOMOGENEOUS WALLS OR ROOFS

Material	Thickness, In.	Over-all Coefficient, Btu per (hr) (sq ft) (°F) U[a]	Thermal Resistance of Solid Material (hr) (sq ft) (°F)/Btu $\frac{L}{k}$	Time Lag, Hr.	Factor, λ, in Equation 5: Horizontal and North	East	South	West
Stone	8	0.67	0.64	5.5	0.51	0.36	0.48	0.42
	12	0.55	0.96	8.0	0.28	0.19	0.26	0.22
	16	0.47	1.28	10.5	0.17	0.10	0.15	0.13
	24	0.36	1.92	15.5	0.06	0.03	0.05	0.04
Solid Concrete	2	0.98	0.17	1.1	0.93	0.87	0.92	0.89
	4	0.84	0.33	2.5	0.79	0.68	0.76	0.72
	6	0.74	0.50	3.8	0.61	0.46	0.58	0.51
	8	0.66	0.67	5.1	0.49	0.33	0.46	0.39
	12	0.54	1.00	7.8	0.29	0.17	0.26	0.22
	16	0.46	1.33	10.2	0.17	0.09	0.15	0.12
Common Brick	4	0.60	0.80	2.3	0.83	0.75	0.81	0.78
	8	0.41	1.60	5.5	0.51	0.39	0.49	0.44
	12	0.31	2.40	8.5	0.26	0.17	0.25	0.21
	16	0.25	3.20	12.0	0.13	0.08	0.12	0.10
Face Brick	4	0.77	0.44	2.4	0.81	0.70	0.78	0.74
Wood	½	0.68	0.62	0.17	1.0	1.0	1.0	1.0
	1	0.48	1.25	0.46	1.0	0.99	0.99	0.99
	2	0.30	2.50	1.3	0.98	0.91	0.96	0.94
Insulating Board	½	0.42	1.51	0.08	1.0	1.0	1.0	1.0
	1	0.26	3.03	0.23	1.0	1.0	1.0	1.0
	2	0.14	6.05	0.77	1.0	1.0	1.0	1.0
	4	0.08	12.1	2.7	0.83	0.74	0.81	0.76
	6	0.05	18.2	5.0	0.64	0.49	0.61	0.55

[a] Based upon an outdoor surface conductance of 4.0 and an indoor surface conductance of 1.65 Btu per (hour) (square foot) (Fahrenheit degree).

different exposures pass through these cycles with maximum points at different times of day.

An example in the use of Tables 9 and 10 follows:

Example 7: Find the instantaneous rate of heat gain through an 8 in. west wall of common brick (α_t = 0.7, f_{cro} = 4.0) located at 40 deg north latitude at 9:30 p.m. sun time. The indoor air temperature is constant at 80 F. Use sol-air data for an industrial atmosphere.

Solution: From Table 10, U = 0.41, the time lag is 5.5 hr, and λ = 0.44. By linear interpolation on the basis of α_t/f_{cro} in Table 9,

$$t_m = 83.1 + \frac{0.175}{0.25}\,(92.9 - 83.1) = 90.0\,\text{F}.$$

The design sol-air temperature at a time earlier than 9:30 p.m. by the time lag (at 4:00 p.m.) is, by interpolation, from Table 9,

$$t_e' = 94 + \frac{0.175}{0.25}\,(137 - 94) = 124.1\,\text{F}.$$

From Equation 5, the instantaneous design rate of heat gain is

$$\frac{q}{A} = 0.41[(90.0 - 80) + 0.44(124.1 - 90.0)] = 10.3 \text{ Btu per (hr) (sq ft).}$$

From Table 10, the time lag is 5.5 hr. From Table 9, the time of maximum rate of heat entry for a west wall is 4:00 p.m. plus 5.5 hr or 9:30 p.m. (*this is sun time*). The computed rate is therefore the maximum rate.

Composite Walls or Roofs, Constant Indoor Temperature

A composite wall or roof is made up of two or more layers of different materials. Since the analytical solution for this type of construction has not been reduced to simple and practical terms, it is necessary at present

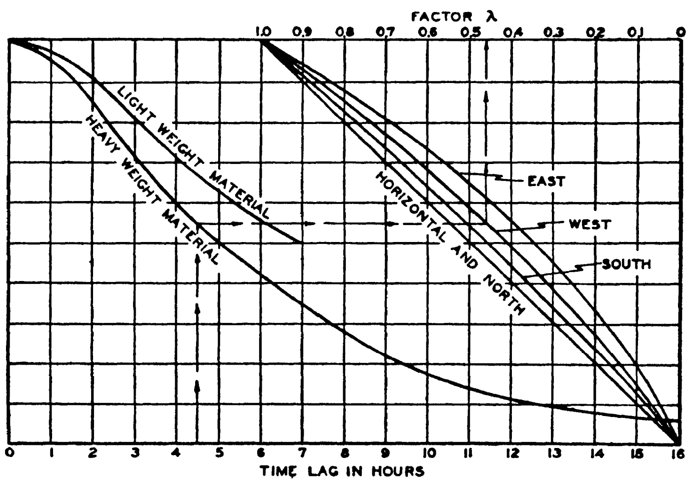


FIG. 2. APPROXIMATE VALUE OF THE AMPLITUDE DECREMENT FACTOR λ FOR USE IN EQUATION 5

to utilize approximate procedures. In accord with the results of comparative calculations, the following procedures are suggested.[11]

To find a time lag for a composite construction:

a. Find the time lag for each layer from Table 10.

b. Add the individual time lags, recognizing that this sum will always be *less* than the true time lag for the actual composite wall.

c. To the sum from (b), *add* an *arbitrary* additional lag of ½ to 1 hr to obtain the estimated lag for the actual construction. For two-layer and light construction walls, the ½-hr value will be suitable while, for walls of three or more layers, or very heavy construction, the 1-hr value is preferred. For intermediate conditions, individual judgment is the only guide. (Computed time lags should not be considered to be accurate closer than to about the nearest hour by this method.)

To find the amplitude decrement factor, λ, *for composite construction:*

Having determined the time lag and the orientation, use Fig. 2. (Note also that the factor λ for a composite construction should never exceed the *product* of the factors for the individual layers.)

One valuable result of the analytical studies made to date on composite walls has been the demonstration of the effect of the *order* of the materials. Other factors remaining the same, the use of the material of lower density on the weather side will increase the time lag and decrease the instantaneous maximum rate of heat gain.

Two examples are given to show how to estimate, roughly, the instantaneous rate of heat gain through sunlit composite walls or roofs.

Example 8: Estimate the maximum instantaneous design rate of heat gain from a horizontal roof in a location having an industrial type of atmosphere. The roof is made up of black, built-up roofing on the weather side (α = 1, f_{oro} = 4, thermal resistance = 0.28), 1 in. of insulating board, and 4 in. of concrete with no ceiling. The temperature of the indoor air is 80 F.

Solution: The overall coefficient of heat transfer for this construction is:

$$U = \frac{1}{0.25 + 0.28 + 3.03 + 0.33 + 0.61} = 0.22 \text{ Btu per (hour) (square foot) (F deg)}$$

If the time lag of the built-up roofing be ignored, the sum of the time lags of the individual layers is, from Table 10, (0.23 + 2.5) or 2.73 hr.

Actually, the time lag will be between 0.5 hr and 1.0 hr greater than this, so assume a time lag of 3.5 hr.

From Table 10, the homogeneous concrete roof having a time lag of 3.5 hr would have a value of λ of about 0.65; use this value for the composite roof.

From Fig. 2, λ is approximately 0.65.

With values of t_m and $t_e{}^*$ found from Table 9, as in previous examples, use Equation 5 and find the maximum *design instantaneous rate* of heat gain as:

$$\frac{q}{A} = 0.22[(103.4 - 80) + 0.65(151 - 103.4)] = 11.9 \text{ Btu per (hr) (sq ft).}$$

The maximum instantaneous rate of heat gain from this roof would occur at about 4:30 p.m., sun time.

Example 9: Estimate the maximum instantaneous design rate of heat gain on August 1 from a south wall in a location at 40 deg north latitude having a clear atmosphere. The wall consists of 4 in. of face brick (α_t = 0.7; f_{oro} = 4.0), 4 in. of common brick, furred, with an air space (thermal resistance = 0.75), and finished on the inside with ¾ in. of plaster on metal lath (thermal resistance = 0.23); the temperature of the indoor air is constant at 75 F.

Solution: The overall coefficient of heat transfer for this construction is:

$$U = \frac{1}{0.25 + 0.44 + 0.80 + 0.75 + 0.23 + 0.61} = 0.32 \text{ Btu per (hr) (sq ft) (F deg).}$$

From Table 10, the sum of the time lags for the face brick and the common brick is (2.4 + 2.3) or 4.7 hr. The actual time lag will be slightly greater than this, and a value of 5.5 hr will be assumed.

From Fig. 2, λ is approximately 0.45.

By interpolation in Table 9, t_m = 89.5 F, and $t_e{}^*$ = 115.4 F. From Equation 5 the maximum instantaneous design rate of heat gain is:

$$\frac{q}{A} = 0.32[(89.5 - 75) + 0.45(115.4 - 89.5)] = 8.4 \text{ Btu per (hr) (sq ft).}$$

The time of this heat gain is about 6:30 p.m., sun time.

Those concerned with a further study of the details of cooling-load estimates in particular relation to periodic heat flow will find much of value and interest in the reports of experimental studies of these problems.[8, 9, 11, 13, 14, 16]

PRACTICAL TABLES FOR CALCULATING SOLAR HEAT GAIN THROUGH WALLS AND ROOFS

Use of Equivalent Temperature Differentials

The preceding paragraphs have explained the principles and methods used in estimating solar heat gain by use of sol-air temperature. This method is rather tedious and is not convenient for every-day use. Some new practical tables[12] have therefore been developed using the basic method reported by Mackey and Wright.[11] These new tables utilize *equivalent temperature differentials* which may be multiplied by the overall heat transmission coefficient U to give directly the total heat transmission, Btu per square foot, from solar radiation and from temperature difference between outside and room air.

These tables were prepared from sol-air data, as shown in Table 11, which

TABLE 11. SUMMER DESIGN SOL-AIR TEMPERATURES USED FOR TABLES 12 AND 13

MEAN SUN TIME	SOL-AIR TEMPERATURE t_e FAHRENHEIT DEGREES								
	Any Surface[b]	Horiz.	North	East		South		West	
Ratio[a]: $\frac{\alpha}{f_{oo}}$	0	0.225	0	0.225	0.125	0.225	0.125	0.225	0.125
12 Midnight	77	77	77	77	77	77	77	77	77
1 AM	76	76	76	76	76	76	76	76	76
2	76	76	76	76	76	76	76	76	76
3	75	75	75	75	75	75	75	75	75
4	74	74	74	74	74	74	74	74	74
5	74	74	74	75	80	74	74	74	74
6	74	76	74	110	93	74	74	74	74
7	75	91	75	123	100	75	75	75	75
8	77	106	77	126	103	82	78	77	77
9	80	119	80	125	104	93	86	80	80
10	83	129	83	117	100	102	93	83	83
11	87	137	87	108	96	110	99	89	87
12 Noon	90	142	90	92	92	114	104	96	92
1 PM	93	144	93	93	93	115	105	110	102
2	94	140	94	95	94	111	104	124	111
3	95	132	95	95	95	104	100	135	119
4	94	120	94	94	94	99	96	141	120
5	93	107	93	93	93	95	94	139	118
6	91	96	91	91	91	91	91	125	111
7	87	90	87	87	87	88	87	103	94
8	85	85	85	85	85	85	85	85	85
9	83	83	83	83	83	83	83	83	83
10	81	81	81	81	81	81	81	81	81
11	79	79	79	79	79	79	79	79	79
24 Hr Avg t_m	83.1	100.5	83.1	93.0	88.4	89.0	86.2	93.0	88.4

[a] α = surface absorptivity, dimensionless: roof = 0.9; dark walls = 0.9, and light walls = 0.5. f_{oo} = unit convective conductance = 4.0 Btu per (hr) (F deg).
[b] values in this column are magnitudes of t_o, the outdoor air temperature.

are approximately the same as data for an industrial atmosphere in Table 9. It is suggested that Tables 12 and 13 be used for general estimating purpose.

These analytical procedures, as well as those using Tables 9 and 10 presented here, yield generally *higher* rates of heat gain than reported for Pittsburgh in early A.S.H.V.E. experimental studies. Current authoritative opinion indicates a preference for analytical calculations.

Table 12. Total Equivalent Temperature Differentials for Calculating Heat Gain Through Sunlit and Shaded Roofs

Description of Roof Construction[a]	Sun Time A.M. 8	10	12	P.M. 2	4	6	8	10	12
Light Construction Roofs—Exposed to Sun									
1″ Wood[b] or 1″ Wood[b] + 1″ or 2″ Insulation	12	38	54	62	50	26	10	4	0
Medium Construction Roofs—Exposed to Sun									
2″ Concrete or 2″ Concrete + 1″ or 2″ Insulation or 2″ Wood[b]	6	30	48	58	50	32	14	6	2
2″ Gypsum or 2″ Gypsum + 1″ Insulation 1″ Wood[b] or, 2″ Wood[b] or, 2″ Concrete or, 2″ Gypsum } + 4″ Rock Wool in Furred Ceiling	0	20	40	52	54	42	20	10	6
4″ Concrete or 4″ Concrete with 2″ Insulation	0	20	38	50	52	40	22	12	6
Heavy Construction Roofs—Exposed to Sun									
6″ Concrete	4	6	24	38	46	44	32	18	12
6″ Concrete + 2″ Insulation	6	6	20	34	42	44	34	20	14
Roofs Covered with Water—Exposed to Sun									
Light Construction Roof with 1″ Water	0	4	16	22	18	14	10	2	0
Heavy Construction Roof with 1″ Water	−2	−2	−4	10	14	16	14	10	6
Any Roof with 6″ Water	−2	0	0	6	10	10	8	4	0
Roofs with Roof Sprays—Exposed to Sun									
Light Construction	0	4	12	18	16	14	10	2	0
Heavy Construction	−2	−2	2	8	12	14	12	10	6
Roofs in Shade									
Light Construction	−4	0	6	12	14	12	8	2	0
Medium Construction	−4	−2	2	8	12	12	10	6	2
Heavy Construction	−2	−2	0	4	8	10	10	8	4

[a] Includes ⅜ in. felt roofing with or without slag. May also be used for shingle roof. [b] Nominal thickness of the wood.

NOTES FOR TABLE 12

Explanation: {Total heat transmission from solar radiation and temperature difference between outside and room air. Btu per (hr) (sq ft) of roof area} = {Equivalent temperature differential from above table} × {Heat transmission coefficient for summer Btu per (hr) (sq ft) (F deg)}

1. *Source.* Calculated by Mackey and Wright method (see reference list) and adjusted after studying ASHVE original test data. Estimated for July in 40 deg north latitude. (For sol-air temperatures used in calculations see Table 11.) For typical design day where the maximum outdoor temperature is 95 F and minimum temperature at night is approximately 75 F (daily range of temperature, 20 F) mean 24 hr temperature 84 F for a room temperature of 80 F. All roofs have been assumed a dark color which absorbs 90 percent of solar radiation, and reflects only 10 percent.

2. *Application.* These values may be used for all normal air conditioning estimates; usually without correction, in latitude 0 deg to 50 deg north or south when the load is calculated for the hottest weather. Note 5 explains how to adjust the temperature differential for other room and outdoor temperatures.

3. *Peaked Roofs.* If the roof is peaked and the heat gain is primarily due to solar radiation, use for the area of the roof, the area projected on a horizontal plane.

4. *Attics.* If the ceiling is insulated and if a fan is used in the attic for positive ventilation, the total temperature differential for a roof exposed to the sun may be decreased 25 percent.

5. *Corrections. For temperature difference when outdoor maximum design temperature minus room is different from 15 deg.* If the outdoor design temperature minus room temperature is different from the base of 15 deg, correct as follows: When the difference is greater (or less) than 15 deg add the excess to (or subtract the deficiency from) the above differentials.

For outdoor daily range of temperature other than 20 deg. If the daily range of temperature is less than 20 deg, add 1 deg for every 2 deg lower daily range; if the daily range is greater than 20 deg, substract 1 deg for every 2 deg higher daily range. For example, the daily range in Miami, Florida is 12 deg or 8 deg less than 20 deg, therefore, the correction is + 4 deg at all hours of the day.

Light Colors. Credit should not be taken for light colored roofs except where the permanence of the light color is established by experience, as in rural areas or where there is little smoke. When the exterior surface of roof exposed to the sun is a light color, such as white or aluminum (which absorb approximately 50 percent and reflect 50 percent of the solar radiation) add to the temperature differential for roof in shade 55 percent of the difference between the roof in sun and roof in shade. When the roof exposed to the sun is a medium color such as light grey, blue or green, or bright red, add 80 percent of this difference.

For solar transmission in latitudes other than 40 deg north, and in other months. The table values of temperature differentials will be approximately correct for a roof in the following months:

North Latitude		South Latitude	
Latitude (deg)	Months	Latitude (deg)	Months
0	All Months	0	All Months
10	All Months	10	All Months
20	All Months except Nov, Dec, Jan	20	All Months except May, June, July
30	Mar, Apr, May, June, July, Aug, Sept	30	Sept, Oct, Nov, Dec, Jan, Feb, Mar
40	April, May, June, July, Aug	40	Oct, Nov, Dec, Jan, Feb
50	May, June, July	50	Nov, Dec, Jan

For other months, the total temperature differential (t_x) may be approximated by the use of the following formula:

$$t_x = t_s + \frac{I_z}{I_y}(t_v - t_s)$$

where t_s = temperature differential for the same wall in shade for desired time of day; obtained from Table 12.

I_y = maximum solar transmission through glass, Btu per (hr) (sq ft) for flat skylight in July, 40 deg north latitude (*Note:* this is maximum value irrespective of time).

I_z = same as I_y except use the maximum value for flat skylight, for month, and latitude desired for t_x.

t_v = temperature differential for particular roof exposed to sun for the desired time of day from Table 12.

(*Note that this makes adjustment only for solar radiation and that there may be additional correction for outdoor temperature.*)

Tables 12 and 13 are based on an equation re-arranged from Equation 5 to read:

$$\frac{q}{A} = U[t_m + \lambda\ (t_e{}^* - t_m) - t_i] \tag{6}$$

Let $t_m + \lambda(t_e{}^* - t_m) = t_p$, a net equivalent outdoor temperature for combined periodic and mean heat flow. Magnitudes of t_p will vary cyclically with time. Then,

$$\frac{q}{A} = U\ (t_p - t_i) \tag{7}$$

which is a simple form analogous to the steady state equations of Chapters 5 and 9. The rate of heat flow is obtained by multiplying the overall heat transmission coefficient of the structure by the equivalent temperature differential obtained from the tables.

Tables 12 and 13 were developed by using an outside surface conductance of 4.0 and an inside film conductance of 1.65 Btu (hr) (sq ft) (F deg). A reduction was made in the temperature differentials for roofs amounting to some 20 percent of solar radiation as explained by Stewart.[12] This was to compensate for several factors, one of which is the radiant heat lost to the

Table 13. Total Equivalent Temperature Differentials for Calculating Heat Gain Through Sunlit and Shaded Walls

North Latitude Wall Facing	Sun Time A.M. 8 D	8 L	10 D	10 L	12 D	12 L	P.M. 2 D	2 L	4 D	4 L	6 D	6 L	8 D	8 L	10 D	10 L	12 D	12 L	South Latitude Wall Facing
	Exterior color of Wall—D = dark, L = light																		
Frame																			
NE	22	10	24	12	14	10	12	10	14	14	14	14	10	10	6	4	2	2	SE
E	30	14	36	18	32	16	12	12	14	14	14	14	10	10	6	6	2	2	E
SE	13	6	26	16	28	18	24	16	16	14	14	14	10	10	6	4	2	2	NE
S	−4	−4	4	0	22	12	30	20	26	20	16	14	10	10	6	6	2	2	N
SW	−4	−4	0	−2	6	4	26	22	40	28	42	28	24	20	6	4	2	2	NW
W	−4	−4	0	0	6	6	20	12	40	28	48	34	22	22	8	8	2	2	W
NW	−4	−4	0	−2	6	4	12	10	24	20	40	26	34	24	6	4	2	2	SW
N (Shade)	−4	−4	−2	−2	4	4	10	10	14	14	12	12	8	8	4	4	0	0	S (Shade)
4 In. Brick or Stone Veneer + Frame																			
NE	−2	−4	24	12	20	10	10	6	12	10	14	14	12	12	10	10	6	4	SE
E	2	0	30	14	31	17	14	14	12	12	14	14	12	12	10	8	6	6	E
SE	2	−2	20	10	28	16	26	16	18	14	14	14	12	12	10	8	6	6	NE
S	−4	−4	−2	−2	12	6	24	16	26	18	20	16	12	12	8	8	4	4	N
SW	0	−2	0	−2	2	2	12	8	32	22	36	26	34	24	10	8	6	6	NW
W	0	−2	0	0	4	2	10	8	26	18	40	28	42	28	16	14	6	6	W
NW	−4	−4	−2	−2	2	2	8	6	12	12	30	22	34	24	12	10	6	6	SW
N (Shade)	−4	−4	−2	−2	0	0	6	6	10	10	12	12	12	12	8	8	4	4	S (Shade)
8 In. Hollow Tile or 8 In. Cinder Block																			
NE	0	0	0	0	20	10	16	10	10	6	12	10	14	12	12	10	8	8	SE
E	4	2	12	4	24	12	26	14	20	12	12	10	14	12	14	10	10	8	E
SE	2	0	2	0	16	8	20	12	20	14	14	12	14	12	12	10	8	6	NE
S	0	0	0	0	2	0	12	6	24	14	26	16	20	14	12	10	8	6	N
SW	2	0	2	0	2	0	6	4	12	10	26	18	30	20	26	18	8	6	NW
W	4	2	4	2	4	2	6	4	10	8	18	14	30	22	32	22	18	14	W
NW	0	0	0	0	2	0	4	2	8	6	12	10	22	18	30	22	10	8	SW
N (Shade)	−2	−2	−2	−2	−2	−2	0	0	6	6	10	10	10	10	10	10	6	6	S (Shade)
8 In. Brick or 12 In. Hollow Tile or 12 In. Cinder Block																			
NE	2	2	2	2	10	2	16	8	14	8	10	6	10	8	10	10	10	8	SE
E	8	6	8	6	14	8	18	10	18	10	14	8	14	10	14	10	12	10	E
SE	8	4	6	4	6	4	14	10	18	12	16	12	12	10	12	10	12	10	NE
S	4	2	4	2	4	2	4	2	10	6	16	10	16	12	12	10	10	8	N
SW	8	4	6	4	6	4	8	4	10	6	12	8	20	12	24	16	20	14	NW
W	8	4	6	4	6	6	8	6	10	6	14	8	20	16	24	16	24	16	W
NW	2	2	2	2	2	2	4	2	6	4	8	6	10	8	16	14	18	14	SW
N (Shade)	0	0	0	0	0	0	0	0	2	2	6	6	8	8	8	8	6	6	S (Shade)
12 In. Brick																			
NE	8	6	8	6	8	4	8	4	10	4	12	6	12	6	10	6	10	6	SE
E	12	8	12	8	12	8	10	6	12	8	14	10	14	10	14	8	14	8	E
SE	10	6	10	6	10	6	10	6	10	6	12	8	14	10	14	10	12	8	NE
S	8	6	8	6	6	4	6	4	6	4	8	4	10	6	12	8	12	8	N
SW	10	6	10	6	10	6	10	6	10	6	10	8	10	8	12	8	14	10	NW
W	12	8	12	8	12	8	10	6	10	6	10	6	10	6	12	8	16	10	W
NW	8	6	8	6	8	4	8	4	8	4	8	4	8	6	10	6	10	6	SW
N (Shade)	4	4	2	2	2	2	2	2	2	2	2	2	2	2	4	4	6	6	S (Shade)
8 In. Concrete or Stone or 6 In. or 8 In. Concrete Block																			
NE	4	2	4	0	16	8	14	8	10	6	12	8	12	10	10	8	8	6	SE
E	6	4	14	8	24	12	24	12	18	10	14	10	14	10	12	10	10	8	E
SE	6	2	6	4	16	10	18	12	18	12	14	12	12	10	12	10	10	8	NE
S	2	1	2	1	4	1	12	6	16	12	18	12	14	12	10	8	8	6	N
SW	6	2	4	2	6	2	8	4	14	10	22	16	24	16	22	16	10	8	NW
W	6	4	6	4	6	4	8	6	12	8	20	14	28	18	26	18	14	10	W
NW	4	2	4	0	4	2	4	4	6	6	12	10	20	14	22	16	8	6	SW
N (Shade)	0	0	0	0	0	0	2	2	4	4	6	6	8	8	6	6	4	4	S (Shade)

TABLE 13. TOTAL EQUIVALENT TEMPERATURE DIFFERENTIALS FOR CALCULATING HEAT GAIN THROUGH SUNLIT AND SHADED WALLS—*Concluded.*

North Latitude Wall Facing	Sun Time																		South Latitude Wall Facing
	A.M.						P.M.												
	8		10		12		2		4		6		8		10		12		
	Exterior color of Wall—D = dark, L = light																		
	D	L	D	L	D	L	D	L	D	L	D	L	D	L	D	L	D	L	
	12 IN. CONCRETE OR STONE																		
NE	6	4	6	2	6	2	14	8	14	8	10	8	10	8	12	10	10	8	SE
E	10	6	8	6	10	6	18	10	18	12	16	10	12	10	14	10	14	10	E
SE	8	4	8	4	6	4	14	8	16	10	16	10	14	10	12	10	12	10	NE
S	6	4	4	2	4	2	4	2	10	6	14	10	16	12	14	10	10	8	N
SW	8	4	8	4	6	4	6	4	8	6	10	8	18	14	20	14	18	12	NW
W	10	6	8	6	8	6	10	6	10	6	12	8	16	10	24	14	22	14	W
NW	6	4	6	2	6	2	6	4	6	4	8	6	10	8	18	12	20	14	SW
N (Shade)	0	0	0	0	0	0	0	0	2	2	4	4	6	6	8	8	6	6	S (Shade)

NOTES FOR TABLE 13

Explanation: {Total heat transmission from solar radiation and temperature difference between outside and room air, Btu per (hr) (sq ft wall area)} = {Equivalent temperature differential from above table} × {Heat transmission coefficient for wall, Btu per (hr) (sq ft) (F deg)}

NOTES:

1. *SOURCE.* Same as Table 12. A north wall has been assumed to be a wall in the shade; this is practically true. Dark colors on exterior surface of walls have been assumed to absorb 90 percent of solar radiation and reflect 10 percent; white colors absorb 50 percent and reflect 50 percent. This includes some allowance for dust and dirt since clean, fresh white paint normally absorbs only 40 percent of solar radiation.

2. *APPLICATION.* These values may be used for all normal air conditioning estimates, usually without corrections, when the load is calculated for the hottest weather. Correction for latitude (Note 3) is necessary only where extreme accuracy is required. There may be jobs where the indoor room temperature is considerably above or below 80 F, or where the outdoor design temperature is considerably above 95 F, in which case it may be desirable to make correction to the temperature differentials shown. The solar intensity on all walls other than east and west varies considerably with time of year.

3. *CORRECTIONS. Outdoor minus room temperature.* If the outdoor maximum design temperature minus room temperature is different from the base of 15 deg, correct as follows: When the difference is greater (or less) than 15 deg, add the excess to (or subtract the deficiency from) the above differentials.

Outdoor daily range temperature. If the daily range of temperature is less than 20 deg, add 1 deg to every 2 deg lower daily range; if the daily range is greater than 20 deg, subtract 1 deg for every 2 deg higher daily range. For example, the daily range in Miami, Florida is 12 deg, or 8 deg less than 20 deg; therefore, the correction is +4 deg.

Color of exterior surface of wall. Use temperature differentials for light walls only where the permanence of the light wall is established by experience. For cream colors use the values for light walls. For medium colors interpolate half way between the dark and light values. Medium colors are medium blue, medium green, bright red, light brown, unpainted wood, natural color concrete, etc. Dark blue, red, brown, green, etc., are considered dark colors.

For latitudes other than 40 deg north, and in other months. These table values will be approximately correct for the east or west wall in any latitude (0 deg to 50 deg North or South) during the hottest weather. In the lower latitudes when the maximum solar altitude is approximately 80 deg to 90 deg (the maximum occurs at noon) the temperature differential for either a south or north wall will be approximately the same as a north, or shade wall. The temperature differential (t_x) for any wall facing, and for any latitude for any month may be approximated as follows:

$$t_x = t_s + \frac{I_2}{I_1} \times (t_w - t_s)$$

where t_s = temperature differential for the same wall in shade for desired time of day; obtained from Table 13.

I_1 = maximum solar radiation intensity transmitted through glass, Btu per (hr) (sq ft) for particular wall facing, in July, 40 deg north latitude (*note:* this is maximum value irrespective of time).

I_2 = same as I_1 except use the maximum value for wall facing, for month, and latitude desired for t_x.

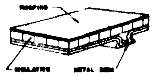
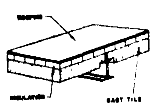
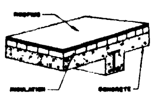
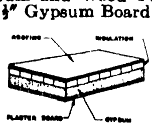
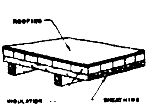
t_w = temperature differential for particular wall facing, for the desired time of day from above table.

(*Note that this makes adjustment only for solar radiation, and that there may be additional correction for outdoor temperature.*)

4. *FOR INSULATED WALLS* use same temperature differentials as used for uninsulated walls.

Table 14. Summer Coefficients of Heat Transmission U of Flat Roofs Covered With Built-Up Roofing[a]

Btu per (hour) (square foot) (F deg difference between the air on the two sides)

Type of Roof Deck Ceiling not shown	Thickness of Roof Deck (Inches)		Insulation on Top of Deck (Covered With Built-Up Roofing) — No Ceiling—Underside of Roof Exposed					Furred Ceiling with Air Space, Metal Lath and Plaster				
			No Insulation	Insulating Board[d] Thickness, In. ½	1	1½	2	No Insulation	Insulating Board[d] Thickness, In. ½	1	1½	2
Flat Metal Roof Deck 	4 Ply Felt Roof		0.73	0.35	0.23	0.17	0.13	0.40	0.25	0.18	0.14	0.12
	Ditto + ½ in. Slag		0.54	0.30	0.20	0.16	0.13	0.34	0.22	0.16	0.13	0.11
Precast Cement Tile 	4 Ply Felt Roof	1⅝	0.67	0.33	0.22	0.17	0.13	0.38	0.24	0.18	0.14	0.12
	Ditto + ½ in. Slag	1⅝	0.50	0.28	0.20	0.15	0.12	0.32	0.21	0.17	0.13	0.11
Concrete 	4 Ply Felt Roof	2	0.65	0.33	0.22	0.16	0.13	0.37	0.24	0.18	0.14	0.12
		4	0.59	0.31	0.21	0.16	0.13	0.36	0.23	0.17	0.13	0.12
		6	0.54	0.30	0.20	0.16	0.13	0.33	0.22	0.17	0.13	0.11
	Ditto + ½ in. Slag	2	0.49	0.28	0.20	0.15	0.12	0.31	0.21	0.16	0.13	0.11
		4	0.46	0.27	0.19	0.15	0.12	0.30	0.21	0.16	0.13	0.11
		6	0.42	0.26	0.19	0.14	0.12	0.29	0.20	0.16	0.13	0.10
Gypsum and Wood Fiber[b] on ½″ Gypsum Board 	4 Ply Felt Roof	2½	0.34	0.23	0.17	0.13	0.12	0.25	0.18	0.14	0.12	0.097
		3½	0.28	0.20	0.15	0.12	0.11	0.21	0.16	0.13	0.11	0.094
	Ditto + ½ in. Slag	2½	0.29	0.20	0.16	0.13	0.11	0.22	0.16	0.13	0.11	0.093
		3½	0.25	0.18	0.14	0.12	0.10	0.19	0.15	0.13	0.10	0.090
Wood[c] 	4 Ply Felt Roof	1	0.43	0.26	0.19	0.15	0.12	0.29	0.20	0.15	0.13	0.11
		1½	0.33	0.22	0.17	0.13	0.11	0.24	0.18	0.14	0.12	0.097
		2	0.29	0.20	0.16	0.13	0.11	0.22	0.16	0.13	0.11	0.094
		3	0.22	0.16	0.13	0.11	0.09	0.17	0.13	0.12	0.10	0.085
	Ditto + ½ in. Slag	1	0.35	0.23	0.17	0.14	0.11	0.25	0.18	0.14	0.12	0.10
		1½	0.29	0.20	0.15	0.12	0.10	0.21	0.17	0.13	0.11	0.093
		2	0.26	0.19	0.14	0.12	0.10	0.20	0.15	0.13	0.10	0.090
		3	0.20	0.15	0.12	0.10	0.09	0.16	0.13	0.11	0.09	0.081

[a] The summer coefficients are considered temporary, and have been calculated with an outdoor wind velocity of 8 mph. For summer an inside surface conductance of 1.2 has been used instead of the regular 1.65 value. In all of these roofs a 4 ply felt roof has been assumed ⅜ in. thick, thermal conductivity = 1.33. Pitch and slag have been assumed as an additional thickness of ½ in. which has been assigned thermal conductivity = 1.0. In both cases thermal conductivity refers to one *inch* thickness.

[b] 87½ percent gypsum, 12½ percent wood fiber. Thickness indicated includes ½ in. gypsum board. This is a poured roof.

[c] Nominal thickness of wood is specified, but actual thickness was used in calculations.

[d] If corkboard insulation is used, the coefficient U may be decreased 10 percent.

sky which is not included in the Mackey and Wright method. The *temperature differentials* for roofs were based on an inside surface conductance of 1.65 because the charts prepared by Mackey and Wright[11] used this value, and it was not considered practicable to repeat their work using a different film coefficient. An examination of the values given in their paper indicates that the *temperature differential* would be changed very little even if a value 1.20 were used instead of 1.65. But to obtain the heat flow rates through roofs, more accurate values will be obtained if the *overall heat transmission coefficient is calculated using* 1.2 as the inside film conductance of heat transfer in summer.

The roof coefficients of transmission for summer shown in Table 14 are based on surface conductances f_{cro} of 4.0 for an outside roof surface and 1.20 for an inside ceiling surface. The outside conductance 4.0 is used for summer because it corresponds to a wind velocity of approximately 7.5 mph averaged for rough and smooth surfaces, and is more representative of summer wind velocities. Also, the lower wind velocity should be used in order to be on the safe side in determining the sol-air temperature. The inside conductance 1.20 is used because the convective portion of the film conductance factor of downward heat flow from a horizontal surface is appreciably less than the winter conductance, which applies when heat is flowing upward.

Since there is little difference in *wall transmission coefficients* for summer, based on the conductances of 4.0 and 1.65, and the winter coefficients, based on 6.0 and 1.65, it is recommended that the overall coefficient *U*, *for walls*, be taken directly from the tables in Chapter 9 in which they are based on an outside film conductance of 6.0, corresponding to a 15 mph wind velocity.

Advantages of Equivalent Temperature Differential Method

The advantages of the *equivalent* temperature differential method of determining the total heat transmission are given in following paragraphs, and are apparent from *Examples 10* to *12*.

1. The total sensible heat flow is obtained by multiplying the overall heat transmission coefficient, *U*, and the equivalent temperature differential indicated in Tables 12 and 13.

2. The temperature differentials listed for a few representative types of construction may be used on all classes of walls and roofs, even though the overall heat transmission coefficient is different, provided the structure has thermal and physical properties similar to one of those listed in Tables 12 and 13.

3. Adjustments can be made, according to instructions given in the footnotes, for room and outdoor conditions different from those on which the tables are based.

Examples of Use of Equivalent Temperature Tables

Example 10: Given: A roof is constructed of 6 in. of stone concrete with 2 in. of insulating board and tar felt roofing $\frac{3}{8}$ in. thick, and is exposed to the sun. The location is the central part of the United States. Find the rate of heat flow into building at 2:00 p.m. during July for an outdoor design temperature 95 F, and an inside temperature 80 F.

Solution: From Table 12 in 2 p.m. column for 6 in. concrete plus 2 in. insulation, find the total equivalent temperature differential 34 deg. The overall heat transmission coefficient for summer is taken from Table 14 and is found to be 0.13. The heat flow rate equals $34 \times 0.13 = 4.42$ Btu per (hr) (sq ft).

Example 11: For the conditions of Example 10, find the rate of heat flow into building at 2:00 p.m. during July for design temperatures of 105 F (outdoor) and 78 F (indoor). Daily range of temperature 30 deg, *i.e.*, outdoor temperature minimum of 75 F which occurs at 4:00 or 5:00 a.m.; this being 30 deg less than the maximum.

Solution: Make correction in equivalent temperature differential in accordance with Note 5 in Table 12 as follows:

The correction for 27 deg design temperature difference is $(27 - 15) = +12$.

The correction for 30 deg daily range is $\left(-\frac{30-20}{2}\right) = -5.$

Net total correction is $+12 - 5 = +7$.

The heat flow rate at 2:00 p.m. therefore is $(34 + 7) \times 0.13 = 5.32$ Btu per (hr) (sq ft).

A method of determining *heat flow rates*, when structure is not given in Tables 12 or 13, is illustrated in *Example 12*.

Example 12: A 4 in. stone concrete roof covered with an average depth of 4 in. cinder concrete ($k = 4.9$) on which is placed a $\frac{3}{8}$ in. thick felt roof with $\frac{1}{2}$ in. pitch and slag surface, is exposed to the sun. The location is the central part of the United States. Design temperatures are: outdoor 95 F; daily range 20 deg; indoor temperature 80 F. Find the heat flow rate at 2:00 p.m. for a day in July.

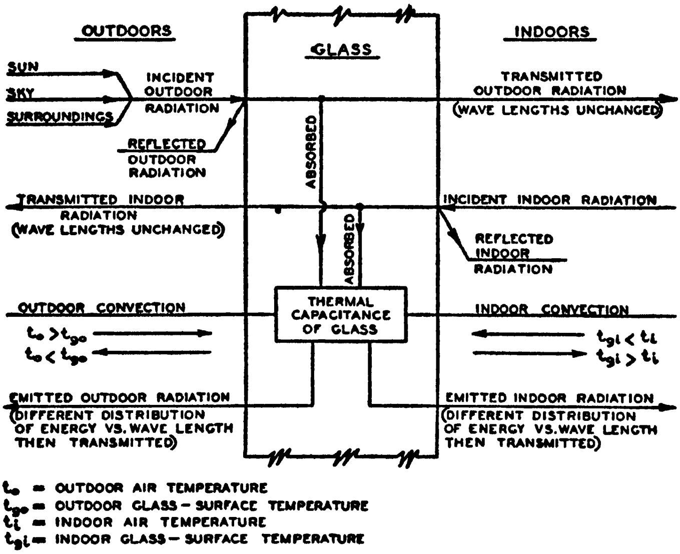


FIG. 3. INSTANTANEOUS HEAT-BALANCE CONDITIONS ON A GLASS SECTION

Solution: For the purpose of selecting the *equivalent temperature differential*, this construction is assumed to be equal approximately to an uninsulated 6 in. concrete roof, for which the equivalent temperature is found to be 38 deg in the 2:00 p.m. column of Table 12. Calculate the overall heat transmission coefficient U of the roof as follows:

$$U = \frac{1}{\frac{1}{1.2} + \frac{4}{12} + \frac{4}{4.9} + \frac{0.375}{1.33} + \frac{0.50}{1.00} + \frac{1}{4.0}} = 0.33.$$

The heat flow rate is then 38×0.33 equals 12.5 Btu per (hr) (sq ft).

TABLES FOR CALCULATING SOLAR HEAT GAIN THROUGH GLASS AREAS

Basic Principles

In order to set forth the principles involved in calculating heat flow through glass areas, the general instantaneous heat-balance relation will be presented. It will be shown schematically in Fig. 3. The net heat gain for the indoor space is the result of several contributing phenomena. Some observations concerning the behavior of glass with respect to radiant energy will lead to a better understanding of the heat-balance relation. To various degrees glass transmits radiation having wave lengths between 0.29 and 4.75 microns. Of the portion not transmitted, part is absorbed, and the remainder is reflected. Outside these limits glass is opaque, absorbing approximately 94 percent and reflecting 6 percent. Only a negligible amount of radiant energy from a surface at 450 F has a wave length shorter than 4.75 microns. It is therefore convenient to treat all forms of solar radiant energy separately from radiant energy from other sources, so long as the temperature of these sources is not over approximately 450 F.

The complete heat-balance for a glass section can be expressed for a unit time interval as follows:

$$\begin{bmatrix}\text{Total heat flow,}\\ \text{through glass section}\end{bmatrix} = \begin{bmatrix}\text{Transmitted,}\\ \text{solar radiation}\end{bmatrix} \pm \begin{bmatrix}\text{Heat flow by convective}\\ \text{and radiative exchanges at}\\ \text{the indoor surface}\end{bmatrix} \quad (7a)$$

The second term of the right side of Equation 7a can also be expressed by a heat balance equation as follows:

$$\begin{bmatrix}\text{Heat flow by convective}\\ \text{and radiative exchanges}\\ \text{at the indoor surface}\end{bmatrix} = \begin{bmatrix}\text{Absorbed}\\ \text{solar}\\ \text{radiation}\end{bmatrix} \pm \begin{bmatrix}\text{Radiative exchanges be-}\\ \text{tween outer surface of glass}\\ \text{and outdoor surroundings}\end{bmatrix}$$
$$\pm \begin{bmatrix}\text{Convective exchanges}\\ \text{between outer surface of}\\ \text{glass and outdoor air}\end{bmatrix} \pm \begin{bmatrix}\text{Heat storage}\\ \text{within the}\\ \text{glass section}\end{bmatrix} \quad (7b)$$

Equations 7a and 7b can be combined and expressed in symbolic terms by Equation 7c. Tabular values of the two bracketed terms of Equation 7a are presented later in this section for various types of glass for specific design conditions.

$$(q/A) = [\tau_D I_D + \tau_d I_d] + [\alpha_D I_D + \alpha_d I_d + \epsilon_{go} R_s - \epsilon_{go} R_{go} - f_{co}(t_{go} - t_o) - S],$$
$$\text{Btu per (hr) (sq ft)} \quad (7c)$$

where

(q/A) = instantaneous rate of heat flow, Btu per (hour)(square foot).
τ_D, τ_d = transmittance of glass for direct and diffuse solar radiation, respectively.
I_D, I_d = incident direct and diffuse solar radiation, respectively, Btu per (hour) (square foot).
α_D, α_d = absorptance of glass for direct and diffuse solar radiation, respectively.
ϵ_{go} = emissivity of glass at temperature t_{go}.
R_s = low temperature radiant energy falling on glass from outdoor surroundings, Btu per (hour)(square foot).
R_{go} = low temperature radiant energy emitted by a surface with emissivity equal to 1.0 at temperature t_{go}.
f_{co} = outdoor convective conductance, Btu per (hour)(square foot) (Fahrenheit degree).
t_{go} = temperature of outdoor surface of glass, Fahrenheit degrees.
t_o = temperature of outdoor air, Fahrenheit degrees.
S = rate at which glass stores energy, Btu per (hour)(square foot).

Transmissivity and absorptivity vary with both wave length of the incident radiation and incident angle. Values of τ and α for a single sheet of the average ordinary drawn window glass are given in Table 15 for a standard distribution of solar energy.[2] Values for two air-spaced sheets are also given. Normal incidence transmittance values for some commonly-used types and combinations are given in Table 18. Some variation in these values can be expected in practice due to variations in manufacture and in solar energy distribution. However, a change in transmissivity causes an approximately equal and opposite change in absorptivity. Hence, the *total heat flow* is not greatly altered. Transmittance data for other types of glass and various patterns of 8-in. glass block are given in A.S.H.V.E. research papers.[16, 17, 18, 19, 20]

As stated earlier in this chapter, present data as to the value of R_s are inadequate, so for the present it is suggested that f_{co} be increased to include radiation, and the term $\epsilon_{go}R_s - \epsilon_{go}R_{go}$ be disregarded. It is not practicable to give values of S in this chapter. However, for ordinary glass, the value of S is small.

Fig. 4 is a graphical solution, for single glass, of Equation 7b. Only

TABLE 15. TRANSMITTANCES AND ABSORPTANCES OF COMMON WINDOW GLASS FOR DIRECT AND DIFFUSE SOLAR RADIATION

Angle of Incidence, θ, deg	Single Sheets		Two Air-Spaced Sheets		
	τ_D	α_D	τ_D	α_D Outdoor Sheet	α_D Indoor Sheet
	For Direct Radiation				
0	0.87	0.05	0.76	0.06	0.04
20	0.87	0.05	0.76	0.06	0.04
40	0.86	0.06	0.74	0.06	0.04
50	0.84	0.06	0.72	0.07	0.05
60	0.79	0.06	0.66	0.07	0.05
70	0.67	0.06	0.52	0.07	0.05
80	0.42	0.06	0.25	0.07	0.05
90	0.0	0.0	0.0	0.0	0.05
	For Diffuse or Sky Solar Radiation				
	0.79	0.06	0.68	0.07	0.05

absorbed solar radiation is considered, although low temperature radiation exchange and heat storage can be added algebraically to $\alpha_t I_t$ if such data are available. The small thermal resistance of the glass has been neglected. The heat flow rates are for a 75 F indoor temperature, an indoor surface conductance for convection f_{ci} as given by Equation 8, and an equivalent surface conductance for radiation f_{ri} as given by Equation 9. Indoor surfaces seen by the glass are assumed to radiate as a black body at room air temperature.

$$f_{ci} = 0.27\ (t_{gi} - t_i)^{0.25} \tag{8}$$

$$f_{ri} = 0.162 \left[\left(\frac{t_{gi} + 460}{100}\right)^4 - \left(\frac{t_i + 460}{100}\right)^4\right] \Big/ (t_{gi} - t_i) \tag{9}$$

where

t_{gi} = temperature of indoor surface of glass, Fahrenheit degrees.
t_i = temperature of indoor air, Fahrenheit degrees.

A more complete treatment of the problem is given in an A.S.H.V.E. research paper.[19]

Example 13: Find the total heat gain at 10 a.m. sun time for a single unshaded sheet of common window glass in a wall facing 18 deg east of south on August 1 at

50 deg north latitude. The indoor temperature is 75 F, the outdoor temperature is 83 F. Use clear atmosphere radiation values and $f_{cro} = 4.0$.

Solution: From *Example* 2, K is 0.557; hence, the angle of incidence, θ, is 56 deg 9 min. From *Example* 4, $I_D = 152.0$, $I_d = 26.6$. By interpolation in Table 15, τ_D is found to be 0.81, α_D is 0.06; τ_d and α_d are 0.79 and 0.06, respectively.

The heat gain due to transmitted solar radiation is

$$(q/A)_\tau = 152.0 \times 0.81 + 26.6 \times 0.79 = 144.1 \text{ Btu per (hr)(sq ft).}$$

The heat gain by convection and radiation from the indoor surface is found from Fig. 4:

$$t_o + \frac{\alpha_t I_t}{f_{co}} = 83 + \frac{0.06\ (152.0 + 26.6)}{4} = 85.7 \text{ F}$$

from which

$$(q/A)_{cr} = 11.5 \text{ Btu per (hr)(sq ft).}$$

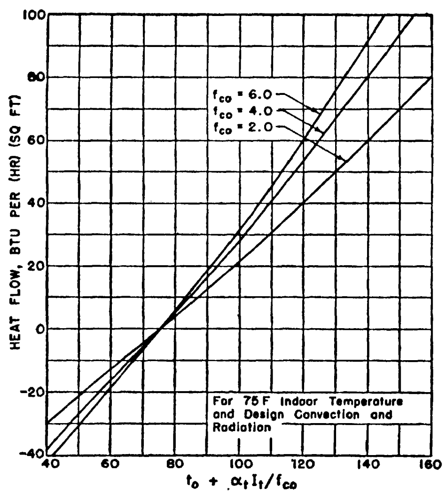


FIG. 4. CONVECTION AND RADIATION HEAT FLOW FOR VERTICAL SINGLE GLASS

From Equation 7a the total heat flow is

$$(q/A) = 144.1 + 11.5 = 155.6 \text{ Btu per (hr)(sq ft).}$$

Design Tables for Flat Glass

Tables 16 and 17 give design values of instantaneous rates of heat gain for *single unshaded common window glass* for a solar declination of 18 deg. This corresponds to a nominal August 1 day. The tables are based upon the solar intensity values for a clear atmosphere as given in Table 5. Table 16 represents the first bracketed term of Equation 7a; therefore, the values are dependent only upon values of I and τ. Table 17 is the second term of Equation 7a, and is based upon a 75 F indoor temperature and a dry-bulb temperature cycle, with a 95 F maximum as tabulated. *The total heat gain is the sum* of the Table 16 and Table 17 values. In preparing Table 17, convection and radiation heat exchange were combined, and a combined surface conductance of 4.0 used. Corrections to be applied for

Table 16. Instantaneous Rates of Heat Gain Due to Transmitted Direct and Diffuse or Sky Solar Radiation by a Single Sheet of Unshaded Common Window Glass

For Clear Atmospheres and 18 Deg Declination, North (August 1)

Note: For total instantaneous heat gain, add these values to the Table 17 values.

Latitude	Sun Time		Instantaneous Heat Gain in Btu per (hr) (sq ft)								
	AM → ↓		N	NE	E	SE	S	SW	W	NW	Horiz.
30 Deg North	6 a.m.	6 p.m.	25	98	108	52	5	5	5	5	17
	7	5	23	155	190	110	10	10	10	10	71
	8	4	16	148	205	136	14	13	13	13	137
	9	3	16	106	180	136	21	15	15	15	195
	10	2	17	54	128	116	34	17	16	16	241
	11	1	18	20	59	78	45	19	18	18	267
	12		18	19	19	35	49	35	19	19	276
40 Deg North	5 a.m.	7 p.m.	3	7	6	2	0	0	0	0	1
	6	6	26	116	131	67	7	6	6	6	25
	7	5	16	149	195	124	11	10	10	10	77
	8	4	14	129	205	156	18	12	12	12	137
	9	3	15	79	180	162	42	14	14	14	188
	10	2	16	31	127	148	69	16	16	16	229
	11	1	17	18	58	113	90	23	17	17	252
	12		17	17	19	64	98	64	19	17	259
50 Deg North	5 a.m.	7 p.m.	20	54	54	20	3	3	3	3	6
	6	6	25	128	149	81	8	7	7	7	34
	7	5	12	139	197	136	12	10	10	10	80
	8	4	13	107	202	171	32	12	12	12	129
	9	3	14	54	176	183	72	14	14	14	173
	10	2	15	18	124	174	110	16	15	15	206
	11	1	16	16	57	143	136	42	16	16	227
	12		16	16	18	96	144	96	18	16	234
		↑ PM →	N	NW	W	SW	S	SE	E	NE	Horiz.

other design temperatures are given in Table 26 in a later section, Effect of Deviation from Design Conditions.

Tables 16 and 17 may be used for other types of glass with good accuracy, by using the factors given in Table 18. Table 16 values are multiplied by the appropriate factor given in Table 18 to obtain heat gain due to transmitted solar radiation. For glasses having a transmittance for normally incident radiation differing from the table values, factors may be found by linear interpolation. To obtain instantaneous rates of heat gain by convection and radiation, two steps are required. First, Table 17 values are multiplied by the appropriate coefficient of X listed in Table 18. Second, Table 19 values are multiplied by the appropriate coefficient of Y listed in Table 18, and added to the first value. All convection and radiation gain values for double glass were computed for a ¼-in. air space. No great error is involved in cooling load estimates if these are used for double glass with other air spaces.

Example 14: **Find the total instantaneous heat gain through a single sheet of regular plate glass in a southwest wall at 2 p.m. sun time and 40 deg north latitude on August 1. The maximum dry-bulb temperature for design is 98 F; the atmosphere is clear. The indoor temperature is 75 F.**

Solution: From Table 16 the heat gain due to transmitted radiation is 148 Btu per (hr) (sq ft) for common window glass; from Table 18, the factor for regular plate glass is 0.87. The coefficient of X in Table 18 is 1.0, while X is found from Table 17 for common window glass for the same hour, orientation and latitude. The coefficient of Y in Table 18 is 0.25, while the Y value is found from Table 19 for a south-

TABLE 17. INSTANTANEOUS RATES OF HEAT GAIN BY CONVECTION AND RADIATION FROM A SINGLE SHEET OF UNSHADED COMMON WINDOW GLASS

For Clear Atmospheres and 18 Deg Declination, North (August 1)

For 75 F Indoor Temperature

Note: For total instantaneous heat gain, add these values to the Table 16 values.

Sun Time	Dry-Bulb Deg F	North Latitude Degrees	Instantaneous Heat Gain in Btu per (hr) (sq ft)								
			N	NE	E	SE	S	SW	W	NW	Hor.
5 a.m.	74		−1	−1	−1	−1	−1	−1	−1	−1	−1
6	74		0	1	1	0	−1	−1	−1	−1	0
7	75		0	3	3	2	0	0	0	0	2
8	77		2	5	6	5	3	2	2	2	5
9	80		5	7	9	8	6	5	5	5	8
10	83		8	9	11	11	10	8	8	8	13
11	87		13	13	15	16	15	14	13	13	18
12	90	30, 40, 50	17	17	17	18	19	18	17	17	21
1 p.m.	93		20	20	20	21	22	22	22	20	25
2	94		21	21	21	21	23	24	24	22	26
3	95		22	22	22	22	24	26	26	24	26
4	94		21	21	21	21	22	25	25	24	24
5	93		20	20	20	20	20	23	24	23	22
6	91		18	18	18	18	18	19	20	20	18
7	87		13	13	13	13	13	13	13	13	13
8	85		11	11	11	11	11	11	11	11	11
9	83		8	8	8	8	8	8	8	8	8

west wall at 2:00 p.m. and 40 deg north latitude. The correction for design dry-bulb temperature is found from Table 26 to be 1.0 Btu per (hr) (sq ft) per degree difference from 95 F design temperature. The total instantaneous heat gain is, from Equation 7a,

$$
\begin{aligned}
q &= 0.87 \times 148 + 1.0 \times 24 + 0.25 \times 27 + 1.0\ (98 - 95) \\
&= 162.7 \text{ Btu per (hr) (sq ft)}.
\end{aligned}
$$

Design Tables for Rolled Figured Glass

Tables 20 and 21 give design values of instantaneous rates of heat gain for a number of common patterns of single vertical sheets of rolled figured glass. The tables are for a solar declination of 18 deg, which corresponds to a nominal August 1 day, and are based upon the solar intensity values for a clear atmosphere as given in Table 5. The values are given in terms of corrections to apply to Tables 16 and 17. The heat gain due to transmitted solar radiation is found by multiplying the Table 16 values by the approximate percentages given in Table 21. To obtain instantaneous rates of heat gain by convection and radiation, Table 19 values are multiplied by the appropriate value of Y from Table 20 and then added to

TABLE 18. APPLICATION FACTORS TO APPLY TO TABLES 16, 17 and 19 to OBTAIN INSTANTANEOUS RATES OF HEAT GAIN FOR VARIOUS TYPES OF SINGLE FLAT GLASS AND COMBINATIONS OF TWO SHEETS OF FLAT GLASS SPACED AT ½ IN.

Glass	Normal Incidence Transmittance	Factor to Apply to Table 16	Factor to Apply to Table 17
Single Common Window	0.87	1.00	$1.0(X)^{b} + 0.0(Y)^{c}$
Single Regular Plate	0.77	0.87	$1.0(X) + 0.25(Y)$
Single Heat Absorbing Plate	0.41	0.46[a]	$1.0(X) + 1.00(Y)$
Double Common Window	0.76	0.85	$0.6(X) + 0.10(Y)$
Double Regular Plate	0.60	0.66[a]	$0.6(X) + 0.55(Y)$
Heat Absorbing Plate Outdoors / Regular Plate Indoors	0.35	0.37[a]	$0.6(X) + 0.75(Y)$

[a] For better precision, increase factors 10 percent when glass is in the shade.
[b] X values are Table 17 values. [c] Y values are Table 19 values.

TABLE 19. VALUES OF Y TO BE USED WITH FACTORS IN TABLE 18 AND TABLE 20 IN THE DETERMINATION OF INSTANTANEOUS RATES OF HEAT GAIN DUE TO CONVECTION AND RADIATION FOR VARIOUS TYPES OF SINGLE GLASS AND COMBINATIONS OF TWO SHEETS OF GLASS SPACED AT ½ IN.

For Clear Atmospheres and 18 Deg Declination, North (August 1)

Sun Time		Values of Y in Btu per (hr) (sq ft)[a]								
		N	NE	E	SE	S	SW	W	NW	Horiz.
5 a.m		0	0	1	0	0	0	0	0	0
6		4	16	18	9	1	1	1	1	3
7		2	24	30	20	2	2	2	2	11
8		2	22	33	25	2	2	2	2	21
9		2	16	30	29	8	3	3	3	32
10	40 Degrees North Latitude	3	5	25	27	14	3	3	3	37
11		3	3	12	21	18	3	3	3	42
12		3	3	3	15	19	12	3	3	45
1 p.m.		3	3	3	3	19	22	10	3	44
2		3	3	3	3	16	27	24	4	41
3		3	3	3	3	10	30	31	15	35
4		3	3	3	3	4	29	36	23	26
5		2	2	2	2	2	23	34	27	17
6		4	1	1	1	1	14	24	21	6
7		0	0	0	0	0	2	3	3	1

Sun Time	Latitude	SE	S	SW	Sun Time	Latitude	SE	S	SW
5 a.m.		0	0	0	5 a.m.		2	0	0
6		7	1	1	6		13	1	1
7		18	2	2	7		22	2	2
8		22	2	2	8		28	3	2
9		24	3	3	9		30	13	3
10	30[b] Degrees North Latitude	22	5	3	10	50[b] Degrees North Latitude	31	20	3
11		16	7	3	11		27	25	5
12		6	9	4	12		20	27	17
1 p.m.		3	9	14	1 p.m		9	25	26
2		3	6	21	2		3	22	32
3		3	5	27	3		3	16	33
4		3	3	26	4		2	7	31
5		2	2	21	5		2	2	26
6		1	1	11	6		1	1	17
7		0	0	0	7		0	0	7

[a] Values of Y for 8 and 9 p.m are zero.
[b] For N, NE, E, W, NW and horizontal use 40 deg North Latitude values.

the corresponding Table 17 values. The total instantaneous heat gain is the sum of the gain due to transmitted solar radiation and the gain by convection and radiation.

The values given in Tables 20 and 21 are based upon an A.S.H.V.E. research paper[20] to which the reader is directed for additional data. The values in Tables 20 and 21 may be used with fair precision for other patterns of similar transmittance and surface characteristics. For example, the data for hammered glass may be used for glass having shallow, closely-spaced ribs or for glass having small, closely-spaced circular indentations. Because some patterns have distinct orientation properties, no attempt has been made to give values for non-vertical glass.

Design Tables for Glass Block Walls

Tables 23 and 24 give design values of instantaneous rates of heat gain for sunlit walls of Type I pattern 8-in. hollow glass block for a solar declination of 18 deg (see Table 22 for description of block patterns). These tables are based upon the solar intensity values for a clear atmosphere as given in Table 5. For solar energy transmittance data the reader is referred to reference 18. Table 23 presents values of transmitted direct

Table 20. Application Factors to Apply to Tables 17 and 19 to Obtain Instantaneous Rates of Heat Gain for Vertical Single Sheets of Rolled Figured Glass Having Normal Incidence Transmittances and Listed Thicknesses

(Smooth Side Indoors, Figured Side Outdoors)

(See Table 21 for Factors to Apply to Table 16 Values)

Glass Pattern	Normal Incidence Transmittance	Thickness, Inches	Factor to Apply to Table 17
Hammered	0.75	7/32	$1.0(X)^a + 0.50(Y)^{b,c}$
Hammered, etched both sides	0.67	7/32	$1.0(X) + 0.65(Y)^d$
Deep ribs on ⅛ in. centers	0.77 approx.	7/32	$1.0(X) + 0.50(Y)^e$
Hammered heat absorbing	0.21	1/4	$1.0(X) + 1.15(Y)^f$
Hammered heat absorbing, etched both sides	0.14	1/4	$1.0(X) + 1.40(Y)$

[a] X values are Table 17 values
[b] Y values are Table 19 values
[c] Use $0.40(Y)$ for east and west glass
[d] Use $0.60(Y)$ for east and west glass
[e] Use $0.35(Y)$ for east and west glass
[f] Use $0.95(Y)$ for south glass

Table 21. Instantaneous Rates of Heat Gain Due to Transmitted Direct and Diffuse Solar Radiation by Unshaded Rolled Figured Glass

Multiply the Table 16 Values by These Percentage Factors

For Clear Atmospheres and 18 Degrees Declination, North (August 1) For 30, 40, and 50 Degrees North Latitude

Note: To obtain total instantaneous heat gain add adjusted Table 16 values to adjusted Table 17 values

Instantaneous Heat Gain Due to Transmitted Solar Radiation as a Percentage of Single Sheet of Common Window Glass

Sun Time		Hammered					Hammered and Etched				
AM → ↓		N, NW W, SW	NE	E	SE	S	N, NW W, SW	NE	E	SE	S
5 a.m.	7 p.m.	80	80	85	70	80	60	65	75	55	60
6	6	80	85	85	75	80	60	65	75	50	60
7	5	80	80	85	80	80	60	60	70	55	60
8	4	80	75	85	80	75	60	55	65	55	60
9	3	80	60	80	80	60	60	50	55	55	50
10	2	80	65	75	75	65	60	55	50	55	50
11	1	80	80	60	70	65[a]	60	60	50	50	50
12		80	80	80	60	65[a]	60	60	60	50	50

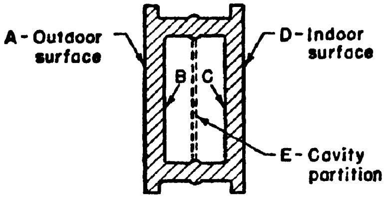
Sun Time		Ribs on ⅛ In. Centers					Hammered Heat Absorbing				
5 a.m.	7 p.m.	60	75	85	25	60	25	20	25	20	25
6	6	60	80	85	30	60	25	20	25	20	25
7	5	60	65	85	40	60	25	20	25	20	25
8	4	60	40	80	50	55	25	20	20	20	25
9	3	60	30	70	55	35	25	20	20	20	20
10	2	60	40	40	55	35	25	20	20	20	20
11	1	60	60	35	40	35[b]	25	25	20	20	20
12		60	60	60	35	45[b]	25	25	25	20	20
	↑ PM →	N, NE E, SE	NW	W	SW	S	N, NE E, SE	NW	W	SW	S

Sun Time		Hammered and Etched Heat Absorbing				
AM → ↓		N, NW W, SW	NE	E	SE	S
5 a.m.	7 p.m.	20	15	15	10	20
6	6	20	15	15	10	20
7	5	20	10	15	10	20
8	4	20	10	15	10	20
9	3	20	10	10	10	10
10	2	20	15	10	10	10
11	1	20	20	10	10	10
12		20	20	20	10	10
	↑ PM →	N, NE E, SE	NW	W	SW	S

[a] Decrease values 10 percent for 30 deg latitude; increase 15 percent for 50 deg latitude

[b] Increase values 30 percent for 50 deg latitude

Table 22. Description of Glass Block Patterns



Elevation Section of Hollow Glass Block to Indicate Location of Surface Patterns

Type I —Smooth Face
- A, D: Smooth
- B: Wide vertical ribs or flutes
- C: Wide horizontal ribs or flutes
- E: None

Type II —Semi-Light Diffusing
- A, D: Narrow vertical ribs or flutes
- B, C: Etched or stippled
- E: None

Type III—Light Diffusing
- A, D: Narrow vertical ribs or flutes
- B, C: Etched or stippled
- E: Glass fiber screen

Type IV—Light Diffusing
- A, D: Close pitch deep horizontal corrugations
- B, C: Vertical light diffusing prisms
- E: None

Type IVA—Light Diffusing
Same as IV except corrugations vertical

Type V —Light Directing
- A, D: Close pitch deep vertical corrugations
- B, C: Horizontal light directing prisms
- E: None

Table 23. Instantaneous Rate of Heat Gain Due to Transmitted Direct and Diffuse Solar Radiation by Unshaded Walls of 8-in. Hollow Glass Block of Type I Pattern

For Clear Atmospheres and 18 Deg Declination, North (August 1)

Note: For total instantaneous heat gain add these values to Table 24 values

Latitude	Sun Time		Instantaneous Heat Gain in Btu per (hr) (sq ft)							
	AM→ ↓		N	NE	E	SE	S	SW	W	NW
30 Deg North	6 a.m.	6 p.m.	4	45	55	12	2	2	2	2
	7	5	5	59	94	29	4	3	3	3
	8	4	5	42	94	38	5	4	4	4
	9	3	5	25	59	34	6	5	5	5
	10	2	6	12	27	24	9	6	6	6
	11	1	6	8	12	13	10	7	6	6
	12		6	6	7	8	9	8	7	6
40 Deg North	5 a.m.	7 p.m.	1	3	3	0	0	0	0	0
	6	6	5	50	67	17	2	2	2	2
	7	5	4	54	98	36	4	4	4	4
	8	4	5	34	90	47	5	4	4	4
	9	3	5	18	59	47	10	5	5	5
	10	2	6	8	29	35	15	5	6	6
	11	1	6	6	13	22	18	7	6	6
	12		6	6	6	13	17	13	6	6
50 Deg North	5 a.m.	7 p.m.	4	28	27	4	1	1	1	1
	6	6	5	53	77	22	3	2	2	2
	7	5	4	44	101	44	4	4	4	4
	8	4	4	26	86	57	7	4	4	4
	9	3	5	12	57	60	16	5	5	5
	10	2	5	6	29	54	25	6	5	5
	11	1	6	6	14	34	32	10	6	6
	12		6	6	6	20	34	20	6	6
		↑ PM→	N	NW	W	SW	S	SE	E	NE

TABLE 24. INSTANTANEOUS RATES OF HEAT GAIN BY CONVECTION AND RADIATION FROM UNSHADED WALLS OF 8-IN. HOLLOW GLASS BLOCK OF PATTERNS TYPE I, II, III, IV, IVA AND V[a]

For Clear Atmospheres and 18 Deg Declination, North (August 1)

For 75 F Indoor Temperature

Note: For total instantaneous heat gain add these values to values in Table 21, or Table 23 adjusted by Table 25 factors.

Sun Time	Dry-Bulb Fahr.		Instantaneous Heat Gain in Btu per (hr) (sq ft)							
			N	NE	E	SE	S	SW	W	NW
5 a.m.	74	40 Degrees North Latitude	−1	−1	−1	−1	−1	−1	−1	−1
6	74		2	5	6[a]	5	0	0	0	0
7	75		3	17	19[a]	15	1	1	1	1
8	77		2	23	28[a]	23	3	2	2	2
9	80		2	22	35	29	7	3	4	3
10	83		4	14	38	34	14	5	5	5
11	87		6	8	31	34	21	7	8	7
12	90		8	10	18	28	25	11	10	9
1 p.m.	93		10	12	16	16	28	27	14	11
2	94		11	12	17	13	27	39	30	13
3	95		13	13	18	13	23	46	43	19
4	94		13	13	17	13	18	47	46	32
5	93		13	12	15	12	15	45	44[a]	37
6	91		15	11	13	11	13	37	37[a]	31
7	87		13	9	10	9	10	21	23[a]	17
8	85		9	6	6	6	6	9	9	6
9	83		6	5	4	5	4	5	6	5

Sun Time	Dry-Bulb	Latitude	SE	S	SW	Latitude	SE	S	SW
5 a.m.	74	30[b] Degrees North Latitude	−1	−1	−1	50[b] Degrees North Latitude	−1	−1	−1
6	74		3	0	0		7	0	0
7	75		13	1	1		15	1	1
8	77		22	2	2		23	2	2
9	80		28	3	4		29	8	3
10	83		31	6	6		34	17	4
11	87		28	10	7		37	24	6
12	90		20	14	9		35	30	19
1 p.m.	93		12	17	18		25	33	33
2	94		13	18	32		14	34	41
3	95		14	16	41		13	30	45
4	94		14	15	45		13	22	46
5	93		13	13	43		12	14	45
6	91		12	11	34		11	12	38
7	87		8	8	17		9	9	25
8	85		6	6	8		6	6	12
9	83		5	5	5		5	5	5

[a] For types III, IV, IVA and V patterns and 30, 40 and 50 deg latitudes, multiply east wall values for 6, 7 and 8 a.m. by 1.40, and west wall values for 5, 6 and 7 p.m. by 1.25.

[b] For N, NE, E, W and NW use 40 deg North Latitude values.

and diffuse solar radiation, while Table 24 gives values of instantaneous rates of heat gain by convection and radiation from the wall. The latter values are for an indoor temperature of 75 F and a 95 F maximum dry-bulb temperature, as indicated in the table, and are based upon experimentally-determined values of solar energy absorption and temperature difference between the two faces. Because the exact dependence of the latter on weather conditions has not been determined, the values of convection and radiation gain cannot be regarded as exact for the assumed design conditions. Indoor and outdoor convection and radiation heat transfer data are the same as those used for common window glass. Table 26 gives corrections to be applied for other design temperatures.

To obtain transmitted direct and diffuse solar radiation for other types of 8-in. block, the Table 23 values are multiplied by the approximate per-

Table 25. Instantaneous Rates of Heat Gain Due to Transmitted Direct and Diffuse Solar Radiation by Unshaded Walls of 8-in Hollow Glass Block of Types II, III, IV, IVA and V Patterns

Multiply the Table 23 Values for Type I by These Percentage Factors

For Clear Atmospheres and 18 Deg Declination, North (August 1)

For 30, 40, and 50 Deg North Latitude

Note: To obtain total instantaneous heat gain add adjusted Table 23 values to the Table 24 values.

Instantaneous Heat Gain Due to Transmitted Solar Radiation as a Percentage of Type I Pattern

Sun Time		Type II Pattern					Type III Pattern				
AM → ↓		N, NW, W, SW	NE	E	SE	S	N, NW, W, SW	NE	E	SE	S
5 a.m.	7 p.m.	100	100	100	100	100	70	70	70	70	70
6	6	100	95	95	90	100	70	65	65	70	70
7	5	100	90	90	90	100	70	65	60	65	70
8	4	100	95	90	90	100	70	65	60	65	70
9	3	100	95	85	90	90	70	65	65	70	65
10	2	100	95	95	95	95	70	70	70	75	65
11	1	100	100	95	95	100	70	70	70	80	75
12		100	100	100	100	100	70	70	70	75	90
	↑ PM →	N, NE, E, SE	NW	W	SW	S	N, NE, E, SE	NW	W	SW	S

Sun Time		Type IV Pattern					Type IVA Pattern				
AM → ↓		N, NW, W, SW	NE	E	SE	S	N, NW, W, SW	NE	E	SE	S
5 a.m.	7 p.m.	55	45	35	75	55	55	45	35	30	55
6	6	55	45	35	80	55	55	35	35	30	55
7	5	55	45	35	60	55	55	35	40	35	55
8	4	55	45	30	40	55	55	45	45	45	55
9	3	55	50	25	30	60	55	50	65	65	50
10	2	55	55	35	30	50	55	55	85	90	60
11	1	55	55	45	35	40	55	55	70	105	95
12		55	55	55	50	45	55	55	55	100	115
	↑ PM →	N, NE, E, SE	NW	W	SW	S	N, NE, E, SE	NW	W	SW	S

Sun Time		Type V Pattern				
AM → ↓		N, NW, W, SW	NE	E	SE	S
5 a.m.	7 p.m.	60	35	35	30	60
6	6	60	35	35	30	60
7	5	60	35	40	35	60
8	4	60	50	65	50	60
9	3	60	80	90	90	60
10	2	60	70	105	105	90[a]
11	1	60	60	80	110	105[a]
12		60	60	60	85	115[a]
	↑ PM →	N, NE, E, SE	NW	W	SW	S

Designation of Block Type
II—Semi-Light Diffusing
III—Light Diffusing
IV—Light Diffusing
IVA—Light Diffusing
V—Light Directing

[a] Reduce by 30% for 30 deg N latitude only.

Table 26. Approximate Corrections to Tables 17 and 24 for Deviations from Indoor and Outdoor Design Temperatures

For each degree the design room temperature exceeds 75 F, subtract correction. For each degree the design outdoor dry-bulb temperature exceeds 95 F, add correction. Apply these corrections to each value in Table 17 or Table 24.

Glass Type	Correction Btu per (hr) (sq ft)
Single Flat or Rolled Figured Glass	1.0
Double Flat Glass and Glass Block	0.5

centages given in Table 25. Note that corrections are the same for all latitudes, and that they vary only on the surfaces exposed to the direct sun. The convection and radiation gain values for all blocks are so nearly the same that a single table suffices. Note, however, that corrections must be made for certain hours for east and west facing walls of some patterns.

Example 15: Find the total instantaneous heat gain through an east wall of 8-in. hollow glass block of Type V pattern at 8 a.m. and 50 deg north latitude. The design temperatures are 75 F indoors and 95 F maximum outdoor dry-bulb, clear atmosphere.

Solution: The gain due to transmitted solar radiation is found from Table 23 for Type I pattern. The factor for Type V is found from Table 25. The convection and radiation gain is found from Table 24 (note the footnote).

The total instantaneous heat gain is, from Equation 7a,

$$q = 86 \times 0.65 + 28 \times 1.4 = 95.2 \text{ Btu per (hr)(sq ft)}.$$

Effect of Deviations from Design Conditions

If the indoor temperature differs from 75 F, or the design outdoor dry-bulb temperature differs from 95 F, corrections can be made to the convection and radiation gain values for flat glass, rolled figured glass, and glass block according to the schedule in Table 26.

The effect of the humid industrial type atmosphere is to cause a considerable reduction in heat gain, if all factors except solar intensity remain the same. Reference 19 gives heat gain values for four orientations at 40 deg north latitude on August 1 for several types of flat glass and glass block, and solar intensities typical of humid industrial atmospheres. These data show that the following approximate reductions, based on total gain for the day, can be expected: 20 percent for all types of glass and glass block in east or west facing walls; 10 percent for south facing flat glass; 5 percent for south facing glass block walls.

Shading of Glass Areas—Design Tables

The effects and possibilities of shading should be carefully investigated whenever the heat gain from glass is a large portion of the cooling load.

Vertical glass, which is not mounted in the plane of the building surface, is partially shaded by the setback. If a vertical window of height l and width w be set back from the plane of the building a distance s, the fraction of the total area of the window which receives direct solar radiation is:

$$G_f = 1 - \frac{r_1 \tan \beta}{\cos \gamma} - r_2 \tan \gamma + \frac{r_1 r_2 \tan \beta \tan \gamma}{\cos \gamma} \tag{10}$$

where

$r_1 = s/l$, $r_2 = s/w$, β = solar altitude, and γ is the wall solar azimuth (see Fig. 1).

Values of β and γ for various latitudes and August 1 are given in Table 7. Special cases not covered by the tabulated data may be solved analytically;[21] however, the design conditions chosen will yield a satisfactory approximation if used without correction for any time during the summer period.

Example 16: Estimate the total instantaneous rate of heat gain for a west window 3 ft wide by 5 ft high, with a setback of 6 in., for August 1 and 40 deg north latitude at 3:00 pm (sun time).

Solution: From Table 16, the instaneous rate of heat gain, due to transmitted direct and diffuse solar radiation, is 180 Btu per hr. From Table 7, β is 45.5 deg and

γ is 16 deg. From Equation 10, the fraction of the total window area that is receiving direct solar radiation is:

$$G_t = 1 - \frac{0.1 \tan 45.5}{\cos 16} - 0.167 \tan 16 + \frac{0.0167 \tan 45.5 \tan 16}{\cos 16}$$
$$= 1 - 0.106 - 0.048 + 0.005 = 0.851$$

In this instance the convection and radiation heat gain is due principally to temperature difference, so that shading has but a small effect on that portion of the absorbed radiation. Hence, the factor 0.851 is applied only to the Table 16 value. Note also a small error results from the fact

TABLE 27. EFFECT OF SHADING UPON INSTANTANEOUS SOLAR HEAT GAIN THROUGH SINGLE THICKNESS OF COMMON WINDOW GLASS

TYPE OF SHADING	FINISH ON SIDE EXPOSED TO SUN	FRACTION OF GAIN THROUGH UN-SHADED WINDOW
Canvas Awning	Dark	0.25-0.35
Inside Roller Shade, Fully Drawn[a]	White	0.45
Inside Roller Shade, Fully Drawn[a]	Medium color	0.63
Inside Roller Shade, Fully Drawn[a]	Dark color	0.80
Inside Roller Shade, Half Drawn[a]	White	0.72
Inside Roller Shade, Half Drawn[a]	Medium color	0.81
Inside Roller Shade, Half Drawn[a]	Dark color	0.90
Inside Venetian Blind, Slats set at 45 deg[b]	White	0.62
Inside Venetian Blind, Slats set at 45 deg[b]	Medium	0.74
Inside Venetian Blind, Slats set at 45 deg[b]	Aluminum	0.70
Inside Venetian Blind, Slats set at 45 deg[b]	Dark color	0.86
Outside Venetian Blind, Slats set at 45 deg[b]	Cream	0.30
Outside Venetian Blind, Slats set at 45 deg, extended as an awning[c]	Any color	0.40
Outside Shading Screen,[d] solar altitude 0-20 deg	Dark color	0.75-0.43
Outside Shading Screen,[d] solar altitude 20-40 deg	Dark color	0.43-0.22
Outside Shading Screen,[d] solar altitude, above 40 deg	Dark color	0.22

[a] Roller shades are assumed to be opaque. Some white shades may transmit considerable solar radiation. For white translucent shades fully drawn use 0.55, and for half drawn use 0.77.

[b] Venetian blinds are fully drawn and cover window. It is assumed that the occupant will adjust slats to prevent direct rays from passing between slats.

[c] Commercial shade with wide slats.

[d] Metal slats 0.05 in. wide, spaced 0.063 in. apart, and set at 17 deg angle with horizontal. At solar altitudes below 38 deg some direct solar rays are allowed to pass between slats, and this amount becomes progressively greater at low solar altitudes.

that the diffuse radiation is not shaded to the same extent as the direct radiation. The *total* instantaneous heat gain therefore is:

$$q = 3 \times 5\ (0.851 \times 180 + 26) = 2690 \text{ Btu per (hr)(sq ft)}$$

A window such as the one used in *Example* 16 would customarily be provided with an additional shading means for use particularly when directly sunlit. Conventional shading devices include *awnings*, *shades*, and *screens* of various types.

Experimental work conducted at the A.S.H.V.E. Research Laboratory,[21,22] and other research[23] to determine the effectiveness of various types of window shades, have been used as the basis for the recommended ratios in column 3 of Table 27. A study of absorptivity of the shade to solar radiation and heat transfer from the shade to the outdoors and indoors, was used to determine these ratios.

There are a number of variables affecting these ratios such as color, fit, solar altitude, and angle of incidence of the solar radiation. These values, therefore, must be considered as approximate, only, and will have to be used with considerable judgment. An inside shade is effective to the extent of its reflectivity, since the portion of the solar radiation directly transmitted by the glass that is absorbed by the shade is transferred by convection to the room air, and by radiation to the solid room surfaces.

INSTANTANEOUS HEAT GAINS VS. INSTANTANEOUS COOLING LOADS

The difference between instantaneous heat gain and instantaneous cooling load has been mentioned previously; its practical importance is sufficient to warrant further consideration.

Fig. 5 offers a simplified schematic illustration showing how the *radiative* part of the instantaneous heat gain is first absorbed by solid objects, and is not encountered by the conditioning equipment as a cooling load until some later time, when it finally appears in the air stream entering the equipment. While it is true that some lag also is inherent in convective heat transfer and the time required to change the air in the conditioned space, this is usually of the order of a few minutes to perhaps half an hour. Heat storage in the interior furnishings and structure increases according to the proportion of the instantaneous heat gain which is in the form of radiation, and also as the thermal capacitance of the objects and materials involved is increased.

Constituents of the total instantaneous heat gain which have appreciable radiation components include those due to glass areas, exposed walls and roofs, lighting, appliances, and people.

No comprehensive data are presently available for use in design load estimates to evaluate the interior load-lag effect, but several investigators[13, 14, 24, 25] have made a study of the problem and have presented many useful data. Tables 12, 13, 16, 17, 21, 23, and 24 are all based on instantaneous rates of heat transfer. Hence, practical judgment and experience offer the only basis of procedure. Until the needed data become available, it is recommended that the non-continuous load be averaged over two or three hours during the time of maximum load, when determining the total instantaneous cooling load where a large portion of the heat gain is radiant. This suggestion applies only to conditions near the time of maximum heat gain, as the heat stored within the structure would necessarily appear in the cooling load eventually; but if it appears at a time when the gain from outside is relatively low, the equipment will be able to maintain satisfactory conditions within the range of maximum capacity.

LOAD FROM INTERIOR PARTITIONS, CEILINGS, AND FLOORS

Whenever a conditioned space is adjacent to another space in which a different temperature prevails, the transfer of heat through the separating structural section must be considered. Calculations are made according to the relation:

$$q = U_i A_i (t_b - t_i) \text{ Btu per hour.} \tag{11}$$

where U_i = coefficient of overall heat transfer between the adjacent and the conditioned space, Btu per (hour) (square foot) (Fahrenheit degree).

A_i = area of separating section concerned, square feet.

t_b = air temperature in adjacent space, Fahrenheit degrees.

t_i = air temperature in conditioned space, Fahrenheit degrees.

Magnitudes of U_i may be obtained from Chapter 9. The temperature

t_b may have any value over a considerable range, according to conditions in the adjacent space. The temperature in a kitchen or boiler room may be as much as 15 to 50 deg above the outdoor air temperature. It is recommended that actual temperatures be measured in adjoining spaces wherever practicable. Where nothing is known, except that the adjacent space is of conventional construction and contains no heat sources, it is recommended that the difference (t_b − t_i) be taken as the difference between the outdoor air and conditioned-space design dry-bulb temperatures *minus* 5 deg. In some cases it may be that the air temperature in the adjacent space will correspond closely to the outdoor air temperature at all times. Under these latter conditions, the heat gain through the partition will be periodic in nature, and the value of a shaded wall should be used from Table 13.

For floors directly in contact with the ground, or over an underground basement that is neither ventilated nor warmed, the heat transfer may be neglected for cooling-load estimates.

LOAD FROM OUTSIDE AIR, VENTILATION AND INFILTRATION

Ventilation. Data for determining the necessary ventilation rate have been presented previously in this chapter. Ventilation required is pri-

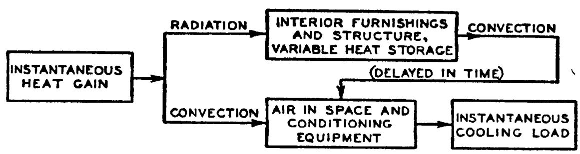


FIG. 5. ORIGIN OF THE DIFFERENCE BETWEEN THE MAGNITUDES OF THE INSTANTANEOUS HEAT GAIN AND INSTANTANEOUS COOLING LOAD

The radiation absorbed by the interior furnishings and structure reaches the conditioning equipment after a considerable delay in time.

marily dependent upon the number of occupants and upon the materials and apparatus within the space which may give off odors. For spaces having ceiling heights 10 ft or less, the total requirement should be checked against the volume, and in no case should the ventilation air rate be less than one air change per hour. In spaces having ceilings higher than 10 ft where the occupant load is low, a check calculation can be made against the volume of the space below an assumed 10-ft ceiling.

Infiltration must never be counted upon to provide ventilation, because on still days there will be little or no infiltration.

Infiltration. The principles of infiltration calculations have been discussed in Chapters 10 and 11, with emphasis on the heating season. For the cooling season, infiltration calculations are usually limited to doors and windows.

To compute cooling-load infiltration for windows by the crack method, use the data of Table 2, Chapter 10, for a wind velocity of 10 mph. Note that for double-hung windows the length of crack is three times the width plus twice the height; while for metal-sash windows the crack length is the total perimeter of the movable or ventilating sections. In calculating window infiltration for an entire structure, it is not necessary to consider the total crack length on all sides of the building, for the wind would not act simultaneously on all sides at once. In no case, however, should less than half of the total crack length be figured. A knowledge of the prevailing wind direction will aid judgment in this consideration.

Cooling-load infiltration for doors[26] may be obtained from Table 3, Chapter 10. For conditions other than those covered, the notes appended to Table 3 will provide a basis for estimates. The tabulated data may also be used as the basis of estimates for interior doors between an air-conditioned and a non-air-conditioned space.

Infiltration load must be included whenever the new air introduced through the system is not sufficient to maintain excess pressure within the enclosure to prevent the infiltration. Whenever economically feasible, it is desirable to introduce sufficient outdoor air through the air-conditioning equipment to maintain a constant outward escape of air, and thus eliminate the infiltration portion of the load. The pressure maintained must, of course, be sufficient to overcome wind pressure through cracks and door openings. When this condition prevails it is not necessary to include any infiltration load. When the quantity of new air introduced through the cooling equipment is not sufficient to build up the required pressure to offset infiltration, the entire infiltration load should be included in the cooling load calculations

Total Outside Air Load. To determine the design cooling load caused by the introduction of outside air, the maximum rate of outside-air entry is first established. In some applications the use of special exhausters from the conditioned space may add to the outdoor-air requirements in determining the maximum rate. Once this design quantity is established, and with the design indoor and outdoor air states known, the cooling load may be computed. There are several methods in use; the more accurate of these require rather detailed calculations. Refer to Chapter 3, and also section on Apparatus-Dew Point in Chapter 29. The following equations are considered to be of sufficient precision for use at usual design conditions, as their accuracy is within 1 percent.

$$\textit{Sensible Load}\quad q_s = Q \times 60 \times 0.244 \times 0.075 \left(1 - \frac{0.00923}{0.62}\right)(t_o - t_i)$$

$$= Q \times 1.08\,(t_o - t_i),\ \text{Btu per hour} \tag{12}$$

$$\textit{Latent Load}\quad q_o = Q \times 60 \times 0.075 \times 1076\,(W_o - W_i)$$

$$= Q \times 4840\,(W_o - W_i),\ \text{Btu per hour} \tag{13}$$

$$\textit{Total Load}\quad q_t = q_s + q_o \tag{14}$$

where

Q = rate of entry of outside air, cubic feet per minute

t_o = outdoor dry-bulb temperature, Fahrenheit degrees.

t_i = indoor dry-bulb temperature, Fahrenheit degrees.

W_o = outdoor humidity ratio, pounds moisture per pound of dry air.

W_i = indoor humidity ratio, pounds moisture per pound of dry air.

0.075 = standard air density, pounds per cubic foot.

0.244 = a constant approximating the specific heat of dry air corrected for moisture Btu per (pound) (Fahrenheit degree).

1076 = a factor approximating the average Btu released in condensing one pound of water vapor from air.

Standard air weight (0.0075) lb per cu ft) is recommended for use in all calculations, as this is the basis for rating fans and its consistent use keeps all parts of the calculations in conformity.

HOW OUTSIDE AIR LOAD AFFECTS ROOM LOAD

Actually, the outdoor air used for ventilation would pass through the conditioning equipment, and be cooled and dehumidified to a lower temperature and humidity ratio than room conditions before entering the room; but for heat-balance purposes the cooling load chargeable to the outdoor air is that corresponding to the difference between the outdoor and indoor air conditions.

One important purpose of the cooling load estimate is to determine the conditions and quantity of air supplied to the space. All the various sensible and latent heat loads within the space must be included. Infiltration must be included in the space load since this air enters the doors and windows, and its heat and moisture load must be offset by the introduction of cooler, dryer air to the space. However, since ventilation air is taken through the conditioning equipment and cooled, this portion does not become a part of the space load. To determine the total load on the refrigeration machine, the ventilation air load must be included in the grand total load.

Example 17: For outdoor design conditions of 95 F dry-bulb and 75 F wet-bulb, and indoor design conditions of 80 F dry-bulb and 67 F wet-bulb, and for the supply of outdoor air at the rate of 1000 cfm and the exhaust of room air at the corresponding rate, calculate the total, sensible and latent heat gains.

Solution: Substituting in Equation 12:

$$q_s = 1000 \times 1.08\ (95 - 80) = 16{,}200 \text{ Btu per hr.}$$

From psychrometric data $W_o = 0.01413$, $W_i = 0.01122$.

Substituting in Equations 13 and 14:

$$q_e = 1000 \times 4840\ (0.01413 - 0.01122) = 14{,}100 \text{ Btu per hr.}$$

$$q_t = q_s + q_e = 30{,}300 \text{ Btu.}$$

Many cooling coil manufacturers publish tables giving psychrometric data based on the *average* conditions of the leaving air for various coil temperatures, air velocities, and entering dry-bulb and wet-bulb conditions. When these tables are used, it is necessary to calculate the mixed air condition entering the coil, and determine from the tables what coil and air velocity will produce the desired leaving air conditions as required for the space to be conditioned. When cooling coils are listed as 80 to 95 percent efficient, the manufacturer indicates that 20 to 5 percent of the air passes through the coil without being cooled. If data of this nature are used, the uncooled portion of the air must be added to the space load before determining the effective air quantity.

HEAT SOURCES WITHIN THE CONDITIONED SPACE

People. The rates at which heat and moisture are given off by human beings under different states of activity are given in Table 28. In many applications these sensible and latent heat gains become a large fraction of the total load. Appreciable variations in heat-emission rates must be recognized according to the age and sex of the individual, state of activity, environmental influences, and duration of occupancy (since for short occupancy the extra heat and moisture brought in by people may be a significant factor).

While Chapter 6 should be referred to for detailed information, Table

28 in this chapter summarizes practical data representing conditions commonly encountered.

Lighting. **In general, the instantaneous rate of heat gain from electric lighting[26] may be calculated from the following relation:**

$$q_{el} = \begin{Bmatrix}\text{total light}\\ \text{wattage}\end{Bmatrix} \times \begin{Bmatrix}\text{use}\\ \text{factor}\end{Bmatrix} \times \begin{Bmatrix}\text{special allow-}\\ \text{ance factor}\end{Bmatrix} \times 3.41, \text{ Btu per hr.} \qquad (15)$$

The *total light wattage* is obtained from the ratings of all fixtures installed, both for general illumination and for display use.

TABLE 28. RATES OF HEAT GAIN FROM OCCUPANTS OF CONDITIONED SPACES[a]

DEGREE OF ACTIVITY	TYPICAL APPLICATION	TOTAL HEAT ADULTS, MALE BTU/HR	TOTAL HEAT ADJUSTED[b] BTU/HR	SENSIBLE HEAT BTU/HR	LATENT HEAT BTU/HR
Seated at Rest	Theater-Matinee	390	330	180	150
	Theater-Evening	390	350	195	155
Seated, Very Light Work	Offices, Hotels, Apartments	450	400	195	205
Moderately Active Office Work	Offices, Hotels, Apartments	475	450	200	250
Standing, Light Work; or Walking Slowly	Department Store, Retail Store				
	Dime Store	550	450	200	250
Walking, Seated	Drug Store				
Standing, Walking Slowly	Bank	550	500	200	300
Sedentary Work	Restaurant[c].	490	550	220	330
Light Bench Work	Factory	800	750	220	530
Moderate Dancing	Dance Hall	900	850	245	605
Walking 3 mph, Moderately Heavy Work	Factory	1000	1000	300	700
Bowling[d]	Bowling Alley				
Heavy Work	Factory	1500	1450	465	985

[a] *Note:* Tabulated values are based on 80 F room dry-bulb temperature. For 78 F room dry-bulb, the total heat remains the same, but the sensible heat values should be increased by approximately 10 percent, and the latent heat values decreased accordingly.

[b] *Adjusted total heat gain* is based on normal percentage of men, women, and children for the application listed, with the postulate that the gain from an adult female is 85 percent of that for an adult male, and that the gain from a child is 75 percent of that for an adult male.

[c] Adjusted total heat value for *sedentary work, restaurant,* includes 60 Btu per hour for food per individual (30 Btu sensible and 30 Btu latent).

[d] For *bowling* figure one person per alley actually bowling, and all others as sitting (400 Btu per hour) or standing (550 Btu per hour).

The *use factor* is the ratio of the wattage in use, for the conditions under which the load estimate is being made, to the total installed wattage. For commercial applications such as stores, the use factor would be unity.

The *special allowance factor* is introduced to care for fluorescent fixtures, and for fixtures which are either ventilated or installed so that only part of their heat goes to the conditioned space. For *fluorescent fixtures*, the special allowance factor is recommended to be taken as 1.20 in order to allow for power consumed in the *ballast*. For *ventilated fixtures*, *recessed fixtures*, and the like, manufacturers' or other data[27] must be sought to establish the fraction of the total wattage which may be expected to enter the conditioned space.

Power. **When equipment of any sort is operated within the conditioned space by electric motors, the heat equivalent of this operation must be considered in the cooling load. The general equation for calculating this load is:**

$$q_{em} = \left(\frac{\text{Horsepower Rating}}{\text{Motor Efficiency}}\right) \times \left(\begin{matrix}\text{Load}\\ \text{Factor}\end{matrix}\right) \times 2544, \text{ Btu per hr.} \qquad (16)$$

It is assumed that both the motor and the driven equipment are within

Table 29. Rate of Heat Gain From Appliances WITHOUT HOODS[a, b]
(Concluded)

Appliance	Capacity	Over-all Dimensions (Less Legs and Handles; Last Dimension is Height) Inches	Control A—Automatic M—Manual	Miscellaneous Data	Manufacturer's Rating Watts	Btu/Hr	Maintaining Rate Btu per Hour	Recommended Rate of Heat Gain Btu per Hour		
								Sensible	Latent	Total
Miscellaneous Electrical Appliances										
Hair Dryer, Blower Type			M	Fan, 165 w; Low, 915 w; High, 1580 w	1580	5400		2300	400	2700
Hair Dryer, Helmet Type			M	Fan, 80 w; Low, 300 w; High, 710 w	705	2400		1870	330	2200
Permanent Wave Machine			M	60 heaters at 25 w each, 36 in normal use	1500	5000		850	150	1000
Neon Sign, per linear ft of tube				½ in. outside diam. ⅜ in. outside diam.				30 60		30 60
Sterilizer, Instrument			A	For physicians; thermostat cuts off 550 w before boiling	1100	3750		650	1200	1850
Miscellaneous Gas-Burning Appliances										
Burners, Laboratory Small Bunsen Small Bunsen Fishtail Fishtail Large Bunsen		 7/16 in. Barrel 7/16 in. Barrel 7/16 in. Barrel 7/16 in. Barrel 1½ in. Mouth	 M M M M M	 Manufactured Gas Natural Gas Manufactured Gas Natural Gas Adjustable orifice		 1800 3000 3500 5500 6000		 960 1680 1960 3080 3350	 240 420 490 770 850	 1200 2100 2450 3850 4200
Cigar Lighter			M	Continuous Flame		2500		900	100	1000
Hair Dryer, 5 helmets			A	Heater and fan blowing air to helmets		33000		15000	4000	19000
Stoves, Oven				Insulated, modern Not insulated		25000 25000	6000 8500	7200 9200	1800 2300	9000 11500

[a] For restaurant appliances, miscellaneous electrical and miscellaneous gas burning appliances.
[b] When these appliances are hooded and provided with adequate exhaust, use 50 percent of recommended rate of heat gain from unhooded appliances.

the conditioned space. If the motor is without the space, then *do not divide* by the motor efficiency in Equation 16. The *load factor* is merely the fraction of the rated load which is being delivered under the conditions of the cooling-load estimate. *Motor efficiencies* may be approximated as follows: about 50 to 60 percent at $\frac{1}{8}$ hp rating, increasing to 80 percent at 1 hp, and to 88 percent at 10 hp and above.

Appliances. Care must be taken in a cooling-load estimate to take into account the heat gain from all appliances, electrical, gas, or steam. Table 29 presents recommended data.[28] Note that the *maintaining rate* in Table 29 is the heat input required to maintain the appliance at the normal operating temperature even though it is not being used, *i.e.*, no coffee is being made, no toast is being made, no food is being cooked in the fry kettle, etc. The maintaining rate is useful in setting up a lower limit to the heat gain to a room from the appliance when in operation.

Experienced judgment must be used in the application of data given in Table 29. Consideration must be given to the heat contributed by appliances which are in use at the time of peak load. The quantity of heat will depend upon whether products of combustion are vented to a flue, whether they escape into the space to be conditioned, or whether appliances are hooded allowing part of the heat to escape through a stack. There are no generally accepted data available on the effects of venting and shielding heating appliances, but it is believed that when they are properly hooded with a positive fan exhaust system through the hood, 50 percent of the heat will be carried away and 50 percent dissipated in the space to be conditioned. The same effectiveness of the hood should be figured for both latent and sensible heat.

LOAD FROM MOISTURE TRANSFER THROUGH PERMEABLE BUILDING MATERIALS

The diffusion of moisture through all common building materials is a natural phenomenon which is always present to a greater or lesser degree.

The permeability values for various building materials are given in Table 22 of Chapter 9, together with an explanation of moisture transmission through these materials.

In the usual comfort air-conditioning application, it is common practice to neglect moisture transfer through walls, for the actual rate is quite small and the corresponding latent-heat load is hardly significant. So-called *vapor barriers* are frequently employed in modern construction for the purpose of keeping moisture transfer to a minimum, and reducing the deteriorating and insulation-destroying effects of moisture.

Industrial jobs, on the other hand, frequently call for a low moisture content to be maintained in a conditioned space. Here the matter of moisture transfer cannot be neglected; indeed, it is quite possible to have the latent-heat load accompanying this transfer be of greater magnitude than any other latent-heat load. The equation for computing this load is:

$$\left(\frac{q_m}{A}\right) = \frac{\mu}{7000} \times \left(\begin{array}{c}\text{Vapor Pressure} \\ \text{Difference, In. Hg}\end{array}\right) \times 1076, \text{ Btu per (hr) (sq ft)} \qquad (17)$$

where

μ = permeability grains per (sq ft) (hr) (in. Hg).

7000 = grains per pound.

The factor 1076 is defined in list of symbols at Equation 14. (Sensible cooling of the water vapor is included in the factor 1076.)

The only means of preventing moisture transfer is to use a vapor proof wall, or to apply a special lining, which is vapor proof. All openings in moisture proof construction must be equipped with special gaskets to prevent entrance of moisture.

When moisture transfer contributes an appreciable part of the latent-heat load, it is recommended that estimates should be made intentionally liberal in order to avoid later difficulties with insufficient dehumidifying capacity. Storage spaces, for example, would require sufficient dehumidifying capacity to handle the moisture brought in with goods to be stored, in addition to moisture leaking in subsequently.

MISCELLANEOUS HEAT LOADS

This designation is intended to cover the various small heat gains from exposed piping, ducts, work done by circulating fan, and unforeseen contingencies. Where sufficient data are available, these various heat gains may be estimated individually. In the majority of cases, however, common practice is to lump these factors together and combine them with a *safety factor* according to the experience and judgment of the estimator. On this basis, a small safety factor is added to the calculated cooling load to compensate for miscellaneous effects. No rules can be given for this procedure, as experience in air conditioning is indispensable for application of suitable safety factors.

REQUIRED AIR QUANTITY THROUGH CONDITIONING EQUIPMENT

The procedure for determining the required air quantity is based upon the thermodynamic principles of Chapter 3 and the use of the A.S.H.V.E. psychrometric chart. Readers are advised to review these principles, paying particular attention to the illustrative examples of cooling load calculations, and to refer to the section on Apparatus Dew-Point in Chapter 29.

Calculation of the cooling load for a conditioned space is equivalent to making, for the space, a heat balance in which all heat, moisture, and infiltration are treated as directly entering the space. As explained in the section, *Load from Outside Air, Ventilation, and Infiltration*, the outside air load normally does not become a part of the space load, because heat and moisture are removed in the air conditioner before this air gets into the conditioned space. The desired conditions are maintained by considering a certain quantity of air to be withdrawn from the space, passed through the conditioning equipment, and returned to the space with such a temperature and humidity ratio that its net effect will be to counterbalance or *remove* the given entering amounts of heat and water vapor. This quantity of *indoor air*, which is considered to be circulated in this manner, is called the *required air quantity* and its determination is normally part of every cooling-load estimate. The procedure is as follows:

1. Determine the total sensible and latent heat loads in Btu per hour for the space.

2. Compute the quantity called the enthalpy-humidity difference ratio (also referred to as heat-moisture ratio) of the room load, $\frac{h - h_s}{W_i - W_s}$. Use the equation:

$$\frac{h_1 - h_s}{W_1 - W_s} = \frac{(\text{Space sensible load} + \text{space latent load})}{\text{Space latent load}/1076} \tag{18}$$

where

h_s = enthalpy of moist air supplied to the space, Btu per pound of dry air.

h_1 = enthalpy of moist air at room design conditions, Btu per pound of dry air.

W_s = humidity ratio of moist air supplied to the space, pounds of vapor per pound of dry air.)

W_1 = humidity ratio of moist air at room design conditions, pounds of vapor per pound of dry air.

Note that the ratio (space latent load/1076) is the equivalent of the required rate of water vapor removal in pounds per hour. If the rate of water removed is known, it may be used directly in Equation 18.

3. Draw a line through the reference point on the A.S.H.V.E. psychrometric chart and the value of $(h_1 - h_s)/(W_1 - W_s)$ determined above. Draw a second line through the state point of the room air (design wet-bulb and dry-bulb temperatures) parallel to this line. This is the *condition line* for the process.

4. Read the temperature where the condition line from step 3 intersects the saturation line. This is called the *apparatus dew-point.*

5. Compute the *required air quantity* from the relation

$$Q_{ra} = \frac{(\text{Space sensible load})}{1.08\left[\begin{pmatrix}\text{Space}\\ \text{dry-bulb}\end{pmatrix} - \begin{pmatrix}\text{Apparatus}\\ \text{dew-point}\end{pmatrix}\right] \times \begin{pmatrix}\text{Coil}\\ \text{efficiency}\end{pmatrix}} \tag{19}$$

The magnitude of Q_{ra} is substantially the quantity, cfm, of cooled and dehumidified air for which the distribution system must be designed.

The numerical factor 1.08 is derived from the product 1 cfm × 60 min × 0.244 × $0.075\left(1 - \frac{0.00923}{0.62}\right) = 1.08$, assuming an average supply air dew-point of 55 F. Since standard air density (0.075) includes the weight of the water vapor, it is desirable to reduce it to the basis of dry air by the last factor where 0.00923 = humidity ratio of air at 55 F dew-point, and 0.62 = ratio of density of water vapor to dry air at same temperature and pressure. Refer to Chapter 35 for coil selection.

Note that the product [(space dry-bulb) − (apparatus dew-point)] × (coil efficiency) is equal to the dry-bulb range through which the conditioned air is cooled. Hence, in rare instances when the *condition line* of the process may not intersect the saturation line, any other convenient reference temperature on the *condition line* may be used instead, provided that the coil efficiency is specified accordingly on the proper basis.

MINIMUM ENTERING AIR TEMPERATURE

Due consideration must be given to the temperature of the air entering the conditioned space in order to prevent objectionable drafts. With ceiling type diffusers or wall grilles with a high aspect ratio (see Chapter 30), many engineers consider 20 deg as the maximum difference for good design under average conditions. This difference can only be exceeded with extremely high ceiling outlets or wall grilles. Thus, if 80 F dry-bulb is to be maintained in a space with average ceiling height, the minimum delivered air temperature would be limited to about 60 F dry-bulb temperature. If the latent heat load is relatively high, it is often necessary to circulate more air with a higher delivered dry-bulb temperature in order to produce a thermodynamic balance. If the dry-bulb temperature of the air supplied to the space is known, the required air quantity can be calcuated from the formula,

$$Q_{ra} = \frac{q_s}{1.08\,(t_1 - t_s)} \tag{20}$$

or the supply temperature t_s can be determined as follows,

$$t_s = t_i - \frac{q_s}{1.08 \times Q_{ra}} \qquad (21)$$

EXAMPLE—COOLING LOAD CALCULATION

An effective means of summarizing the calculation procedure will be the use of an illustrative example. While condensed calculation forms are commonly employed for work of this nature, an outline will be used here in order to facilitate explanatory comments.

Example 18: A one-story office building Fig. 6 is located in an eastern state near 40 deg latitude. The adjoining buildings on the north and west are not conditioned, and the air temperature within them is known to be substantially equal to the outdoor air temperature at any time of the day.

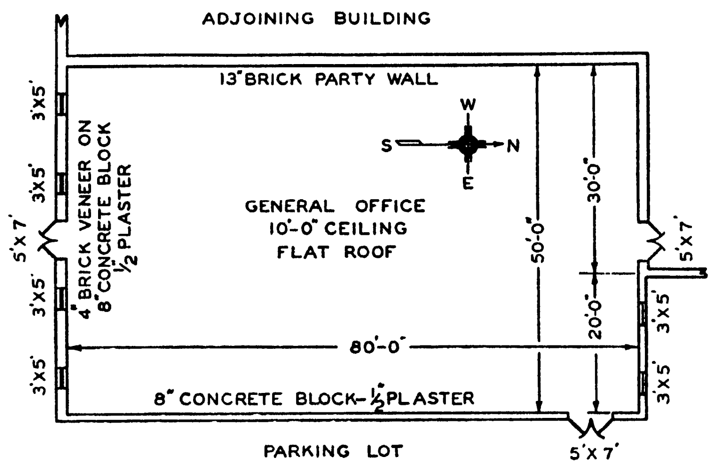


FIG. 6. PLAN OF ONE-STORY OFFICE BUILDING

South wall construction: 8 in. concrete block, 4 in. brick veneer, ½ in. plaster on walls. (Table 9, Chapter 9, No. 92B, $U = 0.41$.)

East wall and outside north wall construction: 8 in. concrete block, painted white, ½ in. plaster on walls. (Table 8, Chapter 9, No. 82B, $U = 0.52$.)

West wall and adjoining north party wall construction: 13 in. solid brick, no plaster:

$$\frac{1}{U} = \frac{1}{1.65} + \frac{13}{5} + \frac{1}{1.65} \qquad \text{or,} \qquad U = 0.263. \quad \text{Use } U = 0.26.$$

Roof construction: 2½ in. flat roof deck of 2 in. gypsum fiber concrete on gypsum board surfaced with built-up roofing. (Table 14, $U = 0.34$ *for summer.*)

Floor construction: 4 in. concrete on ground.

Window: 3 ft x 5 ft, non-opening type, with medium colored Venetian blinds for windows on south wall. Approximately 4 in. reveal on all windows.

Front doors: Two 2 ft-6 in. x 7 ft (glass panels).

Side doors: Two 2 ft-6 in. x 7 ft (½ glass panels).

Rear doors: Two 2 ft-6 in. x 7 ft (wood panels).

Outside design conditions: Maximum dry-bulb 95 F, wet-bulb 78 F; $W_o = 0.0169$ lbs vapor per lb dry air; $h_o = 41.38$ Btu per lb dry air.

Indoor design conditions: Dry-bulb 80 F, wet-bulb 65 F; $W_i = 0.0098$ lb vapor per lb dry air; $h_i = 29.95$ Btu per lb dry air.

Occupancy: 85 office workers.

Lights: 12,000 watts, fluorescent; 4000 watts tungsten.

Fan motor: 7½ hp.

Conditioning equipment to be located in adjoining structure to north.

Find: Total, sensible, and latent maximum cooling loads and required air quantity through conditioning equipment.

Solution: From Table 4, the recommended ventilation rate is 15 cfm per person. Total necessary = 85 × 15 = 1275 cfm or 76,500 cu ft per hr.

As the room volume is 40,000 cu ft, the air changes per hour will be 76,500/40,000 = 1.91 which is more than one air change.

Estimated Time of Maximum Cooling Load:

For this job, judgment indicates that the roof will make the greatest single contribution to the cooling load. Hence, the time of maximum cooling load probably will be the time of maximum heat gain through the roof. From Table 12 the maximum temperature differential for a 2 in. gypsum roof of medium weight construction is 54 deg at 4:00 p.m., and 53 deg at 3:00 p.m. Examination of Table 16 (40 deg N Latitude) shows that solar heat gain through glass on the south wall is 18 Btu per (hr) (sq ft) at 4:00 p.m., and 42 Btu at 3:00 p.m. This indicates that the maximum cooling load occurs at approximately 3:00 p.m. Therefore make load calculations at 3:00 p.m. sun time. (This may be slightly different from 3:00 p.m. local time.) In some cases, there would be no clear-cut evidence of this nature, and consequently, it would be necessary to estimate the load for several successive times, and then to select the maximum.

Heat Gain Through Outer Wall and Roof Areas:

From Table 13 the temperature differential for the south wall (8 in. concrete block with 4 in. brick veneer) may be about the same as a 12 in. brick which is 6 deg at 3:00 p.m. for a dark colored wall. From the same table, the temperature differential for the east wall (8 in. concrete block with plaster) will be 11 deg at 3:00 p.m. for a *light colored* wall (interpolating between 2:00 and 4:00 p.m.). Likewise, the temperature differential for the north exposed wall (8 in. concrete block plus plaster) will be 3 deg at 3:00 p.m. (by interpolation) for a light wall.

The party wall of 13 in. brick on the west side and part of the north side may be treated as if it were an outside wall in the shade which has a temperature differential (from Table 13) of 2 deg.

For the door in north wall estimate $U = 0.59$ from Chapter 9, Table 10, No. 5A. The outdoor temperature at 3:00 p.m. is 95 F. Neglect time lag and any decrement factor. The temperature differential is $(t_o - t_i) = 95 - 80 = 15$ deg. The tabulation of the preceding values at 3:00 p.m. is given in the following table:

Section	Net Area Sq Ft	Temperature Differential F Deg	Heat Transmission Coefficient (U)	Heat Flow Rate per Hour Btu
Roof	4000	53	0.34	72,000
South Wall	405[a]	6	0.41	995
East Wall	765[a]	11	0.52	4,380
North Exposed Wall	170[a]	3	0.52	265
West & North Party Wall	1065[a]	2	0.26	550
Door in North Wall	35	15	0.59	310
Total...				78,500

[a] Calculated from gross wall area, less windows and doors.

Heat Gain Through Glass Areas

In computing the load for 3:00 p.m., only the south windows and doors will be exposed to direct sunlight. Tables 16 and 17 will give the total heat gain from the glass areas. The window reveals will shade the south windows; the fraction of the window area receiving direct radiation is obtained from Equation 10 by substituting values as follows:

$$r_1 = s/l = 4/60;\ r_2 = 4/36;\ \beta = 45.5 \text{ deg},\ \tan\beta = 1.02$$

$$\gamma = 74 \text{ deg},\ \tan\gamma = 3.487,\ \cos\gamma = 0.276$$

$$G_f = 1 - \frac{4}{60}\left(\frac{1.02}{0.276}\right) - \frac{4}{36}(3.487) + \left(\frac{4}{60}\right)\left(\frac{4}{36}\right)\frac{(1.02)\ (3.487)}{0.276} = 0.462.$$

The south doors will be considered entirely sunlit. The outdoor air temperature is 95 F at 3:00 p.m. From Table 27 the inside Venetian blind factor is 0.74. The instantaneous heat gains due to transmitted direct and diffuse solar radiation, and from convection and radiation gain, are found in Tables 16 and 17 as listed below for the south facing doors and windows, the north facing windows and the ½ glass doors in the east wall. The gain through the solid portion of the east doors may be approximated by use of Fig. 4, since the wood panels have little heat capacity. From Table 5 the diffuse radiation value is taken as 18 Btu per (hr)(sq ft) from which $t_o + \alpha_o I_t / f_{cro}$ is found to be 98.2 for α = 0.7 and f_{cro} = 4.0. From Fig. 4, q = 27.0 Btu per (hr)(sq ft). These heat gains are itemized in the following table (note the corrections for difference between 80 F indoor design temperature and the 75 F design temperature used in Table 17).

Location	Area Sq Ft	Fraction Sunlit	Shade Factor	Trans Solar Gain Btu/(hr)(sq ft)	Conv and Rad Gain Btu/(hr)(sq ft)	Corr from 75F to 80F Indoor Temperature Btu/(hr)(sq ft)	Total Gain Btu/(hr)(sq ft)	Total Gain Btu/hr
South Windows	60	0.462	0.74	14	24	−5	33	1980
South Doors	35	1.00	—	42	24	−5	61	2135
East Doors Glass	18	—	—	14	22	−5	31	560
East Doors Wood	18	—	—	—	27	−5	22	395
North Windows	30	—	—	15	22	−5	32	960
Total								6030

In some jobs it would be desirable to increase (or decrease) the instantaneous radiation heat gain by a load-lag factor. The reason for not doing so in this case is that the solar gain is of a low magnitude, and reference to the table indicates that 0.8 of the previous hour would not affect the results materially.

Heat Gain from Ventilation and Infiltration:

Since the necessary ventilation rate 1275 cfm is greater than one air change per hour, it will be satisfactory for determining the ventilation component of the heat gain.

Window infiltration can be taken as negligible since the windows do not open.

Door infiltration requires some judgment. Assume that for each person passing through the double doors, the infiltration will be 100 cu ft of outdoor air, see Chapter 10, Table 3. Assume that the outside doors will be used at the rate of 10 persons per hour and the inside doors at the rate of 30 persons per hour. Total infiltration will then be 40 × 100 = 4000 cfh or 67 cfm.

The design rate of entry of outside air is then:

$$Q = 1275 + 67 = 1342 \text{ cfm}.$$

The sensible, latent and total loads are determined from Equations 12, 13, and 14, respectively, at 3:00 p.m. (Table 11) t_o = 95, t_i = 80, W_o = 0.0169, W_i = 0.0098. All the air entering the room as infiltration becomes a part of the space load.

Infiltration:

$q_s = 67 \times 1.08\ (95-80) = 1085$ Btuh, sensible.

$q_s = 67 \times 4840\ (0.0169-0.0098) = 2300$ Btuh, latent.

$q_t = q_s + q_e = 1085 + 2300 = 3385$ Btuh, total.

Ventilation Air Taken through Cooling Unit Which Does Not Become a Part of the Space Load:

$q_s = 1275 \times 1.08\ (95-80) = 20{,}700$ Btuh, sensible.

$q_e = 1275 \times 4840\ (0.0169-0.0098) = 43{,}800$ Btuh, latent.

$q_t = q_s + q_e = 20{,}700 + 43{,}800 = 64{,}500$ Btuh, total.

Heat Gain from Sources within the Conditioned Space:

For the occupants, use the data of Table 28 for moderately active office work.

Sensible heat gain = 85 × 200 = 17,000 Btu per hr.

Latent heat gain = 85 × 250 = 21,250 Btu per hr.

Total = 38,250 Btu per hr.

For the gain from *lighting*, use Equation 15 with a use factor of unity, and a special allowance factor of 1.20 for the fluorescents and of unity for the tungsten globes.

$$q_{el} = (12{,}000 \times 1.20 + 4000) \times 3.41 = 62{,}700 \text{ Btu per hr.}$$

For the *fan motor*, use Equation 16 with a load factor of unity, and omit term *Motor Efficiency* because the motor is not within the space.

$$q_{em} = 7.5 \times 2544 = 19{,}100 \text{ Btu per hr.}$$

Moisture Permeation, Miscellaneous Allowance, and the Load-Lag Estimate:

Moisture permeation will be negligible, since this is a comfort job with a good building construction.

There would be some heat gain in the ductwork, but this would not be great because of the short run involved. Practical judgment for this job would suggest that no adjustment for load-lag need be made to the load as computed. (Refer to Fig. 5). While it is true that inside radiation forms an important part of the total heat gain, it is advisable to be conservative in recognizing the effect of the large, flat, hot roof on the *comfort sensations of the occupants*. Radiation from the relatively low ceiling, augmented by heat absorption from the lighting fixtures, would produce a sensation of warmth in excess of the nominal effective temperature (see Chapter 6) established by the wet-bulb and dry-bulb temperatures. Hence, it is not desirable to take advantage of every small decrease possible in the peak design load, especially since the peak occurs in mid-afternoon when everything would be rather well warmed.

Total Loads and Required Air Quantity through Conditioning Equipment:

The total loads are summarized in Table 30.

Compute the enthalpy difference ratio from Equation 18.

$$\frac{h_t - h_s}{W_t - W_s} = \frac{(184{,}415 + 23{,}550)}{23{,}550} \times 1076 = 9500.$$

TABLE 30. SUMMARY OF TOTAL LOADS—EXAMPLE 18

LOAD COMPONENT	SENSIBLE Btu/hr	LATENT Btu/hr
All walls, roof and doors	78,500	....
Glass areas	6,030	
Infiltration 67 cfm	1,085	2,300
Occupants	17,000	21,250
Lighting	62,700	
Motor, fan	19,100	
Space Load	184,415	23,550
Ventilation 1275 cfm	20,700	43,800
Totals	205,115	67,350
Grand Total Sensible and Latent		272,465

From the A.S.H.V.E. psychrometric chart, determine that the apparatus dew-point is 54.6 F (refer to Chapters 3 and 29).

In computing the effective air quantity, assume a coil efficiency of 85 percent. Then,

$$Q_{ra} = \frac{184{,}415}{1.08\ (80 - 54.6) \times 0.85} = 7900 \text{ cfm.}$$

(Refer to Chapter 35 for coil selection and efficiency.)

From note under Equation 19 the dry-bulb range will be (80 − 54.6) × 0.85 = 21.6 deg, and the dry-bulb temperature of air leaving the coil will be 80 − 21.6 = 58.4 F. The dry-bulb temperature leaving the fan (including the heat supplied by the fan motor), or delivered into the room, will be (from Equation 21):

$$t_s = 80 - \frac{184{,}415 - 19{,}100}{1.08 \times 7900} = 80 - 19.4 = 60.6 \text{ F.}$$

With good distribution and diffusion, this temperature should not produce objectionable drafts.

Example 18: Summary

Outdoor Conditions	95 DB	78 WB	0.0169 Humidity Ratio
Space Conditions	80 DB	65 WB	0.0098 Humidity Ratio
Difference	15		0.0071

	Btu/Hr	
Sensible Load		
Transmission		
Roof 4000 sq ft × 53° × 0.34 =	72,000	
S. Wall 405 sq ft × 6° × 0.41 =	995	
E. Wall 765 sq ft × 11° × 0.52 =	4,380	
N. Wall Ex. 170 sq ft × 3° × 0.52 =	265	
N. & W. Party Wall 1065 sq ft × 2° × 0.26 =	550	
Floor None		
Door 35 sq ft × 15 × 0.59 =	310	
All Glass and Rest of Doors =	3,020	
Solar Radiation		
S. Glass 60 sq ft × 14 =	840	
S. Glass (Doors) 35 sq ft × 42 =	1,470	
E. Glass (Doors) 18 sq ft × 14 =	250	
N. Glass 30 sq ft × 15 =	450	
Internal Load		
Infiltration 67 cfm × 1.08 × 15° =	1,085	
Lights (12,000 × 1.20 + 4000) 3.41 =	62,700	
People 85 × 200 =	17,000	
Motor, Fan 7.5 hp × 2544 =	19,100	
Total Sensible Space Load		184,415
Latent Load		
Infiltration 67 cfm × 4840 × 0.0071 =	2,300	
People 85 × 250 =	21,250	
Total Latent Space Load		23,550
Ventilation Air		
Sensible 1275 cfm × 1.08 × 15° =		20,700
Latent 1275 cfm × 4840 × 0.0071 =		43,800
Grand Total Load		272,465

LETTER SYMBOLS USED IN CHAPTER 12

α = fraction of incident solar radiation absorbed, dimensionless; subscripts D, d, t refer to direct, diffuse and total, respectively.

β = solar altitude, degrees.

γ = wall solar azimuth, degrees.

ϵ = emissivity, dimensionless.

θ = incident angle, degrees.

λ = amplitude decrement factor, dimensionless.

μ = permeability to moisture transmission, grains per (square foot) (hour) (inch mercury).

τ = fraction of incident solar radiation transmitted, dimensionless. Subscripts D, d, t refer to direct, diffuse and total, respectively.
ϕ = solar azimuth, degrees.
ψ = wall azimuth, degrees.
A = area across which heat is being transferred, square feet.
f = unit surface conductance, Btu per (hour) (square foot) (Fahrenheit degree). Subscripts c, r, o, and i refer to convection, radiation, outdoor, and indoor, respectively.
G_l = fraction of total window area receiving direct solar radiation when shaded by window reveal, dimensionless.
h = enthalpy of air per pound of dry air, Btu per pound. Subscripts i, o, and s refer to indoor, outdoor, and supply air, respectively.
I = incident solar radiation, Btu per (hour) (square foot). Subscripts D, d, Dn, and t refer to direct, diffuse, direct normal and total solar radiation, respectively.
K = cosine of angle of incidence for direct solar radiation striking a surface, dimensionless.
k = thermal conductivity of building material, Btu per (square foot) (hour) (Fahrenheit degree per inch).
L = thickness of building material, inches.
l = height of window, feet.
Q = rate of entry of outdoor air, cubic feet per minute.
Q_{is} = required air quantity through conditioning equipment, cubic feet per minute.
q = instantaneous rate of heat transfer, Btu per hour.
q_e = instantaneous latent heat load, Btu per hour.
q_m = latent heat load due to moisture transmission through materials, Btu per (hr) (sq ft).
q_s = instantaneous sensible heat load, Btu per hour.
q_t = $q_e + q_s$, Btu per hour.
R_s = low temperature radiant energy received from outdoor surroundings (does not include solar radiation), Btu per (hr)(sq ft of receiving surface).
R = radiant energy emitted by a black body, Btu per (hr) (sq ft). Subscripts go and L refer to outdoor surfaces of glass and building, respectively.
S = rate of heat storage within a glass section, Btu per (hr)(sq ft).
t_e = sol-air temperature, Fahrenheit degrees.
t_e^* = sol-air temperature at a time earlier than the time for which heat gain is being found by an amount that is equal to the time lag of the wall or roof, Fahrenheit degrees.
t_{gi} = temperature of indoor glass surface, Fahrenheit degrees.
t_{go} = temperature of outdoor glass surface, Fahrenheit degrees.
t_i = indoor air temperature, Fahrenheit degrees.
t_L = temperature of outer surface of building, Fahrenheit degrees.
t_m = 24-hr cyclic average sol-air temperature, Fahrenheit degrees.
t_o = outdoor air temperature, Fahrenheit degrees.
t_p = $t_m + (t_e^* - t_m)$, *net* equivalent outdoor temperature for combined periodic and mean heat flow, Fahrenheit degrees.
t_s = room supply air dry-bulb temperature, Fahrenheit degrees.
U = overall coefficient of heat transfer of a structural section, Btu per (square foot) (hour) (Fahrenheit degree).
v_o = volume of outdoor air per pound of dry air, cubic feet.
w = width of window, feet.
W = humidity ratio, pounds moisture per pound of dry air. Subscripts i, o, and s refer to indoor, outdoor, and supply air, respectively.

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CHAPTER 13

FUELS AND COMBUSTION

Solid Fuels: Analysis, Classification of Coals, Dustless Treatment, Classification of Cokes, Combustion of Solid Fuels, Firing Methods for Solid Fuels, Secondary Air, Draft Requirements and Regulation, Furnace Volume; Fuel Oils: Classification, Analysis, Combustion, Air Required; Fuel Gases: Classification, Heat Value, Combustion; General Combustion Principles; Air Required; Efficiency from Flue Gas Analysis; Heat Balance; Condensation and Corrosion; Soot

FUELS may be classified according to their physical state as solid, liquid, or gaseous. The principal fuels used for domestic heating are coal, oil, and gas. However, coke, wood, kerosene, sawdust, briquettes, and other substances are used for heating in special applications or in localities where an adequate supply is available. Experiments are in progress in the use of a colloidal suspension of coal particles in fuel oil, but this fuel has not attained wide-spread usage as yet. The choice of fuel is usually based on dependability, cleanliness, availability, economy, operating requirements, and control.

SOLID FUELS

Analysis of Fuels

Coal has a complex composition that makes classification into clear-cut types difficult. Chemically it consists of carbon, hydrogen, oxygen, nitrogen, sulfur, and a mineral residue called ash. A chemical analysis provides some indication of the quality of a coal, but does not define its burning characteristics sufficiently. The coal user is interested principally in the available heat per pound of coal, the handling and storing properties, the amount of ash and dust produced, and the burning characteristics. A description of the relationship between the qualities of coals and these characteristics requires considerable space; a treatment applicable to heating boilers is given in a Bureau of Mines Bulletin.[1]

There are two forms of coal analyses, namely, the *proximate analysis* and the *ultimate analysis*. In the *proximate analysis* the proportions of moisture, volatile matter, fixed carbon, sulfur, and ash are determined. This analysis is more easily made and is satisfactory for indicating most of the characteristics which are of interest to the user. For the proximate analysis the moisture is determined by observing the loss of weight of a sample of coal when dried at about 220 F. To determine the volatile matter, the dried sample is heated to about 1750 F in a closed crucible, and the loss of weight is noted. The remaining sample is then burned in an open crucible, and the accompanying loss of weight represents the fixed carbon. The unburned residue is ash. Although determined separately, the sulfur content is frequently reported with the proximate analysis because the usefulness of a coal for certain purposes depends on its sulfur content.

In the *ultimate analysis*, which is difficult to make, the percentages of carbon, hydrogen, oxygen, nitrogen, sulfur, and ash in the coal sample are determined. It is used for detailed studies of fuels, and in computing a heat balance when required in testing of heating devices. Typical ultimate analyses of the various kinds of coal are shown in Table 1.[2]

Other important qualities of coals are the screen sizes, ash fusion temperature, friability, caking tendency, and the qualities of the volatile matter. In considering these factors the following points are of interest. The volatile products given off by coals when they are heated differ materially in the ratios by weight of the gases to the oils and tars. No heavy oils or tars are given off by anthracite, and very small quantities are given off by semi-anthracite. As the volatile matter in the coal increases to as much as 40 percent of ash and moisture-free coal, increasing amounts of oils and tars are released. For coals of higher volatile content, the relative quantity of oils and tars decreases and is therefore low in the sub-bituminous

Table 1. Typical Ultimate Analyses for Coals

Rank	Btu per Lb		Constituents, Per Cent						
	Moist, Mineral-matter-free[a]	Moist, as Received	Oxygen	Hydrogen	Carbon	Nitrogen	Sulfur	Ash	O_2+H_2+C
Anthracite	14,600	12,910	5.0	2.9	80.0	0.9	0.7	10.5	87.9
Semi-Anthracite	15,200	13,770	5.0	3.9	80.4	1.1	1.1	8.5	89.3
Low-Volatile Bituminous	15,350	14,340	5.0	4.7	81.7	1.4	1.2	6.0	91.4
Medium-Volatile Bituminous	15,200	13,840	5.0	5.0	79.0	1.4	1.5	8.1	89.0
High-Volatile Bituminous *A*	14,500	13,090	9.2	5.3	73.2	1.5	2.0	8.8	87.7
High-Volatile Bituminous *B*	13,500	12,130	13.8	5.5	68.0	1.4	2.1	9.2	87.3
High-Volatile Bituminous *C*	12,000	10,750	21.0	5.8	60.6	1.1	2.1	9.4	87.4
Sub Bituminous *A*	....	....	....	....	....	....	....	....	....
Sub Bituminous *B*	10,250	9,150	29.5	6.2	52.5	1.0	1.0	9.8	88.2
Sub Bituminous *C*	9,000	8,940	35.8	6.5	46.7	0.8	0.6	9.6	89.0
Lignite	7,500	6,900	44.0	6.9	40.1	0.7	1.0	7.3	91.0

[a] (Btu as received) × 100 ÷ (100 − 1.1 Ash)

coals and in lignite. The percentage of ash and its fusion temperature do not indicate the composition or distribution of its constituents.

Classification of Coals

A classification of coals is given in Table 2, and a brief description of the kinds of fuel is given in the following paragraphs, but it should be recognized that there are no distinct lines of demarcation between the kinds, and that they graduate into each other.

Anthracite is a clean, dense, hard coal which creates little dust in handling. It is comparatively hard to ignite, but it burns freely when well started. It is non-caking, it burns uniformly and smokelessly with a short flame, and it requires no attention to the fuel bed between firings. It is capable of giving a high efficiency in the common types of hand-fired furnaces. A tabulation of the quality of the various anthracite sizes will be found in a Bureau of Mines Report.[3] Standard anthracite sizing specifications are shown in Table 3.

Semi-anthracite has a higher volatile content than anthracite. It is not so hard, and ignites somewhat more easily. Otherwise their properties are similar.

Semi-bituminous coal is soft and friable, and fines and dust are created by handling it. It ignites somewhat slowly and burns with a medium length of flame. Its caking properties increase as the volatile matter increases, but the coke formed is relatively weak. Having only half the volatile matter content of the bituminous coals, it can be burned with less production of smoke, and is sometimes called a *smokeless coal.*

The term *bituminous coal* covers a large range of coals and includes many types having distinctly different composition, properties, and burning characteristics. The coals range from the high-grade bituminous coals of the East to the poorer coals of the West. Their caking properties range from coals which melt completely, to those from which the volatiles and tars are distilled without change of form, so that they are classed as non-caking or free-burning. Most bituminous coals are strong

TABLE 2. CLASSIFICATION OF COALS BY RANK[a]

Legend: F.C. = Fixed Carbon. V.M. = Volatile Matter. Btu = British thermal units.

CLASS	GROUP	LIMITS OF FIXED CARBON OR BTU MINERAL-MATTER-FREE BASIS	REQUISITE PHYSICAL PROPERTIES
I. Anthracite	1. Meta-anthracite	Dry F.C., 98 per cent or more (Dry V.M., 2 per cent or less)	Non-agglomerating
	2. Anthracite	Dry F.C., 92 per cent or more and less than 98 per cent (Dry V.M., 8 per cent or less and more than 2 per cent)	
	3. Semi-anthracite	Dry F.C., 86 per cent or more and less than 92 per cent (Dry V.M., 14 per cent or less and more than 8 per cent)	
II. Bituminous[d]	1. Low volatile bituminous coal	Dry F.C., 78 per cent or more and less than 86 per cent (Dry V.M., 22 per cent or less and more than 14 per cent)	Either agglomerating[b] or non-weathering[f]
	2. Medium volatile bituminous coal	Dry F.C., 69 per cent or more and less than 78 per cent (Dry V.M., 31 per cent or less and more than 22 per cent)	
	3. High volatile *A* bituminous coal	Dry F.C., less than 69 per cent (Dry V.M., more than 31 per cent); and moist[c] Btu, 14,000[e] or more	
	4. High volatile *B* bituminous coal	Moist[c] Btu, 13,000 or more and less than 14,000[c]	
	5. High volatile *C* bituminous Coal	Moist Btu, 11,000 or more and less than 13,000[c]	
III. Sub-bituminous	1. Sub-bituminous *A* coal	Moist Btu, 11,000 or more and less than 13,000[c]	Both weathering and non-agglomerating[b]
	2. Sub-bituminous *B* coal	Moist Btu, 9500 or more and less than 11,000[c]	
	3. Sub-bituminous *C* coal	Moist Btu, 8300 or more and less than 9500[c]	
IV. Lignitic	1. Lignite	Moist Btu less than 8300	Consolidated
	2. Brown coal	Moist Btu less than 8300	Unconsolidated

[a] This classification does not include a few coals which have unusual physical and chemical properties and which come within the limits of fixed carbon or Btu of the high-volatile bituminous and sub-bituminous ranks. All of these coals either contain less than 48 percent dry, mineral-matter-free fixed carbon, or have more than 15,500 moist, mineral-matter-free Btu.

[b] If agglomerating, classify in low-volatile group of the bituminous class.

[c] Moist Btu refers to coal containing its natural bed moisture but not including visible water on the surface of the coal.

[d] It is recognized that there may be non-caking varieties in each group of the bituminous class.

[e] Coals having 69 percent or more fixed carbon on the dry, mineral-matter-free basis shall be classified according to fixed carbon, regardless of Btu.

[f] There are three varieties of coal in the high-volatile *C* bituminous coal group, namely, Variety 1, agglomerating and non-weathering; Variety 2, agglomerating and weathering; Variety 3, non-agglomerating and non-weathering.

Adapted from *A.S.T.M.* Standards, 1937, Supplement, p. 145, *American Society for Testing Materials.*

and non-friable enough to permit the screened sizes being delivered free from fines. In general, they ignite easily and burn freely; the length of flame varies with different coals, but it is long. Much smoke and soot are possible, if improperly fired, especially at low rates of burning.

Sub-bituminous coals occur in the western states; they are high in moisture when mined and tend to break up as they dry or when exposed to the weather; they are liable to ignite spontaneously when piled or stored. They ignite easily and quickly, and have a medium length flame; are non-caking and free-burning; the lumps tend to break into small pieces if poked; very little smoke and soot are formed.

Lignite is of woody structure, very high in moisture as mined, and of low heating value; it is clean to handle. It has a greater tendency than the sub-bituminous coals to disintegrate as it dries, and it also is more liable to spontaneous ignition. Freshly mined lignite, because of its high moisture, ignites slowly. It is non-caking. The char left after the moisture and volatile matter are driven off burns very easily, like charcoal. The lumps tend to break up in the fuel bed, and pieces of char falling into the ashpit continue to burn. Very little smoke or soot is formed.

Dustless Treatment

In order to allay the dust, the more friable coals are sometimes sprayed with various petroleum products such as a solution of calcium chloride, or a mixture of calcium and magnesium chlorides.

The coal is usually treated at the mine, but sometimes by the local distributor just before delivery. The salt solutions are sprayed under high

TABLE 3. STANDARD ANTHRACITE SPECIFICATIONS[a]

Test Mesh Round

Size of Coal	Through In.	Over In.	Oversize Max. %	Undersize		Maximum Impurities		
				Max. %	Min. %	Slate[b] %	Bone[b] % or Ash[c] %	
Broken	$4\frac{3}{8}$	$3\frac{1}{4}$ to 3	—	15	$7\frac{1}{2}$	$1\frac{1}{2}$	2	11
Egg	$3\frac{1}{4}$ to 3	$2\frac{7}{16}$	5	15	$7\frac{1}{2}$	$1\frac{1}{2}$	2	11
Stove	$2\frac{7}{16}$	$1\frac{5}{8}$	$7\frac{1}{2}$	15	$7\frac{1}{2}$	2	3	11
Nut	$1\frac{5}{8}$	$\frac{13}{16}$	$7\frac{1}{2}$	15	$7\frac{1}{2}$	3	4	11
Pea	$\frac{13}{16}$	$\frac{9}{16}$	10	15	$7\frac{1}{2}$	4	5	12
Buckwheat	$\frac{9}{16}$	$\frac{5}{16}$	10	15	$7\frac{1}{2}$	...	...	13
Rice	$\frac{5}{16}$	$\frac{3}{16}$	10	17	$7\frac{1}{2}$	.	.	13
Barley	$\frac{3}{16}$	$\frac{3}{32}$	10	20	10		.	15
No. 4	$\frac{3}{32}$	$\frac{3}{64}$	20	30	10	.		15
No. 5	$\frac{3}{64}$	......	30	No limit			.	16

[a] Approved and adopted, effective July 28, 1947, by the Anthracite Committee (Manual of Statistical Information, *Anthracite Institute*).

[b] When slate content in the sizes from Broken to Nut inclusive is less than above standards, bone content may be increased by one and one-half times the decrease in the slate content under the allowable limits, but slate content specified above shall not be exceeded in any event.

[c] Ash determinations are on a dry basis.

A tolerance of 1 percent is allowed on the maximum percentage of undersize and the maximum percentage of ash content. The maximum percentage of undersize is applicable only to anthracite as it is produced at the preparation plant.

Slate is defined as any material which has less than 40 percent of fixed carbon

Bone is defined as any material which has 40 percent or more, but less than 75 percent of fixed carbon

pressure, using from 2 to 4 gal or from 5 to 10 lb of the salt per ton of coal, depending on its friability and size. Oil for the dustless treatment of coal is also applied under high pressure, in concentrations of 1 to 8 qt per ton of coal, depending upon the characteristics of the coal and oil.

Dustless treatments, which are of such a corrosive nature that they may damage coal handling or burning equipment, should not be used.

Classification of Cokes

Coke is produced by the distillation of the volatile matter from coal. The type of coke depends on the coal or mixture of coals used, the temperatures and time of distillation and, to some extent, on the type of retort or oven. Coke is also produced as a residue from the destructive distillation of oil.

High-temperature cokes. Coke, as usually available, is of the high-temperature type, and contains between 1 and 2 percent volatile matter. High-temperature cokes are subdivided into *beehive coke* of which comparatively little is now sold for domestic use, *by-product coke*, which covers the greater part of the coke sold, and *gas-house coke*. The differences among these three cokes are relatively small; their denseness and hardness decrease and friability increases in the order named. In general, the lighter and more friable cokes ignite and burn more readily.

Low-temperature cokes are produced at low coking temperatures, and only a portion of the volatile matter is distilled off. Cokes, as made by various processes under development, have contained from 10 to 15 percent volatile matter. In general, these cokes ignite and burn more readily than high-temperature cokes. The properties of various low-temperature cokes may differ more than those of the various high-temperature cokes because of the differences in the quantities of volatile matter, and because some may be light and others briquetted.

Petroleum cokes, which are obtained by coking the residue left from the distillation of petroleum, vary in the amount of volatile matter they contain, but all have the common property of a very low ash content, which necessitates the use of refractory pieces to protect the grates from being burned.

COMBUSTION OF SOLID FUELS

Firing Methods for Anthracite

An anthracite fire should never be poked or disturbed, as this serves to bring ash to the surface of the fuel bed, where it may melt into clinker.

Egg size is suitable for large fire-pots (grates 24 in. and over) if the fuel can be fired at least 16 in. deep. For best results this coal should be fired deeply.

Stove is the proper size of anthracite for many boilers and furnaces. It burns well on grates at least 16 in. in diameter, on which it is fired about 12 in. deep. The fuel should be fired deeply and uniformly.

Chestnut size coal is in demand for fire-pots up to 20 in. in diameter, and is usually fired to a depth of from 10 to 15 in.

Pea size coal is often an economical fuel to burn. When fired carefully, pea coal can be burned on standard grates. Care should be taken to shake the grates only until the first bright coals begin to fall through the grates. The fuel bed, after a new fire has been built, should be increased in thickness by the addition of small charges until it is at least level with the sill of the fire-door. A satisfactory method of firing pea coal consists of drawing the red coals toward the front end, and piling fresh fuel toward the back of the firebox.

Pea size coal requires a strong draft, and therefore the best results generally will be obtained by keeping the choke damper open, and regulating solely by means of the cold air check and the air inlet damper.

Buckwheat size coal, for best results, requires more attention than pea size coal, and in addition the smaller size of the fuel makes it more difficult to burn on ordinary grates. Greater care must be taken in shaking the grates than with the pea coal on account of the danger of having fuel fall through the grate. In house heating furnaces, the coal should be fired lightly and more frequently than pea coal. When banking a buckwheat coal fire, it is advisable after coaling to expose a small spot of hot fire by putting a straight poker down through the bed of fresh coal. This will serve to ignite the gas that will be distilled from the fresh coal and prevent delayed ignition which, in some cases, depending upon the thickness of the bed of fresh coal, is severe enough to blow open the doors and dampers of the furnace. Where frequent attention can be given and care exercised in manipulation of the grates, this fuel can be burned satisfactorily without the aid of any special equipment, except small mesh grates.

In general, it will be found more satisfactory with buckwheat coal to maintain a uniform heat output and, consequently, to keep the system warm all the time, rather than to allow the system to cool off at times and then to attempt to burn the fuel at a high rate while warming up. A uniform low fire will minimize the clinker formation and keep any clinker formed in an easily broken up condition so that it readily can be shaken through the grate. Forced draft and small mesh grates or, for greater convenience, domestic stokers are frequently used.

Buckwheat anthracite No. 2, or rice size, is used principally in stokers of the domestic, commercial and industrial type. No. 3 buckwheat anthracite, or barley, has no application in domestic heating.

Firing Methods for Bituminous Coal

A commonly recommended procedure for firing domestic heating units, called the side-bank method, requires the movement of live coals to one side or the back of the grate, and placing the fresh fuel charge on the opposite side. The results are a more uniform release of volatile gases, and the subjection of these gases to the high temperature of the red coals. If the fresh charge is covered with a layer of fine coal, still better results may be obtained because of slower release of volatile matter.

Bituminous coal should never be fired over the entire fuel bed at one time. A portion of the glowing fuel should always be left exposed to ignite the gases leaving the fresh charge.

The importance of firing bituminous coal in small quantities at short intervals is discussed in a *U. S. Bureau of Mines* technical paper.[1] Better combustion is obtained by this method in that the fuel supply is maintained more nearly proportional to the air supply.

If the coal is of the caking kind, the fresh charge will fuse into one solid mass which can be broken up with the stoking bar and leveled from 20 min to one hour after firing, depending on the temperature of the firebox. Care should be exercised when stoking not to bring the bar up to the surface of the fuel, as this will tend to bring ash into the high temperature zone at the top of the fire, where it will melt and form clinker. The stoking bar should be kept as near the grate as possible, and should be raised only enough to break up the fuel. With fuels requiring stoking it may not be necessary to shake the grates, as the ash is usually dislodged during stoking.

It is acknowledged that it may be difficult to apply the outlined methods to domestic heating boilers of small size, especially when frequent attendance is impracticable. The adherence to these methods insofar as practicable, however, will result in better combustion.

The output obtained from any heater with bituminous coal will usually exceed that obtained with anthracite, since bituminous coal burns more rapidly than anthracite, and with less draft. Bituminous coal, however, will usually require frequent attention to the fuel bed.

Preventing Smoke

In general, time, temperature and turbulence are the essential requirements for smokeless combustion. Anything that can be done to increase any one of these factors will reduce the quantity of smoke discharged. Special care must be taken in hand-firing bituminous coals.

Checker or alternate firing, in which the fuel is fired alternately on separate parts of the grate, maintains a higher furnace temperature and thereby decreases the amount of smoke.

Coking and firing, in which the fuel is first fired close to the firing door and the coke pushed back into the furnace just before firing again, produces the same effect. The volatiles as they are distilled thus have to pass over the hot fuel bed where they will be burned if they are mixed with sufficient air, and are not cooled too quickly by the heat-absorbing surfaces of the boiler.

Steam or compressed air jets, admitted over the fire, create turbulence in the furnace and bring the volatiles of the fuel more quickly into contact with the air required for combustion. These jets are especially helpful for the first few minutes after each firing. *Frequent firings* of small charges shorten the smoking period, and reduce the density. *Thinner fuel beds* on the grate increase the effective combustion space in the furnace, supply more air for combustion, and are sometimes effective in reducing the smoke emitted, but care should be taken that holes are not formed in the fire. A *lower volatile coal* or a higher *A.P.I.* gravity oil always produces less smoke than a high volatile coal or low *A.P.I.* gravity oil used in the same furnace and fired in the same manner.

The installation of more modern or better designed fuel-burning equipment, or a change in the construction of the furnace, will often reduce smoke. The installation of a Dutch oven, which will increase the furnace volume and raise the furnace temperature, often produces satisfactory results.

In the case of new installations, the problem of smoke abatement can be solved by the selection of the proper fuel-burning equipment and furnace design for the particular fuel to be burned, and by the proper operation of that equipment. Constant vigilance is necessary to make certain that the equipment is properly operated. In old installations the solution of the problem presents many difficulties, and a considerable investment in special apparatus is often necessary.

Lower rates of combustion per square foot of grate area will reduce the quantity of solid matter discharged from the chimney with the gases of combustion. The burning of coke, coking coal, and sized coal from which the extremely fine coal has been removed, will not, as a general rule, produce as much dust and cinders as will result from the burning of non-coking coals and slack coals when they are burned on a grate.

Modern boiler installations are usually designed for high capacity per square foot of ground area, because such designs give the lowest cost of construction per unit of capacity. Designs of this type discharge a large quantity of dust and cinders with the gases of combustion, and if pollution of the atmosphere is to be prevented, some type of dust and cinder catcher must be installed.

Firing Methods for Semi-Bituminous Coal

The *Pocahontas Operators' Association* recommends the central cone method of firing, in which the coal is heaped on to the center of the bed forming a cone, the top of which should be level with the middle of the firing door. This allows the larger lumps to fall to the sides, and the fines to remain in the center and be coked. The poking should be limited to breaking down the coke without stirring. Grates should be rocked gently. It is recommended that the slides in the firing door be kept closed, as the thinner fuel bed around the sides admits the required air.

Firing Methods for Coal and Coke

Coke ignites less readily than bituminous coal and more readily than anthracite, and burns rapidly with little draft. In order to control the air admitted to the fuel it is very important that all openings or leaks into the ashpit be closed tightly. A coke fire responds rapidly to the opening of the dampers. This is an advantage in warming up the system, but it also makes it necessary to watch the dampers more closely in order to prevent the fire from burning too rapidly. In order to obtain the same interval of attention as with other fuels, a deep fuel bed always should be maintained when burning coke. The grates should be shaken only slightly in mild weather, and should be shaken only until the first red particles drop from the grates in cold weather. The best size of coke for general use, for small fire-pots where the fuel depth is not over 20 in., is that which passes over a 1 in. screen and through a 1½ in. screen. For large fire-pots where the fuel can be fired over 20 in. deep, coke which passes over a 1 in. screen and through a 3 in. screen can be used, but a coke of uniform size is always more satisfactory. Large sizes of coke should either be mixed with fine sizes or broken up before using.

SECONDARY AIR

When bituminous coal is hand-fired in a furnace, the volatile matter in the fuel distills off leaving coke on the grate. The product of combustion of the coke is CO_2 and under certain conditions some CO may arise from the bed. The combustion of the volatile matter and the CO may amount to the liberation of from 40 to 60 percent of the heat in the fuel in the combustion space over the fuel bed.

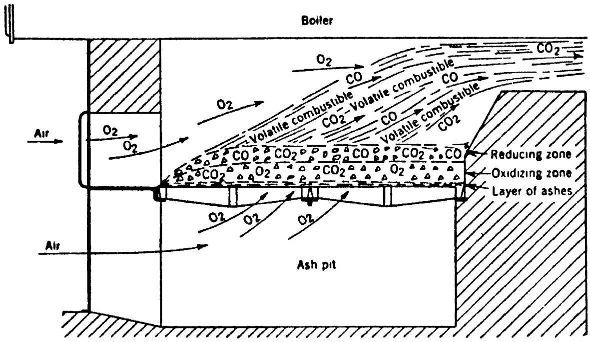


FIG. 1. COMBUSTION OF FUEL IN A HAND-FIRED FURNACE

The air that passes through the fuel bed is called *primary air*, and the air that is admitted over the fuel bed in order to burn the volatile matter and CO is called *secondary air*.

This process of combustion is illustrated in Fig. 1.[4] The free oxygen of the air passes through the grate and the ash above it, and burns the carbon in the lower 3 or 4 in. of the fuel bed forming carbon dioxide. This layer noted as the oxidizing zone, is indicated by the symbols CO_2 and O_2. Some of the carbon dioxide of the oxidizing zone is reduced to carbon monoxide in the upper layer of the fuel bed, noted as the reducing zone and indicated by the symbols CO_2 and CO. The gases leaving the fuel bed are mainly carbon monoxide, carbon dioxide, nitrogen, and a small amount of free oxygen. Free oxygen is admitted through the firing door in an attempt to burn carbon monoxide, as well as the volatile combustible distilled from the freshly fired fuel.

The division of the total into primary and secondary air necessary to produce the same rate of burning and the same excess air, depends on a number of factors which include size and type of fuel, depth of fuel bed, and size of fire-pot.

Size of the fuel is a very important factor in fixing the quantity of secondary air required for non-caking coals. With caking coals it is not so important, because small pieces fuse together and form large lumps. Fortunately, a smaller size fuel gives more resistance to air flow through the fuel bed, and thus automatically causes a larger draft above the fuel bed. More secondary air is drawn through the same slot openings, but, nevertheless, the smallest size of fuel will require the largest secondary air openings. For certain sizes of fuel, no secondary air openings are required, and for large sizes, too much excess air may pass through the fuel bed.

In general, the efficiency of domestic hand-fired furnaces and boilers burning either anthracite or bituminous coal, can be increased for an hour or two after firing, if some secondary air is admitted through the slots of the fire door. However, unless the slots are closed when secondary air is no longer beneficial, the decrease in efficiency during the remainder of the firing cycle, because of excess air, may more than offset the gain resulting from the secondary air at the beginning of the firing period. Unless the secondary air can be readjusted between firings, it is probable that a greater average efficiency will be obtained for domestic hand-fired devices by leaving the secondary air slots closed at all times. There is usually an appreciable amount of air leakage around the firing door and secondary air slots of domestic furnaces and boilers.

When attention is given between firings, the efficiency of combustion can be raised appreciably by admitting secondary air over a bituminous coal fire, to burn the gases and reduce the smoke. The smoke produced is a good indicator, and that opening is best which reduces the smoke to a minimum. Too much secondary air will cool the gases below the ignition point, and prove harmful instead of beneficial.

Secondary air that enters the combustion chamber too far removed from the zone of combustion, will also be harmful, because the oxygen in the secondary air will not react with any unburned gases, unless the mixture is subjected to high temperatures.

Draft Requirements for Coal and Coke Firing

The draft required to effect a given rate of burning the fuel is dependent on the following factors: (1) kind and size of fuel; (2) grate area; (3) thickness of fuel bed; (4) type and amount of ash and clinker accumulation; (5) amount of excess air present in the gases; (6) resistance offered by the boiler passes to the flow of the gases; and (7) accumulation of soot in the passes.

Insufficient draft will necessitate additional manipulation of the fuel bed, and more frequent cleanings to keep its resistance down. Insufficient draft also restricts the control that can be accomplished by adjustment of the dampers. For draft requirements see Chapter 16.

The quantity of excess air present has a marked effect on the draft required to produce a given rate of burning. If the excess is caused by holes in the fuel bed, or an extremely thin fuel bed, it is often possible to produce a higher rate of burning by increasing the thickness of the bed. The thickness of the fuel bed should not, however, be increased too much, because the increased draft resistance will reduce the rate of primary air supply and the rate of burning.

Draft Regulation for Coal and Coke Firing

Because of the varying heating load demands present in most installations, it is necessary to vary the rate of fuel burning. The maintenance of the proper air supply for the various rates of burning is accomplished by regulation of the drafts. Methods of draft regulation used for solid fuel are shown in Fig. 2. The air enters through the ashpit draft door, firing door, and by leaks in the setting, whereas the gases leave only through the outlet. By throttling the gases with the damper in the outlet all the air entering by each of the three intakes is reduced in the same proportion, thus maintaining about the same percent of excess air. If inlet air is controlled by the ashpit draft door, the air admitted through the ashpit is reduced, while it is increased through the other two intake openings, resulting in an increase of excess air. A considerable increase in the efficiency of hand-fired furnaces and boilers can be realized by regulating the air supply by means of the damper in the outlet instead of the ashpit damper. Use of the ashpit damper is required, of course, for low rates of combustion. The cold air check damper is to be used only when chimney draft is excessive. It is normally closed unless closing of the

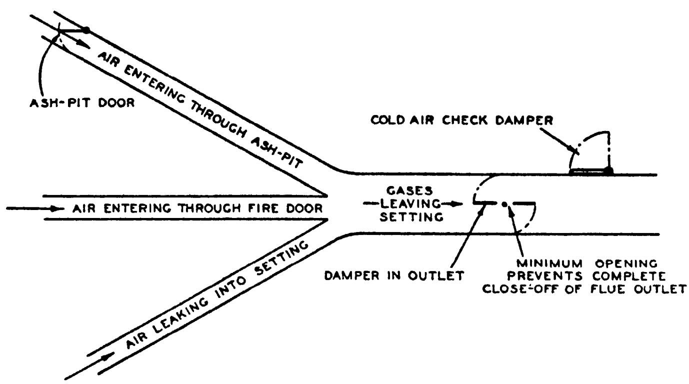


Fig. 2. Methods of Draft Regulation in a Hand-Fired Furnace

outlet damper and ashpit damper is unable to control the rate of combustion.

Furnace Volume for Coal and Coke

The principal requirements for a *hand-fired furnace* are that it shall have enough grate area and correctly proportioned combustion space. The amount of grate area required is dependent upon the desired combustion rate.

The furnace volume is influenced by the kind of coal used. Bituminous coals, on account of their long-flaming characteristic, require more space in which to burn the gases of combustion completely than do the coals low in volatile matter. For burning high volatile coals, provision should be made for mixing the combustible gases thoroughly, so that combustion is complete before the gases come in contact with the relatively cool heating surfaces. An abrupt change in the direction of flow tends to mix the gases of combustion more thoroughly. Anthracite requires comparatively little combustion space.

CLASSIFICATION OF FUEL OILS

Fuel oils are mixtures of hydrocarbons derived from crude petroleum by refining processes designed to produce suitable proportions of naphtha, gasoline, kerosene, fuel oil, and lubricating oil. The processes leave a residue of coke, asphalt, or paraffin depending on the source of the crude oil. In the past, refining processes have been directed toward producing the maximum amount of gasoline, because this product was in greatest demand.

The relative proportions of gasoline and fuel oil produced, per unit volume of crude oil could be varied considerably to suit seasonal changes in demand or gradual trends from year to year. At present gasoline represents about 43 percent and fuel oil (including kerosene) about 22 percent of the yield from crude oil.

Crude oil is distilled in towers at atmospheric pressure to produce gasoline, naphtha, kerosene distillates, and colored distillates, and leave straight-run residues. The colored distillates are distilled further to produce light distillate fuel oils, some lubricating oil, wax, etc., whereas the straight-run residues are distilled under vacuum to produce heavier distillates. The residual fuels remaining can then be passed through cracking plants to produce more gasoline, cracked kerosene, cracked distillates, and cracked residual fuel oils. The exact processes used depend on the proportions of the various end products desired, and to some extent upon the composition and characteristics of the crude oil.

Fuel oils may be described as straight-run fuels, thermally-cracked fuels, catalytically-cracked fuels, or blended fuels depending on the refining process used to produce them. Straight-run fuels are those produced by distillation under atmospheric pressure or a vacuum without decomposition of the hydrocarbons by cracking. Thermally-cracked fuels are those produced by a cracking process involving elevated temperatures (850–1100 F) to decompose some of the heavier hydrocarbons. Catalytically-cracked fuels are those produced with the aid of an alumina-silica catalyst in the cracking process at lower temperatures than those used for thermal cracking. Blended fuel oils are mixtures of any of the above three types.

Analysis of Fuel Oils

Crude oil in its natural state contains primarily paraffin hydrocarbons (chemical formula C_nH_{2n+2}, naphthene hydrocarbons (formula C_nH_{2n}), and aromatic hydrocarbons (formula C_nH_{2n-6}) where n is a whole number. Fuel oils produced by pure distillation, that is the straight-run fuel oils contain essentially these same hydrocarbons. Those produced by cracking processes may contain generally all the hydrocarbon series from C_nH_{2n+2} to C_nH_{2n-14}, and especially do they contain appreciable percentages of the olefin hydrocarbons which are relatively less stable than the paraffin, napthene, and aromatic hydrocarbons. The paraffin hydrocarbons are hydrogen-saturated, are among the most stable, and have the highest hydrogen-carbon ratio of any of the hydrocarbon series. The straight-run fuel oils have the highest paraffin content, the highest hydrogen-carbon ratio and are the most stable of the fuel oils. The thermally-cracked fuel oils have the lowest paraffin content while the catalytically-cracked fuel oils are intermediate in paraffin content and stability. The hydrogen-carbon ratio of straight-run fuel oils ranges from 0.155 to 0.170, and in catalytically-cracked fuel oils ranges from 0.133 to 0.156, while it is somewhat lower for thermally-cracked fuels. The blending of straight-run oils with cracked oils is common practice to improve the paraffin content, stability, and ignition characteristics of fuel oils. A high paraffin content and a high hydrogen-carbon ratio are generally desirable characteristics for domestic fuel oils and consequently, the straight-run distillates are better suited to this use than the fuel oils produced by the various cracking processes. On the other hand, thermally-cracked fuel oils often have a lower pour point and a lower viscosity than comparable straight-run fuel oils. The color and stability of cracked fuel oils can be much improved by treatment with sulfuric aicd, by neutralization, and by redistillation.

Table 4. Detailed Requirements for Fuel Oils[a]

Grade of Fuel Oil[b]	Flash Point F	Pour Point F	Water and Sediment %	Carbon Residue on 10% Residuum %	Ash %	Distillation Temperatures F			Viscosity Seconds							
						10% Point	90% Point	End Point	Saybolt Universal at 100 F		Saybolt Furol at 122 F		Kin. C. S. at 100 F		Kin. C. S. at 122 F	
	Min.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.
1. A distillate oil intended for vaporizing pot-type burners and other burners requiring this grade[d] A.P.I. gravity 35 (min.)	100 or Legal	0	Trace	0.15		420		625					2.2	1.4		
2. A distillate oil for general purpose domestic heating for use in burners not requiring No. 1 A.P.I. gravity 26 (min.)	110 or Legal	20[c]	0.10	0.35		e	675		40				(4.3)			
4. An oil for burner installations not equipped with preheating facilities	110 or Legal	20	0.50		0.10		675		125	45			(26.4)	5.8		
5. A residual type oil for burner installations equipped with preheating facilities	130 or Legal		1.00		0.10					50	40			32.1	(81)	
6. An oil for use in burners equipped with preheaters permitting a high viscosity fuel	150 or Legal		2.00[f]								300	45			(638)	(92)

[a] Recognizing the necessity for low sulfur fuel oils used in connection with heat-treatment, non-ferrous metal, glass and ceramic furnaces and other special uses, a sulfur requirement may be specified in accordance with the following table:

Grade of Fuel Oil	Sulfur, Max. Percent
No. 1	0.5
No. 2	0.5
Nos. 4, 5 and 6	No Limit

Other sulfur limits may be specified only by mutual agreement between the buyer and seller.

[b] It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

[c] Lower or higher pour points may be specified whenever required by conditions of storage or use. However, these specifications shall not require a pour point lower than 0 F under any conditions.

[d] No. 1 oil shall be tested for corrosion in accordance with par. 15 for 3 hours at 122 F. The exposed copper strip shall show no gray or black deposit.

[e] The 10 percent point may be specified at 440 F maximum for use in other than than atomizing burners.

[f] The amount of water by distillation, plus the sediment by extraction, shall not exceed 2.00 percent. The amount of sediment by extraction, shall not exceed 0.50 percent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 percent.

TABLE 5. APPROXIMATE GRAVITY AND CALORIFIC VALUE OF STANDARD GRADES OF FUEL OIL

COMMERCIAL STANDARD NO.	APPROXIMATE GRAVITY RANGE A.P.I.	CALORIFIC VALUE BTU PER GALLON
1	35–45	138,800–132,900
2	26–40	144,300–135,800
4	12–25	153,000–145,000
5	10–23	154,600–146,200
6	8–17.5	156,000–149,700

Grade Classification of Fuel Oils

Fuel oils are most commonly classified by dividing them into grades in accordance with the *Commercial Standard* (CS12-48) entitled *Fuel Oil* published by the *U. S. Department of Commerce.* These specifications, given in Table 4, conform to *ASTM Materials Tentative Specifications for Fuel Oils* D 396 - 48 T. Oils may be classified roughly by specific gravity but it is not an adequate index of the suitability of an oil for a given purpose. Other characteristics of fuel oils which determine their grade classification in the *Commercial Standard,* and their suitability for given uses are the flash point, pour point, water and sediment content, carbon residue, ash, sulfur content, distillation characteristics, and viscosity.

The flash point of an oil is important with regard to safety in storage and ease of ignition in systems employing automatic ignition. The distillation characteristics determine whether or not the oil can be completely evaporated in some types of burners, and whether cracking will be likely to occur prior to combustion. A low pour point and low water content are desirable for outdoor storage in cold climates. Sediment, carbon residue, and ash should be low to prevent clogging of strainers and the accumulation of unburned material in the burner. A low viscosity allows the fuel oil to flow through supply lines readily and to be broken up into small droplets in atomizing type burners. The sulfur content is of importance because sulfur compounds corrode the burner and heating system or because undesirable compounds of sulfur may be formed in certain industrial processes. Some of these characteristics of a fuel oil are required to lie within certain limits for each of the grades of fuel oil listed in *Commercial Standard* CS12-48. Some fuel oils do not fall into any of the grade classifications of the *Commercial Standard* because failure to comply with all of the requirements of one grade does not automatically place the fuel oil in the next lower grade, unless it meets all of the requirements of the lower grade.

By ultimate analysis the No. 1 and No. 2 fuel oils contain 84 to 86 percent carbon, 12.0 to 13.5 percent hydrogen, one to three percent oxygen and nitrogen, and 0.5 percent or less of sulfur. The heavier grades of fuel oil, Nos. 4, 5 and 6, may contain as much as 88 percent carbon, as low as 11 percent hydrogen, and considerably more sulfur than is permissible in the domestic grades.

Due to variation in the constituents of different fuel oils and the different refining methods used, the *API* gravities and calorific values of the different grades of fuel oil cover a range in each grade with some overlapping between domestic grades and between commercial and industrial grades. The relation between the *API* gravity of fuel oils and their calorific value is shown in Table 5. Grades No. 1 and No. 2 are used predominantly in domestic heating equipment whereas grades 5 and 6 are used in commercial and industrial burners. Grade 6 usually requires preheating to increase its fluidity and to permit atomization, whereas grade 5 is used in some burners without preheating. Grade 4 fuel oil does not require preheating and can be burned satisfactorily in a limited number of domestic burners.

COMBUSTION OF FUEL OILS

Many theories have been advanced during the past century to explain the mechanism of combustion of hydrocarbons in oil burners and other devices used for producing heat or light. These theories have been modified from time to time to agree with new experimental evidence. Much still remains unknown about the process of decomposition and combustion of hydrocarbons.

Only three theories will be discussed here: (1) the carbonic combustion theory, (2) the aldehydeous combustion theory, and (3) the chain reaction theory.

The carbonic combustion theory postulates that thermal destruction of hydrocarbon molecules is likely to occur if (1) the oil is suddenly exposed

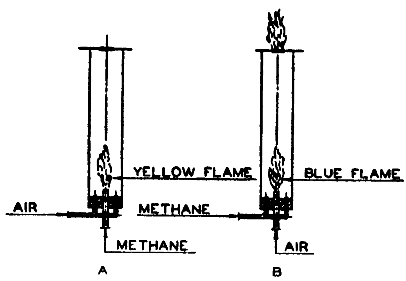


FIG. 3. ILLUSTRATION OF BLUE AND YELLOW FLAME COMBUSTION

to intense heat without allowing time for previous evaporation, (2) the oil and air are inadequately mixed, and (3) there is no preheating of the air or mixture. According to this theory the hydrocarbons would be thermally decomposed into hydrocarbons of lower molecular weight along with some free carbon atoms released under the conditions just described. The free carbon atoms may produce smoky combustion while those carbon atoms that are oxidized to carbon dioxide will produce a yellow luminous flame.

The aldehydeous combustion theory is based on the evidence that aldehydes, alcohols, and possibly peroxides are formed as intermediate products when hydrocarbons are decomposed and oxidized to the final products of combustion. The formation of formaldehyde is certain since it can be identified in the flue gases from blue flame oil burners when insufficient combustion air is provided. Alcohols have been identified by certain investigators during the oxidation of methane and ethane. Aldehydeous combustion is illustrated by the blue flame oil burner and the conditions conducive to this type of hydrocarbon decomposition consist of (1) allowing the oil time and opportunity to evaporate completely prior to combustion, (2) mixing the air and oil vapor thoroughly before combustion, and (3) preheating the air or the mixture.

Blue and yellow flame combustion can be demonstrated by the apparatus illustrated in Fig. 3. If methane is burned in an atmosphere of air, as in burner A, a yellow flame will result, whereas the introduction of the air for combustion in the center of a stream of methane, as in burner B, will result in blue flame combustion. As the center of the flame in burner A is exposed to intense radiation the methane is thermally decomposed and liberates carbon particles which emit a yellow luminous flame during oxidation. In burner B the center of the flame cone is filled with air which cannot decompose under heat, the methane gas at the zone of contact with the air is only moderately heated because of outward radiation, and the air is preheated in the center as it approaches the flame; each of these conditions tend to produce aldehydeous combustion.

It is probable that the chain reaction theory is an extension of the aldehydeous combustion theory since most investigators who have studied the former have observed that the formation of aldehydes is one of the steps in the combustion process. It has been well established that fuel oils must be gasified before combustion can occur, and that molecules of a hydrocarbon and oxygen do not combine directly with each other to form carbon dioxide and water vapor, but pass through intermediate reactions in the process.

Lewis and von Elbe,[5] Pease, and others have advanced the theory that that the reactions between hydrocarbons and oxygen are probably chain reactions. This theory postulates that a great many different reactions take place simultaneously or progressively between molecules, atoms, and radicals in a mixture of hydrocarbons and oxygen. Some of these reactions produce particles or substances that tend to accelerate the reactions while others tend to slow down the process. Time, temperature, pressure, light and certain catalytic agents all may affect the speed and nature of these processes. The kind of intermediate products formed before combustion is complete depends on the physical conditions mentioned, as well as the molecular structure of the particular hydrocarbon participating in the reaction. Aldehydes, methyl and ethyl alcohols, formic acid, and other substances, have all been identified as intermediate products in certain reactions. The chain reaction theory, in reality embraces and elaborates on the aldehydeous combustion theory.

Oil Burning Indexes

A number of indexes have been used, or proposed, as an indication of the burning qualities of fuel oils based on one or more physical measurements made on the oil. These may be summarized as follows:

A. Indexes based on a single physical test: (1) *API* gravity[7], (2) Aniline point[8], (3) *Institute of Petroleum* smoke test, (4) Carbon-Hydrogen ratio based on flue gas analysis or ultimate analysis, and (5) Percent aromatics determined by sulfuric acid absorption tests[9].

B. Indexes based on two or more physical tests: (1) Diesel index[9] based on *API* gravity and aniline point, (2) *Institute of Petroleum* cetane number [10] based on the *API* gravity and 50 percent distillation point, (3) *Universal Oil Products* characterization factor[11] based on specific gravity and average boiling point, (4) Burning index[12] based on *API* gravity and 50 percent distillation point, and (5) Estimated Carbon-Hydrogen ratio[13] based on *API* gravity aniline point, and boiling point.

Various investigators have shown correlation between one or more of these indexes and the performance of fuel oils in oil burners. Experiments conducted with the *Oil Heat Institute* Reference Test Unit[14] indicated good correlation between the smoking tendency of fuel oils and *API* gravity, burning index, Diesel index, and hydrogen-carbon ratio for a limited number of oils in laboratory apparatus simulating a pressure-atomizing burner. These results are shown graphically in Fig. 4. Smoking tendency is given here in terms of smoke spot reflectance, the light reflectance of a smokesoiled filter paper. A high reflectance, relative to a clean filter paper, indicates low smoking tendency. CO_2/U is the observed CO_2 divided by the ultimate or maximum theoretical CO_2 expressed as a percentage. Reid and Hersberger[12] have related burning qualities and burning index for various oils in a wall-flame burner. Cauley and Delgass[13] cite test results on combustion indexes obtained with vaporizing burners. The present experimental data are probably too meager as yet to correlate adequately any one of these indexes with burning qualities of oil fuel for all types of burners. Few attempts have been made to suggest limits for any of these fuel oil indexes for particular applications, even though correlations between them and burning qualities have been observed. In other words,

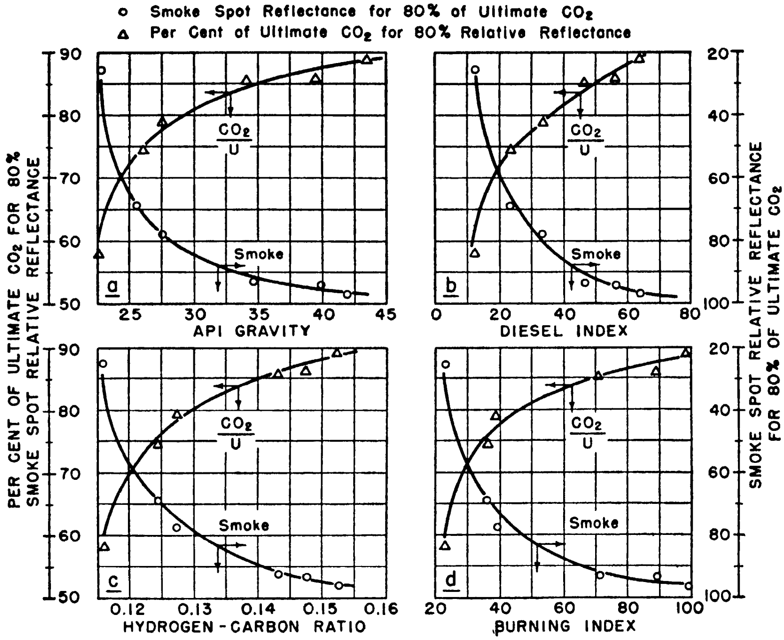


FIG. 4. CORRELATION OF BURNING QUALITIES OF FUEL OILS WITH FOUR COMBUSTION INDEXES

none of the above mentioned indexes has yet gained sufficiently wide usage to replace the grading of oils by *Commercial Standard* CS12-48.

Experiments have shown that thermal decomposition or cracking of hydrocarbons begins at a temperature of approximately 680 F at atmospheric pressure, although the temperature of cracking varies somewhat above and below this value. Thus pure distillate fuel oils, whose end point does not exceed this temperature, can usually be completely evaporated in vaporizing-type oil burners at atmospheric pressure without leaving a residue or without cracking of the hydrocarbons. Fuel oils that cannot be completely evaporated below 680 F are likely to undergo cracking in vaporizing type burners, with the resulting possibilities of smoky combustion and residues in the oil burner. A complete distillation curve cannot usually be determined for fuel oils containing fractions that evaporate above 680 F.

Since No. 1 grade fuel oil in *Commercial Standard* CS12-48 has a maximum end point of 625 F, it can in most cases be completely evaporated in atmospheric vaporizing burners without cracking, although occasionally an oil is found that undergoes cracking at temperatures below 625 F. By the same criterion, No. 2 grade fuel oil in the *Commercial Standard*, which can have a maximum distillation temperature of 675 F at the 90 percent point, would frequently be cracked in a vaporizing burner. However some No. 2 fuel oils do not crack before complete evaporation takes place. Vaporizing-type burners can generally use only No. 1 fuel oil with assurance that thermal decomposition will not occur during combustion. On the other hand either No. 1 or No. 2 fuel oils may be employed in high or low pressure atomizing burners when the temperatures developed in the combustion chamber are high enough to assure complete combustion, even if the fuel oil is thermally decomposed.

In vaporizing burners, preheating of the combustion air and fuel, complete evaporation of fuel before it is exposed to intense heat, and thorough mixing of the air and gasified fuel promote complete combustion without smoke and with a minimum of excess air. In pressure-type burners preheating of the combustion air, a maximum of air turbulence, good atomization of the fuel, and high combustion chamber temperatures (preferably red hot) promote smokeless combustion with a minimum of excess air.

Natural draft burners depend on the motivating force of a chimney to induce enough air into the burner for complete combustion. Forced draft burners are supplied with combustion air by means of a blower or fan; the chimney merely conducts the flue gases outdoors and prevents leakage of flue gases inside the building. More details on the operation of the different kinds of oil burners and on chimneys and draft will be found in Chapters 14 and 16 respectively.

FUEL GASES

Fuel gases employed for various heating and air conditioning processes throughout the United States fall into three broad classifications: natural, manufactured, and liquefied petroleum. Natural gas is a mixture of several combustible gases and, usually, a small percentage of inert gases obtained from geologic formations. Natural gas is produced in significant amounts in 20 states. Texas is by far the largest producer, followed by Louisiana, Oklahoma, California, Kansas, and West Virginia. Manufactured gas is made by the distillation or cracking of oil or coal, by the steam carbon reaction, or by combinations of these processes. Liquefied petroleum gases (propane and butane) are higher hydrocarbon gases normally obtained as a by-product of oil refineries or by stripping natural gas. These two compounds are generally gaseous under usual atmospheric conditions although they can be liquefied by the application of moderate pressures at normal temperatures.

The demand for gaseous fuels has increased so tremendously during the past 25 years that few cities now can be said to depend solely on one source of supply. During peak load periods, heating demands on natural gas distribution systems may necessitate augmenting the base supply with supplemental fuels such as high Btu oil gas or liquefied petroleum gas-air mixtures. The supply of manufactured gases may be similarly increased by adding natural gas, reformed refinery gases, or relatively low heating value mixtures of liquefied petroleum gas and air.

In American gas practice the heating value of a gas and appliance efficiencies are based on the gross heating value. This value is the number of Btu liberated by complete combustion, at constant pressure, of one cubic

TABLE 6. TYPICAL GAS ANALYSES

Constituents of Gas—Percent by Volume

No. Type	Source	CO_2	O_2	N_2	CO	H_2	CH_4	C_2H_6	C_3H_8	C_4H_{10}		Specific Gravity[a]	Btu per Cu Ft[a] Gross	Btu per Cu Ft[a] Net
1 Natural gas	Typical	..	.	0.8	..	.	83.4	15.8	.	..	..	0.61	1129	1021
2 Propane	Commercial (natural gas)	.			.	.	..	2.2	97.3	0.5	..	1.55	2558	2358
3 Propane	Commercial (refinery gas)	.	..	.	..	..	..	2.0	72.9	0.8	C_3H_6 24.3	1.77	2504	2316
4 Butane	Commercial (natural gas)	.	..	..	..	.	..	.	6.0	94.0[c]	..	2.04	3210	2961
5 Butane	Commercial (refinery gas)	..	.	..	..	..	..	..	5.0	66.7[b]	C_4H_8 28.3	2.00	3184	2935
6 Butane-air	......	..	..	..	..	.	..	..	..	17.2	Air 82.8	1.16	550	516
										(Illuminants)				
7 Reformed natural	Straight shot generator	2.1	0.3	1.5	13.6	48.3	33.0	..	..	1.2	..	0.41	559	497
8 Refinery oil	Vapor phase crack	0.2	0.2	0.5	1.2	13.1	23.3	21.9	..	39.6	..	0.89	1475	1351
9 Oil gas	Portland, Ore.	1.2	0.5	2.4	7.7	54.2	30.1	..	..	3.9	..	0.37	570	510
10 Coal gas	Contin. Vert.	3.0	0.2	4.4	10.9	54.5	24.2	..	..	2.8	..	0.42	532	477
11 Coke oven gas	By-product	2.0	0.03	4.8	5.5	51.9	32.3	..	..	3.2	.	0.40	569	509
12 Producer gas	Bituminous	4.5	0.6	50.9	27.0	14.0	3.0	..	..	...	..	0.86	163	153
13 Blast furnace gas	......	11.5	..	60.0	27.5	1.0	..	..	..	...	..	1.02	92	92
14 Blue gas	Bituminous	5.5	0.9	27.6	28.2	32.5	4.6	..	..	0.7	..	0.70	260	239
15 Carburetted water gas	Heavy oil with blow run	6.0	0.9	12.4	26.8	32.2	13.5	..	..	8.2	..	0.66	530	451
16 Carburetted water gas	High Btu	0.7	0.3	5.8	11.7	28.0	36.1	.	..	17.4	..	0.63	840	770
17 Mixed Gas	Natural, Blue, Producer, Reformed Natural	3.9	0.1	15.5	10.3	17.8	46.9	5.4	0.1	...	..	0.63	650	586
18 Mixed Gas	Coal, Natural	1.1	0.4	6.6	4.0	35.8	45.7	4.6	...	Ill. 1.8	..	0.46	700	627
19 Mixed Gas	Natural, Coke Oven	1.0	0.2	5.5	2.0	20.9	66.6	2.6	...	Ill. 1.2	..	0.52	802	721
20 Sewage Gas	Decatur, Ill.	22.0		6.0	.	2.0	68.0	.	..	...	..	0.79	690	621

This table abstracted from *Gaseous Fuels, American Gas Association*, and from *A.G.A. Laboratories* Data.
[c] At 60F and 30 in. mercury, absolute pressure.

foot of gas saturated with water vapor and measured at 60 F and 30 in. of mercury, with air at the same temperature and pressure. Products of combustion are cooled to the initial temperature of the gas and air and the water formed by combustion of free and combined hydrogen is condensed to the liquid state.

Classification of Gases

Representative properties of gaseous fuels commonly employed for domestic heating processes are shown in Table 6.

Natural gas contains from 55 to 98 percent methane with various percentages of higher hydrocarbons, chiefly ethane. In addition to these components, small quantities of non-combustible gases such as carbon dioxide, nitrogen, and helium are sometimes present. Percentages of the different components vary with the area from which natural gas is withdrawn. They may even vary slightly from any given well during its lifetime but these variations are inconsequential insofar as the utilization of the gas is concerned. Heating values of natural gases vary from 900 to 1400 Btu per cu ft but the usual range for use is from 1000 to 1050 Btu (gross) per cu ft. A typical analysis is given in Table 6.

Manufactured gases commonly produced are listed in Table 6. Gross, or higher, calorific values of typical send-out gases made from these manufactured gases generally range from 500 to 600 Btu per cu ft. Due largely to the greatly increased demand for city gases the tendency during recent years has been to increase rather than to decrease heat content of manufactured gases, thus making it possible to serve more customers through the existing distribution system.

Mixed gases are a result of increased distribution of natural gas, through transcontinental transmission lines, into areas having existing manufactured gas facilities. In such instances some gas companies supply a 600 to 800 Btu mixture (See Table 6). In some territories these mixtures are distributed as an intermediate step in changing over from manufactured gas to natural gas. Although the burden of adjusting installed heating and air conditioning equipment and supplying new orifices and burner equipment is generally assumed by the gas companies when the gas is changed, it is advisable to consult the local gas company to insure that equipment is provided with proper orifices and burners when installed.

Most states enforce legislation through their public service commissions to require delivery of a gas of specified average or minimum heating value within their respective limits. Any given heating value within reason fortunately may be maintained and yet permit considerable latitude in the composition of the gas distributed. Hence the constituents of city gases are not necessarily the same in different districts nor even at successive stations in the same district. In every community, however, the objective is to maintain variations in composition and gas pressure within limits which will provide satisfactory operation and performance of all common types of gas burning equipment.

Liquefied petroleum gases, such as propane and butane or mixtures thereof have calorific values ranging from 2500 to 3200 Btu per cu ft. These fuels are often supplied as liquids under pressure in tanks or bottles. In such cases the liquid evaporates when pressure is relieved, the heat necessary for vaporization being obtained from the surrounding air or ground. As butane boils at 32 F some provision is necessary, for maintaining the gas above this temperature or for lowering the partial pressure by dilution if the gas is utilized in colder climates. Propane, with a boiling point of −40 F, may be served in localities where temperatures substantially below freezing are encountered. When employed for heating purposes, these gases are usually stored in high pressure tanks and delivered by tank truck in much the same manner as fuel oil. Both gases, mixed with air, or in undiluted form, are also extensively employed by gas companies to augment their base load supplies during peak load periods. In some smaller communities, where gas manufacturing plants are not economically feasible and natural gas is unavailable, liquefied petroleum gases or liquefied petroleum gas-air mixtures are supplied through mains in much the same manner as manufactured or natural gas.

COMBUSTION OF GASEOUS FUELS

Gas burners employed in domestic heating appliances are generally of the non-luminous flame or Bunsen type. Part of the air required for combustion is inspirated as *primary air* into the burner mixing tube where it mixes with gas, and then takes part in combustion at the burner ports. As the amount of primary air is seldom sufficient to support complete combustion, additional air is supplied to the burner flames around the periphery of its ports. This *secondary air* is induced into the appliance and around the base of each separate burner flame by force of the issuing mixture of gas and primary air and by draft inspiration inherent in the heat of the flames. If a Bunsen type burner is properly adjusted, its flames will generally have a clear, blue appearance. Yellow flames are indicative of insufficient primary air supply, and possibly of incomplete combustion. An appreciable updraft is seldom, if ever, present even in flue-connected gas heating appliances, because most appliances of this kind are equipped with a draft hood which reduces the chimney draft at the appliance. It is important to note that gas furnaces and boilers, as well as most other classes of heating equipment, are designed to create their own draft.

The air-to-gas ratio in a Bunsen burner head has a decided effect on the rate of flame propagation. The gas-air mixture must flow from the burner ports faster than the flame burns, otherwise flashback will occur. The flashback condition normally results either from an excess amount of primary air or insufficient gas or both. Conversely, the velocity of the issuing mixture must not be so high that the flame will be blown from the ports, a condition known as *lifting*. Fortunately, contemporary types of such burners have a rather wide range of flexibility in capacity and adjustment. In addition to this characteristic, gas supply is normally so uniform that if a gas heating appliance is properly adjusted when it is installed, its burners, with occasional cleaning, should provide trouble-free service for years. When problems incident to changeover of the gas supply are involved, they are generally asssmed by the local utility providing the supply of gas. It should be recognized that a change in fuel gas will change the operating CO_2 value. For example, an appliance operating on carburetted water gas at 20 percent excess air will have 14.2 percent CO_2 in the flue gases. If a change is made to coke oven gas at the same gas input rate, with the excess air maintained at 20 percent, the operating CO_2 would drop to 9.2 percent.

Luminous flame burners are occasionally used in central heating gas appliances. With these devices all air required for complete combustion is supplied to the flames as secondary air. Two fundamental advantages of this type of burner are that the possibilities of flashback are eliminated, and that a much higher gas velocity is needed to blow the flames off the ports. On the other hand, if there is any appreciable amount of flame impingement on any portion of the heating surface, or if secondary air is not effectively supplied to the flames, soot may be formed and also combustion may be incomplete.

In some types of gas burners radiant baffles are used to convert part of the energy formed during the process of combustion to radiant heat. These baffles may also serve to direct the flow of products of combustion along the heat-absorbing surface.

Gas designed furnaces and boilers approved by the *American Gas Association* are certified for operation at the rating shown on the nameplate. Considerations relating to safety, performance, and service life, require that such appliances be adjusted at inputs which do not exceed the nameplate input rating. These appliances normally draw in from 20 to 50 percent excess air, depending on the type and general design. As has been indicated, some excess air is necessary to insure complete combustion at all times and also to provide a reasonable degree of flexibility in performance.

Various types of appliances used for gas space heating purposes are described in Chapter 14.

Care must be exercised to insure adequate air supply for combustion equipment installed in buildings or other structures. Where the equipment is closely confined, as in closets or small furnace or boiler rooms, the air supply must also provide for ventilation. Current recommended practices are:

1. Where the equipment is not closely confined (typical cellar installation or equivalent) provide not less than 1 sq in. of free access to outside air for each 1000 Btu per hr heat release of fuel consumed. Infiltration into conventional frame or brick construction, unless unusually tight with storm windows and tight doors, provides adequate air.

2. Where the equipment is closely confined, provide two openings to outside air or from spaces freely communicating to the outside. One opening should be near the top of the equipment enclosure and the other near the bottom. Each opening should have not less than ½ sq in. of free area for each 1000 Btu per hr heat release of fuel burned, and should communicate to air source or outside by suitable ducts. Where the enclosure is in a building of otherwise conventional construction, and the air source will be the normal infiltration into building, each opening shall have not less than 1 sq in. of free area for each 1000 Bu per hr heat release.

3. Clearances from equipment to closely confining enclosures should conform to local codes and to standards listed by recognized agencies such as *AGA Testing Laboratories*[19] and *Underwriters' Laboratories*.[20] See Fig. 5 for details concerning installation of gas burning equipment.[19] For installation of equipment burning other types of fuel, refer to the *National Building Code* of the *N.B.F.U.*

4. Certain additional precautions may be required for certain fuels (such as undiluted liquefied petroleum gas) and consequently provisions of local codes and other authoritative agencies should be followed.

FUNDAMENTAL PRINCIPLES OF COMBUSTION

Regardless of the type of fuel under consideration, its combustion results in the production of gaseous products. Many kinds of solid fuels contain minerals which cannot be burned and are therefore left as a residue commonly called ash. Moreover, unless sufficiently high temperatures are employed and an ample supply of oxygen properly distributed is present, the combustible constituents of solid, liquid, and even gaseous fuels cannot be completely burned. Incomplete or partial combustion of all fuels produces toxic gases, such as carbon monoxide, with smaller quantities of aldehydes, ketones, and other hydroxylated hydrocarbon compounds.

This fact indicates that, while combustion processes involving common types of fuel may be regulated by experienced operators to produce the most efficient results, normal combustion processes can be so unbalanced as to create hazards unless both design and operation are planned with a knowledge of the fundamental principles of combustion.

Combustion may be defined as the chemical combination of a substance with oxygen resulting in the evolution of heat, and usually some light. The rate of combustion depends upon the rate of reaction of the substance with oxygen, upon the rate at which oxygen is supplied, and upon the temperature obtained due to surrounding conditions. This is combustion in its simplest form. All solid, as well as liquid and even gaseous fuels generally contain several combustible elements in combination with others which, depending on their nature, affect oxygen requirements and thus govern

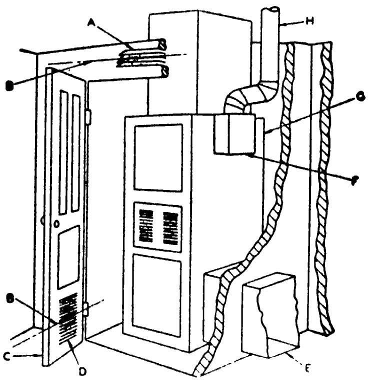


FIG. 5. ILLUSTRATION SHOWING AIR OPENINGS NECESSARY TO SUPPLY AIR FOR COMBUSTION WHEN APPLIANCE IS INSTALLED IN CONFINED SPACE

A. Ventilating air outlet register for furnace room, 1 sq in. free area for each 1000 Btu per hr furnace input located above relief opening of draft hood. Register must not be blocked by drapes or other furnishings.

B. Both registers must either face same large ventilated interior space or extend to such space by means of ducts. Vertical distance C/L to C/L of registers should be not less than 3½ ft.

C. Suggest room access door be not less than 6 ft high by a width sufficient to provide for installation or removal of furnace. At least 2 ft horizontal clearance should be provided in front of furnace when closet door is open, or 18 in. when door is closed.

D. Combustion and ventilation air inlet register for furnace room, 1 sq in. free area for each 1000 Btu per hr furnace input, located at or below combustion air inlet to furnace. Register must not be blocked by drapes or other furnishings.

E. Air circulated by furnace must be handled by ducts which are sealed to furnace casing and are entirely separate from means provided for supplying combustion and ventilation air.

F. Spacing between draft hood and wall at least 6 in. (unless approved for closer spacing). If flue products may be directed toward wall, 12 in. spacing recommended.

G. No part of furnace casing closer than 6 in. to wall (unless approved for closer spacing).

H. Flue should terminate above peak of roof and above nearby walls to assure satisfactory flue performance. In installations where the flue terminal is below nearby walls or roof peaks, an effective vent cowl should be used.

the combustion process. For a continuous reaction, as in heating processes, it is necessary to establish an effective balance between rates of removing heat and of supplying fuel and air or oxygen to keep the reaction going. In establishing such a balance, consideration must be given to the removal or venting of products of combustion, so that the entire process is one of flow wherein draft conditions in the combustion space play an important part.

Complete combustion is obtained when all combustible elements in a fuel are oxidized by all of the oxygen with which they will combine. All oxygen or air supplied is generally not utilized, and this excess portion is

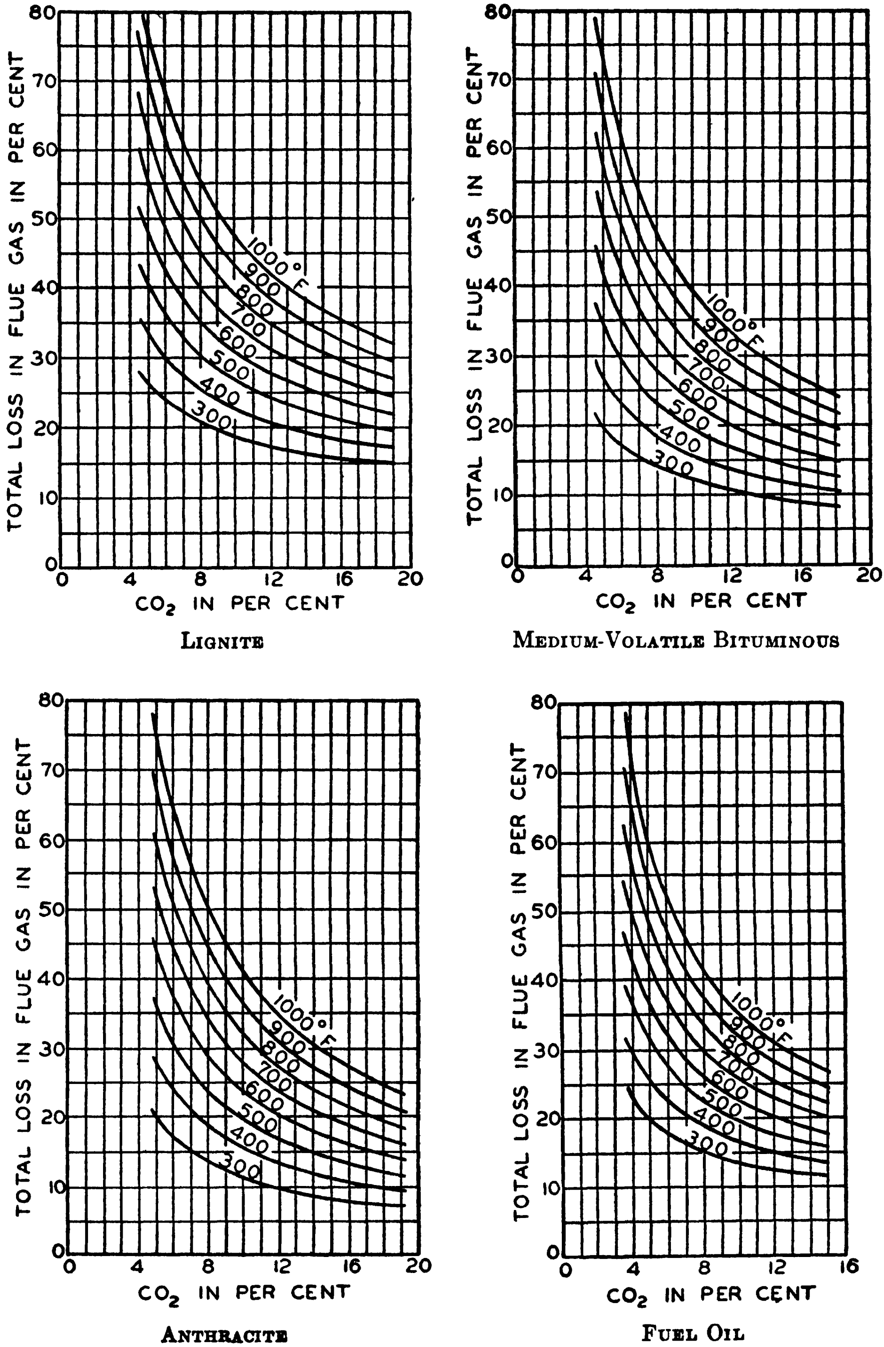
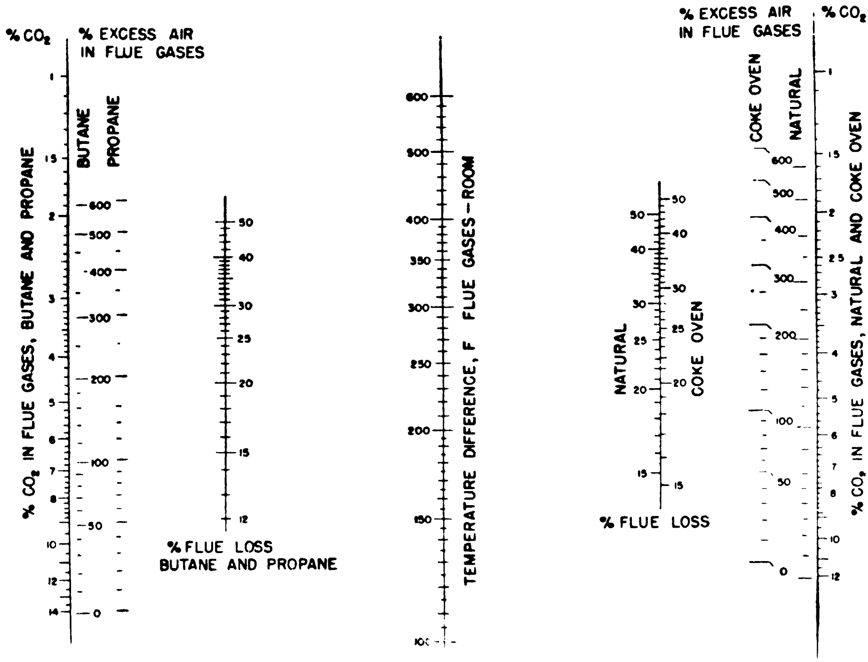


FIG. 6. FLUE GAS LOSSES WITH VARIOUS FUELS[2]

Flue Gas Temperature Shown. Loss is Based on 65 F Room Temperature.

commonly referred to as *excess oxygen* or *excess air*. Excess air is usually expressed as a percentage of the of air required for perfect combustion.

Perfect combustion results when the exact amount of oxygen required for complete combustion of all elements of a fuel is supplied and utilized. The percentage of carbon dioxide contained in the products of combustion from such a reaction is obviously the maximum attainable and is referred to as the ultimate CO_2 or *maximum theoretical percentage of carbon dioxide*. This condition of perfect combustion, without having excess air or oxygen left from the reaction, is seldom, if ever, realized in practice. Most types



Adapted from *American Gas Association Laboratories Flue Loss Charts.*

FIG. 7. ALIGNMENT CHART FOR CALCULATION OF FLUE LOSSES FOR BUTANE, PROPANE, COKE OVEN, AND NATURAL GASES

of heating equipment must be sufficiently flexible in performance to provide complete combustion with not only variations in the quality of a fuel but also changes in the rate at which it is supplied. This situation makes it advisable, from a practical standpoint, to insure *complete combustion* but not *perfect combustion* in the sense expressed above. To attempt to do so would undoubtedly result eventually in unsatisfactory performance especially from a safety standpoint. Consequently, common types of heating equipment are usually designed, installed, and adjusted to operate with some excess air. The exact percentage of such air depends on the type of fuel being utilized, as well as anticipated variations in its quantity and quality. Despite these practical limitations, however, it should not be inferred that common types of fuels cannot be utilized economically. Reference to flue loss charts such as Figs. 6 and 7 for gas burning equipment and to the air requirements discussion in Chapter 14 shows that reasonable quantities of excess air can be used without appreciable reductions in operating efficiencies.

Oxygen combines with the combustible elements and compounds of any fuel in accordance with fixed laws. The reactions and resultant products of perfect combustion of common fuel constituents are set forth in Table 7. All of the oxygen required for combustion is normally obtained from the surrounding air, which is a mechanical mixture of nitrogen and oxygen with small amounts of carbon dioxide, water vapor and inert gases. For practical combustion calculations, air is considered to consist of 20.9 percent oxygen and 79.1 percent nitrogen by volume, and 23.15 percent oxygen and 76.85 percent nitrogen by weight. The nitrogen, being inert, passes through the reaction without change. Table 7 gives the air quantities corresponding to the oxygen required for perfect combustion.

Air supplied to the combustion reaction is in most instances introduced in two ways. *Primary air* is introduced through or with the fuel, and *secondary air* is supplied to the flames issuing from the fuel.

Incomplete combustion is obtained when any of the combustible elements are not completely oxidized in the combustion reaction. This condition not only represents inefficient use of the fuel but also presents a hazard because carbon monoxide is usually one of the products of incomplete combustion. For example, a hydrocarbon may not oxidize completely to carbon dioxide and water, as indicated in Table 7, but may also form alcohols, ketones, aldehydes, or carbon monoxide depending on where and how the reaction is interrupted. Too low a temperature (such as may be caused by flame impingement on a cold surface), a poor oxygen supply to the flames (due to insufficient or poorly located air supply, or smothering by products of combustion not properly vented), or insufficient mixing of the air and fuel, are the primary causes of incomplete combustion.

Heat of Combustion

As previously stated, the process of combustion results in the evolution of heat. The heat generated by the complete combustion of a unit of fuel is constant for a given combination of combustible elements and compounds, and is known as the *heat of combustion*, *calorific value*, or *heating value* of the fuel. The heat of combustion of the several substances found in the more common fuels is given in Table 7.

The calorific value of a fuel may be determined either by direct measurement of the heat evolved during combustion in a calorimeter, or it may be computed from the ultimate analysis and the heat of combustion of the several chemical elements in the fuel. When the heating value of a fuel is determined in a calorimeter, the water vapor is condensed and the latent heat of vaporization is included in the heating value of the fuel. The heating value so determined is termed the *gross* or *higher* heating value, and this is what is ordinarily meant when the heating value of a fuel is specified. In burning the fuel, however, the products of combustion are not cooled to the dew-point and the higher heating value cannot be utilized.

When combustion is complete, the carbon in the fuel unites with oxygen to form carbon dioxide, CO_2, the hydrogen unites with oxygen to form water vapor, H_2O, and the nitrogen, being inert, passes through the reaction without change. When combustion is incomplete, some of the carbon may unite with oxygen to form carbon monoxide, CO, and some of the hydrogen and hydrocarbon gases may not be burned at all. When carbon monoxide or other combustible gases are present in the flue gases, there is a loss of heat produced per unit of fuel consumed, and a lower

TABLE 8. APPROXIMATE AIR REQUIREMENTS FOR THEORETICALLY PERFECT COMBUSTION OF FUELS[a]

TYPE OF FUEL	AIR REQUIRED FOR PERFECT COMBUSTION		APPROXIMATE PRECISION, PER CENT	EXCEPTIONS
	Lbs per Lb Fuel	Cu Ft per Unit[b] Fuel		
Solid	Btu per lb × 0.00073	Btu per lb × 0.0097	3	Fuels containing more than 30% water
Liquid	Btu per lb × 0.00071	Btu per lb × 0.0094	3	Results low for gasoline and kerosene
Gas	Btu per lb × 0.00067	Btu per cu ft × 0.0089	5	Gases of 300 Btu per cu ft or less

[a] Values in table taken from page 276 of *Gaseous Fuels*, 1948, published by *American Gas Association.*
[b] Units for solid and liquid fuels in pounds, for gas in cubic feet.

2. *Fuel Oil* (Pounds air per gallon): Commercial Standard No. 1, 102.6; No. 2, 105.5; No. 5, 112; No. 6, 114.2.

3. *Gaseous Fuels* (Cubic feet of air per cubic foot): Natural, 10.0; Mixed Natural and Manufactured, 8.0; Manufactured, 4.7, Propane, 23.8, Butane, 31.0.

COMBUSTION EFFICIENCY FROM THE FLUE GAS ANALYSIS

Excess Air

A commonly employed index of efficiency of combustion is the relation existing between the amount of air theoretically required for perfect combustion and the amount of air actually supplied. Since the difference between air supplied for combustion and theoretical air required is characterized as excess air, its percentage may be calculated by use of the following equation,

$$\text{Percent excess air} = \left(\frac{\text{Air supplied} - \text{Theoretical air}}{\text{Theoretical air}}\right) \times 100 \qquad (5)$$

The amount of dry air supplied per pound of fuel burned may be obtained from Equation 6 which has reasonable precision for most solid and liquid fuels. Values for CO_2, CO and N_2 are percentages by volume from the flue gas analysis, and C is the weight of carbon burned per pound of fuel, corrected for carbon in the ash.

$$\text{Pounds dry air supplied per pound of fuel} \times \frac{3.04N_2}{(CO_2 + CO)} \times C \qquad (6)$$

Because excess air calculations are almost invariably made from Orsat analysis results, and theoretical air requirements are not always known, another convenient method of expressing the relation of Equation 5 is as follows:

$$\text{Percent excess air} = \frac{100(O_2 - CO/2)}{N_2 \times 0.264 - (O_2 - CO/2)} \qquad (7)$$

As measurement standards for gaseous fuels are almost universally expressed in cubic feet, Equation 8 may be employed for computing excess air on a percentage basis for gases.[21]

$$\text{Percent excess air} = \frac{(U - CO_2)}{CO_2} \times 100\frac{P}{A} \qquad (8)$$

TABLE 9. APPROXIMATE MAXIMUM THEORETICAL CO_2 VALUES, AND CO_2 VALUES FOR VARIOUS FUELS WITH DIFFERENT PERCENTAGES OF EXCESS AIR

Type of Fuel	Maximum Theoretical or Ultimate Percent CO_2	Percent CO_2 at Given Excess Air Values		
		20%	40%	60%
Coke	21.0	17.5	15.0	13.0
Anthracite	20.2	16.8	14.4	12.6
Bituminous Coal	18.2	15.1	12.9	11.3
No. 1 and 2 Fuel Oil	15 0	12.3	10.5	9.1
No. 6 Fuel Oil	16.5	13.6	11.6	10.1
Natural Gas	12.1	9.9	8.4	7.3
Carburetted Water Gas	17.2	14.2	12 1	10 6
Coke Oven Gas	11.2	9.2	7 8	6 8
Mixed Gas (Natural and Carburetted Water Gas)	15.3	12.5	10.5	9.1
Propane Gas (Commercial)	13.9	11.4	9.6	8.4
Butane Gas (Commercial)	14.1	11.6	9.8	8.5

where

- U = ultimate carbon dioxide, percent of flue gases resulting from perfect combustion.
- CO_2 = carbon dioxide content of flue gases, percent.
- P = dry products from perfect combustion, cubic feet per cubic foot of gas burned.
- A = air theoretically required for complete combustion, cubic feet per cubic foot of gas burned.

As the ratio of P/A is approximately 0.9 for most city gases, a value of 90 may be substituted for $100\frac{P}{A}$ in Equation 8 for rough calculation.

Carbon-hydrogen ratios of different fuels vary considerably, hence the maximum or ultimate CO_2 attainable also varies. Where they are unknown, theoretical maximum CO_2 values may be calculated from a flue gas analysis by use of Equation 9.

$$\text{Maximum theoretical \% } CO_2 = \frac{\% \; CO_2 \text{ in flue gas sample} \times 100}{100 - \left(\dfrac{O_2 \text{ in same sample}}{0.21}\right)} \tag{9}$$

Approximate maximum CO_2 values for perfect combustion of several common types of fuel are shown in Table 9 together with values of CO_2 that will be attained with different amounts of excess air. Desirable values to be attained in practice depend upon the fuel, the method of firing, and other considerations. In general, fuels burned in suspension, such as gas, oil, and pulverized coal, can be burned with a lower amount of excess air than fuels burned on grates.

To produce heat efficiently by burning any common fuel a number of basic requirements must be met: (1) adequate heat absorbing surface of proper shape and construction is necessary in the appliance, (2) heat transfer surfaces must be clean, (3) a minimum amount of excess air must be present, (4) air employed for combustion and combustible gases must be properly mixed, and (5) flue gas losses must be reduced to a safe minimum.

If insufficient heating surface is employed, or if heat transfer surfaces are covered with soot, ash, or scale, flue losses will generally be excessive a *heat balance*. Various components of this balance are generally expressed in terms of Btu per pound of fuel burned, or as a percentage of its calorific value. Components of special interest are listed as items 1 to 7 inclusive.

1. Useful heat transferred to heating medium, usually computed by determining the rate of flow of the heating fluid through the heating device, and the change in enthalpy of the fluid (heat added) between the inlet and outlet.

2. Heat loss in the dry chimney gases.

$$h_1 = w_g c_p \ (t_g - t_a) \tag{12}$$

3. Heat loss in water vapor formed by the combustion of hydrogen.

$$h_2 = \frac{9H_2}{100} \ (1091.8 + 0.455 \ (t_g - t_a) \tag{13}$$

4. Heat loss in water vapor in the air supplied for combustion.

$$h_3 = 0.455 \ M \ w_a \ (t_g - t_a) \tag{14}$$

5. Heat loss from incomplete combustion.

$$h_4 = 10143\,C \left(\frac{CO}{CO_2 + CO} \right) \tag{15}$$

6. Heat loss from unburned carbon in the ash or refuse.

$$h_5 = 14600^* \left(\frac{C_u}{100} - C \right) \tag{16}$$

7. Radiation and all other unaccounted for losses.

Radiation and convection losses from a heating appliance are not usually determined by direct measurement. For this reason they, together with any other losses not measured, are determined by subtracting the total of items 1 to 6 from the heat of combustion of the fuel. If the heating appliance is located within the heated space, however, radiation and convection losses may be considered as useful heat rather than lost heat. They may, therefore, be omitted from calculations of heat losses, or added to item 1. If there is *CO* in the flue gases, small amounts of unburned hydrogen and hydrocarbons will probably also be present. The small loses due to incomplete combustion of these latter gases would also be included in item 7.

Symbols used in Equations 12 to 16 inclusive are:

h_1 = heat loss in the dry chimney gases, Btu per pound of fuel.
h_2 = heat loss in water vapor from combustion of hydrogen, Btu per pound of fuel.
h_3 = heat loss in water vapor in combustion air, Btu per pound of fuel.
h_4 = heat loss from incomplete combustion of carbon, Btu per pound of fuel.
h_5 = heat loss from unburned carbon in the ash, Btu per pound of fuel.
w_g = weight of dry flue gas per pound of fuel (from Equation 10), pounds.
c_p = mean specific heat of flue gases at constant pressure (c_p ranges from 0.242 to 0.254 for flue gas temperatures from 300 F to 1000 F)[2], Btu per pound.
t_g = temperature of flue gases at exit of heating device, Fahrenheit.
t_a = temperature of combustion air, Fahrenheit.
H_2 = percentage of hydrogen in fuel by weight from ultimate analysis of fuel burned.
1091.8 = enthalpy of saturated water vapor at a temperature of 70 F, Btu per pound.
M = humidity ratio of combustion air, pounds of water vapor per pound of dry air.

* A value of 14600 applies in calculating ash pit loss; in calculating heat of formation of carbon compounds use 14093 Btu per lb.

w_a = weight of combustion air per pound of fuel used, pounds, from Equations 2, 4, 5, 6, 7 and 8.
CO, CO_2 = percentages of CO, CO_2 in flue gases by volume.
C = weight of carbon burned per pound of fuel corrected for carbon in ash, pounds.

$$C = \frac{WC_u - W_aC_a}{100W} \tag{17}$$

where

C_u = percentage of carbon in the fuel by weight from the ultimate analysis.
W_a = weight of ash and refuse, pounds.
C_a = percent of combustible in ash by weight (combustible in ash is usually considered to be carbon).
W = weight of fuel used, pounds.

Flue gas losses for solid and liquid fuels, listed as items 2, 3 and 4 of the heat balance, may be determined with sufficient precision for most purposes from curves shown in Fig. 6[2], if CO_2 content and temperature of flue gases are known. Values of the losses plotted for fuel oil were computed from the ultimate analysis of a typical fuel oil used in domestic burners, while those presented for the several ranks of coal were computed from the typical ultimate analyses shown in Table 1. The curves for medium volatile bituminous coal may be used for high volatile bituminous coal with negligible error.

Utilization of gaseous fuels, for numerous reasons, is generally a more simple process than is the case with either solid or liquid fuels. Accordingly, the determination of a practical heat balance is also a more simple procedure in that items 5 and 6 do not generally apply to gas installations. A series of typical alignment charts has been combined in Fig. 7 for use in determining flue losses of items 2, 3 and 4 from common types of gas burning appliances. To determine flue losses place a straight edge extending from the corrected temperature reading to the percent CO_2 recorded. Percent flue loss is indicated where the straight edge interesects the flue loss column. The operating efficiency of a gas appliance can then be computed with sufficient precision by application of Equation 18.

Percent Combustion Efficiency =

$$\frac{\begin{pmatrix}\text{Gross Btu of fuel} \\ \text{gas per cubic foot}\end{pmatrix} - \begin{pmatrix}\text{total flue losses per} \\ \text{cubic foot fuel gas}\end{pmatrix}}{\text{Gross Btu of fuel gas per cubic foot}} \times 100 \qquad (18$$

Reference to Table 9 will show that ultimate CO_2 percentage values of fuel gases vary. While personal errors involved in CO_2, temperature, and chart determinations, would doubtless more than offset any inaccuracies due to universal use of the alignment charts shown, precise laboratory work may require a more exact method. For more complete information the reader is referred to *Combustion*, 3rd Edition, and *Gaseous Fuels*, (published by *American Gas Association*) and particularly to tables covering various properties of different commercial gases included in these publications.

CONDENSATION AND CORROSION

Sulfur dioxide or sulfur trioxide, formed by the combustion of sulfur in fuels, are the principal corroding substances in flue gases. They become active whenever sufficient moisture is present for the formation of sulfurous or sulfuric acid,[22] and they lower the dew points of flue gases appreciably.

TABLE 10. AVERAGE FLUE GAS DEW-POINT FOR VARIOUS FUELS[8]

TYPE OF FUEL	AVERAGE DEW-POINT TEMPERATURE, F
Anthracite	68
Semi-Bituminous Coal	84
Bituminous Coal	93
Oil	111
Natural Gas	127
Manufactured Gas	137
Propane Gas (2500 Btu/cu ft)	119
Butane Gas (3200 Btu/cu ft)	124
Butane-Air Gas Mixture (535 Btu/cu ft	121

Therefore, unless heating equipment is designed for operation at flue gas temperatures below the dewpoint, which is seldom the case, it is always advisable to maintain temperatures above this value in all parts of the appliance. Excessive spot temperatures in the combustion chamber or elsewhere, on the other hand, are also destructive in that they may result in rapid oxidation of ordinary heating surfaces. *American Standard Requirements* for gas furnaces, floor furnaces, and recessed heaters, for example, specify that minimum spot heating surface temperatures during normal operation must neither fall below 178 F (50 F above average dewpoint) nor exceed 875 F on any portion of the heating surface. In any event it is usually desirable to maintain flue temperatures within the limits indicated not only throughout the appliance, but in its connecting vent, flue, or chimney as well. Otherwise, excessive condensation and corrosion problems, with resultant customer dissatisfaction, will in all probability be the result. Average dewpoint temperatures of flue gases resulting from the combustion of various fuels, when burned with the amount of excess air normally supplied to insure complete combustion, are shown in Table 10.

SOOT

The deposit of soot on the flue surfaces of a boiler or heater acts as an insulating layer over the surface, and reduces the heat transmission to the water or air. The *Bureau of Mines Report of Investigations* No. 3272[23] shows that the loss of seasonal efficiency is not so great as has been believed, and usually is not over 6 percent because the greater part of the heat is transmitted through the combustion chamber surfaces. The *Bureau of Standards Report* BMS 54[24] points out that, although the decrease in efficiency of an oil fired boiler, due to soot deposits, is relatively small, the attendant increase in stack temperature may be considerable.

The soot accumulation clogs the flues, reduces the draft, and may prevent proper combustion. Soot can probably be most effectively removed by a jet of compressed air, by means of a brush, or a vacuum cleaner. However, it has been found that copper chloride, lead chloride, tin chloride, zinc chloride, common salt and some other salts are partially effective in removing soot from furnaces and boilers when properly used.[25] A discussion of instruments and methods of evaluating smoke will be found in Chapter 49.

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CHAPTER 14

AUTOMATIC FUEL BURNING EQUIPMENT

Classification of Stokers, Combustion Process and Adjustments, Furnace Design, Rating; Classification of Oil Burners, Combustion Process, Combustion Chamber Design; Classification of Gas-Fired Heating Equipment, Combustion Process, Ratings; Sizing of Gas Piping, Fuel Burning Rates

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AUTOMATIC mechanical equipment for the combustion of solid, liquid, and gaseous fuels is considered in this chapter.

MECHANICAL STOKERS

A mechanical stoker is a device that feeds a solid fuel into a combustion chamber, provides a supply of air for burning the fuel under automatic control and, in some cases, incorporates a means of removing the ash and refuse of combustion automatically. Coal can be burned more efficiently by a mechanical stoker than by hand firing because the stoker provides a uniform rate of fuel feed, better distribution in the fuel bed and positive control of the air supplied for combustion.

Classification of Stokers According to Capacity

Stokers may be classified according to their coal feeding rates. The following classification has been made by the *U. S. Department of Commerce*, in cooperation with the *Stoker Manufacturers Association.*

Class 1. Capacity under 61 lb of coal per hour.
Class 2. Capacity 61 to 100 lb of coal per hour.
Class 3. Capacity 101 to 300 lb of coal per hour.
Class 4. Capacity 300 to 1200 lb of coal per hour.
Class 5. Capacity 1200 lb of coal per hour and over.

Class 1 Stokers

These stokers are used primarily for home heating and are designed for quiet, automatic operation. Simple, trouble-free construction and attractive appearance are desirable characteristics of these small units.

A common stoker in this class (Fig. 1) consists essentially of a coal hopper, a screw for conveying the coal from the hopper to the retort, a fan which supplies the air for combustion, a transmission for driving the coal feed worm, and an electric motor for supplying power for coal feed and air supply.

Air for combustion is admitted to the fuel through tuyeres at the top of the retort which may be either round or rectangular. Stokers in this class are made for burning anthracite, bituminous, semi-bituminous, and lignite coals, and coke. The *U. S. Department of Commerce* has issued commercial standards for household anthracite stokers.[1]

Units are available in either the hopper type, as shown in Fig. 1, or in the bin-feed type as shown in Figs. 2 and 3. Some stokers, particularly those designed for use with anthracite, automatically remove ash from the ash pit and deposit it in an ash receptacle as shown in Fig. 3. Most of the bituminous models, however, require removal of the ash from the fuel bed after it is fused into a clinker.

Stokers in this class feed coal to the furnace intermittently in accordance with temperature or pressure demands. A special control is used to insure sufficient stoker operation to maintain a fire during periods when no heat is required. Where year-round domestic hot water is supplied by a boiler and indirect water heater connected to a storage tank, the stoker will usually be called on to operate often enough to maintain the fire.

Stoker-Fired Boiler and Furnace Units

Boilers, air conditioners, and space heaters especially designed for stokers are available having design features closely coordinating the heat absorber and the stoker. Although efficient and satisfactory performance can be obtained from the application of stokers to existing boilers and

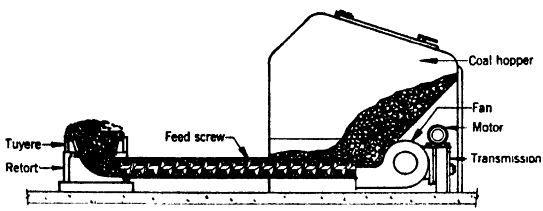

FIG. 1. UNDERFEED STOKER, HOPPER TYPE, CLASS 1

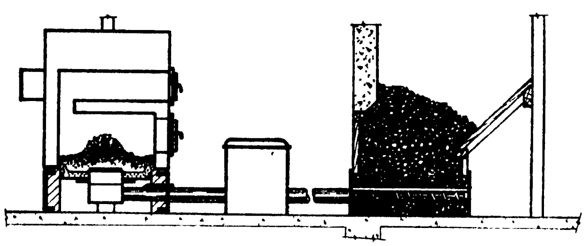

FIG. 2. UNDERFEED STOKER, BIN FEED TYPE, CLASS 1

furnaces, some of the combination stoker-fired units (Fig. 4) are more compact and attractive in appearance.

Class 2 and 3 Stokers

Stokers in this class are usually of the screw feed type without auxiliary plungers or other means of distributing the coal. They are used extensively for heating plants in apartments and hotels, also, for industrial plants. They are of the underfeed type and are available in both the hopper type, as illustrated in Fig. 5, and the bin feed type, shown in Fig. 6. These units also are built in plunger feed type with an electric motor or a steam or hydraulic cylinder coal feed drive.

Stokers in this class are available for burning all types of anthracite, bituminous and lignite coals. The tuyere and retort design varies according to the fuel and load conditions Stationary type grates are used on bituminous models, and the clinkers formed from the ash accumulate on the grates surrounding the retort.

Anthracite stokers in this class are equipped with moving grates which discharge the ash into a pit below the grate. This ash pit may be located on one or both sides of the grate and, in some installations, is of sufficient capacity to hold the ash for several weeks' operation.

Class 4 Stokers

Stokers in this group vary widely in details of design, and several methods of feeding coal are employed. The underfeed stoker is widely used, although a number of the overfeed types are used in the larger sizes. Bin-feed, as well as hopper models, are available in both underfeed and overfeed types.

Class 5 Stokers

The prevalent stokers in this field are: (1) underfeed side cleaning, (2) underfeed rear cleaning, (3) overfeed flat grate, and (4) overfeed inclined grate.

Underfeed side cleaning stokers are made in sizes up to approximately 500 boiler horsepower. They are not so varied in design as those in the smaller classes, although the principle of operation is similar. A stoker of this type is illustrated in Fig. 7.

The rear cleaning underfeed stoker is usually of the multiple retort design, and is used in some of the largest industrial plants and central power stations. Zoned air control has been applied to these stokers, both longitudinally and transversely of the grate surface.

The overfeed flat grate stoker is represented by the various chain—or traveling-grate stokers. A typical traveling-grate stoker is illustrated in Fig. 8.

Another distinct type of overfeed flat-grate stoker is the spreader (Figs. 9 and 10) type in which coal is distributed either by rotating paddles or by air over the entire grate surface. This type of stoker is adapted to a wide range of fuels and has a wide application on small sized fuels, and on fuels such as lignites, high-ash coals, and coke breeze.

The overfeed inclined-grate stoker operates on the same general combustion principle as the flat-grate stoker, the main difference being that rocking grates, set on an incline, are provided in the former to advance the fuel during combustion.

Combustion Process

In anthracite stokers of the Class 1 underfeed type, burning takes place entirely within the stoker retort. The refuse of combustion spills over the edge of the retort into an ash pit or receptacle from which it may be removed either manually or automatically.

Larger underfeed anthracite stokers operate on the same principle, except that the retort is rectangular and the refuse spills over only one or two sides of the grate. Anthracite for stoker firing is usually the No. 1 buckwheat or No. 2 buckwheat size.

Because the majority of the small bituminous coal stokers operate on the underfeed principle, a general description of their operation is given. When the coal is fed into the retort, it moves upward toward the zone of combustion and is heated by conduction and radiation from the burning fuel in the combustion zone. As the temperature of the coal rises, it gives off moisture and occluded gases, which are largely non-combustibles. When the temperature increases to around 700 or 800 F the coal particles become plastic, the degree of plasticity varying with the type of coal.

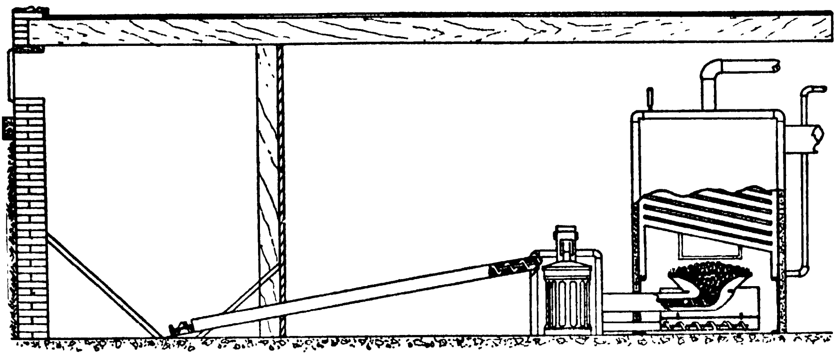

FIG. 3. UNDERFEED ANTHRACITE STOKER WITH AUTOMATIC ASH REMOVAL, BIN TYPE

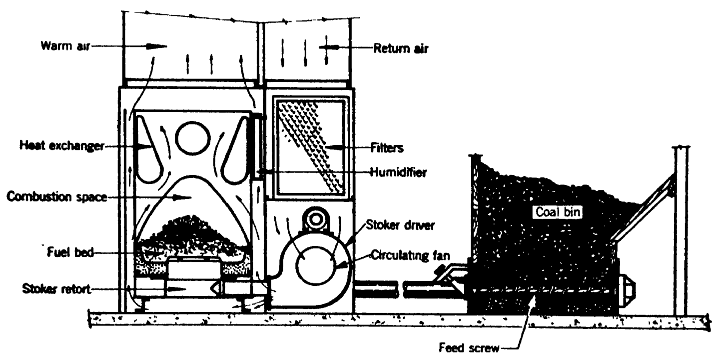

FIG. 4. STOKER-FIRED WINTER AIR CONDITIONING UNIT

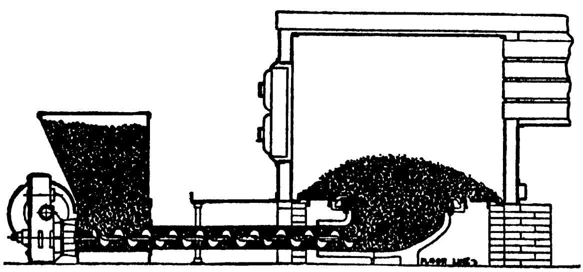

FIG. 5. UNDERFEED SCREW STOKER, HOPPER TYPE, CLASS 2, 3 OR 4

A rapid evolution of the combustible volatile matter occurs during and directly after the plastic stage. The distillation of volatile matter continues above the plastic zone where the coal is coked. The strength and porosity of the coke formed will vary according to the size and characteristics of the coal. While some of the ash fuses into particles on the surface of the coke as it is released, most of it remains on the hearth or grates and, as this ash layer becomes thicker with time, that portion exposed to the higher temperatures surrounding the retort fuses into a clinker. The

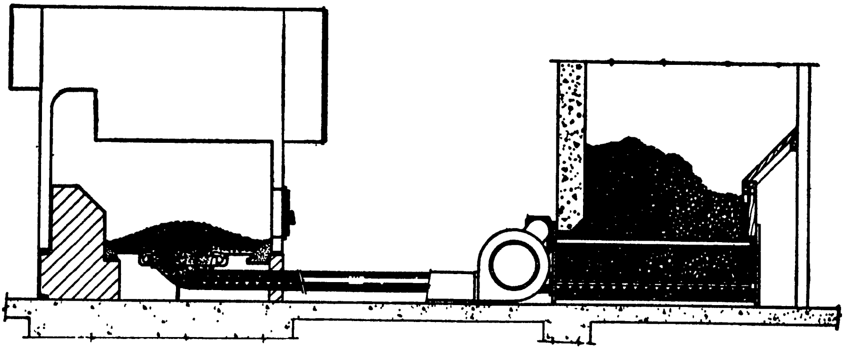

Fig. 6. Underfeed Screw Stoker, Bin Type, Class 2, 3 or 4

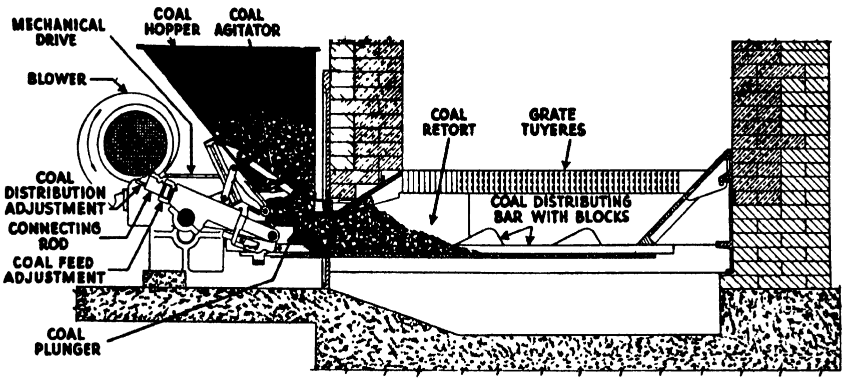

Fig. 7. Underfeed Side Cleaning Stoker

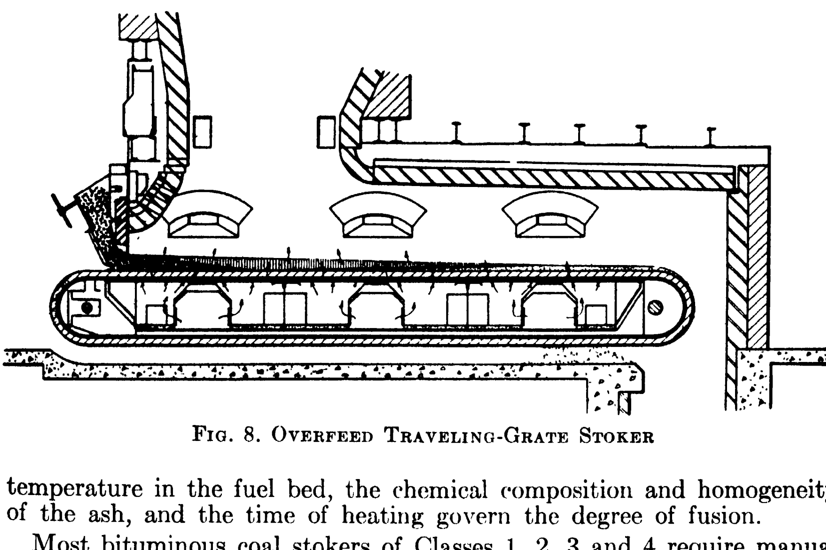

Fig. 8. Overfeed Traveling-Grate Stoker

temperature in the fuel bed, the chemical composition and homogeneity of the ash, and the time of heating govern the degree of fusion.

Most bituminous coal stokers of Classes 1, 2, 3 and 4 require manual removal of the ash in clinker form.

In the underfeed side-cleaning stokers the fuel is introduced at the front of the furnace to one or more retorts, and is advanced away from the retort as combustion progresses, while finally the ash is disposed of at the sides. This type of stoker is suitable for all bituminous coals, while in the smaller sizes it is suitable for small sizes of anthracite. In this type of stoker the fuel is delivered to a retort beneath the fire and is raised into the fire. During this process the volatile gases are released, are mixed with air, and pass through the fire where they are burned. The ash may be continuously or periodically discharged at the sides.

The underfeed rear-cleaning stoker accomplishes combustion in much the same manner as the side-cleaning type, but consists of several retorts placed side by side and filling up the furnace width, while the ash disposal is at the rear. In principle, its operation is the same as the side-cleaning underfeed type.

Overfeed flat-grate stokers receive fuel at the front of the grate in a

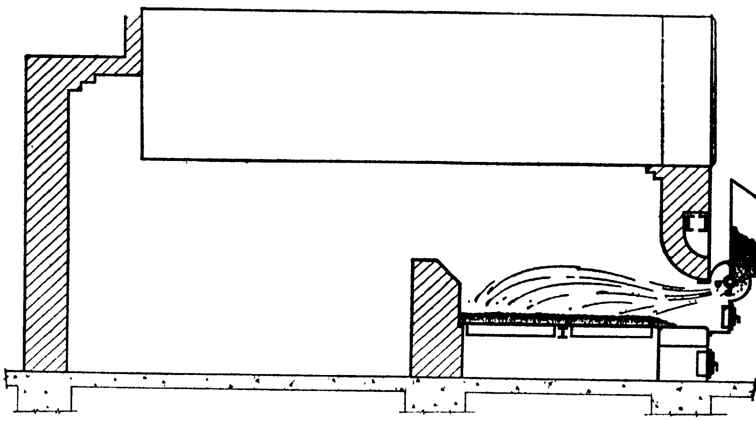

FIG. 9. OVERFEED SPREADER STOKER (ROTOR TYPE)

layer of uniform thickness and move it horizontally to the rear of the furnace. Air is supplied under the moving grate to carry on combustion at a sufficient rate to complete the burning of the coal near the rear of the furnace. The ash is carried over the back end of the stoker into an ash pit beneath. This type of stoker is suitable for small sizes of anthracite or coke breeze, and also for bituminous coals, the characteristics of which make it desirable to burn the fuel without disturbing it. This type of stoker requires an arch over the front of the fuel bed to maintain ignition of the incoming fuel, and frequently a rear combustion arch.

In addition to the use of rocking grates, the overfeed inclined-grate stoker is provided with an ash plate on which ash is accumulated and dumped periodically. This type of stoker is suitable for all types of coking fuels, but preferably for those of low volatile content. Its grate action keeps the fuel bed broken up, thereby allowing free passage of air. Because of its agitating effect on the fuel, it is not desirable for badly clinkering coals. It usually should be provided with a front arch to ignite the volatile gases.

Combustion Adjustments

The coal feeding rate and air supplv to the stoker should be regulated so as to maintain a balance between the load demand and the heat liberated by the fuel. Under such conditions, no manual attention to the fuel bed should be required other than the removal of clinker in stokers which operate on this principle of ash removal.

As in all combustion processes, the maintenance of the correct proportions of air and fuel is essential. It is desirable to supply the minimum amount of air required to properly burn the fuel at the rate of feed.

While there may be only slight variations in the rate at which the coal is being fed, due to variations in the size or density of the coal, there may be wide variations in the rate of air flow as the result of changes in fuel bed resistance. These changes in resistance may be caused by changes in the porosity of the fuel bed due to variations in size or friability of the coal, ash and clinker accumulation, and variations in depth of the fuel bed. Because of this variable fuel bed resistance, many bituminous stokers, even in the smaller domestic sizes, incorporate air controls which automatically com-

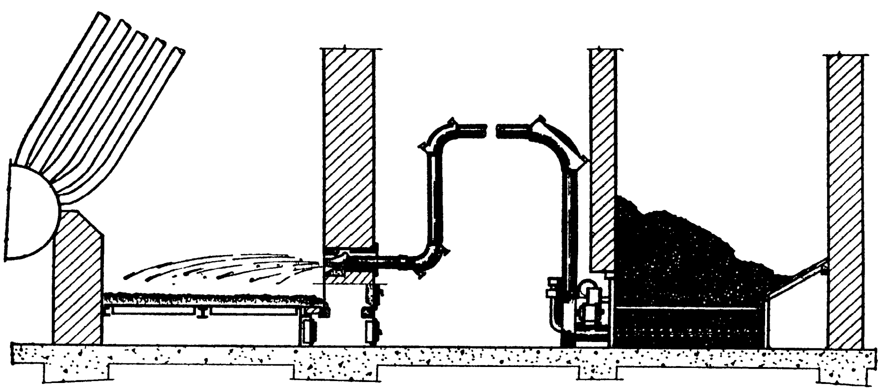

FIG. 10. OVERFEED SPREADER STOKER (PNEUMATIC TYPE)

pensate for these changes in resistance and maintain a constant air fuel ratio. The efficiency of combustion may be determined by analyzing the flue gases, as explained in Chapters 13 and 49.

It is desirable on most stoker installations to provide automatic draft regulation in order to reduce air infiltration and provide better control during the banking, or off, periods of the stoker.

Furnace Design

Although there is considerable variation in stoker, boiler, and furnace design, the stoker industry, from long-time experience, has established certain rules for the proportioning of furnaces for domestic and commercial stokers. The stoker installer and designer of stoker-fired equipment should give careful consideration to these factors.

The *Stoker Manufacturers Association* has published standard recommendations on setting heights for stokers having capacities up to 1200 lb of coal per hour.[2]

The empirical formulas for determining these setting heights are:

For burning rates up to 100 lb coal per hour

$$H = 0.1125\,B + 15.75 \qquad (1)$$

For burning rates from 100 to 1200 lb coal per hour

$$H = 0.03\,B + 24$$

where

H = minimum setting height, inches, measured from dead plates to crown sheet for steel boilers. For cast-iron boilers height may be $\frac{7}{8}\,H$.

B = burning rate coal per hour, pounds.

Standards for minimum firebox dimensions and base heights have been formulated by the *Stoker Manufacturers Association* as shown in Fig. 11.[2]

In considering these recommendations, it should be understood that they show the average recommended minimum. There are many factors affecting the proper application of stokers to various types of boilers and furnaces, and, in certain instances, setting height or firebox dimensions shown in the standards may be modified without impairing performance. Such modification will depend upon the experience of the installer or designer with a particular stoker, the type of fuel used, and the construction of the boiler or furnace.

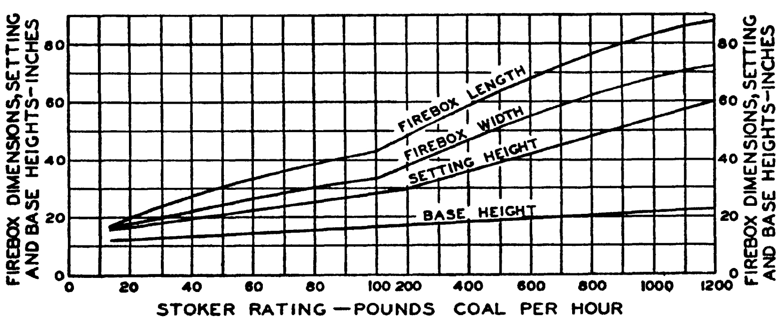

FIG. 11. SUGGESTED MINIMUM FIREBOX DIMENSIONS AND BASE HEIGHTS[a]

[a] For reference in selecting or designing boilers and furnaces for stoker firing. Dimensions shown are for net inside clearance at grate level using coal with heating value of not less than 12,000 Btu per pound. Under certain conditions smaller fireboxes will permit satisfactory performance but these dimensions are preferred normal minimums.

Installation of stokers (particularly smaller sizes) on the side of the boiler or furnace will sometimes facilitate clinker removal.

Rating and Sizing Stokers

The capacity or rating of small underfeed stokers is usually stated as the burning rate in pounds of coal per hour. Codes for establishing uniform methods of rating anthracite and bituminous coal stokers have been adopted by the *Stoker Manufacturers Association.*[3]

The Association also has adopted a uniform method of selecting stokers that is published in convenient tables and charts.[2] The required capacity of the stoker is calculated as follows:

$$\frac{\text{Load (Btu per hour)}}{\text{Heating value of coal (Btu per pound)} \times \text{overall efficiency of stoker and boiler or furnace}} = \text{Stoker burning rate required (pounds of coal per hour)}$$

In determining the total load placed on a stoker-fired boiler by a steam or hot water heating system, a piping and pick-up factor of 1.33 is commonly used in sizing the stoker, but this factor should be increased at times due to unusual conditions.

Controls

The heat delivery from the stoker of the smallest household type to the largest industrial unit can be regulated accurately with fully automatic controls. The smaller heating applications are controlled normally by a thermostat placed in the building to be heated. Limit controls are supplied to prevent excessive temperature or pressure from being developed in the furnace or boiler, and refueling controls are used to maintain ignition during periods of low heat demand. Automatic low water cut-outs are recommended for use with all automatically-fired steam boilers. (See Chapter 38.)

DOMESTIC OIL BURNERS

An oil burner is a mechanical device for producing heat automatically from liquid fuels. Two methods are employed for the preparation of the

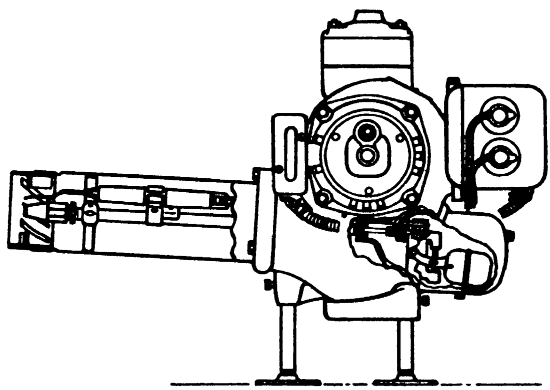

FIG. 12. LOW PRESSURE ATOMIZING OIL BURNER

oil for the combustion process; atomization, and vaporization. The simpler types of burners depend upon the natural chimney draft for supplying the air for combustion. Other burners provide mechanical air supply or a combination of atmospheric, and mechanical. Ignition is accomplished by an electrical spark or hot wire, or by an oil or gas pilot. Some burners utilize a combination of these methods. Continuously operating burners may use manual ignition. Burners of different types operate with luminous or non-luminous flame. Operation may be intermittent, continuous with high-low flame, or continuous with graduated flame.

Classification of Burners

Domestic oil burners may be classified by type of design or operation into the following groups: pressure atomizing or gun, rotary, and vaporizing or pot. These are further classified as mechanical draft, and natural draft.

Pressure Atomizing (Gun Type) Burner

Gun type burners may be divided into two classes: low-pressure and high-pressure atomization. In the first group, a mixture of oil and primary air is pumped as a spray through the nozzle at a pressure of 2 to 7 psi. Secondary air is supplied by a fan. Ignition is obtained by means of a high-voltage electric spark used alone, or as primary ignition for a gas pilot. Various features of a low pressure atomizing burner are shown in Fig. 12.

The high-pressure atomizing type, illustrated in Fig. 13, is characterized by an air tube, usually horizontal, with oil supply pipe centrally located in the tube and arranged so that a spray of atomized oil is introduced at about 100 psi, and mixed in the combustion chamber with the air stream emerging from the air tube. A variety of patented shapes is employed at the end of the air tube to influence the direction and speed of the air, and thus the effectiveness of the mixing process.

This type of burner utilizes a fan to supply the air for combustion, and ignition is established by a high-voltage electric spark that may be operative continuously while the burner is running, or just at the beginning of the

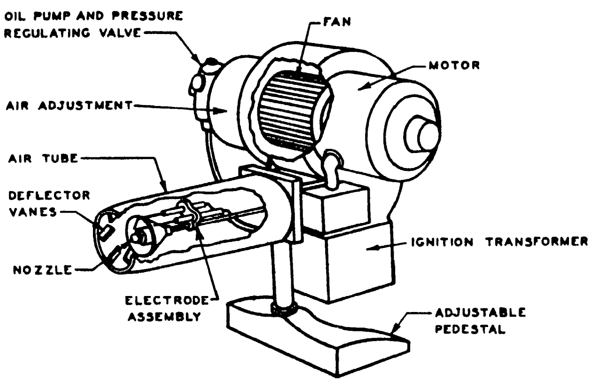

FIG. 13. HIGH-PRESSURE ATOMIZING OIL BURNER

running period. Gun type burners operate on the intermittent *on-off* principle, and with a luminous flame.

The combustion process is completed in a chamber constructed of refractory material, or stainless steel, this being a part of the installation. Pressure-atomizing burners generally use the distillate oils, No. 1 or 2 grade. (See Chapter 13.)

Rotary Type Burner

This class of burners may be divided into two groups: vertical and horizontal. Most of the smaller rotary burners are of the vertical type, and use a light distillate oil of No. 1 grade.

The most distinguishing feature of vertical rotary burners is the principle of flame application. These burners are of two general types: the center flame and wall flame. In the former type (Fig. 14), the oil is atomized by being thrown from the rim of a revolving disc or cup, and the flame burns in suspension with a characteristic yellow color. Combustion is supported by means of a bowl-shaped chamber or hearth. The wall flame burner (Fig. 15) differs in that combustion takes place in a ring of stainless steel or refractory material which is placed around the hearth. Dependent upon combustion adjustment, these burners may operate with either a semi-luminous or non-luminous flame.

Both types of vertical rotary burners are further characterized by their installation within the ash pit of the boiler or furnace. Various types of ignition are utilized, gas and electric, either spark or hot wire. The air for combustion is supplied partially by natural draft, and partially by fan effect of the central spinner element.

Horizontal rotary burners are used principally to burn the heavier oils, Nos. 5 and 6 grades, principally in larger commercial and industrial installations, although domestic sizes are available. Such burners are of the mechanical atomizing type, using rotating cups which throw the oil from the edge of the cup at high velocity into the surrounding stream of air delivered by the blower (Fig. 16).

Horizontal rotary burners commonly use a combination electric-gas ignition system, or are lighted manually. Primary air for combustion is supplied by a blower, and secondary air, often introduced through a

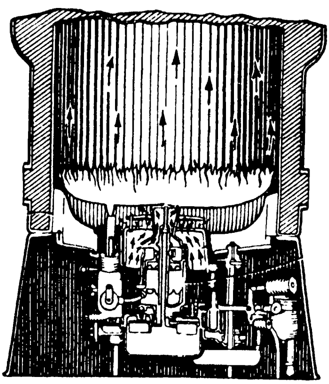

FIG. 14. CENTER FLAME VERTICAL ROTARY BURNER

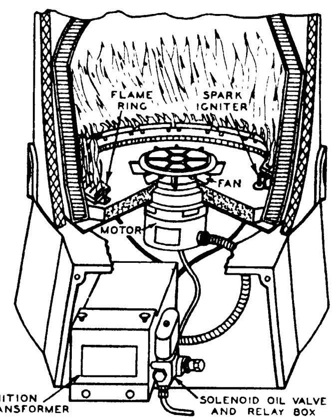

FIG. 15. WALL FLAME VERTICAL ROTARY BURNER

checkerwork in the combustion chamber, is controlled by chimney draft. These burners operate with a luminous flame, usually on high-low or continuous setting.

In larger installations, burners may be installed in multiple in a common combustion chamber. Because of the high viscosity oils used in these burners, it is customary to preheat the oil between the tank and the burner. Preheating when delivering from tank car, or truck, is often required in cold weather.

Vaporizing Burners

In the vaporizing burner, fuel oil is ignited (manually or electrically) and vaporized in a vessel or pot which is open at the top or one side. Heat for vaporization is supplied by the combustion process. Openings in the side walls of the burner admit primary air which forms a rich mixture of air and oil vapors in the burner. Adjacent to the outlet opening, sufficient additional or secondary air is admitted to complete combustion. The openings for admitting air are arranged to obtain gradual and intimate mixing of air and oil vapor for combustion, with a minimum amount of excess air and resulting high combustion efficiency.

Fuel is fed by gravity from a constant level control valve, and the flow is either *on* (at rated capacity) or *off* (at pilot flow), according to the demand of the thermostat. However, the high fire can be reduced and the

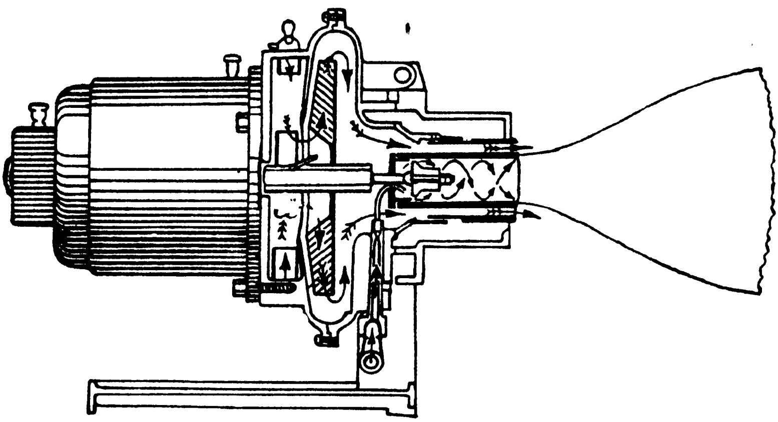

FIG. 16. HORIZONTAL ROTATING CUP OIL BURNER

pilot fire can be increased to give almost any desired control characteristic within the range of the burner. The majority of vaporizing burners are manufactured in sizes up to one gallon per hour input. Most vaporizing burners are limited to use with No. 1 fuel oil having a maximum end point of 625 F and a minimum A.P.I. gravity of 35 deg.

A barometric draft regulator is required to maintain the recommended draft. A draft of not more than 0.06 in. of water column is recommended for most natural draft burners. When burners are equipped with mechanical forced draft, a slightly lower chimney draft can be used. A burner of this type is illustrated in Fig. 17.

Vaporizing burners are adaptable to water heaters, space heaters, and furnaces. Some types have also been applied successfully to conversion installations. The heat output is in the range of requirements for the average or small home.

The modulating flame allows simple manual control by regulation of a metering valve, and simplifies the control equipment. Quiet combustion and the absence of moving parts contribute to quiet operation when the heating device is located in the living quarters.

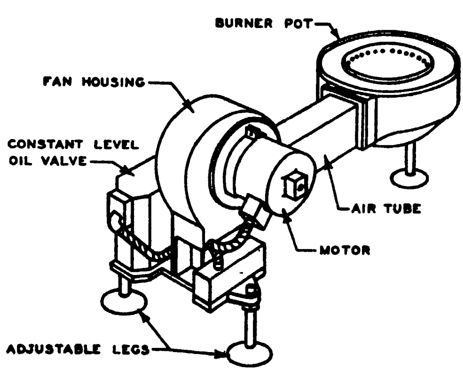

FIG. 17. VAPORIZING POT-TYPE BURNER

The ability to operate on natural draft and gravity feed of the fuel, makes possible the use of these burners where electric current is not available or is unreliable. However, most furnaces are thermostatically controlled, and many are provided with mechanical draft.

Oil-Fired Boiler and Furnace Units

A number of types of specially designed oil-fired boiler-burner and furnace-burner units are available. Various locations of burners will be noted in such units; some having the combustion chamber and burner at the top, some at the bottom, and some at the center of the appliance. One type of boiler-burner unit is shown in Fig. 18. The coordinated design of boiler (or furnace) and burner elements insures the optimum in operating characteristics, and the maintenance of balanced performance. This type of equipment usually has more heating surface, and better flue proportions and gas travel than conventional boilers or furnaces. Some

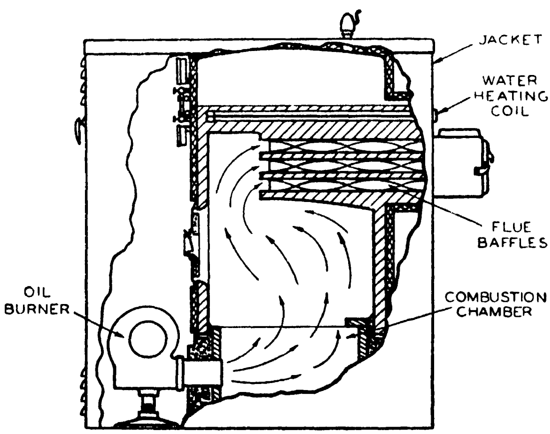

FIG. 18. TYPICAL BOILER-BURNER UNIT

of the better conversion installations, however, may equal the unit type in performance.

Operating Requirements for Oil Burners

The U. S. Department of Commerce, in conjunction with the oil burner and heating appliance industries, has established commercial standards for conversion burners and burner-appliance units which cover installation, construction and performance tests.[4]

Combustion Process

Efficient combustion must produce a clean flame and use a relatively small excess of air, *i.e.*, between 25 and 50 percent. This can be done only by vaporizing the oil quickly and completely, and mixing it vigorously with air in a combustion chamber hot enough to support the combustion. A vaporizing burner prepares the oil for combustion by transforming the liquid fuel to the gaseous state by the application of heat before the oil vapor mixes with air to any extent and, if the air and oil vapor temperatures are high and the fire pot hot, a clear blue flame is produced.

In an atomizing burner, the oil is mechanically separated into very fine particles so that the surface exposure of the liquid to the radiant heat of the combustion chamber is vastly increased and vaporization thereby promoted. The result is the ability to burn more and heavier oil within a given combustion space. Because the air enters the combustion chamber with the liquid fuel particles, mixing, vaporization and burning occur all at once in the same space. This produces a luminous flame. A deficient amount of air is indicated by a dull red or dark orange flame with smoky tips.

An excessive supply of air may produce a brilliant white flame or a short ragged flame with incandescent sparks flashing through the combustion space. While extreme cases may be detected, it is not possible to distinguish, by eye, the effect of the finer adjustment which competent installation requires.

Combustion Adjustments

The present-day oil burner with mechanical oil and air supply, properly installed and equipped with an automatic draft regulator, is capable of maintaining efficient combustion for a considerable period following the initial adjustments of oil and air. Eventually, certain changes will occur, and may be such that the amount of excess air will decrease below allowable limits. A decrease in air supply while the oil delivery remains constant, or an increase in oil delivery while the air supply remains constant, will make the mixture of oil and air too rich for clean combustion. The more efficient the adjustment, the more critical it will be. The oil and air supply rates must remain constant.

The following factors may influence the oil delivery rate: (1) changes in oil viscosity due to temperature change or variations in grade of oil delivered; (2) erosion of atomizing nozzle; (3) fluctuations in by-pass relief pressures; and (4) possible variations in methods of atomization. Any change due to partial stoppage of oil delivery will increase the proportion of excess air. This will result in less heat, reduced economy; and possibly a complete interruption of service.

The following factors may influence the air supply: (1) changes in combustion draft due to a variety of causes (*i.e.*, changes in chimney draft because of weather changes, seasonal changes, back drafts, failure or inadequacy of automatic draft regulator, use of chimney for other purposes, possible stoppage of the chimney, and changes in draft resistance of boiler due to partial stoppage of the flues); and (2) changes in air inlet adjustments at the fan.

Air leakage into the boiler or furnace setting should be reduced to a minimum. The amount of air leakage will be determined by the draft in the combustion chamber. It is important that this draft should be reduced as low as is consistent with the proper disposal of the gases of combustion. When using mechanical draft burners with average conditions, the combustion chamber draft should not be allowed to exceed 0.02–0.05 in. water. An automatic draft regulator is very helpful in maintaining such values.

Even though a fan is generally used to supply the air for combustion, in most oil burners, the importance of a proper chimney should not be overlooked. The chimney should have sufficient height and size to insure that the draft will be uniform within the limits given, if maximum efficiency throughout the heating season is to be maintained.

Measurement of the Efficiency of Combustion

Since efficient combustion is based upon a clean flame and definite proportions of oil and air employed, it is possible to determine the results by analyzing the combustion gases. It is usually sufficient to analyze only for carbon dioxide (CO_2). A showing of 10 to 12 percent indicates the best adjustment, if the flame is clean. Most of the good installations show from 8 to 10 percent CO_2. Taking into account the potential hazard of low excess air (high CO_2), a setting to give 10 percent CO_2 constitutes a reasonable standard for most oil burners.

Combustion Chamber Design

With burners requiring a refractory combustion chamber, the size and shape should be in accordance with the manufacturer's instructions. It is important that the chamber be as nearly air tight as is possible, except when the particular burner requires a secondary supply of air for combustion.

The atomizing burner is dependent upon the surrounding heated refractory or firebrick surfaces to vaporize the oil and support combustion. Unsatisfactory combustion may be due to inadequate atomization and mixing. A combustion chamber can only compensate for these things to a limited extent. If liquid fuel continually reaches some part of the firebrick surface, a carbon deposit will result. The combustion chamber should enclose a space having a shape similar to the flame, but large enough to avoid flame contact. The nearest approach in practice is to have the bottom of the combustion chamber flat, but far enough below the nozzle to avoid flame contact, the sides tapering from the air tube at the same angle as the nozzle spray, and the back wall rounded. A plan view of the combustion chamber resembles in shape the outline of the flame. In this way as much firebrick as possible is close to the flame so it may be kept hot. This insures quick vaporization, rapid combustion and better mixing by eliminating dead spaces in the combustion chamber. An overhanging arch at the back of the fire pot is sometimes used to increase the flame travel and give more time for mixing and burning, and sometimes to prevent the gases from going too directly into the boiler flues. When good atomization and vigorous mixing are achieved by the burner, combustion chamber design becomes a less critical matter. Where secondary air is used, combustion chamber design is quite important. When installing some of the vertical rotary burners, the manufacturer's instructions must be followed carefully when installing the hearth, as in this class successful performance depends upon this factor.

Boiler Settings

As the volume of space available for combustion is a determining factor in oil comsumption, it is general practice to remove grates and extend the combustion chamber downward to include or even exceed the ash pit volume; in new installations the boiler may be raised to make added volume available. Approximately 1 cu ft of combustion volume should be provided for every developed boiler horsepower, and in this volume from 1.5 to 2.5 lb of oil per hour can properly be burned. This corresponds to an average liberation of about 38,000 Btu per cubic foot per hour. At times much higher fuel rates may be satisfactory. For best results, care should be taken to keep the gas velocity below 40 fps. Where checkerwork of brick is used to provide secondary air, good practice calls for about 1 sq in. of opening for each pound of oil fired per hour. Such checkerwork is best adapted to flat flames, or to conical flames that can be spread over the floor of the combustion chamber. The proper bricking of a large or even medium sized boiler for oil firing is important, and frequently it is advisable to consult an authority on this subject. The essential in combustion chamber design is to provide against flame impingement upon either metallic or firebrick surfaces. Manufacturers of oil burners usually have available detailed plans for adapting their burners to various types of boilers, and such information should be utilized.

Controls for oil burner operation, including devices for the safety and protection of a boiler or furnace, are fully described in Chapter 38.

GAS-FIRED HEATING EQUIPMENT

A gas burner is defined by the *American Gas Association* as "a device for the final conveyance of the gas, or a mixture of gas and air, to the combustion zone." Burners used for domestic heating are of the atmospheric injection, luminous flame, or power burner types.

Because of the ease with which gas fuel may be controlled, automatic gas-fired heating equipment has become very widely used, and is available in a number of types of domestic gas heating appliances and systems. These may be classified in types designed for central heating plants and those for unit application. Gas designed units and conversion burners are available for the several kinds of central systems in which gravity and forced warm air funaces, steam and hot water boilers are used, and for other applications where warm air floor furnaces and room heaters are installed in the space being heated.

Central Heating Systems

Boilers and furnaces specially designed for gas-firing incorporate design features for obtaining maximum efficiency and performance. Small flue passes to secure good heat transfer, the use of materials resistant to the corrosive effects of products of combustion, and draft hoods are notable features. Control equipment includes gas pressure regulators, automatic pilots, and limit controls designed to protect the appliance and to insure safety of operation. A boiler designed for gas-burning is illustrated in Fig. 19.

Conversion burners are usually complete burner and control units designed for installation in existing boilers and furnaces. Burner heads are of circular or rectangular shape in order to fit in the space available. Single port burners, discharging the flame against a ceramic, stainless steel, or cast-iron target, have become popular in the past few years. The control equipment is generally the same as for gas boilers and furnaces. Various baffles made of clay radiants or metal are used for the purpose of guiding the products of combustion along the heating surface in the firebox or flues. Automatic air dampers are supplied on many models to prevent flow of air into the firebox when the burners is not operating. A typical gas conversion burner is shown in Fig. 20.

Burners of this type are available in sizes ranging from 50,000 to 400,000 Btu per hour capacity. Burners of larger capacity, for use with natural gas in large boilers, are usually engineered by the local utility or contractor. They are available in an infinite number of sizes because the burner may be an assembly of multiple burner heads filling the entire firebox.

Domestic sizes of conversion burners should conform to American Standard Listing Requirements for Conversion Burners, A.S.A. Z21.17–1948, and installation should be made in accordance with American Standard Requirements for Installation of Domestic Gas Conversion Burners, A.S.A. Z21.8-1948.[5]

Draft hoods, conforming to American Standard Requirements, should be installed in place of the dampers used with a solid fuel.

One form of central heating system is the warm air floor furnace.[6] The use of these furnaces is adaptable to mild climates, or for auxiliary heating or heating of single rooms in colder climates. They are used for heating first floors, or where heat is required in only one or two rooms. A number may be used to provide heat for the entire building where all rooms are on the ground floor, thus giving the heating system flexibility.

In floor-furnace applications, the heating element, gas piping, and flue

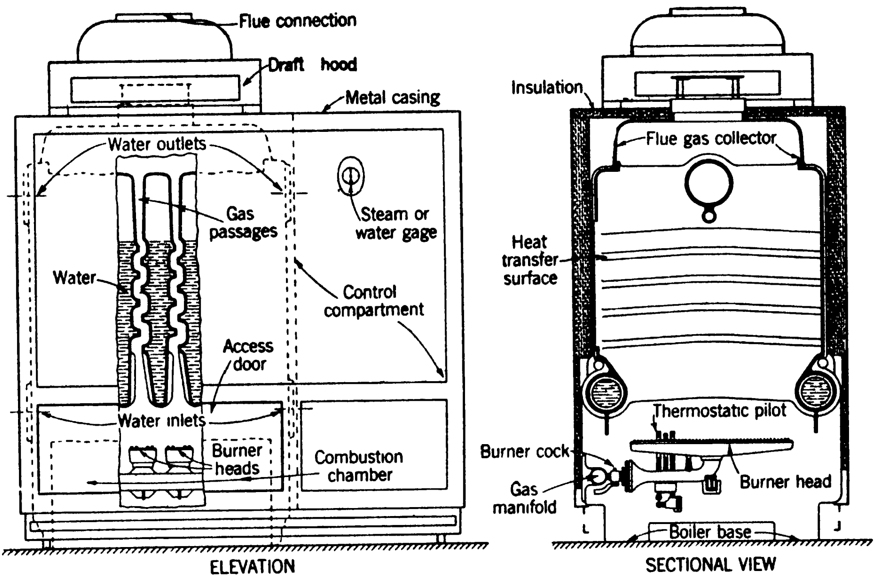

FIG. 19. GAS-FIRED BOILER

vent piping are suspended below the floor, and the only part in the room being heated is the circulating air register which may be of the single-type installed in the floor, or of the dual-type installed in a partition, and heating two rooms.

A recent type of central heating, used in mild climates, is the recessed heater which is either a gravity or forced-air furnace designed for installation in the interior partition of a building, and having stub ducts conducting air to two or more rooms. This type of heater is usually installed in new homes, and is plastered into the wall, becoming a permanent part of the building.

Space Heaters

Space heaters are defined as heating units that take the air for combustion from the space being heated. They may be broadly classified as room heaters and unit heaters.

Room heaters are used for heating single rooms or connecting rooms with good circulation between them and, except for wall-type heaters, they are semi-portable. Unvented-type room heaters should not be used in residences, unless provision is made to remove the excess moisture caused by release of the flue gases into the living quarters. All types of room heaters are capable of automatic control, although they are generally controlled manually. When equipped for automatic control, they must have an automatic pilot as part of the control equipment. Room heaters may be classified as follows:

Circulators, vented and unvented, are small warm air furnaces that heat the room mainly by convection, although some have radiants over the burner, with windows in the front to allow some heat by radiation.

Radiant heaters, usually unvented, although some vented types are available, have a refractory directly above the burners which is heated to incandescence, and gives off at least 30 percent of its heat in the form of radiant energy.

Gas-fired steam and hot water radiators, vented and unvented, are similar in appearance to an ordinary steam or hot water radiator, but are self-contained, and the gas input is controlled by steam pressure or water temperature within the radiator.

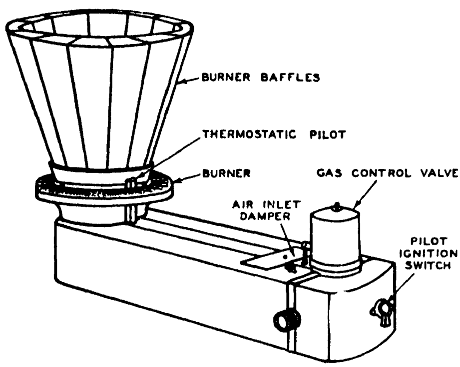

FIG. 20. TYPICAL GAS CONVERSION BURNER

Warm air radiators are vented or unvented circulators whose heating elements are constructed in the form of a steam radiator.

Wall heaters, vented or unvented, are usually a type of radiant heater constructed with sufficient insulation (either solid or circulating air) to prevent overheating of the casing, and are built into a wall with the front flush with the wall surface.

Unit heaters are used extensively for heating large spaces such as stores, garages, and factories. These heaters consist of a burner, heat exchanger, fan for distributing the air, draft hood, automatic pilot, and controls for burners and fan. They are usually mounted in an elevated position from which the heated air is directed downward by louvers. Some unit heaters are suspended from the ceiling, and others are free-standing floor units of the heat tower type.

Unit heaters are available in two types, classified according to their use, with, or without ducts. Only those types of unit heaters tested and approved as warm air furnaces can be connected safely to ducts, as they have sufficient blower capacity to deliver an adequate air supply against duct resistance, and are equipped with limit controls.

Duct furnaces are usually of the unit heater type without the fan, and are used for heating air in existing duct systems where blowers are provided for moving the air through the system. Duct furnaces are tested for operation at much higher static pressures than are obtained in unit heaters.

Combustion Process and Adjustments

Most domestic gas burners are of the atmospheric injection (Bunsen) type in which primary air is introduced and mixed with the gas in the throat of the mixing tube. A ratio of about 5 parts air to 1 part gas for manufactured gas, and a 10 to 1 ratio for natural gas, are generally used as theoretical values of air required for complete combustion. For normal operation of most atmospheric type burners, 40 to 60 percent of the theoretical value as primary air will give best operation. The amount of excess air required in practice depends upon several factors, notably: uniformity of air distribution and mixing, direction of gas travel from burner, and the height and temperature of combustion chamber.

Secondary air is drawn into gas appliances by natural draft. As with other fuels, excess secondary air constitutes a loss, and should be reduced to a proper minimum, which usually cannot be less than 25 to 35 percent, if the appliance is to meet *ASA* approval. Yellow flame burners depend upon secondary air, alone, for combustion.

The flame produced by atmospheric injection burners is non-luminous. Air shutter adjustments for manufactured gas should be made by closing the air shutter until yellow flame tips appear, and then by opening the air shutter to a final position at which the yellow tips just disappear. This type of flame obtains ready ignition from port to port, and also favors quiet flame extinction. When burning natural gas, the air adjustment is generally made to secure as blue a flame as is obtainable without lifting of flames from burner ports.

Little difficulty should be had in maintaining efficient combustion when burning gas. The fuel supply is normally held to close limits of variation in pressure and calorific value, and the rate of heat supply is nominally constant. Because the force necessary to introduce the fuel into the combustion chamber is an inherent factor of the fuel, no draft by the chimney is required for this purpose. The use of a draft hood insures the maintenance of constant low draft conditions in the combustion chamber with a resultant stability of air supply. A draft hood is also helpful in controlling the amount of excess air and preventing back drafts that might extinguish the flame. (See Chapter 13.)

Due to the use of draft hoods and gas pressure regulators, both the input and combustion conditions of gas appliances are maintained quite uniform until deposits of dirt, corrosion, or scale accumulate in the air inlet openings, burner ports, or on the heating surface. Periodic cleaning is necessary to keep any gas appliance in proper operating condition.

Measurement of the Efficency of Combustion

The efficiency of combustion may be judged from the percentage of carbon dioxide (CO_2), oxygen (O_2) and carbon monoxide (CO) in the flue gases. The CO_2 and O_2 may be obtained by means of an Orsat apparatus, but the CO must be determined by more accurate equipment. It is customary to use simple indicators to determine whether CO is present, and to make adjustments of the appliances to reduce the CO below 4/100 of one percent before continuing tests in which the CO_2 and O_2 can then be found by use of the Orsat apparatus. Since the ultimate CO_2 for any gas depends on the carbon-hydrogen ratio, the quality of the combustion should not be judged from the value of the CO_2 in the flue gas without reference to the ultimate CO_2 obtainable. Practical values of CO_2 will usually be from 8 to 14 percent, depending on the gas used.

Ratings for Gas Appliances

Input rating for a gas appliance is established by demonstrating that the appliance can meet the Approval Requirements of the *ASA*. The tests are conducted at the *A.G.A. Laboratories*. Output rating is determined from the approved input and an average efficiency stated in the Approval Requirements, and is the heat available at the outlet.

Sizing Gas-Fired Heating Plants

Although gas-burning equipment usually is completely automatic,

Table 1. Capacity of Gas Piping

Length of Pipe in Feet	Nominal Diameter of Pipe in Inches				
	¾	1	1¼	1½	2
	Capacity—Cu Ft Per Hr with a 0.6 Sp Gr Gas and Pressure Drop of 0.3″ Water Column				
15	172	345	750		
30	120	241	535	850	
45	99	199	435	700	
60	86	173	380	610	
75	77	155	345	545	
90	70	141	310	490	
105	65	131	285	450	920
120		120	270	420	860
150		109	242	380	780
180		100	225	350	720

maintaining the temperature of rooms at a predetermined figure, there are some manually controlled installations. In order to overcome effectively the starting load and losses in piping, a manually-controlled gas boiler should have an output as much as 100 percent greater than the equivalent standard radiation which it is expected to serve.

Boilers under thermostatic control, however, are not subject to such severe pick-up loads and consequently, it is possible to use a lower selection factor. For a gas-fired boiler or furnace under thermostatic control. a factor of 20 to 25 percent is usually sufficient for pick-up allowance.

In those installations, in mild climates where 100 percent outside air is used, furnaces should be of larger size in order to provide adequate capacity and quick pick-up under intermittent heating conditions.

The factor to be allowed for loss of heat from piping will vary somewhat, the proportionate amount of piping installed being greater for small installations than for large ones. For selection factors to be added to installed radiation under thermostatic control, see Chapter 15.

Appliances used for heating with gas should bear the approval seal of the *A.G.A. Laboratories* on the manufacturer's nameplate, together with the official input and output ratings. It is not permissible to operate a gas heating unit above its stated rating. It may be necessary to operate below this rating at elevations above 2000 ft, unless the appliance has been tested and approved for operation at altitudes up to 5200 ft, in which case such approval will be shown on the manufacturer's nameplate.

Installations should be made in accordance with recommendations shown in the publications of the *American Gas Association.*

Controls

Temperature controls for gas burners are described in Chapter 38. Some central heating plants are equipped with push-button or other manual control. The main gas valve may be of either the snap action or throttling type. Automatic electric ignition is available.

Sizing of Gas Piping

Piping for gas appliances should be of adequate size, and so installed as to provide a supply of gas sufficient to meet the maximum demand without

TABLE 2. MULTIPLIERS FOR VARIOUS SPECIFIC GRAVITIES
For Use With Table 1

Specific Gravity	Multiplier	Specific Gravity	Multiplier
.35	1.31	1.00	.775
.40	1.23	1.10	.740
.45	1.16	1.20	.707
.50	1.10	1.30	.680
.55	1.04	1.40	.655
.60	1.00	1.50	.633
.65	.962	1.60	.612
.70	.926	1.70	.594
.75	.895	1.80	.577
.80	.867	1.90	.565
.85	.841	2.00	.547
.90	.817	2.10	535

undue loss of pressure between the point of supply (the meter) and the burner. The size of gas pipe required depends upon the following:

1. Maximum gas consumption to be provided.
2. Length of pipe and number of fittings.
3. Allowable loss in pressure from the outlet of the meter to the burner.
4. Specific gravity of the gas.

To obtain the cubic feet per hour of gas required by the burner, divide the Btu input at which the burner will be adjusted, by the average Btu heating value per cubic foot of the gas.

Capacities of different sizes and lengths of pipe, in cubic feet per hour, with a pressure drop of 0.3 in. of water column for a gas of 0.60 sp gr, are shown in Table 1. In adopting a 0.3 in. pressure drop, due allowance for an ordinary number of fittings was made.

To convert the figures given in Table 1 to capacities for another gas of different specific gravity, multiply the tabular values by the multipliers shown in Table 2.

FUEL BURNING RATES

The burning rate for automatic fuel burning devices is determined by the gross heat output required of the boiler, or furnace, to carry the net heating load, plus allowances for system losses and pick-up. General

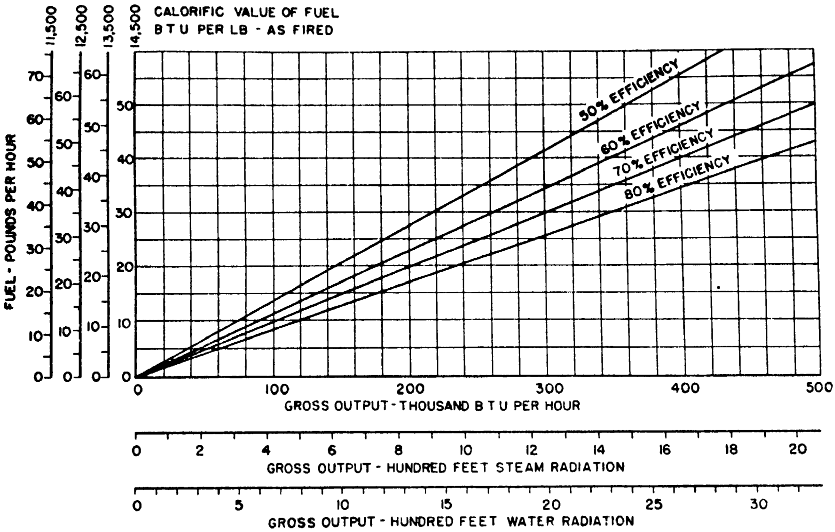

FIG. 21. COAL FUEL BURNING RATE CHART

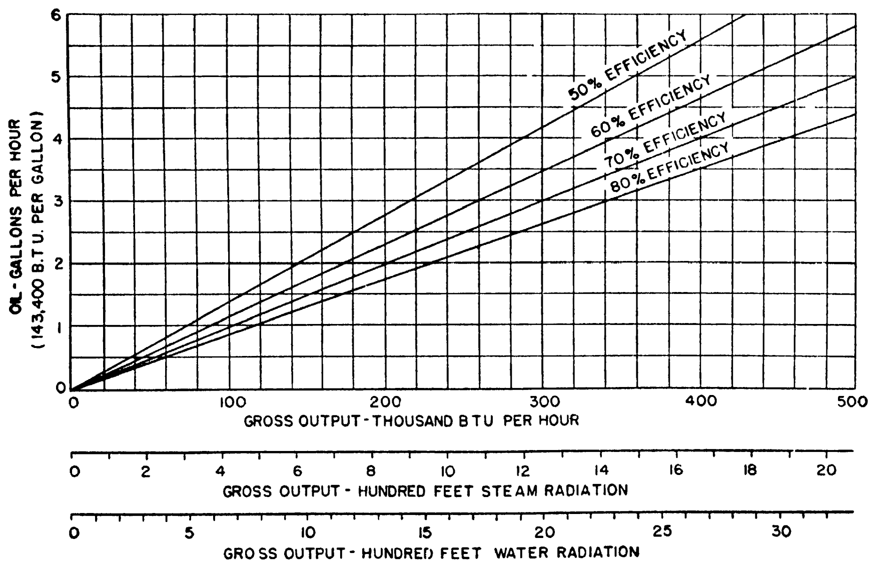

FIG. 22. OIL FUEL BURNING RATE CHART[a]

[a] This chart is based upon No. 2 oil having a heat content of 143,400 Btu per gallon. If other grades of oil are used multiply the value obtained from this chart by the following factors: No. 1 oil (139,000 Btu per gallon) 1.032; No. 4 oil (144,500 Btu per gallon) 0.992; No. 5 oil (146,000 Btu per gallon) 0.982; and No. 6 oil (150,000 Btu per gallon) 0.956.

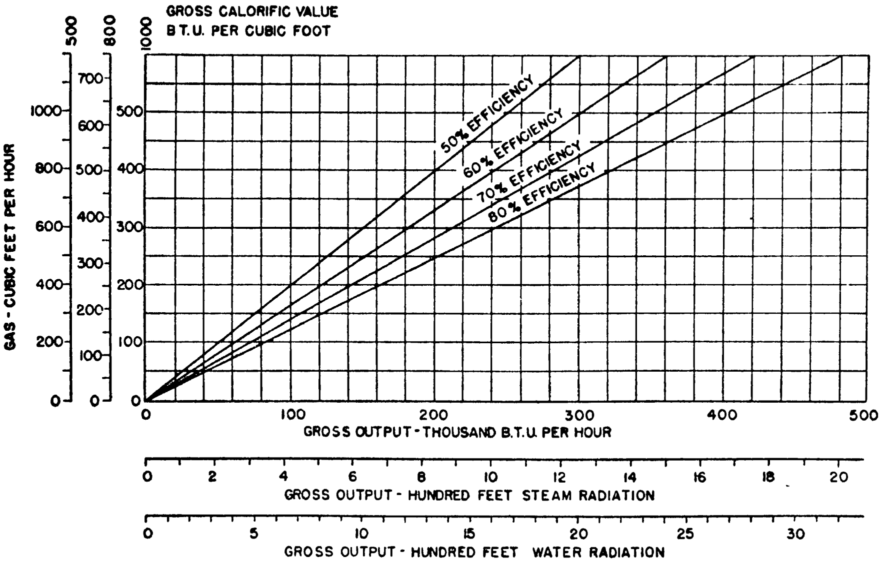

FIG. 23. GAS FUEL BURNING RATE CHART

values for these allowances have been given in preceding text. Detailed information for piping and pick-up allowances for steam and hot water systems, is given in Chapter 15, and for warm air systems, in Chapters 18 and 19.

When the gross output, operating efficiency, and heat value of the fuel are known, the required rate of burning can be determined by means of Figs. 21, 22 and 23 for the several fuels.

As the rate of fuel burning is directly proportional to the load for a given efficiency, these charts can be extended by moving the decimal points the same number of digits in both vertical and horizontal scales.

The correct fuel burning rate can be determined directly from the several charts for oil or gas burning installations, as these customarily operate on a strictly intermittent basis. These fuel burning devices usually introduce the fuel at a single fixed rate during the *on* periods, and this rate should be sufficient to carry the gross load. In the case of coal stokers, which are usually capable of variable rates of firing, it is desirable to operate at as low a rate as weather conditions will permit, but the maximum firing rate of the stoker should be sufficient to carry the gross load. This rate may be determined by the same method as used for oil or gas.

REFERENCES

[1] Domestic Burners for Pennsylvania Anthracite (Underfeed Type), (*U. S. Department of Commerce, National Bureau of Standards,* Commercial Standard No. CS48-40).

[2] *Stoker Manufacturers Association Manual:* Industry Standards, Recommended Practices, Technical Information. Published by *Stoker Manufacturers Association,* 307 N. Michigan Ave., Chicago 1, Ill.

[3] Code for Determination of Rated Capacities of Anthracite Underfeed Stokers, adopted June 1, 1944, and a Code for Determination of Rated Capacities of Bituminous Underfeed Stokers, adopted May 3, 1944. See *Stoker Manufacturers Association Manual.*

[4] Automatic Mechanical Draft Oil Burners Designed for Domestic Installations (*U. S. Department of Commerce, National Bureau of Standards,* Commercial Standard No. CS75-42). Flue Connected Oil Burning Space Heaters Equipped with Vaporizing Pot Type Burners (*U. S. Department of Commerce, National Bureau of Standards,* Commercial Standard No. CS101-43). Warm-Air Furnaces Equipped with Vaporizing Pot-Type Oil Burners (*U. S. Department of Commerce, National Bureau of Standards,* Commercial Standard No. CS104-46). Oil-Burning Floor Furnaces Equipped with Vaporizing Pot-Type Burners (*U. S. Department of Commerce, National Bureau of Standards,* Commercial Standard No. CS113-44).

[5] American Standard Requirements for Installation of Domestic Gas Conversion Burners (A.S.A. Z21.8-1948 American Standard Association).

[6] Gas Floor Furnaces, Gravity Circulating Type (*U. S. Department of Commerce, National Bureau of Standards,* Commercial Standard No. CS99-42).

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Gaseous Fuels, American Gas Association.

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Heat Losses and Efficiencies of Fuels in Residential Heating, by R. A. Sherman and R. C. Cross (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 185).

CHAPTER 15

HEATING BOILERS, FURNACES, SPACE HEATERS

BOILERS: Construction, Types, Design Considerations, Testing and Rating Codes, Efficiency, Rating, Selection, Space Limitations, Connections and Fittings, Erection, Operation and Maintenance. FURNACES: Types, Materials and Construction, Ratings, Testing and Rating Codes, Efficiency, Design Considerations, Humidification Equipment. SPACE HEATERS: Types: Solid Fuel, Oil, Gas; Materials and Construction, Testing and Rating, Design Considerations, Installation

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IN presenting the subject of Boilers, Furnaces and Space Heaters this chapter is divided into three parts; the first dealing with boilers, the second treating warm air furnaces, and the third covering space heaters.

HEATING BOILERS

Steam and hot water boilers for low pressure heating are built of steel or cast-iron in a wide variety of types and sizes, many of which are illustrated in the *Catalog Data Section*.

CONSTRUCTION

The nationally recognized code governing the construction of low-pressure steel and cast-iron heating boilers is the *ASME* Boiler Construction Code for Low Pressure Heating Boilers. Some states and municipalities have their own codes which apply locally, but these are usually patterned after the *ASME* Code.

The maximum allowable working pressures are limited by the *ASME* Code to 15 psi for steam and 30 psi for hot water heating boilers. Hot water boilers may be used for higher working pressures, for heating purposes or for hot water supply, when designed and tested for the higher pressure.

TYPES OF HEATING BOILERS

Heating boilers are classified in a number of different ways, such as:

1. According to materials of construction. These are steel and cast-iron. Very few non-ferrous boilers are made.
2. According to the fuels for which the boilers are designed. These are coal, hand-fired or stoker-fired; oil; gas; or wood. Some boilers are designed specifically for one fuel, but many boilers are designed for more than one fuel.
3. According to the specific purpose or application for which the boiler is used, such as space heating or domestic hot water supply.
4. According to the design or construction of the boiler such as sectional, round, fire-tube, water-tube, magazine feed, Scotch, etc.

Cast-Iron Boilers

Cast-iron boilers are generally classified as:

1. Square or rectangular boilers with vertical sections and rectangular grates, commonly known as sectional boilers.
2. Round boilers with horizontal pancake sections and circular grates.

Cast-iron boilers are usually shipped in sections, and assembled at the place of installation. However some small boilers are shipped factory assembled. In the majority of boilers the sections are assembled with push nipples and tie rods. Many sectional boilers are provided with large push nipples at top to permit the circulation of water between adjacent sections at both the water line and bottom of the boiler, which is necessary to enable the use of an indirect water heater with the boiler for summer-winter hot water supply. Round and sectional boilers may be increased in size by the addition of sections and corresponding plate work.

Small sectional type boilers are available with *wet-base* construction, wherein the ashpit or combustion chamber sides and bottom are surrounded by extensions of the water legs of the boiler sections, and thus no separate base is required. This type of construction permits the boiler to be set directly on a wood or composition floor without danger of fire. The *wet-base* also provides some additional heating surface.

Capacities of cast-iron boilers range generally from capacities required for small residences up to about 12,000 sq ft of steam radiation. There are a few boilers made with capacities up to 18,000 sq ft of steam radiation. For larger loads, boilers must be installed in multiple.

Steel Boilers

Steel boilers may be of the fire-tube type, in which the gases of combustion pass through the tubes and the boiler water circulates around them, or of the water-tube type, in which the gases circulate around the tubes and the water passes through them. Either the fire-tube or water-tube type may be designed with integral water jacketed furnaces, or arranged for refractory lined brick or refractory lined jacketed furnaces. Those with integral water jacketed furnaces are called portable firebox boilers, and are the most commonly used type. They are usually shipped in one piece, ready for piping connections. Refractory furnaces are usually installed in refractory lined furnace boilers after they are set in place.

Capacities of steel boilers range from those required for small residences up to about 35,000 sq ft of steam radiation.

Boilers for Special Applications

One of these is known as the *magazine feed boiler* developed for the burning of small sizes of anthracite and coke, and has a large fuel carrying capacity, which results in longer firing periods than would be the case with the standard types burning coal of buckwheat size. Special attention must be given to proper chimney sizes and connections in order to insure adequate draft.

Boilers for *hot water supply* are classified as direct, if the water heated passes through the boiler, and as indirect, if the water heated does not come in contact with the water or steam in the boiler.

Direct heaters are built to operate at the pressures found in city supply mains, and are tested at pressures from 200 to 300 lb per square inch. The life of direct heaters depends almost entirely on the scale-forming properties of the water supplied and the temperatures maintained. If low water temperatures are maintained, the life of the heater will be much longer due to decreased scale formation and minimized corrosion. Direct water heaters in some cases are designed to burn refuse and garbage.

Indirect heaters generally consist of steam boilers in connection with heat exchangers of the coil or tube types which transmit the heat from the steam to the water. This type of installation has the following advantages:

1. The boiler operates at low pressure.
2. The boiler is protected from scale and corrosion.
3. The scale is formed in the heat exchanger in which the parts to which the scale is attached, can be cleaned or replaced. The accumulation of scale does not affect efficiency, although it will affect the capacity of heat exchanger.
4. Discoloration of water may be prevented if the water supply comes in contact with only non-ferrous metal.

Where a steam or a forced circulation hot water heating system is installed, the domestic hot water may be heated by an indirect heater attached to the boiler. For most satisfactory performance in the steam system, this heater is placed just below the water line of the boiler. In a forced circulation hot water system, it should be located as high as possible with respect to the boiler.

BOILER DESIGN CONSIDERATIONS

Furnace Design

Good efficiency and proper boiler performance are dependent on correct furnace design. There must be sufficient volume for burning the particular fuel which is used, and means to obtain a thorough mixing of air and gases at a high temperature and at a velocity low enough to permit complete combustion of all the volatiles. For hand-fired boilers, the furnace volume should be large enough to hold sufficient fuel for reasonably long firing periods. (See Chapters 13 and 14.)

Heating Surface

Boiler heating surface is that portion of the surface of the heat transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other side. Heating surface on which the fire shines, is known as *direct* or radiant surface, and that in contact with hot gases only, as *indirect* or convection surface. The amount of heating surface, its distribution, and the temperatures on either side thereof, influence the capacity of any boiler.

Direct heating surface is more valuable than indirect per square foot because it is subjected to a higher temperature and also, in the case of solid fuel, because it is in position to receive the full radiant energy of the fuel bed.

The effectiveness of the heating surface depends on its cleanliness, its location in the boiler, and the shape of the gas passages. The area of the gas passages must not be so small as to cause excessive resistance to the flow of gases, where natural draft is employed. Inserting baffles so that the heating surface is arranged in series with respect to the gas flow, increases boiler efficiency and reduces stack temperature, but increases the draft loss through the boiler.

Heat Transfer Rate

Practical average overall heat transfer rates, expressed in Btu absorbed per square foot of surface per hour, will average about 3300 for hand-fired boilers, and 4000 for mechanically-fired boilers when operating at *design load.* When mechanically-fired boilers are operating at *maximum load,* as defined in this chapter under heading *Selection of Boilers,* these values will run between 5000 and 6000. Boilers operating under favorable conditions at these heat transfer rates, will give exit gas temperatures that are considered consistent with good practice, although there are boilers which have high efficiencies and also operate at higher transmission rates.

TESTING AND RATING CODES

The Society has adopted four solid fuel testing codes, a solid fuel rating code, and an oil fuel testing code.

ASHVE Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers—Codes 1 and 2—(Revision of June, 1929),[1] provide a method for conducting and reporting tests to determine heat efficiency and performance characteristics.

ASHVE Performance Test Code for Steam Heating Solid Fuel Boilers—Code No. 3—(Edition of 1929)[1] is intended for use with ASHVE Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers.[2] The object of this test code is to specify the tests to be conducted, and to provide a method for conducting and reporting tests to determine the efficiencies and performance of the boiler.

The ASHVE Standard Code for Testing Steam Heating Boilers Burning Oil Fuel,[3] (Adopted June, 1932), provides a standard method for conducting and reporting tests to determine the heating efficiency and performance characteristics when oil fuel is used with steam heating boilers.

The ASHVE Standard Code fot Testing Stoker-Fired Steam Heating Boilers,[4] (Adopted June, 1938), provides a test method for determining the efficiency and performance characteristics of any stoker and boiler combination burning any type of solid fuel, such as anthracite or bituminous coal.

The *Steel Boiler Institute, Inc.* has adopted a Rating Code for Commercial Steel Boilers and Residential Steel Boilers, and for Testing Oil-Fired Residential Steel Boilers (Fifth Edition as Revised Jan. 1, 1948). The commerical boilers (defined as those having 129 to 2500 sq ft of heating surface) are rated in square feet (steam) on the basis of heating surface with limitations set for grate area, furnace volume, and furnace height.

TABLE 1. *SBI* NET RATING DATA FOR RESIDENTIAL STEEL BOILERS—OIL FIRED[a]

SBI Net Rating			Minimum Furnace Volume Cu Ft	Heating Surface Sq Ft
Sq Ft Steam	Sq Ft Water	Btu		
275	440	66000	2.5	16
320	510	77000	2.9	19
400	640	96000	3.6	24
550	880	132000	5.0	32
700	1120	168000	6.4	41
900	1440	216000	8.2	53
1100	1760	264000	10.0	65
1300	2080	312000	11.8	77
1500	2400	360000	13.6	88
1800	2880	432000	16.4	106
2200	3520	528000	20.0	129
2600	4160	624000	23.6	153
3000	4800	720000	27.3	177

[a] Stoker-fired and Gas-fired *SBI* Net Rating not greater than Oil-fired. Hand-fired, *SBI* Net Rating (Steam) not greater than 14 times the square feet of heating surface.

TABLE 2. *SBI* RATINGS FOR COMMERCIAL STEEL BOILERS

Mechanically Fired								Heating Surface Sq Ft	Hand Fired						Minimum Grate Area Sq Ft	Flow Tappings	
SBI Rating			*SBI* Net Rating			Minimum Furnace Volume Cu Ft[a]	Minimum Furnace Height[b] In.		*SBI* Rating			*SBI* Net Rating				Steam Outlet I.P.S. In.	Steam Return I.P.S.[c] In.
Sq Ft Steam	Sq Ft Water	Btu	Sq Ft Steam	Sq Ft Water	Btu				Sq Ft Steam	Sq Ft Water	Btu	Sq Ft Steam	Sq Ft Water	Btu			
2,190	3,500	526,000	1,800	2,880	432,000	15.7	26	129	1,800	2,880	432,000	1,500	2,400	360,000	7.9	6	3
2,680	4,280	643,000	2,200	3,520	528,000	19.2	28	158	2,200	3,520	528,000	1,830	2,930	439,000	8.9	6	3
3,160	5,050	758,000	2,600	4,160	624,000	22.6	29¼	186	2,600	4,160	624,000	2,170	3,470	521,000	9.7	6	3
3,650	5,840	876,000	3,000	4,800	720,000	26.1	29½	215	3,000	4,800	720,000	2,500	4,000	600,000	10.5	6	3
4,250	6,800	1,020,000	3,500	5,600	840,000	30.4	30	250	3,500	5,600	840,000	2,920	4,670	700,000	11.4	6	3
4,860	7,770	1,166,000	4,000	6,400	960,000	34.8	30½	286	4,000	6,400	960,000	3,330	5,330	800,000	12.2	6	3
5,470	8,750	1,313,000	4,500	7,200	1,080,000	39.1	31¼	322	4,500	7,200	1,080,000	3,750	6,000	900,000	13.4	6	3
6,080	9,720	1,459,000	5,000	8,000	1,200,000	43.5	31¾	358	5,000	8,000	1,200,000	4,170	6,670	1,000,000	14.5	6	3
7,290	11,660	1,750,000	6,000	9,600	1,440,000	52.1	32¾	429	6,000	9,600	1,440,000	5,000	8,000	1,200,000	16.4	8	4
8,500	13,600	2,040,000	7,000	11,200	1,680,000	60.8	34	500	7,000	11,200	1,680,000	5,830	9,330	1,400,000	18.1	8	4
10,330	16,520	2,479,000	8,500	13,600	2,040,000	73.8	35½	608	8,500	13,600	2,040,000	7,080	11,330	1,700,000	20.5	8	4
12,150	19,440	2,916,000	10,000	16,000	2,400,000	86.8	37½	715	10,000	16,000	2,400,000	8,330	13,330	2,000,000	22.5	8	4
15,180	24,280	3,643,000	12,500	20,000	3,000,000	108.5	40¼	893	12,500	20,000	3,000,000	10,420	16,700	2,500,000	25.6	8	4
18,220	29,150	4,373,000	15,000	24,000	3,600,000	130.2	43	1,072	15,000	24,000	3,600,000	12,500	20,000	3,000,000	28.4	8	4
21,250	34,000	5,100,000	17,500	28,000	4,200,000	151.8	46	1,250	17,500	28,000	4,200,000	14,580	23,330	3,500,000	30.9	8	4
24,290	38,860	5,830,000	20,000	32,000	4,800,000	173.5	48¾	1,429	20,000	32,000	4,800,000	16,670	26,670	4,000,000	33.2	8	4
30,360	48,570	7,286,000	25,000	40,000	6,000,000	216.9	54½	1,786	25,000	40,000	6,000,000	20,830	33,330	5,000,000	37.4	10	6
36,430	58,280	8,743,000	30,000	48,000	7,200,000	260.3	60½	2,143	30,000	48,000	7,200,000	25,000	40,000	6,000,000	41.2	10	6
42,500	68,000	10,200,000	35,000	56,000	8,400,000	303 6	66	2,500	35,000	56,000	8,400,000	29,170	46,670	7,000,000	44.7	10	6

[a] Oil, gas or bituminous stoker-fired coal. Minimum furnace volumes for anthracite, stoker-fired, are not specified in this code.

[b] Bituminous, stoker-fired.

[c] The tapping sizes shown for boilers having 129 to 500 sq ft of heating surface, inclusive are adequate for forced hot water.

The residential boilers (defined as those having not more than 177 sq ft of heating surface) are rated from tests of oil-fired boilers, with limitations in relation to heating surface and testing conditions. Stoker-fired and gas-fired *residential* boilers are rated (*SBI* Net Rating) not in excess of the oil-fired rating. Hand-fired residential boilers are rated (*SBI* Net Rating) not greater than 14 times the heating surface.

Tables 1 and 2 show the *SBI* ratings of residential and commercial steel boilers, respectively.

The *Institute of Boiler and Radiator Manufacturers* has adopted a Code[5] for rating cast-iron heating boilers, based upon performance obtained under controlled test conditions. This Code applies to all sectional cast-iron heating boiler except those of magazine-feed type.

The *Gross I = B = R* Output is obtained by test, and is subject to certain limiting factors. For hand-fired boilers, the number of boilers of a series to be tested, the minimum overall efficiency, the minimum time limit (the time an *Available Fuel Charge* will last when burned at a rate which will produce the *Gross I = B = R Output*), the chimney area and height, and the draft in the stack are all subject to the limits established in the Code. Tests are run using anthracite coal of standard specification. Bituminous coal and coke ratings are the same as for anthracite coal.

For automatically-fired boilers, the number of boilers of a series to be tested, the flue gas temperature and analysis, the minimum overall efficiency, the draft loss through the boiler, and the heat release in the combustion chamber are subjected to limitation by the Code.[5] Automatically-fired boiler ratings are established by oil-fired tests using gun type oil burners and commercial grade No. 2 fuel oil. Stoker-fired and gas-fired ratings (where no *A.G.A.* Rating is published) are based on the *Gross I = B = R Output* obtained by oil-fired tests.

The *Net I = B = R Rating* is determined from the *Gross I = B = R Output* by applying specified *Piping and Pickup Factors* which range from 2.36 to 1.40 for hand-fired boilers, and from 1.56 to 1.288 for automatically-fired steam boilers, and from 1.333 to 1.288 for automatically-fired hot water boilers. In all cases, the factor decreases as the boiler size increases. Table 3 is abstracted from the 1951 *I = B = R* Boiler Rating Tables in the Code and illustrates the relationship between *Net I = B = R Rating* and *Gross I = B = R Output*.

The *American Gas Association* rates gas designed boilers at 80 percent of the *A.G.A.* Input Rating. These ratings are determined by performance tests described in the *A.G.A.* Approval Requirements for Central Heating Appliances.

The *Heating, Piping and Air Conditioning Contractors National Association* has adopted a method, based on their physical characteristics for rating boilers that are not rated in accordance with the *SBI* or *I = B =* R Codes. Ratings are expressed on a Net Load basis in square feet of steam radiation.

BOILER EFFICIENCY

The term efficiency, as used for guarantee of boiler performance, is usually construed as follows:

1. *Solid Fuels.* The efficiency of the *boiler* alone is the ratio of the heat absorbed by the water and steam in the boiler per pound of combustible burned on the grate, to the calorific value of 1 lb of combustible as fired. The *combined* efficiency of *boiler, furnace and grate* is the ratio of the heat absorbed by the water and steam in the boiler per pound of fuel as fired, to the calorific value of 1 lb of fuel as fired.

TABLE 3. I=B=R BOILER RATING TABLE[a]

Hand-Fired							Automatic-Fired							
Gross I=B=R Output	Net I=B=R Rating		Piping and Pickup Factor	Time Available Fuel Will Last,	Maximum Stack Height	Minimum Stack Area[b]	Gross I=B=R Output	Net I=B=R Rating			Piping and Pickup Factor		Minimum Stack Area[b]	Maximum Allowable Draft Loss
	Steam and water	Steam						Water	Steam		Water	Steam		
1000 Btu	1000 Btu	Sq Ft		Hr	Ft	Sq In.	1000 Btu	1000 Btu	1000 Btu	Sq Ft			Sq In.	In. W.G.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
57	24	100	2.360	7.50	29.0	50	37	28	24	100	1.333	1.560	50	0.044
218	96	400	2.268	6.32	36.5	50	146	110	96	400	1.333	1.525	50	0.058
359	168	700	2.139	5.73	42.0	54	251	188	168	700	1.333	1.492	50	0.072
489	240	1000	2.039	5.32	46.0	82	352	264	240	1000	1.333	1.466	52	0.084
614	312	1300	1.967	4.99	49.0	110	451	338	312	1300	1.333	1.444	73	0.096
735	384	1600	1.914	4.73	52.0	135	547	410	384	1600	1.333	1.424	95	0.108
854	456	1900	1.872	4.49	54.5	160	642	482	456	1900	1.333	1.408	115	0.118
970	528	2200	1.837	4.34	56.5	185	736	552	528	2200	1.333	1.394	135	0.130
1083	600	2500	1.805	4.24	59.0	207	829	622	600	2500	1.333	1.382	155	0.141
1192	672	2800	1.774	4.15	61.0	231	920	690	672	2800	1.333	1.369	173	0.152
1298	744	3100	1.744	4.07	63.0	253	1011	758	744	3100	1.333	1.359	192	0.162
1401	816	3400	1.717	4.00	64.5	275	1101	826	816	3400	1.333	1.349	211	0.172
1502	888	3700	1.691	4.00	66.5	296	1189	892	888	3700	1.333	1.339	230	0.183
1599	960	4000	1.666	4.00	68.0	315	1278	960	960	4000	1.331	1.331	249	0.192
1726	1056	4400	1.634	4.00	70.0	337	1394	1056	1056	4400	1.320	1.320	274	0.206
1849	1152	4800	1.605	4.00	71.5	358	1509	1152	1152	4800	1.310	1.310	297	....
1968	1248	5200	1.577	4.00	73.0	376	1624	1248	1248	5200	1.301	1.301	318	....
2086	1344	5600	1.552	4.00	74.5	393	1739	1344	1344	5600	1.294	1.294	339	....
2197	1440	6000	1.526	4.00	76.0	409	1858	1440	1440	6000	1.290	1.290	359	....
2307	1536	6400	1.502	4.00	77.5	425	1978	1536	1536	6400	1.288	1.288	377	....
2415	1632	6800	1.482	4.00	79.0	438	2102	1632	1632	6800	1.288	1.288	395	....
2470	1680	7000	1.470	4.00	79.5	446	2164	1680	1680	7000	1.288	1.288	405	....
2605	1800	7500	1.447	4.00	81.5	464	2318	1800	1800	7500	1.288	1.288	426	....
2738	1920	8000	1.426	4.00	83.0	481	2473	1920	1920	8000	1.288	1.288	446	....
2874	2040	8500	1.409	4.00	85.0	497	2628	2040	2040	8500	1.288	1.288	467	....
3024	2160	9000	1.400	4.00	87.0	515	2782	2160	2160	9000	1.288	1.288	486	....
3192	2280	9500	1.400	4.00	89.0	535	2937	2280	2280	9500	1.288	1.288	504	....
3360	2400	10000	1.400	4.00	91.5	555	3091	2400	2400	10000	1.288	1.288	522	....
4032	2880	12000	1.400	4.00	99.0	634	3709	2880	2880	12000	1.288	1.288	596	....
4704	3360	14000	1.400	4.00	106.5	712	4328	3360	3360	14000	1.288	1.288	663	....
5376	3840	16000	1.400	4.00	112.5	789	4946	3840	3840	16000	1.288	1.288	740	....
6048	4320	18000	1.400	4.00	118.0	863	5564	4320	4320	18000	1.288	1.288	810	....
6720	4800	20000	1.400	4.00	120.0	900	6182	4800	4800	20000	1.288	1.288	877	....

[a] Extracted from Rating Table in 1951 edition of I=B=R Testing and Rating Code for Low Pressure Cast Iron Heating Boilers.
[b] To be specified in catalog.

2. *Liquid and Gaseous Fuels*. The *combined* efficiency of boiler, furnace and burner is the ratio of the heat absorbed by the water and steam in the boiler per pound or cubic foot of fuel, to the calorific value of 1 lb or cubic foot of fuel, respectively.

The following efficiencies apply to current designs of boilers operated under favorable conditions at their gross output ratings. Some older boilers, designed primarily for hand firing, may have lower efficiencies when automatically fired.

Anthracite, hand-fired	60 to 75 percent
Bituminous coal, hand-fired	50 to 65 percent
Stoker-fired	60 to 75 percent
Oil and gas-fired	70 to 80 percent

Higher efficiencies for hand-fired bituminous coal may be obtained by careful firing of either a regular or a smokeless boiler.

RATING OF BOILERS

In referring to boiler rating, it is necessary to know the basis on which the rating has been established in order to understand the exact meaning of the term. The following example will illustrate the meaning of three ratings which might be established for the same boiler.

Assume that an installation has the following loads determined in accordance with the section *Selection of Boilers:*

Net Load	1000 sq ft of steam radiation
Piping Tax	200 sq ft of steam radiation
Design Load	1200 sq ft of steam radiation
Pickup Allowance	240 sq ft of steam radiation
Maximum or Gross Load	1440 sq ft of steam radiation

A boiler that is just large enough to carry this system might be said to have a *net load rating* of 1000 sq ft, a *design load* rating of 1200 sq ft, or a *gross load* rating of 1440 sq ft, depending on the basis on which the boiler is rated.

On a *net load* basis the boiler would be rated 1000 sq ft of steam radiation and would have sufficient excess capacity to supply the *normal piping* and *pickup load*. Net $I = B = R$ Ratings, *SBI* Net Ratings, and Net Load Ratings of the *Heating, Piping and Air Conditioning Contractors National Association* are established on this basis.

On a *design load* basis the boiler would be rated 1200 sq ft of steam radiation and would have sufficient excess capacity to supply the *pickup load*. It would be of adequate size for a system in which the *sum* of the *net load* and the *piping heat loss* did not exceed 1200 sq ft of steam radiation. The *SBI* Ratings shown in columns 1, 2, 3, 10, 11 and 12 of Table 2 (not to be confused with *SBI* Net Rating) are established on a *design load* basis.

On a *gross output* basis of rating, the boiler would be rated 1440 sq ft of steam radiation and would be of adequate size for a system in which the sum of the *net load, piping load*, and *pickup load* did not exceed 1440 sq ft of steam radiation. Gross $I = B = R$ Output and *A.G.A.* Ratings are established on a *gross output* basis.

In the determination of boiler ratings, the *Gross Output* is the quantity of heat available at the boiler nozzle, with the boiler normally insulated and when operating under limitations stipulated in the code or method by which the boiler is rated. The boiler may be capable of producing a greater nozzle output, but in doing so would exceed some of these limitations.

SELECTION OF BOILERS

General Factors

The *Maximum Load* or *Gross Load* on the boiler is the sum of the four following items.

The *Design Load* is the sum of items 1, 2, and 3.

The *Net Load* is the sum of items 1 and 2.

1. *Radiation Load.* The estimated heat emission in Btu per hour of the connected radiation (direct, indirect, or forced convection coils) to be installed.

The connected radiation is determined by calculating the heat losses for each room in accordance with data given in Chapters 9, 10 and 11. The sum of the calculated heat losses for all the rooms represents the total required heat emission of the connected radiation, expressed in Btu per hour. As practically all boilers are now rated on a Btu basis, it is unnecessary to convert the radiation load to square feet of equivalent direct radiation.

2. *Hot Water Supply Load.* The estimated maximum heat in Btu per hour required to heat water for domestic use.

$I = B = R$ recommends that allowance for hot water supply load be made only for bathrooms in excess of two, as follows: Instantaneous Coil 12,000 Btu per hour, and for Storage Tank installation 120 Btu per (hour) (gallon of tank capacity). For instantaneous coil installations the boiler capacity should not be less than required to heat 2 to 3 gal of water, 100 deg per min. See also Chapter 48.

3. *Piping Tax.* The estimated heat emission in Btu per hour of the piping connecting the radiation and other apparatus to the boiler.

As the heating industry as a whole is not entirely agreed upon piping tax allowances for different sizes of installations, it is better to compute the heat emission from both bare and covered pipe surface in accordance with data in Chapter 27. In average house heating systems, it is common practice to consider the piping tax to be equal to 25 percent of the Net Load. In determining Net $I = B = R$ Ratings from Gross $I = B = R$ Output, the piping factor allowed varies from 30 percent for small boilers to 12 percent for larger boilers.

4. *Warming-Up or Pick-Up Allowance.* The estimated increase in the normal load in Btu per hour caused by the heating up of the cold system.

The warming-up allowance represents the load due to heating the boiler and contents to operating temperature, and heating up cold radiation and piping. The factors to be used for determining the allowance to be made should be selected from Table 4.

TABLE 4. WARMING-UP ALLOWANCES FOR HAND-FIRED LOW-PRESSURE STEAM AND HOT WATER HEATING BOILERS[a, b, c]

DESIGN LOAD (REPRESENTING SUMMATION OF ITEMS 1, 2, AND 3)		PERCENTAGE CAPACITY TO ADD FOR WARMING-UP[c]
Btu per Hour	Equivalent Square Feet of Radiation[d]	
Up to 100,000	Up to 420	65
100,000 to 200,000	420 to 840	60
200,000 to 600,000	840 to 2500	55
600,000 to 1,200,000	2500 to 5000	50
1,200,000 to 1,800,000	5000 to 7500	45
Above 1,800,000	Above 7500	40

[a] This table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings, except that the second column has been added for convenience in interpreting the design load in terms of equivalent square feet of radiation.

[b] See also Time Analysis in Starting Heating Apparatus, by Ralph C. Taggart (A.S.H.V.E. TRANSACTIONS, Vol. 19, 1913, p. 292); Report of A.S.H.V.E. Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers (A.S.H.V.E TRANSACTIONS, Vol. 36, 1930, p. 35); Selecting the Right Size Heating Boiler, by Sabin Crocker (*Heating, Piping and Air Conditioning*, March, 1932).

[c] This table refers to *hand-fired, solid fuel boilers.* A factor of 20 percent over design load is adequate when *automatically-fired fuels* are used.

[d] 240 Btu per square foot.

Other items to be considered in boiler selection are:

a. Efficiency with hard or soft coal, gas, or oil firing, as the case may be.
b. Grate area with hand fired coal, or fuel burning rate with stokers, oil, or gas.
c. Combustion space in the furnace.
d. Type of heat liberation, whether continuous or intermittent, or a combination of both.
e. Convenience in firing and cleaning.
f. Adaptability to changes in fuel and kind of attention.
g. Height of water line.
h. Miscellaneous items such as draft available, possibility of future extension, possibility of break-down, and head room in the boiler room.
i. The most economical size of boiler is usually one that is just the right size for the load. Either larger or smaller boilers may be less economical.

Cast-Iron Boilers

Net load ratings of cast-iron boilers are usually available from manufacturers' catalogs. They may also be obtained conveniently from published tables of $I = B = R$ ratings,[6] or from recommendations of the *Heating, Piping and Air Conditioning Contractors National Association,*[7] and can be used in selection of boilers, unless the heating system contains an unusual amount of bare pipe, or the nature of the connected load is such that the normal allowances for pipe loss and pickup do not apply. In such a case, the selection must be based on the gross output.

Steel Heating Boilers

SBI catalog ratings, in accordance with the previously mentioned *Steel Boiler Institute, Inc.* code, are intended to correspond with the estimated

TABLE 5. PRACTICAL COMBUSTION RATES FOR COAL-FIRED HEATING BOILERS OPERATING AT MAXIMUM LOAD ON NATURAL DRAFT OF FROM ⅛ IN. TO ¼ IN. WATER[a]

KIND OF COAL	SQ FT GRATE	LB OF COAL PER SQ FT GRATE PER HOUR
No. 1 Buckwheat Anthracite	Up to 4	3
	5 to 9	3½
	10 to 14	4
	15 to 19	4½
	20 to 25	5
Anthracite Pea	Up to 9	5
	10 to 19	5½
	20 to 25	6
Anthracite Nut and Larger	Up to 4	8
	5 to 9	9
	10 to 14	10
	15 to 19	11
	20 to 25	13
Bituminous	Up to 4	9.5
	5 to 14	12
	15 and above	15.5

[a] Steel boilers usually have higher combustion rates for grate areas exceeding 15 sq ft than those indicated in this table.

design load. When the heat emission of the piping is not known, the net load to be considered for the boiler may be determined from Tables 1 and 2. The difference between design load and net load represents an amount which is considered normal for piping loss of the ordinary heating system.

Boilers with less than 177 sq ft of heating surface, and having *SBI* net ratings (steam) of not more than 3,000 sq ft if mechanically-fired and 2,480 sq ft if hand-fired, are classified as residence size. An insulated residence boiler for oil, gas, or stoker firing may carry a *net load* expressed in square feet of steam radiation of not more than 17 times the square feet of heating surface in the boiler, provided the boiler has been tested in accordance with the *SBI* Code for Testing Oil-Fired Steel Boilers at output rates of 125, 150, and 175 percent of the *SBI* Net Rating. The *SBI* Net Rating (square feet steam) for hand-fired residence boilers is not greater than 14 times the heating surface. If the heat loss from the piping system exceeds 20 percent of the installed radiation, the excess is to be considered as a part of the *net load.*

Heating Surface and Grate Area Basis

Where neither the *net load* nor *gross output* ratings based upon performance tests are available, a good general rule for conventionally designed boilers is to provide 1 sq ft of boiler heating surface for each 14 sq ft of equivalent radiation (240 Btu per square foot) represented by the design load. This is equivalent to allowing 10 sq ft of boiler heating surface per boiler horsepower. In this case it is assumed that the *maximum load* including the warming-up allowance will be provided for by operating the boiler in excess of the *design load,* that is, in excess of the 100 percent rating on a boiler-horsepower basis. *SBI* ratings for hand firing are based on 10 sq ft of heating surface per boiler horsepower.

Due to the wide variation which may be encountered in manufacturers' ratings for boilers of approximately the same capacity, it is advisable to check the grate area required for heating boilers burning solid fuel by means of the following formula:

$$G = \frac{H}{C \times F \times E} \tag{1}$$

where

G = grate area, square feet.

H = required *gross output* of the boiler, Btu per hour (see Selection of Boilers).

C = desirable combustion rate for fuel selected, pounds of dry coal per square foot of grate per hour (see Table 5).

F = calorific value of fuel, Btu per pound.

E = efficiency of boiler, usually taken as 0.60.

Example 1. Determine the grate area for a required *gross output* of the boiler of 500,000 Btu per hour, a combustion rate of 6 lb per hour, a calorific value of 13,000 Btu per pound, and an efficiency of 60 percent.

$$G = \frac{500{,}000}{6 \times 13{,}000 \times 0.60} = 10.7 \text{ sq ft}$$

The boiler selected should have a grate area not less than that determined by Equation 1. With small boilers, where it is desired to provide sufficient coal capacity for approximately an eight-hour firing period plus a 20 percent reserve for igniting a new charge, more grate area may be required depending upon the depth of the fuel pot.

Gas-Fired Boilers

After determining the *net load* for the installation, gas designed boilers can usually be selected from manufacturers' tables of *net load ratings* which are based on piping and pickup allowances varying from 56 percent for small steam boilers and 33.3 percent for small hot water boilers to 28.8 percent for very large boilers. If the piping and pickup load or other factors create an unusual load, a boiler should be selected which has an *A.G.A. output rating* equal to the maximum output required. Detailed recommendations for selection of gas designed boilers are given in the *A.G.A.* publication, Comfort Heating.[8]

SPACE LIMITATIONS

Boiler rooms should, if possible, be situated at a central point with respect to the building, and should be designed for a maximum of natural light. The space in front of the boilers should be sufficient for firing, stoking, ash removal and cleaning or renewal of flue tubes, and should be at least 3 ft greater than the length of the tubes.

A space of at least 3 ft should be allowed on at least one side of every boiler for convenience of erection and for accessibility to the various dampers, cleanouts, and trimmings. The space at the rear of the boiler should be ample for the chimney connection and for cleanouts. With large boilers the rear clearance should be at least 3 ft in width.

The boiler room height should be sufficient for the location of boiler accessories, and for proper installation of piping. In general, the ceiling height for small steam boilers should be at least 3 ft above the normal boiler water line. With vapor heating, especially, the height above the boiler water line is of vital importance.

CONNECTIONS AND FITTINGS

Steam outlet connections should be the full size of the manufacturers' tappings, in order to keep the velocity of flow through the outlet reasonably low, and to avoid fluctuation of the water line and undue entrainment of moisture, and should extend vertically to the maximum height available above the boiler. A steam velocity in boiler outlets not exceeding 25 to 30 fps at maximum load is recommended, unless data are available to show that a higher velocity is satisfactory. See further data on pipe connections to boilers in Chapters 20 and 21 and in the *ASME* Boiler Construction Code for Low Pressure Heating Boilers.

Where a *return header* is used on a cast-iron sectional boiler to distribute the returns to both rear tappings, it is advisable to provide full size plugged tees instead of elbows where the branch connections enter the return tappings. This aids in cleaning of sludge from the bottom of the boiler sections through the large plugged openings. An equivalent cleanout plug should be provided in the case of a single return connection.

Blow-off or drain connections should be made near the boiler, and so arranged that the entire system may be drained of water by opening the drain cock. In the case of two or more boilers separate blow-off connections must be provided for each boiler, on the boiler side of the stop valve on the main return connection.

Water service connections must be provided for both steam and water boilers, for refilling and for the addition of make-up water to boilers. This connection is usually of galvanized steel pipe, and is made to the return main near the boiler or boilers.

Fitting connections for pressure gage piping, water gage connections, and safety valves, should be made in accordance with the *ASME* Boiler Construction Code for Low Pressure Heating Boilers.

Smoke Breeching and Chimney Connections. The breeching or smoke pipe from the boiler outlet to the chimney should be air-tight and as short and direct as possible, preference being given to long radius and 45-deg instead of 90-deg bends. The breeching entering a brick chimney should not project beyond the flue lining, and where practicable it should be grouted from the inside of the chimney. A thimble or sleeve usually is provided where the breeching enters a brick chimney.

Where a battery of boilers is connected into a breeching, each boiler should be provided with a tight damper. The breeching for a battery of boilers should not be reduced in size as it goes to the more remote boilers. Good connections made to a good chimney will usually result in a rapid response by the boilers to demands for heat.

ERECTION, OPERATION, AND MAINTENANCE

The directions of the boiler manufacturer should always be read before the assembly or installation of any boiler is started, even though the contractor may be familiar with the boiler. All joints requiring boiler putty or cement, which cannot be reached after assembly is complete, must be finished as the assembly progresses.

Five precautions that should be taken in all installations to prevent damage to the boiler are:

1. There should be provided paper and convenient drainage connections for use if the boiler is not in operation during freezing weather.
2. Strains on the boiler, due to movement of piping during expansion, should be prevented by suitable anchoring of piping, and by proper provision for pipe expansion and contraction.
3. Direct impingement of too intense local heat upon any part of the boiler surface, as with oil burners, should be avoided by protecting the surface with firebrick or other refractory material.
4. Condensation in steam systems must flow back to the boiler as rapidly and uniformly as possible. Return connections should prevent the water from backing out of the boiler.
5. Automatic boiler feeders and low water cut-off devices which shut off the source of heat if the water in the boiler falls below a safe level, are recommended for mechanically-fired boilers.

Boiler Troubles

A complaint regarding boiler operation generally will be found to be due to one of the following:

1. *The boiler fails to deliver enough heat.* The cause of this condition may be: (*a*) poor draft; (*b*) poor fuel; (*c*) inferior attention or firing; (*d*) boiler too small; (*e*) improper piping; (*f*) improper arrangement of sections; (*g*) heating surfaces covered with soot; (*h*) insufficient radiation installed; and (*i*) with mechanical firing, fuel burning equipment too small.
2. *The water line is unsteady.* The cause of this condition may be: (*a*) grease and dirt in boiler; (*b*) water column connected to a very active section and, therefore, not showing actual water level in boiler; and (*c*) boiler operating at excessive rate of output.
3. *Water disappears from the gage glass.* This may be caused by: (*a*) priming due to grease and dirt in boiler; (*b*) too great pressure difference between supply and

return piping preventing return of condensation; (*c*) valve closed in return line; (*d*) connection of bottom of water column into a very active section or thin waterway; and (*e*) improper connections between boilers in battery permitting boiler with excess pressure to push returning condensation into boiler with lower pressure.

4. *Water is carried over into steam main.* This may be caused by: (*a*) grease and dirt in boiler; (*b*) insufficient steam dome or too small steam liberating area; (*c*) outlet connections of too small area; (*d*) excessive rate of output; and (*e*) water level carried higher than specified.

5. *Boiler is slow in response to operation of dampers.* This may be due to: (*a*) poor draft resulting from air leaks into chimney or breeching; (*b*) inferior fuel; (*c*) inferior attention; (*d*) accumulation of clinker on grate; and (*e*) boiler too small for the load.

6. *Boiler requires too frequent cleaning of flues.* This may be due to: (*a*) poor draft; (*b*) smoky combustion; (*c*) too low a rate of combustion; and (*d*) too much excess air in firebox causing chilling of gases.

7. *Boiler smokes through fire door.* This may be due to: (*a*) defective draft in chimney or incorrect setting of dampers; (*b*) air leaks into boiler or breeching; (*c*) gas outlet from firebox plugged with fuel; (*d*) dirty or clogged flues; and (*e*) improper reduction in breeching size.

8. *Low carbon dioxide.* This may be due on oil burning boilers to: (*a*) improper adjustment of the burner; (*b*) leakage through the boiler setting; (*c*) improper fire caused by a fouled nozzle; or (*d*) to an insufficient quantity of oil being burned.

Cleaning Boilers

All boilers are provided with flue clean-out openings through which the heating surface can be reached by means of brushes or scrapers. Flues of solid fuel boilers should be cleaned often to keep the surfaces free of soot or ash. Gas boiler flues and burners should be cleaned at least once a year. Oil burning boiler flues should be examined periodically to determine when cleaning is necessary.

The grease used to lubricate the cutting tools during erection of new piping systems serves as a carrier for sand and dirt, with the result that a scum of fine particles and grease accumulates on the surface of the water in all new boilers, while heavier particles may settle to the bottom of the boiler and form sludge. These impurities tend to cause foaming, preventing the generation of steam and causing an unsteady water line.

This unavoidable accumulation of oil and grease should be removed by blowing off the boiler as follows: If not already provided, install a surface blow connection of at least $1\frac{1}{4}$ in. nominal pipe size with outlet extended to within 18 in. of the floor or to sewer, inserting a valve in line close to boiler. Bring the water line to center of outlet, raise steam pressure, and while fire is burning briskly open valve in blow-off line. When pressure recedes, close valve and repeat process adding water at intervals to maintain proper level. As a final operation bring the pressure in the boiler to about 10 lb, close blow-off, draw the fire or stop burner, and open drain valve. After boiler has cooled partly, fill and flush out several times before filling it to proper water level for normal service. The use of soda, or any alkali, vinegar or any acid is not recommended for cleaning heating boilers because of the difficulty of complete removal and the possibility of subsequent injury, after the cleaning process has been completed.

Insoluble compounds have been developed which are effective, but special instructions on the proper cleaning compound and directions for its use, as given by the boiler manufacturer, should be carefully followed.

Care of Idle Heating Boilers

Heating boilers are often seriously damaged during summer months due chiefly to corrosion resulting from the combination of sulfur in the

soot with the moisture in the cellar air. At the end of the heating season the following precautions should be taken:

1. All heating surfaces should be cleaned thoroughly of soot, ash and residue, and the heating surfaces of steel boilers should be given a coating of lubricating oil on the fire side.

2. All machined surfaces should be coated with oil or grease.

3. Connections to the chimney should be cleaned, and in case of small boilers, the pipe should be placed in a dry place after cleaning.

4. If there is much moisture in the boiler room, it is desirable to drain the boiler to prevent atmospheric condensation on the heating surfaces of the boiler when they are below the dew-point temperature. Due to the hazard that some one may inadvertently build a fire in a dry boiler, however, it is safer to keep the boiler filled with water, particularly in residential installations. Air can be excluded from a steam boiler by raising the water level into the steam outlets. A hot water system usually is left filled to the expansion tank.

5. The grates and ashpit should be cleaned.

6. Clean and repack the gage glass if necessary.

7. Remove any rust or other deposit from exposed surfaces by scraping with a wire brush or sandpaper. After boiler is thoroughly cleaned, apply a coat of preservative paint where required to external parts normally painted.

8. Inspect all accessories of the boiler carefully to see that they are in good working order. In this connection, oil all door hinges, damper bearings, and regulator parts.

WARM AIR FURNACES

Warm air heating furnaces of a number of types and a wide range of sizes are listed and illustrated in the *Catalog Data Section*.

Warm air furnaces may be classified in several different ways:

1. According to method of heat distribution—these are either gravity or mechanical (blower) furnaces.

2. According to fuels for which the furnaces are designed—these are coal hand-fired or stoker-fired, oil, gas, or wood.

3. According to materials of construction—they are cast-iron, low carbon steel, and occasionally high temperature steel alloys.

4. According to design or construction, such as drum and radiator, tubular, horizontal, etc.

Gravity Warm Air Furnaces

A gravity furnace is one in which the motive head producing air flow depends upon the difference in density between the heated air leaving the top of the casing and cooled air entering the bottom of the casing. Since this gravity head is relatively low, the furnace must have low internal resistance to the flow of air, and relatively large areas must be available for free circulation within the furnace casing. It is common practice to provide approximately 50 percent free air area through gravity type furnaces.

Furnaces for gravity type systems are available in designs suitable for central heating, pipeless furnace, or unit floor furnace installations. Booster fans are sometimes used in conjunction with gravity design systems, to increase air circulation. Where a fan is to be used with a furnace casing sized for gravity air flow, some form of baffling must be employed to restrict the free area within the casing and to force impingement of the air against the heating surfaces. Where square casings are used, the corners must be baffled.

Mechanical Warm Air Furnaces

Mechanical or forced warm air furnaces include fans or blowers as integral parts, for the purpose of circulating the air, and usually include air filters.

Centrifugal fans with either backward or forward curved blades are the type most commonly used. Motors may be mounted on the fan shaft or connected to the fan by a belt drive. Adjustable pulleys are desirable to provide means of regulating the quantity of air distributed to the heated spaces. Either the motor load or the noise considerations may limit the maximum operating fan speed. Two-speed motors have given successful operating results. Motors and mountings must be carefully selected for quiet operation. Electrical conduit and water piping must not be fastened to, nor make contact with the fan housing.

Filters

Several types of filters are available for mechanical warm air furnace applications, and are discussed in Chapter 33. For maximum efficiency and life under operating conditions, filters should not be subjected to a temperature in excess of 150 F. Filters should have at least 80 percent average efficiency on an 8-hr test at a maximum resistance of 0.25 in. of water. Filter resistance rises rapidly with the accumulation of dirt, and may reduce the air circulation over heating surfaces. In domestic furnaces, the maximum velocity, based on nominal filter area, should not exceed 300 fpm.

Fuel Utilization

A combustion rate of from 5 to 8 lb of coal per (square foot of grate) (hour) is recommended for residential furnaces. A higher combustion rate is permissible with larger furnaces for buildings other than residences, depending upon the ratio of grate surface to heating surface, firing period, and available draft.

In residential furnaces for coal burning, the ratio of heating surface to grate area will average about 20 to 1; in commercial sizes the ratio may be as high as 50 to 1, depending on fuel and draft. Furnaces may be installed singly, each furnace with its own fan, or in batteries of a number of furnaces, using one or more fans.

Where oil fuel is used, care must be exercised in selecting the proper size and type of burner for the particular size and type of furnace used. Furnaces for burning oil fuel are usually designed for blow-through installations so that the pressure in the air space is higher than that in the combustion chamber or flues. The *National Warm Air Heating and Air Conditioning Association* has prepared a Tentative Code for Testing and Rating of Oil-Fired Furnaces. Compact fan-furnace-burner units are available, suitable for basement, closet, or attic installations.

Gas-fired forced air furnaces should conform in construction and performance to *A.G.A.* Approval Requirements.

Heavy Duty Fan Furnaces

Fan furnaces for large commercial and industrial buildings, churches, schools, etc., are available in sizes ranging from 300,000 to 6,000,000 Btu per (hour) (unit). Heavy duty furnace heaters may be arranged in battery combinations of one or more units.

Most manufacturers of heavy duty furnaces rate their furnaces in Btu per hour, and also in the number of square feet of heating surface. Conservative practice indicates that at no time in the heating-up period should the furnace surface be required to emit more than an average of 3500 Btu per square foot. A higher rate of heat emission tends to increase the heat loss up the chimney, and raise fuel consumption, to shorten the life of the furnace, and to overheat the air. The ratio of heating surface to grate area of furnaces for this type of work should never be less than 30 to 1 and, as indicated previously, may run as high as 50 to 1.

Control of temperature is secured through (1) controlling the quantity of heated air entering the room, (2) using mixing dampers, or (3) regulating the fuel supply.

The design of heavy duty fan furnace heating systems is in many respects similar to that of the central fan heating systems described in Chapter 29. Ducts are designed by the method outlined in Chapter 31.

MATERIALS AND CONSTRUCTION

Cast-Iron Furnaces

Cast-iron furnaces are made in a multiplicity of designs or shapes. For solid fuels they are generally of round sectional construction, the sections being cemented or bolted together. Various types of radiators for secondary convection heat transfer are employed. Such radiators are of the circular, doughnut type, or tubular type.

Cast-iron is frequently used in the construction of gas or oil-fired furnaces, designs varying considerably with two general types in common use: multi-sectional type, and those with single combustion chambers having auxiliary secondary surface.

Cast-iron furnaces are made in capacities ranging from those for small insulated residence application with inputs of 40,000 Btu per hour or less, to capacities as large as 600,000 Btu per hour.

Cast-iron furnaces are usually constructed with a minimum sectional thickness of $\frac{1}{4}$ in., and effectively resist high temperatures and corrosion. They usually have a fairly large heat capacity because of their mass, which provides a distinct *fly wheel* or carry-over heating effect.

Steel Furnaces

Formed sheet steel construction is frequently used in furnace design. Welding, riveting, or both are used to join the formed metal. The use of steel castings, however, is rare, because of the cost, and because high stresses are not encountered in normal furnace construction. Types of design employed vary greatly, although perhaps the most common type consists of a drum and circumferential or rear radiator. Steel gas furnaces may also be sectional in design, or may be combinations of common combustion chambers and sectional or tubular radiation surfaces connected to a flue gas collector.

Steel furnaces are made in capacities ranging from 40,000 Btu per hour to capacities as large as 600,000 Btu. Steel furnaces have low heat capacities as a result of their relatively low mass and, therefore, deliver heat rapidly on demand.

FURNACE RATING

Warm air furnaces are generally rated in Btu per hour output at the bonnet (point of heat generation) or at the register (point of heat delivery).

Rating Equations for Gravity Warm Air Furnaces[8]

Until a method of testing and rating gravity warm air furnaces has been developed, the following empirical rating equations are recommended by the *National Warm Air Heating and Air Conditioning Association.*

Gravity warm-air furnaces of conventional design, having ratios (of heating surface to grate area) of 15 to 1 or greater, and having a ratio of casing area to face area not less than 0.4, are rated by the following equations:

1. *Hand-fired furnaces converted to Stoker, Gas, or Oil Firing.*

$$\text{Bonnet Capacity in Btu per hour} = 1785 \times S \times 1.333 \qquad (2)$$

2. *Hand-fired furnaces, with ratios of heating surface to grate area greater than 15 to 1 and less than 25 to 1.*

$$\text{Bonnet Capacity in Btu per hour} = 1785 \times S \times 1.333 \qquad (3)$$

3. *Hand-fired furnaces with ratios of heating surface to grate area in excess of 25 to 1.*

$$\text{Bonnet Capacity in Btu per hour} = 1785 \times 25 \times G \times 1.333 \qquad (4)$$

where

S = heating surface, in square feet.
G = actual grate area, in square feet.

The *Register Delivery Rating* is equal to 0.75 x (*Bonnet Capacity*). The *Leader Pipe Rating* in square inches, formerly used as a rating unit, may be found by dividing the *Register Delivery Rating* by 136.

Heating Surface of Furnace

Prime heating surface is defined[9] as surface above the top of the grate having *hot gases or live fuel on one side and circulating air over the other*, and in all cases is measured on the exterior or air side. The areas of the outer casing, the inner liner, and any radiation shields shall not be considered as heating surface.

In determining the amount of heating surface, extended surfaces are considered to be prime heating surface subject to the following limitations:

1. Extended heating surface may consist of fins, ribs, webs, lugs, or other projections from the prime heating surface. Projections less than ½ in. thick at the base, and extending more than 1 in. from the prime surface are classified as fins.

2. Integral fins are continuously welded to, or cast as a part of, the prime heating surface. Both sides are included as heating surface, subject to the following allowances:

Distance from Prime Surface	1st inch	2nd inch	3rd inch	Over 3 in.
Ratio of Effective Area to Total Area	0.40	0.30	0.20	None

3. Non-integral fins are spot welded to, or otherwise held in line contact with the prime heating surface. Both sides are included as heating surface, subject to the following allowances:

Distance from Prime Surface	1st inch	2nd inch	3rd inch	Over 3 in.
Ratio of Effective Area to Total Area	0.30	0.20	0.15	None

4. In the case of ribs, webs, or lugs more than ¼ in. thick at the base and extending less than 1 in. from the prime surface, the entire surface in contact with circulating air in included as heating surface.

5. In the case of ribs, webs, or lugs more than ¼ in. thick at the base and extending more than 1 in. from the prime heating surface, the areas of both sides of the first inch are included as prime heating surface. The portions projecting beyond 1 in. are treated as integral fins.

Grate Area

Grate area is defined[9] and treated for purpose of rating as follows:

1. The *nominal grate area* is defined as the total cross-sectional area of the bottom of the firepot. In steel furnaces the nominal grate area is the cross-sectional area inside the firebrick lining.

2. The *actual grate area*, used for calculating the ratios of heating surface to grate area, is the nominal grate area minus certain areas that cannot be considered as part of the grate itself. The following rules govern these deductions: (1) If a solid, continuous ledge extends around the grate and inside the firepot, any area of this ledge extending inside of a circle, the diameter of which is 1 in. less than the diameter of the bottom of the firepot, shall be deducted. (2) If separate, solid projections extend from the firepot towards the grate, the areas of any portions of these projections extending inside of a circle, the diameter of which is 3 in. less than the diameter of the bottom of the firepot, shall be deducted. (3) In the case of grates which are inclined, or are conical, the projected area is the same as the nominal grate area. The latter should, therefore, be used after making any necessary deductions.

Ratings for Forced Air Furnaces

For solid fuel burning, forced air furnaces having bonnet capacities between 80,000 and 250,000 Btu per hour, no standard method of test has been accepted, although eventually such codes will be developed. The *National Warm Air Heating and Air Conditioning Association* recommends the following empirical equations for use in rating solid fuel forced air furnaces:

1. *Hand-fired furnaces converted to Stoker, Gas, or Oil Firing.*

$$\text{Bonnet Capacity in Btu per hour} = 2265 \times S \times 1.177 \tag{5}$$

2. *Hand-fired furnaces, with ratios of heating surface to grate area greater than 15 to 1 and less than 25 to 1.*

$$\text{Bonnet Capacity in Btu per hour} = 2265 \times S \times 1.177 \tag{6}$$

3. *Hand-fired furnaces with ratios of heating surface to grate area in excess of 25 to 1*

$$\text{Bonnet Capacity in Btu per hour} = 2265 \times 25 \times G \times 1.177 \tag{7}$$

where

S = heating surface, in square feet.

G = actual grate area, in square feet.

The *Register Delivery Rating* is equal to 0.85 x (*Bonnet Capacity*).

The following testing and rating codes have been generally accepted in the industry:

Commercial Standards CS-109-44 for rating solid fuel-burning, forced-air furnaces having bonnet outputs of 80,000 Btu per hour or less. This provides a method of rating small coal-fired forced-air furnaces by test.

A Tentative Code for Testing Oil-Fired Furnaces. This code has been adopted by the *National Warm Air Heating and Air Conditioning Association* for rating oil-fired furnaces by test.

The American Gas Association method of rating gas-fired furnaces on performance under tests. This is described in the Approval Requirements for Central Heating Gas Appliances.

Commercial Standards 113-44 is a method of rating oil-burning floor furnaces by test.

Commercial Standards CS 104-46 is a method of rating warm air furnaces equipped with pot-type oil burners by test.

Various codes covering the construction and performance of appliances as related to fire hazards have been developed by Underwriter Laboratories, Inc. In addition, there are many municipal codes[10] which regulate construction and installation of furnace equipment.

The yardstick of the *National Warm Air Heating and Air Conditioning Association* provides criteria for evaluating a furnace design and installation against industry accepted standards.

FURNACE EFFICIENCY

Rating formulas of the *National Warm Air Heating and Air Conditioning Association* are based on 55 percent efficiency for gravity coal furnaces and 65 percent efficiency for forced-air coal furnaces. In the tentative Oil Testing Code the contemplated minimum efficiency is 70 percent for oil-fired forced-air furnaces. Gravity gas furnaces approved by the *American Gas Association* are assigned a rating based on 75 percent efficiency. All *forced-air gas-fired furnaces* approved by *American Gas Association* are assigned a rating based on 80 percent efficiency.

DESIGN CONSIDERATIONS

Considerations of prime importance in the design of warm air furnaces and some general suggestions to be observed in connection with each, are as follows:

1. *Adequate heat transfer surface.*
   a. Heat transfer rates of 2,000 to 4,500 Btu per (hour) (square foot) of heating surface may be obtained without unduly high metal temperatures.
   b. Fins, pins and bosses are frequently used to add surface and to break down superficial gas films, both on gas-to-metal and metal-to-air surfaces.
   c. Surface and stack (flue gas) temperatures are good indications of the amount and effectiveness of the heating surfaces.
2. *Safe and efficient combustion of fuel.*
   a. Proper mixture of fuel and air is necessary for efficient combustion. This necessitates careful attention to the design of grates, nozzles, burners, air inlet areas and location, and combustion chamber baffling.
   b. Regulation of the quantity and the distribution of the air for combustion should be provided by use of check dampers, draft regulators, draft hoods, air shutters and air orifices.
   c. Total draft loss through appliances should not exceed that available from chimneys which would normally be obtainable in the size of building which the appliance will supply with heat.
   d. The use of ignition safety devices such as safety pilots, hold-fire controls, and the like is recommended.
3. *Fuel capacity of appliance.*
   a. With solid fuels adequate coal capacity should be provided for at least 5 hr of operation at the maximum rated combustion rate.
4. *Adequate circulation of air over heating surface.*
   a. In gravity furnaces, free air space between casing and heat exchanger should be great enough to permit free flow over all surfaces.
   b. Forced air furnace design must include fans having proper capacity and suitable performance characteristics. Internal static pressures must be minimized without losing the advantages of high velocity circulation over the heat exchanger surfaces.
   c. The air flow over the heating surface must be directed to obtain maximum efficiency and to eliminate hot spots and air noises.
   d. Air velocities at bonnet should not be much in excess of 1,000 fpm, and air temperature distribution at the furnace outlet should be uniform within approximately $\pm$ 30 deg.

5. *Durability.*
   - a. A minimum metal weight for gas-fired heat exchangers is established as No. 20 U.S. Gage for plain carbon steel by the *A.G.A.* Approval Requirements for Central Heating Gas Appliances, with some municipal codes specifying 18 gage. Cast-iron sectional thicknesses of ¼ in. to ⅜ in. are recommended.
   - b. Added strength and reinforced designs may be required to preclude damage in shipment, burning out from overfiring, or corrosion from condensation.
   - c. Maximum heat exchanger surface temperatures which may be used vary with the metal. The *American Gas Association* Approval Requirements for Central Heating Gas Appliances specify a maximum of 875 F for cast-iron or steel gas furnaces, and the *National Bureau of Standards* CS 109-44 Code for Forced Air Solid Fuel-Burning Furnaces specifies 1000 F as a maximum surface temperature. These temperatures define the range in which oxidation of non-alloy ferrous metal begins. The use of proper alloy additions increases the temperature resistance properties of metals.
   - d. Casing temperatures should be controlled so that they do not become hazards to burn those who touch them, or to create fires.
6. *Serviceability.*
   - a. Those parts of the furnace which may be subject to soot, fly-ash, or condensation deposits should be accessible for cleaning.
   - b. Parts which may require adjustments or replacements, such as grates, baffles, liners, controls, should be removable.
   - c. Furnaces should be so designed that they can be installed with a minimum of difficulty.
7. *Control.*
   - a. Thermostatic controls of various types should be used to correlate space temperatures with unit operation.
   - b. Controls should be provided wherever possible, to prevent the occurrence of excessive temperatures or other conditions in any part of the unit which might cause unsafe operation.
8. *General design considerations.*
   - a. Furnace casings are normally constructed of formed and painted sheet steel or of galvanized iron. The casing should be protection from excessive radiation losses and temperatures by use of insulation or sheet steel air space liners. Liners should extend from the grate level to the top of the furnace and should be spaced from 1 in. to 1½ in. from the outer casing.
   - b. The hood or bonnet of the casing above the furnace should be as high as basement conditions will allow, to form a plenum chamber over the top of the furnace. This tends to equalize the pressure and temperature of the air leaving the bonnet through the various openings. It is generally considered advisable to take off the warm air pipes from the side of the bonnet near the top, as this method of take-off allows the use of a higher bonnet and thus provides a larger plenum chamber.
   - c. Warm air outlet and return air connections should be designed so that the ductwork may be easily attached. A ⅞ in. flange is normally used for this purpose.
   - d. Suitable provision should be made in appliances so that the controls and humidifiers may be installed in the proper location. When these auxiliary units are installed in the ductwork, detailed instructions should be provided to insure their proper location.
   - e. The flue connection should be of integral flue pipe size, and provision should be made to attach the flue pipe to the flue outlet of the furnace.

HUMIDIFICATION EQUIPMENT

Evaporating pans are usually located in the outlet air. There is a present trend toward heating the water. Equipment for doing this may make use of sprays, or it may take the form of water circulating coils placed within the combustion chamber, and connected by pipes to the humidifier pans where a constant water level is maintained by some separate float device. All humidifiers require provision for removal of dirt and lime.

SPACE HEATERS

Space heaters may be classified in several ways, such as:

1. By the type of fuel used as coal, wood, gas, and fuel oil.
2. According to the method of heat distribution as circulators or radiant types. A radiant heater is one in which the heat exchanger surface is exposed directly to the room atmosphere, and the generated heat is dissipated primarily by radiation. A circulating heater is essentially a jacketed radiant heater from which circulation of room air is promoted by the chimney effect caused by the movement of air passing upward between the jacket and heat exchanger surface.
3. According to method of design for particular fuel types, such as: (a) surface-fired and magazine-feed for solid fuels, (b) vaporizing pot-type and blue-flame heater for oil, and (c) vented and unvented heaters for gas. (The type of gas burner design, such as injection, yellow flame, power, and pressure, may also be mentioned.)

SOLID FUEL-FIRED HEATERS

Surface-fired heaters normally have a front firing door and are operated with relatively shallow fuel beds. A magazine-feed heater includes a deep reservoir of fuel to lengthen the attention intervals. In a true magazine feed heater the rate of fuel ignition would be equal to the rate of burning; and self-feeding should operate to move the unburned fuel by gravity flow from the magazine section into the hearth area. However, the ideal balance between rate of ignition and rate of burning is virtually impossible to attain for any solid fuel under normal usage, although self-feeding may be obtained with wood and some free burning coals. Thus, a magazine-type space heater is essentially a deep surface-fired heater, its principal difference being increased fuel capacity.

Materials and Construction

There is no accepted code governing the construction of solid fuel burning space heaters. In past years cast-iron was used predominantly in the construction of coal and wood heaters, with the exception of the so-called *air-tight* heaters designed as low-cost wood-burning units with little consideration for long life, and stoves were priced on a poundage basis. The present trend is toward fabricated steel parts and welded assemblies, although cast-iron is still used for grates, firebox liners, and parts subject to high temperatures. Refractory firebox liners are also used quite extensively.

Formed sheet steel is used predominantly for the outer jacket of circulator heaters, although heaters with outer casings formed from cast-iron are still readily available. Circulator cabinets normally have surfaces finished with a porcelain enamel, while the casing of a radiant heater is finished with an air-dried japan or a baked enamel.

Both welding and stove bolts are used in unit assembly, and stove cement is used on section joints to prevent air leakage. This latter is extremely important to obtain a low rate of combustion when desired.

Testing and Rating

There is no accepted code governing the method of testing and rating solid-fuel space heaters. A tentative procedure, TS-3443, has been issued by the *Division of Trade Standards, National Bureau of Standards*, but is based on use of anthracite as a rating fuel, although bituminous coals are used predominantly as heating fuels. This procedure, which has been the basis of published ratings, consists of determining the heater output, expressed in Btu per hour, by the indirect method. The measurable heat losses: (1) loss due to moisture in the fuel, (2) loss due to heat in the dry flue gases, (3) loss due to unburned carbon monoxide, and (4) loss due to unburned combustible in the ash and refuse, are measured by test. The total of these four measurable losses, plus an assumed value for unaccounted for losses, are then subtracted from the heat input. The difference multiplied by the burning rate in pounds per hour is the heater output.

When using anthracite as the rating fuel, the unaccounted for loss has been assumed to be zero. It has been accepted practice to use a 20 percent allowance for the unaccounted for losses when burning a bituminous coal. The value of this factor has been under study and, although test work is incomplete, a value of 12 percent of the heat input has been determined as representing the losses due to smoke and unburned hydrocarbons for a surface-fired heater, when burning a high volatile bituminous coal.

No allowance is made for a radiation loss, as this is useful heat.

Design Considerations

Some important considerations in the design of solid fuel space heaters are:

1. Suitable protection by baffling or insulation against overheating of floors and walls. Although there are no industry-accepted standards by which floor and wall temperatures may be determined, some indication of heater performance, with regard to overheating, may be found by the use of the corner booth test arrangement described in *National Bureau of Standards* Commercial Standard CS 103-43 and *Underwriter's Laboratories* Standard, Subject 896, mentioned in following section on Oil Heaters.
2. Tight heater construction to prevent air leakage, and to enable maintenance of a suitably low minimum burning rate. This includes a ground, paper tight joint between the ashpit door and door frame.
3. Sufficient free air space, between the casing and heat exchanger of circulating heaters, to permit free air flow over all surfaces and maintain a suitably low casing temperature.
4. Protection of all metal parts from deterioration due to high temperature. This may be accomplished by: (a) fabrication of certain parts from cast-iron or an alloy iron, (b) protection by a refractory liner, (c) use of high temperature enamel coatings, (d) directing air against hot spots.
5. Proper admission of secondary air to complete the combustion process. Care should be taken to prevent this air from also functioning as primary air.
6. Strength in assembly to prevent transportation and use damage.

Considerable work has been directed in the past few years towards improvement of the performance of bituminous coal-fired space heaters, with particular reference to smokeless operation under conditions of normal operation such as obtained in homes. A recent paper describes the smokeless coal heater developed by *Bituminous Coal Research, Inc.*, wherein smokeless combustion is obtained by admitting secondary air through a narrow slot, extending from side to side, above the edge of the fuel bed where the gas leaves to enter a vertical gas passage.[11] Complete mixing of the volatile material, released from the coal in the magazine, with the secondary air supplied is obtained as both streams pass under the bottom of the arch. Complete combustion results from this intimate mixing in a region which maintains itself at high temperatures even during banking periods.

OIL HEATERS

Vaporizing pot-type oil heaters consist of: (1) a metal pot in the bottom of which the oil is vaporized, the vapors burning at or near the top of the pot; (2) a secondary combustion chamber, or heat exchanger, in which combustion is completed. The flue connection is made to this chamber or through a second heat exchanger which may be of the diving-flue type installed to increase efficiency. The burner may be designed for operation both with and without mechanical draft.

Blue-flame oil burners differ from the pot-type variety in that removable perforated sleeves are provided above an oil pan instead of a metal pot, and lighting rings or kindlers are used for easy lighting.

Both types of oil burners operate by the burning of the oil vapor rather than the oil itself, the oil being first fed to a chamber in which the oil is entirely vaporized, then mixed with air introduced through suitably located ports and burning at the top of the pot or perforated sleeves. Such heaters are designed to burn No. 1 oil (See Table 4, Chapter 13) or kerosene (coal oil). At no time should oil heavier than that for which the burner is designed be used, as heavier oils may cause excessive carbonization in the burner or fuel feed line.

Materials and Construction

Formed sheet steel and welded assemblies are used primarily in oil heater construction. Standards governing construction which have been generally accepted are:

1. Commercial Standard for Flue-Connected Oil-Burning Space Heaters equipped with Vaporizing Pot-Type Burners, CS101-43 (*National Bureau of Standards*).
2. Standard for Oil-burning Stoves, Subject 896 (*Underwriters' Laboratories, Inc.*).
3. Standard for Construction and Performance of Oil Burners for Installation in Stoves and Ranges, Subject 865 (*Underwriters' Laboratories, Inc.*).

Some States or municipalities have codes which apply locally, but these usually apply primarily to installation, and the *Underwriters' Laboratories* label of approval is sufficient to cover acceptance of the unit.

Testing and Rating

Commercial Standard CS101-43 is intended to provide a uniform standard method for ascertaining the maximum practical heat output in Btu per hour of flue-connected oil-burning space heaters under approximately normal service conditions. This method is based upon the following equations:

$$H_r = A - B \tag{8}$$

and

$$E = H_r/A \tag{9}$$

where

A = total heat of fuel used.

B = heat lost in flue gases.

H_r = net heat delivered to the room.

E = unit efficiency.

The following minimum performance requirements are stipulated:

1. Adequate provision for ease of lighting and insurance against loss of ignition prior to heating of burner.
2. Ease of operation of controls.

3. Proper operation of burner without excessive carbonization with grades of oil recommended by the manufacturer.

4. The heater shall be capable of passing the 6 percent *ICHAM* smoke test.

5. The heater shall be capable of operating with an overall efficiency of not less than 70 percent under conditions of test, or at a lower stack draft recommended by the manufacturer.

Design Considerations

Some factors important in the design of oil-burning heaters are:

1. Proper pitch of oil lines from the sump to the burner, thus preventing vapor and air lock.
2. Proper positioning of the oil sump or constant-level valve to maintain the proper oil level in the burners, if factory assembled
3. Tight construction, not only of oil lines, but of oil tank, sump, and burner to prevent a hazardous condition due to oil leakage
4. Provision for leveling and aligning the entire heater for maintenance of proper operation. If a separate fuel tank is used, the heater should have provision for secure fastening to the floor to prevent excessive strain on oil supply line, and the consequent danger of leakage of oil.
5. Provision of a draft regulator to prevent abnormal draft fluctuations.
6. Proper shielding of an attached fuel oil supply tank to prevent excessive oil temperatures.
7. All metal parts subjected to the corrosive action of the oil shall be made of noncorrodible metal, or of metal suitably coated to resist corrosion.
8. The heater should have suitable baffling or insulation to prevent overheating of floors and walls.
9. Strength in assembly to prevent transportation and use damage.

GAS HEATERS

Vented gas heaters are defined as those capable of removing 90 percent of the flue gases through a single flue outlet. All heaters having a gas input rating in excess of 50,000 Btu per hour must be of the vented type in order to meet *ASA* Approval Requirements for Gas-Fired Room Heaters.[12]

Space heaters may be classified by burner type as follows:

1. *Injection Burner* type which employs the energy of a jet of gas to inject air for combustion into the burner and mix it with the gas.
2. *Yellow Flame Burner* type in which secondary air only is depended on for the combustion of the gas.

Materials and Construction

Standards covering materials and accessories used in the construction of gas heaters are described in ASA Approval Requirements for Gas-Fired Room Heaters[12] and in applicable Listing Requirements.[13]

Efficiency Requirement

Vented space heaters having input ratings in excess of 20,000 Btu per hour are required to have a heating efficiency of not less than 70 percent,[12] based on the total heating value of the gas. Vented space heaters having input ratings of 20,000 Btu per hour or less are required to have a heating efficiency of not less than 65 percent.[12] These efficiencies are based upon the following equation:

$$e_t = 100 - \frac{H_f}{q} \times 100 \tag{10}$$

where

q = hourly gas heat input, Btu per hour.

H_f = heat above room temperature carried away by the flue products, Btu per hour.

e_t = heating efficiency, percent.

Radiant heaters are required to have a radiant efficiency of not less than 28 percent.

Design Considerations

Some factors important in the design of gas heaters are:

1. Proper design of the burner head, port sizes and locations so that the flame will not lift, float, or flash back, and be excessively noisy in operation.
2. Proper venting of combustion chamber for relief of forces resulting from ignition of an explosive mixture of gas and air.
3. Protection of valve handles to prevent excessive temperature rise during operation.
4. Insulation and baffling of heater to prevent over-heating of walls and floor.

INSTALLATION OF SPACE HEATERS

The two most important considerations involved in the installation of a space heater are safety and chimney draft. The items of chimney details and flue connections which should have special attention, are outlined in Chapter 16, Chimneys and Draft Calculations. In all cases, it is recommended that installation be made in accordance with the current National Building Code.

REFERENCES

[1] See A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, pp. 332 and 332.

[2] See A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 42.

[3] See A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 23.

[4] See A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 366.

[5] $I=B=R$ Testing and Rating Code for Low Pressure Heating Boilers, 1950 (*Institute of Boiler and Radiator Manufacturers*).

[6] $I=B=R$ Ratings for Cast-Iron Boilers (*Institute of Boiler and Radiator Manufacturers*).

[7] Engineering Standards, Part II, Net Square Feet Radiation Loads in 70 Deg Fahr, Recommended for Low Pressure Heating Boilers, 1948 (*Heating, Piping and Air Conditioning Contractors National Association*).

[8] Comfort Heating, 1938, pp. 35 to 39 (*American Gas Association*).

[9] Gravity Code and Manual for the Design and Installation of Gravity Warm Air Heating Systems, Section No. 5, Third Edition, Jan. 1947 (*National Warm Air Heating and Air Conditioning Association*).

[10] Recommended forms for municipal installation and fire codes are included in Manual 7—Code and Manual for design and Installation of Warm Air Winter Air Conditioning Systems, Second Edition, 1947 (*National Warm Air Heating and Air Conditioning Association*).

[11] The Development of a Design of Smokeless Stove for Bituminous Coal, by B. A. Landry and R. A. Sherman, presented at the 1948 Annual Meeting of the *ASME*.

[12] American Standard Approval Requirements for Gas Fired Room Heaters, *ASA* Z21.11, 1950 with addenda Jan. 1, 1951 (*American Standards Association*).

[13] American Standard Listing Requirements for: Automatic Pilots, Z21.20, 1951, Gas Appliance Thermostats Z21.23, 1941, Domestic Gas Pressure Regulators, Z21.18, 1936, with addenda effective June 15, 1935, July 8, 1938, Automatic Main Gas-Control Valves, Z21.22, 1949 (*American Standards Association*).

CHAPTER 16

CHIMNEYS AND DRAFT CALCULATIONS

Theoretical Draft; Factors Affecting Required Draft; Industrial Chimneys, Available Draft, Determining Chimney Size; Residential Chimneys, Available Draft, Determining Chimney Size; Draft Requirements of Residential Appliances; Chimneys for Gas Heating; Recommendations of the National Board of Fire Underwriters; General Considerations

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A DRAFT to the layman, is a current of air, and the draft of a furnace or boiler is the current of air which flows through the firebox and furnishes the oxygen for combustion. To the engineer, however, the word *draft* has come to mean the pressure difference which causes this current of air to flow.

The engineering concept of draft will be used in this chapter; hence, *draft* will be defined as a negative differential pressure, constituting the absolute pressure at some point in the flue, less the absolute atmospheric pressure. The opposite of draft will be called *positive pressure* and will be defined as a positive differential pressure.

Draft is usually measured in inches of water. It is most commonly measured at the thimble, where the breeching enters the chimney proper, although it may be measured in the firebox, the smoke breeching, the base of the chimney, or elsewhere, depending upon the type of chimney installation.

Draft may be classified as either natural or mechanical, depending on whether it is produced by a chimney or by a blower. Mechanical draft is further classified as induced or forced, depending on whether the air is drawn through or forced through the combustion chamber.

THEORETICAL DRAFT

If the air in one of two equal chimneys is heated, while that in the other is not, the air in the heated chimney will be lighter than that in the other chimney, and a manometer or other pressure gage connecting the two at the bottom will indicate a pressure difference, called natural draft. The pressure of the air at the tops of the two chimneys will be equal, so that the pressure difference between them at the bottom will depend only on their height and the difference in density of the air they contain. As the density of the air in either chimney is inversely proportional to its absolute temperature, the difference in pressure between them at the bottom will be proportional to their height and to the difference between the reciprocals of the absolute temperatures within them.

As the pressure at the bottom of an unheated (and uncooled) chimney will be the same as that of the air outside, the unheated chimney can be dropped from the foregoing illustration. The manometer reading will be the same if its free connection is left open to the atmosphere.

These considerations, in conjunction with those of barometric pressure and the difference in density of flue gases from that of air, lead to the following formula:

$$D_t = 2.96\, HB_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) \quad (1)$$

where

H = height of chimney, feet.

B_o = existing barometric pressure, inches of mercury.

W_o = density of air at 0 F and 1 atmosphere pressure, pounds per cubic foot.

W_c = density of flue gas at 0 F and 1 atmosphere pressure, pounds per cubic foot.

T_o = temperature of air surrounding the chimney, Fahrenheit degrees absolute.

T_c = average or effective temperature of the gases in the chimney, Fahrenheit degrees absolute.

The quantity D_t, found by the formula, is the pressure difference between the gas inside and air outside of the chimney, in inches of water, when no flow occurs in the chimney. The quantity is variously known as

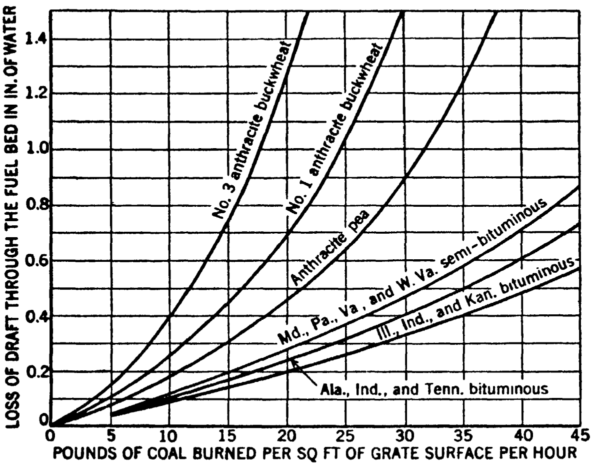

FIG. 1. DRAFT REQUIRED AT DIFFERENT RATES OF COMBUSTION FOR VARIOUS KINDS OF COAL

the theoretical draft, the static draft, or the computed draft. It is very useful in predicting and analyzing chimney performance, but it is seldom, if ever, attained in an actual chimney because of the friction incident to gas flow and the effects of wind.

The efficiency of a chimney is defined as the ratio of the observed draft or available draft, produced by the chimney for a given inlet temperature and flow rate, to the ideal draft that would be observed if the flue gases traversed the chimney without cooling and without friction. The chimney efficiency may be calculated as follows:

$$\text{Efficiency} = \frac{\text{final measured draft}}{\text{ideal draft calculated from the inlet temperature}} \quad (2)$$

FACTORS AFFECTING REQUIRED DRAFT

Before the proper chimney can be selected for an installation, the required draft of the combustion unit must be known. The required draft is, of course, equal to the sum of all the resistances to gas flow from the ash pit door to, and including, the chimney connection.

Fig. 1 presents information on the fuel-bed draft loss for various kinds of coal burned at different rates. Rough generalizations can be given for the losses in the flue passages of boiler or furnace, but, on account of the great differences in such devices, more reliable data on their flue gas volume, temperature and flue resistance should be obtained for design purposes from their respective manufacturers.

Flue gases encounter resistance to flow in breechings or smoke pipes, and this can probably be treated with sufficient accuracy by means of the method used for air ducts. (See Chapter 31.) The friction in straight ducts can be estimated by means of the last term of Equations 3 and 4.

Also, the temperature of flue gases falls during passage through breechings or flue pipes. For uninsulated surfaces this probably can be adequately estimated by assuming a loss of heat from the flue gas of 3 Btu per (hr) (sq ft) (Fahrenheit deg temperature difference between the gases and surrounding air).

INDUSTRIAL CHIMNEYS

Chimneys can be classified as residential and industrial, the chief difference being their sizes and the types of draft. Chimneys over approximately $1\frac{1}{2}$ ft in diameter are in the industrial chimney class, and their requirements should be treated accordingly. The majority of industrial chimneys operate under induced or forced draft, resorting to natural draft operation only in the case of emergencies. They are built of brick, concrete, or steel, depending upon economy and the type of installation needed. Proper height is of importance because of removal of waste products, inasmuch as the products of combustion are often deflected downward around the chimney and, with the large amount of gases that are exhausted to the atmosphere through the industrial chimney, downwash can be very objectionable.[1]

AVAILABLE DRAFT FOR THE INDUSTRIAL CHIMNEY

The available draft, D_a, *for large chimneys and stacks* has been estimated with apparent satisfaction in the past by means of formulas which in effect deduct an estimated friction loss from a theoretical draft determined as in Equation 1. The friction loss can be estimated by means of one of the formulas available for ducts, such as the Fanning equation. This procedure results in formulas for the available draft as follows:

For a *cylindrical* stack:

$$D_a = 2.96\,HB_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.00126W^2T_c fL}{D^5B_oW_c} \tag{3}$$

and for a ***rectangular*** stack:

$$D_a = 2.96\,HB_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.000388W^2T_c fL(x + y)}{(xy)^3B_oW_c} \tag{4}$$

where

D_a = available draft, inches water gage.

H = height of chimney above inlet, feet.

B_o = existing barometric pressure, inches of mercury.

W_o = density of air at 0 F, 1 atmosphere pressure.

W_c = density of flue gas at 0 F, 1 atmosphere pressure.

T_o = temperature of atmosphere, Fahrenheit degrees, absolute.

T_c = temperature of flue gas, Fahrenheit degrees, absolute.
W = flue gas flow rate, pounds per second.
f = coefficient of friction.
L = length of friction duct (approximately equal to H), feet.
D = minimum diameter of round chimney, feet.
x and y = length and width of cross-section of rectangular chimney, feet.

The following notes facilitate the use of Equations 3 and 4.

1. The *barometric pressure*, represented by B_o, is the actual pressure at the site of the chimney and not the pressure reduced to sea level datum.

In general, the barometric pressure decreases approximately 0.1 in. Hg per 100 ft increase in elevation.

2. The *unit weight of a cubic foot* of chimney gases at 0 F and sea level barometric pressure is given by the equation:

$$W_c = 0.131CO_2 + 0.095O_2 + 0.083N_2 \tag{5}$$

In this equation CO_2, O_2 and N_2 represent the percentages of the parts by volume of the carbon dioxide, oxygen and nitrogen content, respectively, of the gas analysis. For ordinary operating conditions, the value of W_c may be assumed at 0.09.

The density effect on the chimney gases, due to superheated water vapor resulting from moisture and hydrogen in the fuel, or due to any air infiltration in the chimney proper, is disregarded. Though water vapor content is not disclosed by Orsat analysis, its presence tends to reduce the actual weight per cubic foot of chimney gases.

3. The *atmospheric temperature* is the actual observed temperature of the outside air at the time the analysis of the operating chimney is made. The mean atmospheric temperature in the temperate zone is approximately 62 F.

4. The *chimney gas temperature* decreases from the breeching connection to the top of the stack. This drop in temperature depends upon the material and construction of the stack, its tightness or freedom from leaks, its area, its height, and the velocity of the gases through it. The same chimney will suffer different temperature losses depending upon the capacity under which it is working, and the variable atmospheric conditions. No general equation covering all these variables has been suggested, but from observations on chimneys varying in diameter from 3 to 16 ft, and in height from 100 to 250 ft, Equation 6 was deduced:[2]

$$T_c = \frac{3.13T_1\left[\left(\frac{H_b}{3}\right)^{0.96} - 1\right]}{H_b - 3} \tag{6}$$

where

T_1 = temperature at the center of the connection from the breeching, Fahrenheit degrees, absolute.

H_b = the height of the stack above center line connection to breeching, feet.

5. The *coefficient of friction* between the chimney gases and a sooted surface has been taken by many workers in this field as a constant value of 0.016 for the conditions involved. This value, of course, would be less for a new unlined steel stack than for a brick or brick-lined chimney, but in time the inside surface of all chimneys, regardless of the materials of construction, becomes covered with a layer of soot, and thus the coefficient of friction has been taken the same for all types of chimneys and generally constant for all conditions of operation. For reasons of simplicity and convenience to the reader, this constant value of 0.016 has been employed in the development of the various special equations and charts shown in this chapter.

In important large chimney design, especially when the construction or the materials are unusual, it is recommended that use be made of the Reynolds number[3] in determining the friction factor, f.

The following problem illustrates the use of Equation 3:

Example 1: Determine the available draft of a natural draft chimney 200 ft in height and 10 ft in diameter, operating under the following conditions: atmospheric temperature, 62 F; chimney gas temperature, 500 F; sea level atmospheric pressure, $B_0 = 29.92$ in. Hg; atmospheric and chimney gas density, 0.0863 and 0.09, respectively; coefficient of friction, 0.016; length of friction duct, 200 ft. The chimney discharges 100 lb of gases per second.

Solution: Substituting these values in Equation 3 and reducing,

$$D_a = 2.96 \times 200 \times 29.92 \times \left(\frac{0.0863}{522} - \frac{0.09}{960}\right) - \frac{0.00126 \times 100^2 \times 960 \times 0.016 \times 200}{10^6 \times 29.92 \times 0.09}$$

$$= 1.27 - 0.14 = 1.13 \text{ in.}$$

Fig. 2 shows the variation in the available draft of a typical 200 ft by 10 ft chimney operating under the general conditions noted in *Example 1*. When the chimney is under static conditions and no gases are flowing, the available draft is equal to 1.27 in. of water, the theoretical intensity. As

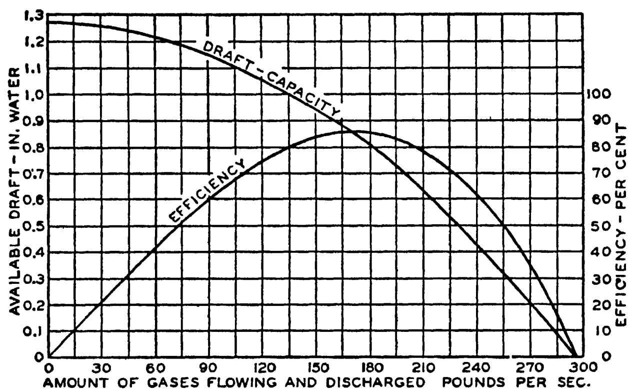

FIG. 2. TYPICAL SET OF OPERATING CHARACTERISTICS OF A NATURAL DRAFT CHIMNEY

the amount of gases flowing increases, the available draft decreases until it becomes zero at a gas flow of 297 lb per second, at which point the draft loss, due to friction, is equal to the theoretical intensity. The point of maximum draft and zero capacity is called shut-off draft, or point of impending delivery, and corresponds to the point of shut-off head of a centrifugal pump. The point of zero draft and maximum capacity is called the wide open point, and corresponds to the wide open point of a centrifugal pump. A set of operating characteristics may be developed for any size chimney operating under any set of conditions by substituting the proper values in Equation 3, and then plotting the results in the manner shown in Fig. 2.

Fig. 3 is a typical chimney performance chart giving the available draft for various gas flow rates and sizes of chimney. This chart is based on an atmospheric temperature of 62 F, a chimney gas temperature of 500 F, a unit chimney gas weight of 0.09 lb per cu ft, sea level atmospheric pressure, a coefficient of friction of 0.016, and a friction duct length equal to the height of the chimney above the grate level. These curves may be used for general operating conditions. For specific conditions, a new chart may be prepared from Equations 2 or 4.

DETERMINING INDUSTRIAL CHIMNEY SIZES

If the required performance for a proposed chimney is known, and *if a chimney-gas velocity is assumed*, Equation 3 can be transposed to yield the necessary height, and an equation can be developed for the required diameter. These operations result in the following equations:

$$H = \frac{D_r}{2.96B_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.184fW_cB_oV^2}{T_cD}} \tag{7}$$

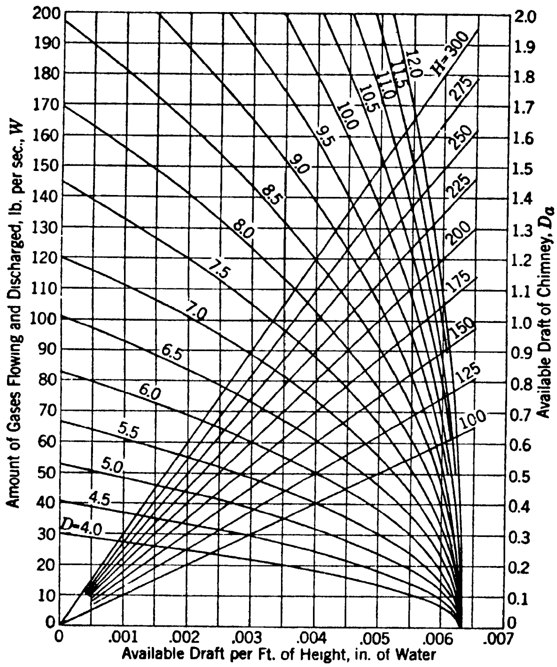

FIG. 3. CHIMNEY PERFORMANCE CHART

To solve a typical example: Proceed horizontally from a Weight Flow Rate point to intersection with diameter line; from this intersection follow vertically to chimney height line; from this intersection follow horizontally to the right to Available Draft scale. Starting from a point of Available Draft, take steps in reverse order.

The weight of gas per second, $W = 12.075 \frac{D^2VB_oW_c}{T_c}$ from which

$$D = 0.288\sqrt{\frac{WT_c}{B_oW_cV}} \tag{8}$$

where

H = required height of chimney above grate, feet.

D = required minimum diameter of chimney, feet.

V = chimney gas velocity, feet per second.

D_r = total required draft, inches of water.

For large chimneys, it is usual to assume that total construction cost is least when the product HD (height × diameter) is minimum. On this assumption, the product of Equations 7 and 8 can be differentiated, and the differential set equal to zero to find the minimum. Solution for velocity then yields the following equation:

$$V_e = \left(\frac{0.772 T_c \left(\frac{W_o}{T_o} - \frac{W_c}{T_c} \right) \sqrt{\frac{W T_c}{B_o W_c}}}{f W_c} \right)^{2/5} \tag{9}$$

where

V_e = economical chimney gas velocity, feet per second.

Equations 7, 8 and 9 can of course be simplified if values are assumed for some of the factors in it. Some typical figures for boiler plants are:

Average chimney gas temperature 500 F	T_c = 960 F absolute
Average atmospheric temperature 62 F	T_o = 522 F absolute
Average coefficient of friction 0.016	f = 0.016
Average chimney, gas density, 0 F, 1 Atmosphere	W_c = 0.09 lb per cu ft
Barometer reading, sea level	B_o = 29.92 in. Hg

When these values are substituted in Equations 7, 8 and 9, respectively, the results are:

$H = 190D_r$ (10) $D = 1.5W^{2/5}$ (11) $V_e = 13.7W^{1/5}$ (12)

These equations should be used for general operating conditions only, or where the required data necessary for an exact determination are difficult or impossible to secure. Whenever it is possible to obtain accurate data, or the anticipated operating conditions are fairly well known, the required size should be determined from Equations 7, 8, and 9.

Additional construction data for large industrial chimneys, whether brick, concrete, or steel, may be found in *Kent's Mechanical Engineers' Handbook*[4] or the *Handbook of Building Construction*.[5]

RESIDENTIAL CHIMNEYS

A residential chimney, to provide satisfactory performance, must have adequate height and area, be of permanently tight construction, be as smooth as practicable internally, and be of such construction as to present no fire hazard to the building. The height of a residence or apartment chimney is usually limited by the height of the building, and by cost. The chimney height and location that are best suited to a building from an architectural standpoint, will sometimes be unsatisfactory for the proper operation of the heating equipment. Chimney height is likely to be critical in one-story ranch-type or rambler-type houses, and therefore, it is important to compare carefully the available draft of the chimney and the required draft of the heating appliance to determine whether or not they will operate together satisfactorily.

Most residential chimneys are constructed of brick with a clay flue liner, but recently several lightweight, prefabricated chimneys have been marketed. These chimneys were primarily designed for use with gas equipment, but recently several have been approved by the *National Board of Fire Underwriters* for use with all types of fuels. The advantages of the lightweight, prefabricated chimney are ease of installation, somewhat lower cost, and reduced weight on the supporting structure.

AVAILABLE DRAFT FOR THE RESIDENTIAL CHIMNEY

Equations 3 and 4 cannot readily be used for computing the available draft for residential chimneys because of the relatively greater importance of friction losses, cooling of the gases, and soot deposits in small chimneys. Eddy currents and simultaneous flow both upward and downward can occur in a residential chimney for very low flue-gas velocities.

At present, it is best to rely on actual test data for determining the available draft of residential chimneys. Fig. 4 shows the available draft of nominal 9 × 9-in. and 9 × 13-in. masonry chimneys with an ambient temperature of 0 F for a range of effective heights from 5 to 32 ft, a range of entering flue-gas temperatures from 200 to 1000 F, and for mass flow rates of 83 and 300 lb per hr. Fig. 5 shows the same information for an

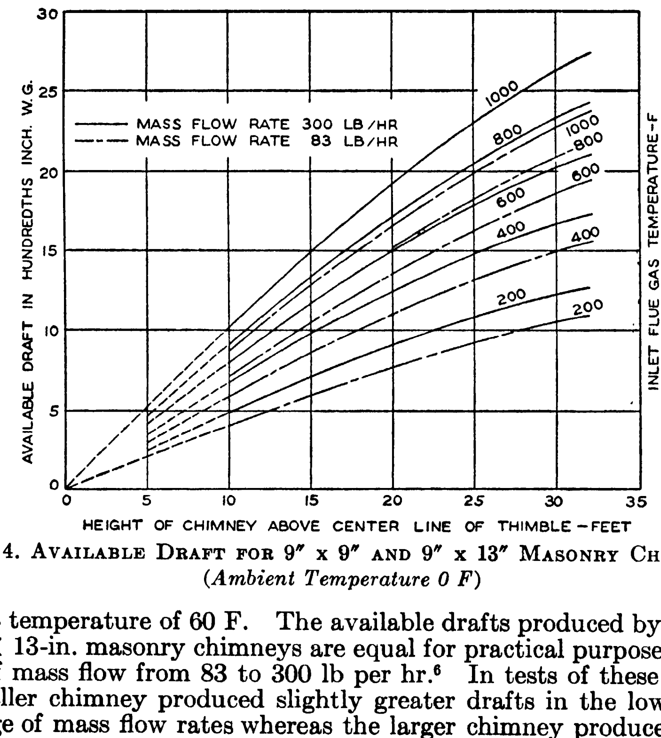

FIG. 4. AVAILABLE DRAFT FOR 9″ x 9″ AND 9″ x 13″ MASONRY CHIMNEYS
(Ambient Temperature 0 F)

ambient temperature of 60 F. The available drafts produced by 9 × 9-in. and 9 × 13-in. masonry chimneys are equal for practical purposes over the range of mass flow from 83 to 300 lb per hr.[6] In tests of these chimneys the smaller chimney produced slightly greater drafts in the lower end of the range of mass flow rates whereas the larger chimney produced slightly higher drafts in the upper end of the range.

The chimney height for heating plants that operate on an *on-off* or *high-fire low-fire* cycle should be selected to produce the desired draft from Fig. 5 since this class of heating system is required to operate at rated input for short periods when the outdoor temperature is 60 F. Heating plants whose fuel-burning rate is gradually increased as the outdoor temperature decreases are not required to operate at rated input except when outdoor temperatures approach design conditions. For such systems, the chimney height should be selected from Fig. 4 which shows the available draft for an outdoor temperature of 0 F since this is likely to be the more critical condition with respect to chimney draft. The available draft for outdoor temperatures between 0 F and 60 F can be obtained by interpolation from Figs. 4 and 5 with only slight error.

Fig. 6 is a graphical representation of the available draft for a 13-ft brick chimney with a nominal 8 × 8-in. flue liner[7] over a wide range of mass flow rates and for inlet flue-gas temperatures ranging from 200 to 1000 F. This family of curves is a typical group of performance curves showing that there is a certain mass flow rate that produces a maximum available draft for any flue-gas temperature.

The following approximate method may alternately be used to determine the available draft for small residential chimneys, from 10 to 25 ft in height and with internal cross-section areas from 35 to 55 sq in., with a maximum probable error of ± 15 percent at the same flow and temperature conditions. This method is based on the chimney efficiencies[7] shown in Fig. 7 and the ideal draft computed from the chimney inlet temperature.

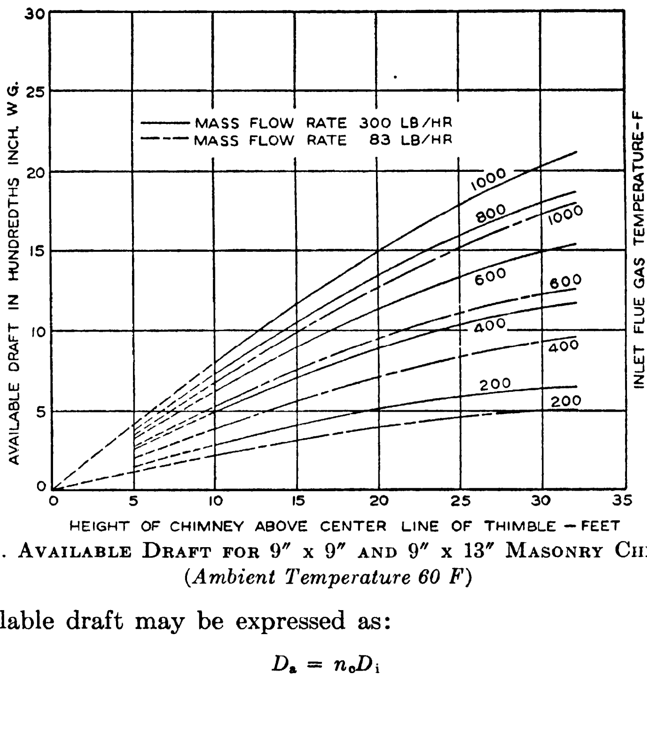

FIG. 5. AVAILABLE DRAFT FOR 9″ x 9″ AND 9″ x 13″ MASONRY CHIMNEYS
(Ambient Temperature 60 F)

The available draft may be expressed as:

$$D_a = n_c D_i \tag{13}$$

where

n_c = chimney efficiency taken from Fig. 7 at the desired conditions of temperature and flow.

D_i = ideal draft, calculated from Equation 14, assuming that the barometric pressure is 29.92 in Hg, and the ambient temperature is 60 F.

$$D_i = 0.2554 B_o H \left(\frac{1}{T_o} - \frac{1}{T_i}\right) \tag{14}$$

where

B_o = barometric pressure, inches of mercury.

H = chimney height, feet.

T_i = chimney inlet temperature, Fahrenheit degrees, absolute.

T_o = ambient air temperature, Fahrenheit degrees, absolute.

The clay-lined brick chimney is the most commonly used chimney, but recently other building materials have been used for reasons of economy or convenience. Investigations have established that the results shown in Figs. 4 and 5 for brick chimneys can be used, with slight error, for chimneys made of shale tile, concrete block, or cinder block.[8]

There are several lightweight chimneys that have recently been approved by the *National Board of Fire Underwriters* for use with all types of fuel. These chimneys are constructed of precast masonry or vitreous enameled steel surrounded by an insulating material. The characteristics of these chimneys can be assumed to be approximately those presented in Figs. 4 and 5.

DETERMINING RESIDENTIAL CHIMNEY SIZES

The flue sizes for small residential chimneys are governed by the National Building Code of the *National Board of Fire Underwriters* for gas-burning

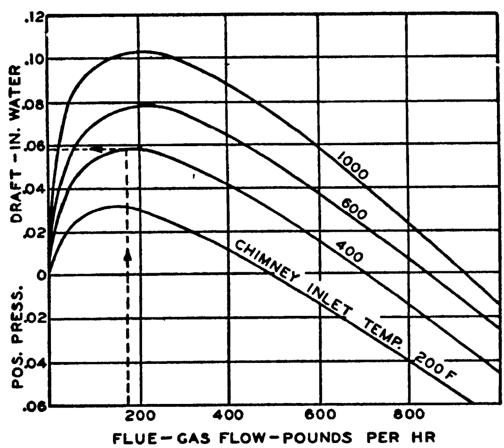

FIG. 6. AVAILABLE CHIMNEY DRAFT FOR 13 FT BRICK CHIMNEY[a, b]

[a] Square Flue Liner 6¾ x 6¾ in. inside.
[b] Barometric Pressure 29.92 in. Hg. Air Temperature 60F.

appliances. Chimney areas for liquid- and solid-fuel-burning devices are selected primarily to meet the requirements of local building codes, but these requirements are not determined by any rigorous formula based on physical principles.

By calculating the available draft for the chimney in question, and comparing it with the performance values of the heating unit (either natural or forced draft type) at the desired output, it is possible to determine whether the chimney is adequate in height for the particular heating unit it serves.

For calculations where the fuel rate and the percentage CO_2 are the only known factors, the flue-gas rate can easily be determined for coal, oil and gas from Fig. 8. By entering Fig. 8 at the percentage CO_2, moving vertically to the curve for the type of fuel, and then moving horizontally to the fuel rate, the flue-gas rate in pounds per hour may be determined for any fuel.[7]

Any of the described methods of determining available draft may be used, but a graphical solution to the problem may be had for the 8 $\times$ 8-in.

chimney from Figs. 6, 7 and 8. This solution can be best explained by a numerical example.

Example 3: Determine if a 13-ft, 8 x 8-in. nominal-size flue is sufficient for a coal-heating unit rated at 0.03 in. of water draft at 400 F inlet temperature, fuel rate being 10 lb per hr of bituminous coal, with 10 percent CO_2.

Solution: From Fig. 8, a flue-gas rate of approximately 180 lb per hr is obtained. The available draft for a 180 lb per hr fuel rate, and an inlet temperature of 400 F, obtained from Fig. 6, is 0.056 in. of water. This indicates that the chimney is adequate.

The selection of chimney areas for liquid- and solid-fuel-burning devices is difficult because of the variability in efficiency of different models, the possibility that soot on the lining will restrict the chimney area, and the variation in combustion air requirements of different solid fuels. Figs. 6 and 7 show that a given chimney produces a maximum available draft and a maximum efficiency for some intermediate mass flow rate for any selected inlet flue-gas temperature. For mass flow rates lower than the optimum

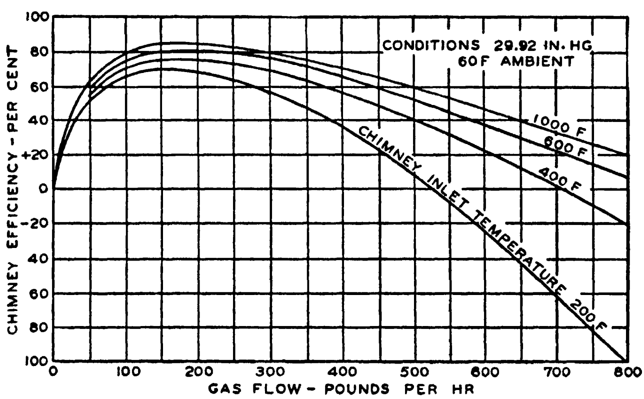

FIG. 7. EFFECT OF GAS FLOW ON CHIMNEY EFFICIENCY[a]

[a] Derived from temperature plots. Liner 8 x 8 in. outside, 6¾ x 6¾ inside. Height 13 ft.

the greater cooling of the gases in the chimney causes lower available draft, whereas for mass flow rates above the optimum the greater friction losses reduce the available draft. A chimney for a given heating system should probably be designed to operate at its point of maximum efficiency and maximum available draft for its full rated output. A chimney would have an accelerating effect[9] on the combustion rate of a solid-fuel burning device if it were operating to the left of the optimum point in Fig. 6 because an additional increment in mass flow rate would increase the available draft a small amount and tend to increase the mass flow still more. On the other hand, a chimney operating to the right of the optimum point in Fig. 6 would tend to decelerate the combustion rate for any small increase in mass flow rate.

Data are not complete for the selection of proper chimney areas for heating plants of different capacities, but some information on the effect of cross-section area on the capacity of masonry chimneys is provided by recent tests[8, 10] on several chimneys with liners having nominal outside dimensions: 9 in. diameter, 12 in. diameter, 9 × 9 in., and 9 × 13 in. These tests showed that for flue gas rates up to 200 lb per hr, and entering flue gas temperatures from 200 to 1000 F, the 9-in. round liner provided an available draft equal to, or greater than, that produced by the other three larger liners. When the flue gas rate was increased to 320 lb per hr, the three larger liners produced a little more draft than the smallest one for entering flue gas temperatures above 600 F.

Table 1 shows approximate values of the mass flow rates and flue-gas velocities at the chimney inlet that produce the maximum available draft for masonry chimneys[8] of several conventional sizes and with an effective height of 15 ft. This table shows that a 9-in. round chimney is best suited to mass flow rates from 130 to 170 lb per hr, a 9 × 9-in. chimney performs best for flow rates from about 200 to 300 lb per hr, and a 9 × 13-in. and 12 in. round chimneys are best suited to flow rates above 300 lb per hr. These results were obtained with clean chimneys, so that conclusions about chimney areas require some modification if soot deposits are taken into consideration.

Soot deposits in chimneys reduce the effective area of the liner and may in some cases entirely close the passage. Soot deposits are likely to be

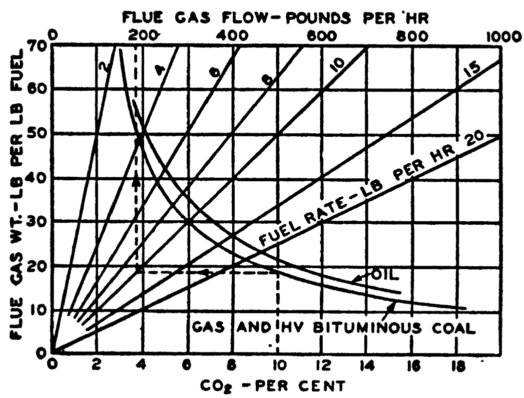

FIG. 8. GRAPHICAL EVALUATION OF RATE OF FLUE GAS FLOW FROM PERCENT CO_2 AND FUEL RATE[a]

[a] Average density gas 0.045 lb per cu ft, 62 F. Average weight oil 7.08 lb per gal, 60 F.

greater in the horizontal passages in the heating plant, the breeching, and the smokepipe, than in the vertical chimney liner. An increase in the liner diameter of one inch above that required for a clean chimney will probably make adequate allowance for soot deposits in all but the worst cases. Where smoky combustion is likely to occur and the mass flow rates on a clean chimney basis approach those listed in Table 1, the next larger commercial size liner should be used. Smoky combustion with oil-burning devices, particularly with vaporizing oil burners, is likely to be caused by inadequate draft which is often due to insufficient chimney height. For coal, inadequate chimney height or chimney area may be a contributing cause, but smoky combustion with coal is related to fuel characteristics and firing methods. Therefore, an increase in chimney area will usually not cure smoky combustion of a coal-burning device, but can be expected to lengthen the interval between cleanings.

New standard sizes of clay flue linings were recently developed by the industry and approved by the *American Standards Association*, in order to effect economies that can be derived from coordination of the dimensions of building materials. These linings are known as modular clay flue

TABLE 1. APPROXIMATE FLUE-GAS FLOW RATES FOR MAXIMUM AVAILABLE DRAFT IN MASONRY CHIMNEYS

Nominal External Liner Dimensions	Internal Area of Liner, Sq In	Flue Gas Temperature at Chimney Inlet, Fahr. 200	600	1000	200	600	1000
		Mass Flow Rate, LB/HR			Flue Gas Velocity at Chimney Inlet, FPM		
9 in. Dia.	38.5	170	150	130	175	250	300
9 in. x 9 in.	49	215	306	334	175	400	600
12 in. Dia.	78.5	290	Above 320	Above 320	150	—	—
9 in. x 13 in.	77	295	Above 320	Above 320	150	—	—

linings, and the dimensions and tolerances are summarized in *ASA* Standard A62.4-47. As the effective areas of these liners are somewhat smaller than those of the corresponding linings used previously, they cannot be used in certain municipalities where building codes specify minimum areas based on the older dimensions.

DRAFT REQUIREMENTS OF APPLIANCES

Typical flue-gas temperatures and drafts required at rated output for several kinds of domestic heating appliances[11] are contained in Table 2. Chimney height and chimney area for cast iron boilers are specified in the *I = B = R* Testing and Rating Code of the *Institute of Boiler and Radiator Manufacturers.*

Mechanically-fired devices such as oil burners and stokers are equipped with blowers, and therefore, the chimney is not required to overcome the resistance of the fuel bed or burner. Nevertheless, a draft in the firebox, of about 0.03 in. of water, is considered desirable so that any small openings in the firebox or flue passages will result in leakage of air inward, and not leakage of combustion products outward. Firebox leakage should be kept to a minimum, however, since such leakage adversely affects heating plant efficiency.

Difficulty in obtaining enough draft for natural draft appliances is likely to occur for attic installations or in one-story houses without basements. Automatic oil-burning space heaters, floor furnaces, and warm air furnaces employing natural-draft vaporizing burners require a draft of 0.06 to 0.08

TABLE 2. DRAFTS REQUIRED BY TYPICAL RESIDENTIAL HEATING DEVICES OR APPLIANCES

Device	Draft, Inches Water	Stack Temperature[c] F Deg
Space Heater, Oil Burning, Pot Burner	0.06 to 0.08	1000
Warm Air Furnace, Oil Burning, Pot Burner	0.06	860
Warm Air Furnace, Hand Fired	0.06[b]	900
Floor Furnace, Oil Burning, Pot Burner	0.06	860
Mechanical Oil Burner, Less than 5 gph	0.03[a]	—
Mechanical Oil Burner, More than 5 gph	0.05[a] or less	—
Cooking Stove, Solid Fuel	0.04[b]	400
Space Heater, Coal Burning	0.06[b]	900

[a] Draft in firebox. [b] For chestnut sized anthracite. [c] 18 in. from heater.

in. water for outdoor temperatures of 60 F. Taking into account the temperature drop of the flue gases between heater and chimney, Fig. 5 shows that an effective chimney height of 10 to 13 ft above the center line of the thimble is the minimum that should be employed for these types of equipment. Coal-burning heaters and furnaces that are required to attain rated output at a design outdoor temperature of 0 F would probably perform satisfactorily with an effective chimney height of about 8 ft as shown in Fig. 4. Application of adequate insulation to the smokepipe and breeching, might reduce these minimum values by 1 to 2 ft. When these

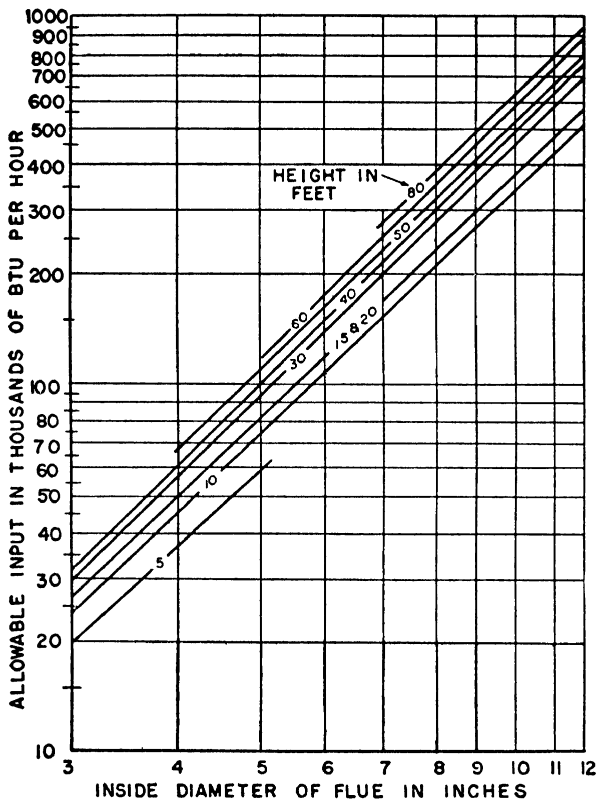

FIG. 9. CAPACITY IN BTU PER HOUR FOR GAS APPLIANCE FLUES OR VENTS

minimum chimney heights cannot be provided for architectural reasons or where flue gas temperatures lower than those shown in Table 2 exist, forced draft or induced draft should be employed. Since most residential heating plants are not designed for pressurized combustion chambers, forced draft should be used only to overcome the fuel bed or burner friction unless it is known that the plant has been constructed with no possibility of leakage in the combustion chamber and flue gas passages.

The use of barometric dampers in the smokepipe of natural-draft oil heating appliances employing vaporizing burners, is not recommended when these appliances are connected to chimneys having effective heights less than 15 ft, since most barometric dampers permit enough cold air to leak into the chimney even in the closed position to reduce the draft appreciably.

CHIMNEYS FOR GAS HEATING

Since a gas-designed appliance must be able to operate at rated input (plus 10 or 15 percent) without chimney connection, and without producing carbon monoxide, the only function of the chimney is to remove the products of combustion from the room. The chimney provides draft to overcome the friction in the flue pipe and chimney, but does not draw air into the appliance.

Chimneys for venting appliances designed for burning gas, can therefore be low in height, but must have adequate area. The height is usually established by the building height. Chimney sizes are usually selected on the basis of Btu input of the appliance. The chimney sizes adopted by

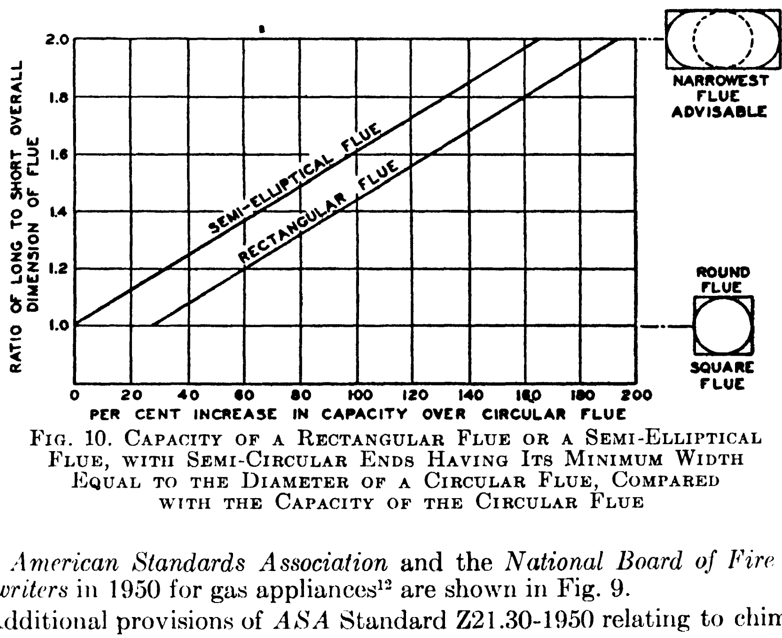

FIG. 10. CAPACITY OF A RECTANGULAR FLUE OR A SEMI-ELLIPTICAL FLUE, WITH SEMI-CIRCULAR ENDS HAVING ITS MINIMUM WIDTH EQUAL TO THE DIAMETER OF A CIRCULAR FLUE, COMPARED WITH THE CAPACITY OF THE CIRCULAR FLUE

the *American Standards Association* and the *National Board of Fire Underwriters* in 1950 for gas appliances[12] are shown in Fig. 9.

Additional provisions of *ASA* Standard Z21.30-1950 relating to chimney size are:

1. In no case shall the vent area be less than the area of a 3-in. diameter pipe.
2. When more than one appliance vents into a flue or vent, the flue or vent area shall be not less than the area of the largest flue or vent connector plus 50 per cent of the areas of the additional flue or vent connectors.
3. An elliptical flue or vent may be used provided its flue gas venting capacity is equal to the capacity of the round pipe for which it is substituted.

Since Fig. 9 has been prepared for circular flues, relative capacities for rectangular and semi-elliptical flues[13] are shown in Fig. 10.

Heating appliances designed to burn gas, as well as appliances converted to gas burning, except those equipped with power type burners and excepting conversion burner installations in excess of 400,000 Btu per hour input in large steel boilers, are always equipped with a draft hood attached to the flue outlet of the appliance. This draft hood is required if the appliance is to meet the approval requirements of the *American Gas Association* and the *American Standards Association*, and is essential for safe operation. It is designed to prevent excessive chimney draft which would lower appliance efficiency, to prevent a blocked flue or a down-draft in the chimney from impairing combustion, to provide a relief opening for the products of combustion during down-draft or blocked flue conditions, and to prevent spillage of the products of combustion to the space surrounding the appliance, if there is a chimney draft equivalent to that provided by a 3-ft chimney. As the draft hood is designed without moving parts, the relief opening is always open, and consequently some air is drawn into the chimney. This air lowers the gas temperature in the chimney, but it also lowers the dew-point of the gases and tends to prevent condensation.

The installation of conversion burner equipment in large boilers is usually made in accordance with regulations of the local gas company. In such installations a definite chimney draft may be required for proper combustion, and consequently the foregoing reference to the use of draft hoods would not apply.

The products of complete combustion of gas are water vapor (H_2O) and carbon dioxide (CO_2). In the case of manufactured gas, the presence of organic sulfur compounds, generally between 3 and 15 grains per hundred cubic feet, gives rise to minute percentages of sulfur dioxide and sulfur trioxide. The volume of water vapor in the flue products from natural or coke oven gas is about twice the volume of carbon dioxide. It is extremely important that the chimney be tight and resistant to corrosion, not only from moisture, but also from dilute sulfur trioxide.

Clay linings with joints which prevent retention of moisture and linings made of non-corrosive materials, are advantageous. The protection of unlined chimneys has been investigated and the results indicate that after the loose material has been removed, spraying with a water emulsion of asphalt chromate will provide excellent protection.

Advice regarding recommended practice and materials for flue connections and chimney linings can usually be obtained from the local gas company, and should be given careful consideration.

RECOMMENDATIONS OF THE NATIONAL BOARD OF FIRE UNDERWRITERS

For general data on the construction of chimneys, reference should be made to the National Building Code, recommended by the *National Board of Fire Underwriters*, Article X, Section 1001 to 1006, in which the following are some of the important provisions listed in the 1951 edition:

1. *Flue Connections Required.* Every heating apparatus or heat producing appliance requiring a flue connection shall be connected with a flue conforming to the provisions of this article. This shall not include electric appliances; gas appliances, except as specifically required in this article; nor oil fired appliances especially designed for use without flue connection.

2. *Use of Nonconforming Flues.* Flues not conforming to the requirements of this article for chimneys, metal smokestacks or vents for gas appliances, shall not be used unless listed by *Underwriters' Laboratories, Inc.*, installed in full compliance with the listing and the manufacturer's instructions, and approved for such use by the building official.

3. *Smoke Pipe Connections.*

a. No flue shall have smoke pipe connections in more than one story of a building, unless provision is made for effectively closing smoke pipe openings with devices made of noncombustible materials whenever their use is discontinued temporarily, and completely closing them with masonry when discontinued permanently.

b. Two or more smoke pipes shall not be joined for a single connection, unless the smoke pipes and flue are of sufficient size to serve all the appliances thus connected.

c. The smoke pipe of a heating appliance shall not be connected into the flue of an incinerator which has the rubbish chute identical with the smoke flue.

4. *Construction of Chimneys.*

a. Chimneys hereafter erected within or attached to a structure shall be constructed in compliance with the provisions of this section.

b. Chimneys shall extend at least 3 ft above the highest point where they pass through the roof of the building, and at least 2 ft higher than any ridge within 10 ft of such chimney.

c. Chimneys shall be wholly supported on masonry or self-supporting fireproof construction.

d. No chimney shall be corbeled from a wall more than 6 in.; nor shall a chimney be corbeled from a wall which is less than 12 in. in thickness, unless it projects equally on each side of the wall; provided that in the second story of 2-story dwellings corbeling of chimneys on the exterior of the enclosing walls may equal the wall thickness. In every case the corbeling shall not exceed 1 in. projection for each course of brick projected.

e. No change in the size or shape of a chimney, where the chimney passes through the roof, shall be made within a distance of 6 in. above or below the roof joists or rafters.

5. *Chimneys for Heating Appliances, Low Heat Industrial Appliances and Portable Type Incinerators.*

a. Chimneys for stoves, cooking ranges, warm air, hot water and low pressure steam heating furnaces, fireplaces, and low heat industrial appliances, other than chimneys for incinerators of nonportable type, shall be constructed of solid masonry units or of reinforced concrete. The walls shall be properly bonded or tied with corrosion-resistant metal anchors. In dwellings and buildings of like heating requirements, the thickness of the chimney shall be not less than 4 in. In other buildings the thickness shall be not less than 8 in., except that rubble stone masonry shall be not less than 12 in. thick.

b. Every such chimney hereafter erected or altered shall be lined with a flue lining conforming to the requirements below.

c. Flue linings shall be made of fire clay or other refractory clay which will withstand the action of flue gases and resist, without softening or cracking, the temperatures to which they will be subjected, but not less than 2000 F. Flue linings may be of cast iron of approved quality, form and construction.

d. Required clay flue linings shall be not less than $\frac{5}{8}$ in. thick for the smaller flues, and shall increase in thickness for the larger flues.

e. Flue linings shall be installed ahead of the construction of the chimney as it is carried up, carefully bedded one on the other in Type A, Type B, or fire clay mortar with close fitting joints left smooth on the inside.

f. Flue linings shall start from a point not less than 8 in. below the intake, or, in the case of fireplaces, from the throat of the fireplace. They shall extend, as nearly vertically as possible, for the entire height of the chimney, and be extended 4 in. above the top of cap of the chimney.

g. Cleanouts for flues or fireplaces shall be equipped with cast-iron doors and frames arranged to remain tightly closed when not in use.

h. When two or more flues are contained in the same chimney, at least every third flue shall be separated by masonry at least 4 in. thick bonded into the masonry wall of the chimney. Where flue linings are not so separated, the joints of adjacent flue linings shall be staggered at least 7 in.

6. *Sizes of Flues.*

a. The cross-sectional area of smoke flues shall be designed and proportioned to meet the conditions of temperature, within and without flue, thickness of masonry, exposure, shape and material of flue, and other influences.

The National Building Code specifies lined chimneys and metal smoke stacks for all gas appliances which may be converted readily to the use of solid or liquid fuel, and also for all boilers and furnaces, except those having a flue-gas temperature not exceeding 550 F at the outlet of the draft hood when burning gas at the manufacturer's rating and which may, therefore, be connected to *Type B* vent piping. Approved *Type B* vent piping is noncombustible, corrosion-resistant piping of adequate strength and heat-insulating value, and having bell and spigot or other acceptable joints. Fig. 9 may be used for selection of vent-pipe size.

Important points to be considered in the use of *Type B* vent piping are:

1. *Type B* flues must be plainly and permanently marked at the point where the vent connection enters the flue: *For Use of Gas Appliances Only.*

2. *Type B* gas vents shall be installed with a clearance to combustible material or construction, whether plastered or unplastered, of not less than one inch, provided that for vents of floor furnaces, such clearance shall be not less than 3 ft from the outlet of the draft hood, measured along the center line of the vent piping.

Other important points that should be considered for flues and vents for gas appliances are as follows:

1. Clearances from combustible material to gas appliance vent piping other than approved *Type B* gas vents shall be in accordance with the Building Code Standards of the *National Board of Fire Underwriters* for the Installation of Heat Producing Appliances, Heating, Ventilating, Air Conditioning, Blower and Exhaust Systems.

2. Every flue-connected appliance, except an incinerator, unless its construction serves the same purpose, shall be equipped with an effective draft hood which either (a) has been approved as part of the appliance or (b) complies with nationally recognized standards for draft hoods. The draft hood shall be attached to the flue collar of the appliance as conditions permit, and in a position for which it is designed with reference to horizontal and vertical planes. The draft hood shall be so located that the relief opening is not obstructed by any part of the appliance or adjacent construction.

3. No vent pipe from a gas appliance shall be interconnected with any other vent pipe, smoke pipe, or flue, unless such gas appliance is equipped with an automatic device to prevent the escape of unburned gas at the main burner or burners. Where a gas appliance vent pipe is joined with a smoke pipe from an appliance burning some other type of fuel, for connection into a single flue opening, they shall be joined by a Y fitting located as close as practicable to the chimney. With liquefied petroleum gases, the automatic device to prevent the escape of unburned gas shall shut off the pilot light, as well as the main burner or burners.

Recent tests of masonry chimneys[14] made of a variety of materials have developed additional recommendations regarding the construction of masonry chimneys that will decrease the hazard to surrounding combustible materials.

GENERAL CONSIDERATIONS FOR CHIMNEYS

The draft of domestic chimneys may be subject to a variety of influences not usually encountered in power chimneys[15] because of the low available draft often supplied by a short chimney. Horizontal winds have an aspirating effect as they cross the chimney and are an aid to draft. However, surrounding objects, such as trees or other buildings, may affect the direction of the wind at the chimney top, and may even direct it down the chimney, tending to reduce the draft or even to cause it to change to a positive pressure.

It is not to be assumed that increasing the cross-sectional area of a chimney will always effect a cure for poor draft. The opposite result may occur because of the cooling effect of the larger area, and the effect of recirculation of the flue gases. The flow of gases into the chimney top has been observed at low rates, and recirculation in small residential chimneys has been noted throughout the entire length of a chimney and smoke pipe, with the greater amount of recirculation occurring at the thimble. The effect of recirculation decreases with chimney height and the increase in flue-gas velocity.

It is also important to consider the course of the air supply for proper combustion. The boiler or furnace is usually located in the basement. In the majority of cases, the furnace room has windows and doors opening to the outside on two or more sides of the house. Through these enough air leaks into the furnace room to sustain combustion. In some cases, however, windows and doors are so tight as to restrict the flow of combustion air, and thereby affect the correct operation of the chimney. In case the boiler room is fairly tight and is open to the outside on only one side of the house, then the draft will be affected in windy weather even with windows or doors open. If the wind is blowing toward the boiler room, the draft will be increased, but if blowing in the opposite direction, the draft may be decreased.

The surrounding of a heating appliance with a restrictive enclosure that will limit entrance of combustion air is more likely to occur in a small utility closet installation on the first floor than in a basement installation. An opening with a free area approximately twice the area of the smokepipe should therefore, be provided between the utility closet and the living space or the outdoors. If the utility closet is connected to the outdoors, greater difficulty with wind pressures will be encountered. Where a draft regulator is used it should have ample communication with the same space from which the combustion air is taken. Where forced warm air furnaces are enclosed in utility closets, care should be used to make the return air connection inside the utility closet airtight so that the blower cannot reduce the pressure in the closet and cause a downdraft in the chimney.

Two or more chimneys, either large or small, should never be connected together. If connected at the bottom, hot gases in the U-tube thus formed would be in unstable equilibrium. Cold air from the top would descend through one such chimney and drive the hot gases out of the other, thus annulling the draft.

More than one device can be served by one chimney. Batteries of boilers are commonly connected to a single chimney in power plants. However, if two or more chimneys are used, each chimney should be used separately for part of the boilers, and not connected in manifold with another chimney, in order to avoid the difficulty described previously.

In domestic installations it is sometimes necessary to serve a space heater or cooking stove and a water heater with the same chimney flue. This is not desirable, especially for low chimneys, since doors left open on one device, while it is unfired, will tend to annul the draft on another device. Gas burning devices, with their draft hoods and lack of draft dampers, are especially bad in this respect. The traditional method of avoiding this with brick chimneys has been to construct multiple-flue chimneys, so that each fuel-burning device could be served by a separate opening. If two devices must be served by one flue-opening in a chimney, their connections to the chimney should not be located opposite each other. The connection from the larger device should be reasonably low, and that from the smaller, up near the ceiling, so that each device can be serviced as well as possible, regardless of the treatment of the other.

Excessive height in a chimney does no harm, but means for controlling the draft are more than ordinarily essential if the chimney is too large in capacity. Coal-burning devices often have air leaks around the firebox, and the draft doors sometimes fit so poorly that the fire cannot be controlled at a low rate. The simplest remedy for such cases is the barometric damper which admits air into the flue pipe and thus reduces draft.

Where a chimney serves a fireplace, it is important that no other heating device be connected to it unless the fireplace is effectively sealed.

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CHAPTER 17

ESTIMATING FUEL CONSUMPTION FOR SPACE HEATING

Bases of Fuel Estimates; Season Efficiency; Calculated Heat Loss Method: Computation and Application, Examples and Solutions, Short Methods for Estimating Heat Loss; Degree Day Method: Computation and Application, Unit Fuel Consumption per Degree Day, Estimating Consumption for Various Fuels, Examples and Solutions, Degree Day as an Operating Unit; Maximum Demand and Load Factors

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IT IS often necessary to estimate the anticipated heat requirements and fuel consumptions of heating plants for either short or long terms of operation. There are various general methods for estimating these conditions, and frequently the methods can be so modified as even to become useful in evaluating the effectiveness of heat production or fuel utilization during plant operation.

In applying a consumption-estimating method to a particular building, it is well to note that the bases of the methods may vary as to reliability. For example:

1. Records of past heat requirements or fuel consumption of the particular building are a better basis for estimates than are averages from records of similar buildings.
2. In the absence of past records for a particular building, the data from similar plants in the same locality may become very helpful.
3. Averages of consumption taken from many types of plants in many types of buildings in various localities can only produce an average estimate which may prove to be very inaccurate as applied to the particular building being considered.
4. Estimates based upon computed heat losses (without benefit of operating data) are wholly dependent, of course, on the degree to which the computation represents the actual facts.

Where unusual operating conditions exist due to factors such as excessive ventilation, abnormal inside temperatures and heat gains from external sources, or where, in the case of proposed buildings of unusual design, no information is available regarding former consumption, it is necessary to estimate fuel consumption from the computed heat losses.

In preparing fuel consumption estimates it is well to realize that any estimating method used will produce a more reliable result over a long period operation than over a short period. Nearly all of the methods in common use will give trustworthy results over a full *annual* heating season, and in some cases such estimates will prove consistent within themselves for monthly periods. As the period of the estimate is shortened, there is more chance that some factor not allowed for in the estimating method will become dominant, and thus give discrepant and even ridiculous results.

The Calculated Heat-Loss Method, and the Degree-Day Method of estimating fuel requirements are illustrative of all methods used. Both methods are based upon an estimate of seasonal efficiency. The former is also based upon an estimate of average seasonal temperature. Neither method takes into account factors which are difficult to evaluate, such as opening of windows, abnormal heating of the building, poor design of heating system, sun effect, and other variations. The Degree-Day Method is the more practical since it is based, in part, on actual consumption data. The presentation of the two illustrative estimating methods will be preceded by a discussion of *seasonal efficiency.*

SEASONAL EFFICIENCY

The seasonal efficiency differs from the measured efficiency of the fuel-fired heating unit because it is affected by the many minor sources of extraneous heat gain and heat loss. Throughout a season, useful heat is supplied to a building not only by the heating unit, but also by the external surfaces of the heating unit, the flue pipe, and the chimney. In addition, heat is gained from lights, occupants, cooking and other appliances, and from the sun. Indeterminable amounts of heat are lost through radiation directly to basement or utility-room walls and floors, from the heating unit, the distribution system, and the flue pipe, from stand-by operation of the unit, from opening of doors and windows, and from faulty adjustment and operation of the combustion unit. Fortunately, data which are available permit the making of reasonable estimates of seasonal efficiency for residences.

The average fuel consumption of various types of approved gas-fired equipment has been obtained from a large number of heating systems.[1] Corresponding seasonal efficiencies can be calculated from these data. They show a variation from approximately 72 to 88 percent, depending upon the type and size of system. Laboratory tests on a gas conversion burner in a heating boiler operated with *on* and *off* cycles gave about 72 percent efficiency.[2] Other tests on coal-fired room heaters indicate seasonal efficiencies of 65 to 75 percent, when the heat from the flue pipe was included.[3] A survey of 30 residences in one locality showed a variation of 45 to 75 percent in utilization efficiency, depending upon the condition of the equipment and the fuel used.[4] A summary[5, 6] of many tests in two research residences at the *University of Illinois*, using many fuels and systems, gave values of 67 to 90 percent for overall house efficiency (the ratio of the heat loss from the structure to the heat input to the unit for an average test day).

These data were used by the *Small Homes Council*[7] of the *University of Illinois* in a tabulation of the comparative costs of heating the *same* building with various fuels.

The approximate seasonal efficiencies shown in Table 1 are suggested as a guide.

TABLE 1. APPROXIMATE SEASONAL EFFICIENCY

Type of Fuel-Burning Unit	Approximate Seasonal Efficiency, Percent	Type of Fuel-Burning Unit	Approximate Seasonal Efficiency, Percent
Gas, designed unit	75–80	Anthracite, hand-fired with controls	60–80
Gas, conversion unit	60–80	Anthracite, hand-fired without controls	50–65
Oil, designed unit	65–80	Anthracite, stoker-fired	60–80
Oil, conversion unit	60–80	Coke, hand-fired with controls	60–80
Bituminous coal, hand fired with controls	50–65	Coke, hand-fired without controls	50–65
Bituminous coal, hand fired without controls	40–60	Direct electric heating	100
Bituminous coal, stoker-fired	50–70		

CALCULATED HEAT LOSS METHOD

In the Calculated Heat-Loss Method a constant average outdoor temperature is assumed throughout the heating season. This average temperature is considerably above the design temperature given in Chapter 11. The method becomes unreliable whenever data on the seasonal average temperature are not available for the particular locality. If the length of time is shown over which the degree-day data apply in the particular locality, it is possible to calculate the average temperature from the definition of the degree-day (see Chapter 1). When this is done, the two methods described in this chapter become identical. The average temperature for the period October to April inclusive, is listed in Table 1, Chapter 11, for U. S. and Canadian cities.

Computation and Application

In order to apply the Calculated Heat-Loss Method, the hourly heat loss from the building under maximum load, or design condition, is computed following the principles discussed in Chapters 9, 10, and 11. The fuel requirement is then computed by the equation

$$F = \frac{H(t - t_a)N}{E(t_d - t_o)C} \tag{1}$$

where

F = quantity of fuel or energy required (in the units in which C is expressed).

H = calculated heat loss including infiltration loss, Btu per hour, during the design hour, based on t_o and t_d.

t = average inside temperature maintained during heating period, Fahrenheit degrees.

t_a = average outside temperature through estimate period, Fahrenheit degrees (for cities with an Oct. 1–May 1 heating season—see Table 1, Chapter 11).

t_d = inside design temperature, Fahrenheit degrees (usually 70 F).

t_o = outside design temperature, Fahrenheit degrees (see Table 1 in Chapter 11).

N = number of heating hours in estimate period (for an Oct. 1–May 1 heating season, 212 days × 24 hr = 5088).

E = efficiency of utilization of the fuel over the period, expressed as a decimal; not the efficiency at peak or rated load condition.

C = heating value of one unit of fuel or energy.

Although the assumption of an Oct. 1–May 1 heating season is reasonably accurate in the well-populated New York-Chicago zone, it is not valid as far north as Minneapolis nor farther south than Washington, D. C. and St. Louis. Consequently, it is suggested that allowance be made for this variation, especially in the far north or southern cities.

Example 1: A residence building is to be heated to 70 F from 6 a.m. to 10 p.m. and 65 F from 10 p.m. to 6 a.m. The calculated hourly heat loss is 120,000 Btu per hour based on 70 F inside at −10 F outside. If the building is to be heated by metered steam, how many pounds would be required during an average heating season?

Solution: The heating value of steam may be taken as 1000 Btu per lb, and since it is purchased steam, the efficiency can be assumed as 100 percent. Assume average outside temperature as 36.4 F. The average inside temperature is:

$$\frac{(16 \times 70) + (8 \times 65)}{24} = 68.3\text{ F.}$$

Substituting in Equation 1

$$F = \frac{120{,}000\ (68.3 - 36.4)\ 5088}{1.00[70 - (-10)]1000} = 243{,}000 \text{ lb.}$$

Example 2: What would be the fuel cost to heat the building in *Example 1* during an average heating season, using stoker-fired bituminous coal at $14.00 per ton having a calorific value of 13,000 Btu per lb, assuming that the seasonal efficiency of the system is 65 percent?

Solution: Substituting in Equation 1

$$F = \frac{120{,}000\ (68.3 - 36.4)\ 5088}{0.65[70 - (-10)]13{,}000} = 28{,}800 \text{ lb.}$$

The fuel cost is then (28,800 ÷ 2000) (14) = $201.00

Example 3: What will be the estimated fuel cost per year of heating a building with gas, assuming that the calculated hourly heat loss is 92,000 Btu based on 0 F, which includes 26,000 Btu for infiltration? The design temperatures are 0 F and 72 F. The normal heating season is 210 days, and the average outside temperature during the heating season is 36.4 F. The seasonal efficiency will be 80 percent. The heating plant will be thermostatically controlled, and a temperature of 65 F will be maintained from 11 p.m. to 7 a.m. Assume that the price of gas is 7.5 cents per 100,000 Btu of fuel consumption, and disregard the loss of heat through open windows and doors.

Solution: The average hourly temperature is

$$t_a = \frac{(72 \times 16) + (65 \times 8)}{24} = 69.7 \text{ F.}$$

The maximum hourly heat loss will be

$$H = 92{,}000 \text{ Btu.}$$

The seasonal heat loss is

$$M = \frac{92{,}000\ (69.7 - 36.4) \times 24 \times 210}{100{,}000 \times 0.80 \times (72 - 0)} = 2697 \text{ hundred thousand Btu.}$$

The estimated seasonal fuel cost will be

$$2697 \times \$0.075 = \$202.00$$

It should be noted that savings from night setback may not result as calculated. Room temperature may not decrease and combustion efficiency may be poor during morning pickup. See Reference 8.

Several time-saving procedures have been devised for quickly estimating the hourly Btu loss of one and two-story residences in order that fuel estimates can be predicted more quickly from Equation 1. A graphical method of calculating heat losses has been developed[9] which makes possible a quick solution if the gross wall, ceiling, or floor areas and respective transmission coefficients are known.

The *Federal Housing Administration has* originated a short-cut formula for residential heat loss determinations which makes use of the floor area and three selected transmission coefficients. The formula was developed to apply to detached houses approximately rectangular in shape with total exterior door and window areas equal to about 25 percent of the floor area, and with a floor area not greater than about 1500 sq ft. Equation 2 is for a one-story residence, and Equation 3 is intended for two-story structures.

$$H_1 = A\ (G + U_w + U_c + U_f)\ (t_d - t_o) \qquad (2)$$

$$H_2 = A\ (G + 1.2\ U_w + 0.5 U_c + 0.5\ U_f)\ (t_d - t_o) \qquad (3)$$

where

H_1 = heat loss from one-story residence, Btu per hour.
H_2 = heat loss from two-story residence, Btu per hour.
A = floor area, square feet, measured to the inside faces of enclosing walls and is the sum of the following areas: (1) all the area on each principal floor level; (2) the area of all finished habitable attic rooms, including bathrooms, toilet compartments, closets, and halls; (3) all other areas intended to be heated and not located in the basement.
G = glass and infiltration factor for ordinary construction: (0.45 for no weatherstripping or storm windows), (0.40 for weatherstripping), (0.30 for storm windows with or without weatherstripping).
U_w = coefficient of transmission for outside wall.
U_c = coefficient of transmission for ceiling.
U_f = coefficient of transmission for floor.
t_d = inside design temperature, Fahrenheit degrees.
t_o = outside design temperature, Fahrenheit degrees.

Notes for application of Equations 2 and 3.

1. The calculation of heat loss from heated spaces into adjacent spaces such as attics, basementless areas, and heated or unheated garages shall be based on the assumption that the temperature of such adjacent spaces is the same as the outside design temperature.
2. For all floors over basements or other warmed spaces assume $U_f = 0$.
3. For structures having concrete slab floors laid on the ground a modified application of the formula may be made. Assume $U_f = 0$ and calculate the heat loss in accordance with the check formula. Then add the slab loss determined in accordance with the procedure developed by the *National Bureau of Standards* and described in BMS Report 103.
4. No basement area is to be included in the formula calculation. If finished habitable rooms in the basement are to be heated, the additional heat loss should be calculated separately and added to the amount obtained by the formula.

Both the graphical method and short-cut formulas, when used within the limitations established, have been found to give reasonably accurate results for the average residence, but if precise estimates are required, the procedure outlined in Chapter 11 should be used.

In the case of gravity warm air heating installations, the load was formerly expressed in square inches of leader pipe which can be converted into Btu per hour by multiplying the square inches of leader area by 111, 167, and 200 for first, second, and third floors, respectively.

DEGREE-DAY METHOD

This method is based on consumption data which have been taken from buildings in operation, and the results have been computed on a degree-day basis. While this method may not be as theoretically correct as the Calculated Heat Loss Method, it is considered by many to be of more value for practical use.

The amount of heat required in a building depends upon the outdoor temperature, if other variables are eliminated. Theoretically it is proportional to the difference between the outdoor and indoor temperatures. The *American Gas Association*[10] determined from records in the heating of residences that the gas consumption varied directly as the degree-days, or as the difference between 65 F and the mean outside temperature. In other words, on a day when the mean temperature was 20 deg below 65 F, twice as much gas was consumed as on a day when the temperature was 10 deg below 65 F. For any one day, when the mean temperature is less than 65 F, there are as many degree-days as there are degrees difference in temperature between the mean temperature for the day and 65 F. Degree-days may be calculated on other than the 65 F base, but are seldom used and are of little value except for example, in warehouses where the inside temperature to be maintained differs greatly from the usual inside temperature range of 68 F to 72 F.

Studies made by the *National District Heating Association*[11] of the metered steam consumption of 163 buildings located in 22 different cities and served with steam from a district heating company, substantiate the approximate correctness of the 65 F base chosen by the gas industry.

Table 2 lists the average number of degree-days which have occurred over a long period of years, by months, and the yearly totals for various cities in the United States, Canada and Newfoundland. The values for United States cities were calculated by taking the difference between 65 F and the daily mean temperature computed as half the total of the daily maximum and the daily minimum temperatures. The monthly averages were obtained by adding daily degree-days for each month each year and dividing by the number of days in the month; then totaling the respective calendar monthly averages for the number of years indicated and dividing by the number of years. The total or long term yearly average degree-day value is the summation of the 12 monthly averages. Degree days for Canadian cities were supplied by the *Canadian Meteorological Division* of the *Department of Transport*, and were computed from the mean temperature normals on record for the various stations.

Any attempt to apply the degree-day method of estimating fuel consumption for less than one month would be of very little value. It should be noted that this method of calculation is based on a long term average and cannot be expected to coincide with any single year in calculating fuel requirement. Individual yearly degree-day calculations may vary as much as 20 percent above and below the long term average.

If the degree-days occurring each day are totaled for a reasonably long period, the fuel consumption during that period as compared with another period will be in direct proportion to the number of degree-days in the two periods. Consequently, for a given installation, the fuel consumption can be calculated in terms of fuel used per degree-day for any sufficiently long period, and compared with similar ratios for other periods to determine the relative operating efficiencies with the outside temperature variable eliminated.

Computation and Application

The general equation for calculating the probable fuel consumption by the degree-day method is:

$$F = U \times N \times D \times C_t \tag{4}$$

where

F = fuel consumption for the estimate period.

U = unit fuel consumption, or quantity of fuel used per (degree-day) (*building load unit*).

N = number of *building load units* (when available use calculated hourly heat loss instead of actual amount of radiation installed).

D = number of degree-days for the estimate period.

C_t = temperature-correction factor from Table 3.

Values of N depend on the particular building for which the estimate is

TABLE 2. AVERAGE MONTHLY AND YEARLY DEGREE-DAYS FOR CITIES IN THE UNITED STATES, CANADA AND NEWFOUNDLAND[a, b] (BASE 65F)

State	Station		Years	No. of Sea-sons	July	Aug.	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Yearly Total
Ala.	Anniston		05/06–40/41	36	0	0	10	135	388	600	609	513	361	152	37	1	2806
	Birmingham	A	98/99–45/46	48	0	1	10	111	348	586	591	497	313	130	23	1	2611
	Mobile		98/99–45/46	48	0	0	1	44	203	377	397	314	175	52	3	0	1566
	Montgomery		98/99–45/46	48	0	0	4	71	279	484	494	405	239	85	10	0	2071
Ariz.	Flagstaff		98/99–40/41	43	43	70	244	573	847	1111	1167	970	889	668	469	190	7241
	Phoenix		98/99–45/46	48	0	0	0	18	166	384	402	263	154	47	7	0	1441
	Yuma		98/99–40/41 45/46	44	0	0	0	9	113	306	318	182	85	22	1	0	1036
Ark.	Bentonville		06/07–40/41	35	1	1	38	216	516	810	879	716	519	247	86	7	4036
	Fort Smith	A	98/99–45/46	48	0	0	12	128	410	717	763	615	390	154	36	1	3226
	Little Rock	A	98/99–45/46	48	0	0	11	120	383	668	704	579	367	145	31	1	3009
Calif.	Eureka		98/99–45/46	48	281	269	274	344	411	518	541	478	504	440	391	307	4758
	Fresno	A	98/99–45/46	48	0	0	5	77	309	562	573	380	289	152	52	4	2403
	Independence		98/99–40/41	43	0	0	28	216	512	778	799	619	477	267	120	18	3834
	Los Angeles		98/99–45/46	48	1	0	5	43	110	225	272	235	212	158	103	27	1391
	Needles		17/18–38/39	22	0	0	0	19	217	416	447	243	124	26	3	0	1495
	Point Reyes		98/99–40/41	43	350	336	263	282	317	425	467	406	437	413	415	363	4474
	Red Bluff	A	98/99–33/34 38/39–40/41 44/45–45/46	41	0	0	12	97	345	592	601	419	328	178	72	9	2653
	Sacramento		98/99–45/46	48	2	1	15	98	332	582	595	405	326	202	101	21	2680
	San Diego	A	98/99–45/46	48	5	1	9	60	143	252	300	257	230	172	118	49	1596
	San Francisco		98/99–45/46	48	196	179	121	139	241	420	460	340	317	272	255	197	3137
	San Jose		06/07–40/41	35	21	21	52	151	329	512	527	383	339	249	167	72	2823
Colo.	Denver		98/99–45/46	48	8	8	126	411	716	1005	1023	897	790	516	275	64	5839
	Durango		04/05–40/41	37	25	37	201	535	861	1204	1271	1002	859	615	394	139	7143
	Grand Junction		98/99–45/46	48	1	1	59	347	743	1138	1218	883	671	377	152	23	5613
	Leadville		07/08–40/41	34	280	332	509	841	1139	1413	1470	1285	1245	990	740	434	10678
	Pueblo	A	98/99–45/46	48	3	4	91	377	730	1042	1042	875	724	446	195	29	5558
Conn.	Hartford	A	04/05–45/46	42	3	16	105	370	692	1065	1157	1062	859	524	213	47	6113
	New Haven	A	98/99–45/46	48	3	11	88	341	658	1017	1109	1023	840	522	221	47	5880
D. C.	Washington		98/99–45/46	48	0	2	42	251	553	872	928	834	624	340	101	14	4561
Fla.	Apalachicola		13/14–45/46	33	0	0	1	23	154	300	323	252	159	38	2	0	1252
	Jacksonville		98/99–45/46	48	0	0	0	25	144	294	302	244	131	42	3	0	1185
	Key West		98/99–45/46	48	0	0	0	0	2	14	21	15	7	0	0	0	59
	Miami		11/12–45/46	35	0	0	0	0	15	41	53	45	28	3	0	0	185
	Pensacola		13/14–45/46	33	0	0	0	25	159	305	332	255	162	39	4	0	1281
	Tampa	A	98/99–45/46	48	0	0	0	6	60	149	157	126	62	11	0	0	571
Ga.	Atlanta		98/99–45/46	48	0	0	12	128	392	644	660	563	382	169	33	2	2985
	Augusta		98/99–45/46	48	0	0	4	85	312	529	533	448	274	107	13	1	2306
	Macon		99/00–45/46	47	0	0	5	91	322	532	538	449	278	108	14	1	2338
	Savannah	A	98/99–45/46	48	0	0	1	45	206	390	395	332	194	66	6	0	1635
	Thomasville		05/06–40/41	36	0	0	2	48	208	361	359	299	178	52	5	1	1513
Idaho	Boise	A	98/99–45/46	48	9	17	136	385	717	1025	1077	840	688	440	252	92	5678
	Lewistown		00/01–32/33	33	5	9	107	378	688	932	992	779	603	371	193	52	5109
	Pocatello		98/99–45/46	48	12	21	176	475	821	1159	1224	1004	845	550	330	124	6741
Ill.	Cairo		98/99–45/46	48	0	0	26	181	493	823	878	748	512	232	60	4	3957
	Chicago		98/99–45/46	48	6	7	88	337	712	1116	1218	1080	861	531	259	67	6282
	Peoria	A	05/06–45/46	41	4	8	88	350	730	1126	1231	1035	790	436	178	28	6004
	Springfield		98/99–45/46	48	0	3	65	286	664	1056	1151	977	719	377	132	16	5446
Ind.	Evansville	A	98/99–45/46	48	0	1	35	211	544	888	948	822	582	288	85	6	4410
	Fort Wayne	A	11/12–45/46	35	6	13	106	374	737	1107	1211	1052	864	504	217	41	6232
	Indianapolis		98/99–45/46	48	1	4	66	297	660	1032	1102	973	737	410	154	22	5458
	Royal Center		18/19–31/32	14	11	19	116	373	740	1104	1239	976	860	502	245	54	6239
	Terre Haute		12/13–45/46	34	0	3	62	270	627	993	1072	897	687	358	133	15	5117
Iowa	Charles City		04/05–45/46	42	8	24	164	480	906	1362	1535	1281	995	552	255	62	7624
	Davenport		98/99–45/46	48	2	6	91	344	748	1176	1291	1111	835	448	171	29	6252
	Des Moines		98/99–45/46	48	1	6	102	354	767	1204	1320	1132	843	446	171	29	6375
	Dubuque		98/99–45/46	48	3	12	123	402	808	1249	1380	1190	915	493	204	41	6820
	Keokuk		98/99–41/42	44	1	3	71	303	680	1077	1191	1025	761	397	136	18	5663
	Sioux City	A	98/99–45/46	48	3	11	128	402	844	1273	1402	1206	909	485	202	40	6905
Kan.	Concordia		98/99–45/46	48	1	3	68	288	670	1060	1144	954	712	365	142	18	5425
	Dodge City	A	98/99–45/46	48	1	3	59	275	641	998	1046	868	668	351	139	20	5069
	Iola		05/06–40/41	36	0	1	40	236	579	930	1026	817	599	282	98	8	4616
	Topeka		98/99–45/46	48	0	2	56	254	623	1013	1096	917	659	326	116	13	5075
	Wichita	A	98/99–45/46	48	0	1	41	221	576	947	1016	836	604	290	103	9	4644
Ky.	Louisville		98/99–45/46	48	0	1	35	217	549	881	931	816	588	298	93	8	4417
	Lexington		98/99–40/41	43	1	3	48	258	601	916	964	862	650	352	123	14	4792
La.	New Orleans		98/99–45/46	48	0	0	0	23	145	304	323	247	129	31	1	0	1203
	Shreveport	A	98/99–45/46	48	0	0	4	71	275	506	531	415	241	79	10	0	2132
Me.	Eastport		98/99–45/46	48	158	146	271	528	827	1224	1364	1238	1080	778	530	301	8445
	Greenville		07/08–40/41 42/43–45/46	38	69	113	315	643	1012	1464	1625	1443	1251	842	468	194	9439
	Portland	A	98/99–45/46	48	28	48	182	466	794	1182	1309	1188	997	671	376	136	7377
Md.	Baltimore		98/99–45/46	48	0	1	33	227	526	855	921	837	637	343	95	12	4487

[a] Computed from daily temperatures recorded by *United States Weather Bureau* stations over a varied number of seasons as indicated in the 3rd and 4th column of the table. Degree-day data for airport stations are not included in this table. The data for United States cities were computed by the *United States Weather Bureau* in 1946 and 1947 in accordance with the requirements of the *National Joint Committee on Weather Statistics*. The data for a number of the cities listed are based on readings taken at more than one official city weather station during the periods of analysis, but the slight difference in the readings would not appreciably affect the resultant. [b] Letter A indicates city office and airport records combined.

TABLE 2. AVERAGE MONTHLY AND YEARLY DEGREE-DAYS FOR CITIES IN THE UNITED STATES, CANADA AND NEWFOUNDLAND[a] (CONTINUED)

State	Station	Years	No. of Seasons	July	Aug.	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Yearly Total
Mass.	Boston A	98/99–45/46	48	7	15	98	338	647	1008	1108	1025	841	538	245	66	5936
	Fitchburg	98/99–40/41	43	12	29	144	432	774	1139	1240	1137	940	572	254	70	6743
	Nantucket	98/99–45/46	48	15	15	87	315	590	904	1010	967	866	619	366	121	5875
Mich.	Alpena	98/99–45/46	48	53	79	235	548	874	1238	1388	1321	1162	764	448	168	8278
	Detroit A	98/99–45/46	48	7	15	111	388	749	1124	1230	1134	927	566	253	56	6560
	Escanaba	98/99–45/46	48	54	84	256	572	927	1329	1499	1382	1219	808	477	170	8777
	Grand Rapids	03/04–45/46	43	8	20	128	422	764	1136	1248	1143	944	569	263	57	6702
	Houghton	00/01–40/41 42/43–45/46	45	70	94	268	582	965	1355	1535	1421	1251	820	474	195	9030
	Lansing	10/11–45/46	36	18	36	167	467	818	1190	1306	1178	995	600	294	80	7149
	Ludington	12/13–40/41	29	41	55	182	472	794	1135	1271	1183	1056	698	418	153	7458
	Marquette	98/99–45/46	48	86	99	258	555	926	1306	1465	1349	1193	794	494	220	8745
	Sault Ste. Marie A	98/99–45/46	48	91	110	285	616	965	1379	1572	1470	1291	826	487	215	9307
Minn.	Duluth	98/99–45/46	48	80	97	292	631	1066	1539	1714	1497	1254	804	515	234	9723
	Minneapolis	98/99–45/46	48	8	23	167	481	942	1415	1587	1372	1072	577	260	62	7966
	Morehead	98/99–40/41	43	20	47	240	607	1105	1609	1815	1555	1225	679	327	98	9327
	St. Paul	98/99–32/33 37/38–40/41	39	11	24	169	488	942	1412	1589	1371	1078	573	258	60	7975
Miss.	Corinth	09/10–40/41	32	0	1	13	142	418	669	696	570	396	149	32	1	3087
	Meridian	00/01–45/46	46	0	0	6	99	322	525	539	440	274	107	17	1	2330
	Vicksburg	98/99–45/46	48	0	0	5	76	267	483	503	407	236	82	10	0	2069
Mo.	Columbia	98/99–45/46	48	0	3	62	266	621	1000	1076	916	655	337	120	14	5070
	Hannibal	98/99–40/41	43	1	3	66	288	652	1037	1139	980	710	374	128	15	5393
	Kansas City A	98/99–45/46	48	0	2	51	239	598	993	1077	909	651	322	108	12	4962
	Saint Louis	98/99–45/46	48	0	1	38	215	558	925	998	855	607	300	91	8	4596
	Springfield A	98/99–45/46	48	1	2	48	232	561	908	971	827	596	302	109	12	4569
Mont.	Billings A	09/10–45/46	37	14	31	223	530	889	1215	1310	1102	923	555	315	106	7213
	Havre	98/99–45/46	48	27	54	275	592	1012	1376	1532	1358	1102	614	341	133	8416
	Helena A	98/99–45/46	48	43	66	291	596	944	1252	1347	1157	990	639	413	192	7930
	Kalispell	99/00–45/46	47	66	102	332	636	968	1235	1339	1135	956	630	413	220	8032
	Miles City A	98/99–45/46	48	7	20	188	510	918	1322	1461	1267	997	545	275	81	7591
	Missoula	92/93–45/46	54	37	56	275	606	951	1235	1331	1072	903	590	369	179	7604
Neb.	Drexel	15/16–25/26	11	4	6	95	405	788	1271	1353	1096	843	493	219	38	6611
	Lincoln	98/99–45/46	48	1	5	85	325	732	1144	1242	1056	792	407	166	25	5980
	North Platte	98/99–45/46	48	4	9	131	410	799	1163	1227	1039	846	480	227	49	6384
	Omaha A	98/99–45/46	48	1	4	84	324	744	1169	1280	1088	810	410	157	24	6095
	Valentine	98/99–45/46	48	8	19	167	479	877	1246	1349	1166	962	563	287	74	7197
Nev.	Reno A	05/06–45/46	41	8	18	140	407	697	959	1007	791	702	498	301	93	5621
	Tonopah	14/15–40/41	27	5	7	105	388	713	1010	1075	870	749	522	286	82	5812
	Winnemucca	98/99–45/46	48	10	23	188	494	801	1090	1128	884	768	536	321	114	6357
N. H.	Concord A	03/04–45/46	43	18	49	189	497	823	1228	1345	1213	993	637	308	100	7400
N. J.	Atlantic City	98/99–45/46	48	1	2	39	247	546	867	946	887	750	485	208	37	5015
	Cape May	98/99–31/32	34	1	2	38	221	527	852	936	876	737	459	188	33	4870
	Newark A	98/99–23/24 35/36–40/41	32	1	6	65	295	635	980	1083	1002	794	448	162	29	5500
	Sandy Hook	15/16–40/41	26	1	2	40	268	579	921	1016	973	833	499	206	31	5369
	Trenton	14/15–45/46	32	1	6	63	301	604	957	1033	923	748	441	154	25	5256
N. M.	Albuquerque A	19/20–45/46	27	0	0	27	258	646	913	955	708	592	322	91	5	4517
	Roswell	05/06–45/46	41	0	0	26	191	512	781	773	585	459	199	50	2	3578
	Santa Fe	98/99–45/46	43	12	15	129	451	772	1071	1094	892	786	544	297	60	6123
N. Y.	Albany	98/99–45/46	48	4	15	117	411	753	1143	1271	1169	948	551	220	46	6648
	Binghamton	98/99–45/46	48	15	37	148	448	767	1137	1236	1155	950	584	267	74	6818
	Buffalo A	98/99–45/46	48	15	24	126	413	745	1110	1226	1165	995	668	348	90	6925
	Canton	06/07–45/46	40	27	61	219	550	898	1368	1516	1385	1139	695	340	107	8305
	Ithaca	99/00–42/43	44	17	40	156	451	770	1129	1236	1156	978	606	292	83	6914
	New York	98/99–45/46	48	1	4	50	272	594	940	1028	953	771	465	172	30	5280
	Oswego	98/99–45/46	48	20	33	147	440	762	1151	1275	1188	1015	665	366	124	7186
	Rochester A	98/99–45/46	48	10	26	132	423	751	1123	1227	1155	967	605	282	71	6772
	Syracuse A	03/04–45/46	43	13	32	146	437	760	1147	1255	1167	972	611	283	76	6899
N. C.	Asheville	02/03–45/46	44	2	3	49	279	565	800	817	719	558	315	114	15	4236
	Charlotte	98/99–45/46	48	0	1	17	148	420	684	700	600	413	198	39	4	3224
	Hatteras	98/99–45/46	48	0	0	1	61	273	500	570	530	391	193	34	1	2554
	Manteo	04/05–28/29	25	0	0	7	113	358	595	642	594	469	249	75	7	3109
	Raleigh	98/99–45/46	48	0	1	17	153	415	681	702	615	429	210	46	6	3275
	Wilmington	98/99–45/46	48	0	0	5	90	306	520	531	479	320	144	24	1	2420
N. D.	Bismarck	98/99–45/46	48	21	44	244	595	1057	1520	1704	1479	1181	643	340	109	8937
	Devils Lake	04/05–45/46	42	42	76	295	687	1186	1676	1906	1615	1313	752	411	145	10104
	Grand Forks	12/13–40/41 42/43–45/46	33	32	60	274	663	1160	1681	1895	1608	1298	718	359	123	9871
	Williston	98/99–45/46	48	28	61	285	637	1104	1545	1733	1513	1226	671	368	130	9301
Ohio	Cincinnati	98/99–45/46	48	1	3	53	273	611	960	1008	898	668	369	130	16	4990
	Cleveland	98/99–45/46	48	7	14	93	354	684	1045	1143	1067	876	553	252	56	6144
	Columbus	98/99–45/46	48	2	6	68	314	670	1019	1081	975	749	432	165	25	5506
	Dayton	11/12–42/43 45/46	33	2	6	71	309	660	992	1079	939	752	416	163	23	5412
	Sandusky	98/99–45/46	48	3	8	82	347	695	1067	1155	1066	862	536	232	42	6095
	Toledo	98/99–45/46	48	5	13	100	370	718	1097	1189	1083	887	533	227	47	6269
Okla.	Broken Arrow	18/19–30/31	13	0	0	28	169	513	805	881	646	506	212	61	5	3826
	Oklahoma City	98/99–45/46	48	0	0	22	153	455	792	846	684	457	200	58	3	3670
Ore.	Baker	98/99–45/46	48	54	72	259	534	848	1161	1222	984	827	601	418	217	7197
	Medford A	11/12–40/41 45/46	31	7	10	99	345	632	837	844	636	556	387	223	74	4650

TABLE 2. AVERAGE MONTHLY AND YEARLY DEGREE-DAYS FOR CITIES IN THE UNITED STATES, CANADA AND NEWFOUNDLAND[a] (CONTINUED)

STATE	STATION	YEARS	NO. OF SEA-SONS	JULY	AUG.	SEPT.	OCT.	NOV.	DEC.	JAN.	FEB.	MAR.	APR.	MAY	JUNE	YEARLY TOTAL
Ore.	Portland.	98/99-45/46	48	27	28	106	292	538	725	775	616	533	368	237	108	4353
	Roseburg	98/99-45/46	48	23	26	116	316	541	714	730	582	530	386	257	111	4332
Pa.	Erie	98/99-45/46	48	8	17	101	367	692	1049	1159	1105	922	591	284	68	6363
	Harrisburg A	98/99-45/46	48	1	6	67	314	637	990	1073	973	757	425	146	23	5412
	Philadelphia	98/99-45/46	48	0	2	36	235	544	884	962	881	685	378	115	17	4739
	Pittsburgh	98/99-45/46	48	3	7	69	322	651	982	1042	964	751	444	166	29	5430
	Reading	12/13-45/46	34	1	5	69	301	606	957	1038	929	736	423	144	23	5232
	Scranton	00/01-45/46	46	6	22	117	400	717	1074	1162	1071	862	523	213	51	6218
R. I.	Block Island	98/99-45/46	48	10	11	79	313	596	919	1030	984	875	618	354	108	5897
	Narragansett Pier	98/99-17/18	20	1	26	121	366	691	1012	1113	1074	916	622	342	113	6397
	Providence	04/05-45/46	42	6	16	101	358	668	1020	1106	1027	847	538	237	60	5984
S. C.	Charleston	98/99-45/46	48	0	0	1	47	225	428	452	384	239	83	7	0	1866
	Columbia	98/99-45/46	48	0	0	6	95	327	560	568	482	305	126	18	1	2488
	Due West	21/22-31/32	11	0	0	9	142	393	594	651	491	411	158	39	2	2890
	Greenville A	17/18-45/46	29	0	1	13	127	410	650	684	551	403	179	40	1	3059
S. D.	Huron	98/99-45/46	48	10	20	159	502	962	1409	1572	1353	1039	573	271	70	7940
	Pierre	98/99-40/41 42/43-45/46	47	4	11	136	438	887	1317	1460	1253	971	516	238	52	7283
	Rapid City	98/99-45/46	48	15	28	192	495	842	1178	1280	1140	981	598	339	109	7197
Tenn.	Chattanooga	98/99-45/46	48	0	0	13	150	432	691	711	604	412	185	39	1	3238
	Knoxville A	98/99-45/46	48	0	0	20	189	498	756	774	666	470	226	56	3	3658
	Memphis A	98/99-45/46	48	0	0	14	126	387	670	716	600	386	157	33	1	3090
	Nashville	98/99-45/46	48	0	0	20	170	469	748	788	675	467	218	55	3	3613
Texas	Abilene A	98/99-45/46	48	0	0	10	96	332	603	619	483	296	110	23	1	2573
	Amarillo A	98/99-45/46	48	1	2	42	221	548	854	861	719	546	284	107	11	4196
	Austin A	26/27-45/46	20	0	0	2	31	227	410	458	315	185	46	5	0	1679
	Brownsville A	08/09-45/46	38	0	0	0	8	65	176	191	111	65	11	1	0	628
	Corpus Christi	98/99-45/46	48	0	0	0	11	102	255	282	204	93	17	1	0	965
	Dallas A	13/14-45/46	33	0	0	6	70	293	574	600	437	281	91	15	0	2367
	Del Rio	05/06-45/46	41	0	0	2	35	203	413	413	262	139	31	3	0	1501
	El Paso A	98/99-45/46	48	0	0	6	88	366	615	615	432	291	104	14	1	2532
	Fort Worth A	98/99-45/46	48	0	0	5	79	285	553	586	463	270	97	16	1	2355
	Galveston	98/99-45/46	48	0	0	0	14	123	290	334	255	130	27	1	0	1174
	Houston	09/10-45/46	37	0	0	1	27	160	331	361	247	150	36	2	0	1315
	Palestine	98/99-45/46	48	0	0	4	67	261	496	512	401	236	80	11	0	2068
	Point Arthur	17/18-45/46	29	0	0	1	27	177	328	375	254	151	37	2	0	1352
	San Antonio A	98/99-45/46	48	0	0	1	31	171	366	390	287	148	37	4	0	1435
	Taylor	01/02-40/41	40	0	0	2	56	234	462	494	375	214	64	8	0	1909
Utah	Modena	00/01-45/46	46	6	11	156	499	832	1142	1190	944	816	567	338	97	6598
	Salt Lake City	98/99-45/46	48	3	5	98	371	712	1033	1093	871	716	446	236	66	5650
Vt.	Burlington	06/07-45/46	40	23	51	209	530	870	1313	1467	1338	1111	694	339	106	8051
	Northfield	98/99-42/43	45	62	112	283	602	947	1389	1524	1384	1176	754	405	166	8804
Va.	Cape Henry	98/99-45/46	48	0	0	7	125	398	676	731	682	526	301	86	6	3538
	Lynchburg A	98/99-45/46	48	1	2	37	230	521	799	829	732	537	287	81	12	4068
	Norfolk	98/99-45/46	48	0	0	9	129	392	668	712	650	483	254	62	5	3364
	Richmond	98/99-45/46	48	0	1	27	196	486	780	814	722	538	278	72	8	3922
	Wytheville	02/03-40/41	39	7	13	82	352	662	916	945	836	677	410	168	35	5103
Wash.	North Head	02/03-45/46	44	251	229	255	350	491	642	697	597	610	505	428	312	5367
	Seattle	98/99-45/46	48	67	69	170	365	554	704	759	637	595	436	299	160	4815
	Spokane A	98/99-45/46	48	20	37	184	480	817	1061	1139	931	756	490	285	118	6318
	Tacoma	98/99-45/46	48	71	75	190	390	581	737	786	658	612	455	313	171	5039
	Tatoosh Island	98/99-45/46	48	301	295	325	421	534	654	716	627	643	537	454	350	5857
	Walla Walla	98/99-45/46	48	5	10	90	315	662	910	981	770	571	354	186	56	4910
	Yakima	09/10-45/46	37	8	17	124	395	778	1050	1125	837	624	374	193	60	5585
W. Va.	Elkins A	98/99-45/46	48	15	23	115	403	722	1003	1033	947	763	489	229	58	5800
	Parkersburg	98/99-45/46	48	1	3	56	286	617	930	977	882	660	369	129	18	4928
Wis.	Green Bay	98/99-40/41	48	17	38	179	494	889	1329	1493	1329	1087	658	327	91	7931
	La Crosse	98/99-45/46	48	7	22	157	454	864	1339	1492	1281	990	531	232	52	7421
	Madison	04/05-45/46	42	8	20	145	452	857	1296	1451	1246	1002	588	274	66	7405
	Milwaukee	98/99-45/46	48	13	17	124	411	786	1203	1329	1177	959	617	341	102	7079
	Wausau	15/16-40/41	26	26	58	216	568	982	1427	1594	1381	1147	680	315	100	8494
Wyo.	Cheyenne A	98/99-45/46	48	40	46	251	587	876	1144	1187	1064	996	720	460	165	7536
	Lander	98/99-45/46	48	27	43	265	623	1021	1400	1427	1197	1006	669	410	155	8243
	Yellowstone Park	04/05-40/41	37	125	173	424	759	1079	1386	1464	1252	1165	841	603	334	9605
Alta.	Calgary	.	..	108	167	432	722	1122	1426	1609	1355	1215	750	480	264	9,650
	Edmonton	.	.	70	167	441	750	1224	1593	1810	1504	1299	777	428	222	10,285
B. C.	Vancouver	.		43	65	234	459	657	818	893	736	682	498	326	162	5,573
	Victoria	. .		155	158	264	446	462	738	815	689	651	504	366	237	5,485
	Prince Rupert	.	.	282	229	342	546	702	893	933	804	809	645	521	357	7,063
Man.	Churchill			350	391	696	1187	1773	2356	2604	2288	2204	1530	1097	672	17,148
	Winnipeg	..	.	27*	34	330	744	1302	1829	2111	1775	1531	822	397	78	10,980
N. B.	Moncton			8*	56	282	595	936	1373	1528	1392	1190	798	474	180	8,812
	Saint John	.....	...	124	112	270	558	870	1271	1417	1266	1132	792	505	261	8,578
N. S.	Halifax	... .	.	12	12	189	493	783	1141	1280	1168	1065	768	493	210	7,614
Ont.	Fort William	. .....	.	62	155	354	722	1143	1596	1807	1582	1386	888	567	234	10,496
	Hamilton	....	. .	3*	9*	108	471	816	1178	1305	1187	1063	651	322	6	7,119
	London	....	.....	34*	11*	126	508	843	1200	1336	1240	1073	642	307	105	7,425
	Ottawa	....	.....	29*	22*	204	595	984	1494	1646	1459	1256	726	310	91	8,816
	Toronto	....	.....	16	35	168	504	817	1155	1304	1206	1072	669	341	87	7,374
	Windsor	....	.....	19*	5*	54	425	795	1172	1283	1148	995	582	251	73	6,802

TABLE 2. AVERAGE MONTHLY AND YEARLY DEGREE-DAYS FOR CITIES IN THE UNITED STATES, CANADA AND NEWFOUNDLAND* (CONCLUDED)

STATE	STATION	YEARS	NO. OF SEA-SONS	JULY	AUG.	SEPT.	OCT.	NOV.	DEC.	JAN.	FEB.	MAR.	APR.	MAY	JUNE	YEARLY TOTAL
P.E.I.	Charlottetown			5*	57*	222	539	858	1246	1463	1338	1203	858	536	213	8,538
P.Q.	Montreal			7*	20*	180	561	948	1407	1587	1392	1209	702	298	88	8,399
	Quebec			17*	43	276	651	1050	1534	1696	1481	1311	849	428	102	9,438
Sask.	Saskatoon			16*	68	396	781	1305	1767	2027	1635	1445	725	397	138	10,700
Y. T.	Dawson			167	322	687	1209	1908	2440	2666	2159	1879	1092	580	246	15,355

Degree-days for cities in Canada and Newfoundland were supplied by the *Canadian Meteorological Division, Department of Transport*, and were computed from mean temperature normals. * Indicates actual degree days for 1947.

being prepared and must be found by surveying plans, by observation, or by measurement of the building. Values of U for use in this equation are the unit fuel consumptions per degree-day, obtained as a result of the collection of operating information and listed in Tables 4, 5 and 6. Attention is directed to the nature of these units in the next following sections.

Unit Fuel Consumptions per Degree-Day

The quantity of fuel used per degree-day in a given heating plant can be reduced to a unit basis in terms of quantity of fuel or steam per degree-day per thousand Btu hourly heat loss at design conditions. A less frequently used basis is quantity of fuel per (degree-day) (square foot of floor area). In fact any convenient unit can be used to relate the consumption to the degree-day and to the building.

The choice of these units requires explanation, and some discrimination and judgment. If the volume basis is used, the net heated space is preferable to the gross building cubage, since gross cubage includes outer walls and certain portions of attic and basement space which are usually unheated. In the absence of data on net heated volume, a figure of 80 percent of the gross volume may be used to obtain the estimated net heated volume. The volume basis has been rather widely used primarily because it is simple to apply. In industrial buildings it is usually easier to obtain the correct volume of a given building than to measure and evaluate the heating capacity of its heating system, or calculate its maximum hourly Btu loss. The comparison of buildings on a straight volume basis does not allow for variation in exposure, type of construction, ratio of exposed area to cubical contents, and type of occupancy. It is inaccurate for estimating purposes unless the buildings are of very similar nature.

The calculated heat loss or the heating capacity of the installed radiation may be used as the unit. The use of the heating capacity of the installed radiation is of questionable value when referring to heat transfer surfaces

TABLE 3. CORRECTION FACTORS FOR OUTSIDE DESIGN TEMPERATURES*

OUTSIDE DESIGN TEMP F	−20	−10	0	+10	20
Correction Factor	0.778	0.875	1.000	1.167	1.400

* The multipliers in Table 3, which are high for mild climates and low for cold regions, are not in error as might appear. The unit figures in Tables 4, 5, and 6 are per square foot of radiator or thousand Btu heat loss per degree-day. For equivalent buildings and heating seasons, those in warm climates have lower design heat losses and smaller radiator quantities than those in cold cities. Consequently, the *unit* figure in quantity of fuel per (square foot of radiator) (degree-day), is larger for warm localities than for colder regions. Since the northern cities have more radiator surface per given building and a higher seasonal degree-day total than cities in the south, the total fuel per season will be larger for the northern city.

used in warm air furnace or central air conditioning systems. Where steam or hot water radiation is already installed, care should be exercised in using the installed radiation as the basis for estimating, since actual installed radiation may differ considerably from the exact radiation requirements. In view of all these considerations, it is believed that the unit based on *thousands of Btu of hourly calculated heat loss for the design hour* is probably the most desirable.

Estimating Gas Consumption

Values of the Unit Fuel Consumption Constant (U) for gas are given in Table 4 for various gas heating values, and different types and sizes of heating plants. They are based on an inside design temperature of 70 F and an outside design temperature of 0 F, and apply only to these conditions. For other outside design conditions corrections must be made by applying factors given in Table 3.

The factors in Table 4, as corrected if necessary, are satisfactory for regions having 3500 to 6500 degree-days per heating season. In regions with less than 3500 degree-days the unit gas consumption is higher than given; where over 6500, the unit is less than given. Ten percent addition or deduction in these cases is recommended by *A.G.A.* publications. This table cannot be used for making estimates for industrial buildings where low inside temperatures are maintained.

For gas heat values other than those given in Table 4, simply interpolate or extrapolate. It will also be noted that Table 4 applies only to small installations. In general, the larger the installation, the smaller the unit gas consumption becomes, and the values in the table should be used with care, if at all, in large gas-burning installations.

Example 4: Estimate the gas required to heat a building located in Chicago, Ill., where the heating season has 6282 degree-days and the gas heating value is 800 Btu per cu ft. The calculated heating requirements are 1000 sq ft of hot water radiation based on design temperature of -10 F and 70 F.

Solution: From Table 4, the fuel consumption for a design temperature of 0 F with 800 Btu gas is found to be 0.087 cu ft of gas per (degree-day) (square foot of hot water radiation). From Table 3, the correction factor is 0.875 for -10 F outside design temperature, hence, $0.875 \times 0.087 = 0.076$. By Equation 4,

$$F = 0.076 \times 1000 \times 6282 = 478{,}000 \text{ cu ft.}$$

Estimating Oil Consumption

Table 5 gives unit fuel consumption factors for oil, similar to those given for gas in Table 4.

The factors in Table 5 apply only to an inside design temperature of 70 F and an outside design temperature of 0 F. For other outside design temperatures, the constants in Table 5 must be multiplied by the values in Table 3 as explained under Estimating Gas Consumption.

Values given in Table 5 assume the use of oil with a heating value of 141,000 Btu per gallon. For other heating values, multiply the values in Table 5 by the ratio of 141,000 divided by the heating value per gallon of fuel being used.

Example 5: Estimate the seasonal oil consumption of a boiler, designed for oil firing, in a building located in Toledo, Ohio. The building has a calculated heat loss of 240,000 Btu per hr. The oil heat value is 144,000 Btu per gal, and the assumed seasonal efficiency is 80. The outside design temperature for Toledo is -10 F, and the inside design temperature is 70 F.

Solution: From Table 5, under 80 percent efficiency and in the bottom line, the value of U is found to be 0.00383 gal per 1000 Btu hourly heat loss for 0 F outside temperature. The correction factor for -10 F outside design temperature from Table 3 is 0.875. Solving 0.875 × 0.00383 = 0.00335. Making a further correction for the heating value,

$$0.00335 \times \frac{141{,}000}{144{,}000} = 0.00328 \text{ gal per 1000 Btu per hr calculated heat loss per degree-day.}$$

From Table 2, the total normal degree-days for Toledo are 6269. Since U is expressed in 1000 Btu, N is equal to 240. Substituting in Equation 4

$$F = 0.00328 \times 6269 \times 240 = 4930 \text{ gal.}$$

Estimating Coal or Coke Consumption

Coal or coke consumption estimates are made by following exactly the same procedure as for oil. Values of U are given in Table 6 which only apply to an inside design temperature of 70 F and an outside design temperature of 0 F. A correction must be made for other conditions by use of the multiplying factors in Table 3. Data in Table 6 are based on 12,000 Btu per lb coal, and for other heating values of coal they must be multiplied by the ratio of 12,000 divided by the heating value of fuel used.

TABLE 4. UNIT FUEL CONSUMPTION CONSTANTS (U) FOR GAS[a]
Based on 0 F Outside Temperature, 70 F Inside Temperature

Heating Value of Gas Btu per Cu Ft	Hot Water			Steam			Warm Air	
	Cu Ft Gas per Degree-Day per Sq Ft EDR			Cu Ft Gas per Degree-Day per Sq Ft EDR			Cu Ft Gas per Degree-Day per 1000 Btu Hourly Design Heat Loss	
	Up to 500 Sq Ft	500 to 1200 Sq Ft	Over 1200 Sq Ft	Up to 300 Sq Ft	300 to 700 Sq Ft	Over 700 Sq Ft	Gravity	Fan Systems
500	0.149	0.142	0.134	0.254	0.242	0.231	0.896	0.861
535	0.139	0.132	0.126	0.237	0.226	0.216	0.840	0.805
800	0.094	0.087	0.085	0.159	0.151	0.144	0.560	0.538
1000	0.075	0.071	0.068	0.127	0.121	0.116	0.449	0.431
1 Therm = 100,000 Btu	Gas Consumption in Therms per Degree-Day							
	0.000743	0.000709	0.000675	0.00127	0.00121	0.00116	0.00450	0.00430

[a] Abstracted from Comfort Heating, *American Gas Association*, 1938 and 5 percent added for operation without night reduction of temperature.

Example 6: A building in Scranton, Pa. has a calculated heat loss of 240,000 Btu per hr based on an inside design temperature of 70 F and an outside design temperature of −10 F. What will be the estimated normal seasonal anthracite or coke consumption for heating if 13,000 Btu per lb fuel is burned in a hand-fired boiler, with automatic control, at a seasonal efficiency of 80 percent, and what part of the total will be used during November, December, and January?

Solution: From Table 6, U is 0.0444 of coal or coke per 1000 Btu per hr heat loss per degree-day. Correcting for the outside design temperature of −10 F from Table 3, the value of U is 0.875 × 0.0444 = 0.0389. From Table 2, D is 6218 and from the problem, N is 240.

Substituting in Equation 4,

$$F = 0.0389 \times 240 \times 6218 = 58{,}100 \text{ lb.}$$

Fuel used over any period is, according to the theory of the degree-day, proportional to the number of degree-days during the period. From Table 2, the average numbers of degree-days for November, December, and January in Scranton are 717, 1074, and 1162 respectively, a total of 2953. The yearly total is 6218, so that during these three months the estimated consumption is

$$\frac{2953}{6218} \times 58{,}100 = 27{,}600 \text{ lb.}$$

TABLE 5. UNIT FUEL CONSUMPTION[a] CONSTANTS (U) FOR OIL[b]
Based on 0 F Outside Temperature, 70 F Inside Temperature

UNIT[c]	EFFICIENCY IN PERCENT	
	70	80
Gal Oil per Sq Ft Steam Radiator	0.00105	0.00092
Gal Oil per Sq Ft Hot Water Radiator	0.00066	0.00058
Gal Oil per 1000 Btu per Hour Heat Loss	0.00437	0.00383

[a] Based on a heating value of 141,000 Btu per gallon.
[b] Abstracted by permission from *Degree-Day Handbook* (Second Edition, 1937), by C. Strock and C. H. B. Hotchkiss. Seven percent added for operation without night reduction of temperature and change to heating value of 141,000 Btu per gallon.
[c] Per degree day.

Estimating Steam Consumption

In estimating steam consumption the efficiency is generally assumed at 100 percent. If for low-pressure steam an average heating value of 1000 Btu per pound of steam is used, no correction is necessary. In comparing values from different cities, correction should be made for design temperature (see Table 3) when the unit figures are in terms of heat loss but not when the values are in terms of building volume or floor space.

Where the heat loss is calculated in Btu per (hour) (*degree difference in temperature*) the simple Equation 5 may be used:

$$F = \frac{H \times 24 \times D}{1000} \tag{5}$$

where

F = pounds of steam required for estimate period.
H = calculated heat loss, Btu per (hour) (degree difference).
24 = hours in one day.
D = number of degree-days for the period of estimation.
1000 = Btu delivered per pound of steam condensed.

In this method the number of degree-days automatically takes care of average inside and outside temperature difference. When degree-days are taken from Table 2, an average inside temperature of approximately 70 F is assumed throughout the period. If an average inside temperature other

TABLE 6. UNIT FUEL CONSUMPTION[a] CONSTANTS (U) FOR COAL[b]
Based on 0 F Outside Temperature, 70 F Inside Temperature

UNIT[c]	EFFICIENCY IN PERCENT				
	40	50	60	70	80
Lb Coal per Sq Ft Steam Radiator	0.0216	0.0172	0.0143	0.0123	0.0108
Lb Coal per Sq Ft Hot Water Radiator	0.0135	0.0108	0.0091	0.0078	0.0068
Lb Coal per 1000 Btu per Hour Heat Loss	0.0889	0.0717	0.0592	0.0507	0.0444

[a] Based on a heating value of 12,000 Btu per pound.
[b] Abstracted by permission from *Degree-Day Handbook* (Second Edition, 1937), by C. Strock and C. H. B. Hotchkiss. Eight percent added for operation without night reduction of temperature.
[c] Per degree-day.

TABLE 7. STEAM CONSUMPTION OF BUILDINGS WITH VARIOUS TYPES OF OCCUPANCY[a]

Type of Building	No. Bldgs.	Average Volume Heated Space 1000 Cu Ft	Steam for Heating Lb per DD per 1000 Cu Ft	Average Hours of Occupancy
Office	334	2160	0.685	12.1
Office and Bank	49	3000	0.577	13.1
Office and Printing	8	1895	1.230	17.7
Office and Theater	7	4950	0.412	12.9
Office and Stores or Shops	26	1615	0.617	13.2
Bank	16	806	0.786	11.7
Department Store	63	3400	0.385	11.1
Stores	73	310	0.624	10.4
Loft	63	865	0.588	10.0
Warehouse	24	2230	0.459	9.4
Hotel and Club	73	1795	0.990	22.3
Apartment or Residence	51	1425	0.962	21.8
Theater	22	1240	0.482	12.9
Garage	13	1540	0.202	21.4
Manufacturing	19	1350	0.808	9.5
Church	9	656	0.532	7.9
Hospital	4	3306	1.194	22.0
School	8	1115	0.592	11.5
Municipal or Federal	15	3215	0.587	15.6
Lodge, Gym, Hall or Auditorium	12	880	0.390	12.4
Miscellaneous	7	1387	0.479	21.4

[a] Principles of Economical Heating, *National Association of Building Owners and Managers.*

than approximately 70 F is to be used, the number of degree-days should be obtained for the new base.

Example 7: An eight-story building in Pittsburgh is operated with a daytime temperature of 70 F. The calculated heat loss is 10,500 Btu per (hr) (*degree temperature difference*). What is the estimated average yearly steam consumption for building heating?

Solution: Since the average inside temperature is approximately 70 F, the degree-days from Table 2, based on 70 F may be used. Therefore, from Table 2, Pittsburgh has 5430 degree-days per normal season. Inserting in Equation 5:

$$F = \frac{10{,}500 \times 24 \times 5430}{1000} = 1{,}368{,}000 \text{ lb of steam.}$$

Consideration has been given to the difference in steam utilization of different types of buildings, and Table 7 shows actual average units for these various types. These figures were obtained from operating results in 896 buildings located in all sections of the United States. Being averages, and for small groups in each type, the figures may need considerable modification to allow for local variations. It should be especially noted that the steam used for heating water for service is not included in the values given in Table 7.

Example 8: A store in Philadelphia with a heating system designed to maintain 70 F inside in 0 F weather has 250,000 cu ft of heated space. What would be the estimated average yearly steam consumption of purchased steam for heating?

Solution: According to Table 7, a store would use 0.624 lb of steam per degree-day per 1000 cu ft heated space. From Table 2, Philadelphia has 4739 degree-days per normal year. Inserting in Equation 4:

$$F = 0.624 \times 250 \times 4739 = 739{,}000 \text{ lb of steam.}$$

Degree-Day as an Operating Unit

The degree-day is also widely used as a means of comparing the efficiency of the fuel consumption of one period with another *for the same building.*

TABLE 8. HEAT CONSUMPTION RECORD FOR COMPARISON

Heating Season		Col. 1	Col. 2 Total Consumption	Col. 3 Consumption for Heating	Col. 4 Avg Mean Temp.	Col. 5 Deg Days 65 F Base	Col. 6 Lb/Deg Day	Col. 7 Lb/Deg Day/ M Cu Ft
	1942–43	Sept.	337,500	170,500	65	146	1,170	0.575
		Oct.	834,200	667,200	53	339	1,966	0.970
		Nov.	1,446,600	1,279,600	44	641	1,990	0.982
		Dec.	2,176,400	2,009,400	25	1,233	1,630	0.804
		Jan.	2,332,200	2,165,200	22	1,297	1,670	0.822
		Feb.	2,131,100	1,964,100	28	1,106	1,775	0.888
		Mar.	2,021,900	1,854,900	31	1,032	1,799	0.885
		Apr.	1,241,500	1,074,500	43	647	1,660	0.818
		May	672,500	505,500	55	303	1,670	0.822
		June	258,600	91,600	. .	50	1,830	0.905
		July	188,400	......	...	....	..	.....
		Aug.	180,100	...	..	....	...	.....
		Total	13,821,000	......	..	. .	.....	. . .
	1943–44	Sept.	330,200	146,200	61	167	875	0.431
		Oct.	887,100	703,100	52	410	1,718	0.845
		Nov.	1,525,200	1,341,200	39	812	1,653	0.815
		Dec.	2,045,500	1,861,500	28	1,120	1,660	0.817
		Jan.	1,933,400	1,749,400	30	1,044	1,670	0.825
		Feb.	1,990,200	1,806,200	30	1,111	1,624	0.800
		Mar.	1,984,100	1,800,100	31	1,021	1,760	0.868

If, for example, the heat consumption in March, 1943, is compared with that in March, 1944, it will be found that in the latter the steam consumption is 1799 − 1760 = 39 lb less which is a decrease of 2.2 percent.

Since the fuel consumption is proportional to the weather (degree-days), and since the periods to be compared may not have the same weather conditions, the comparison can be made only after the fuel consumptions have been computed on a comparable weather basis, that is, upon the actual number of degree-days occurring for a given month and year in the city under consideration. Since fuel consumption is proportional to the number of degree-days, plant operators frequently compute each month the fuel burned per degree-day by the heating plant. The resulting unit value, by eliminating the outside temperature variable, indicates whether the operating efficiency of the plant is above or below the previous month or year.

The figures in Table 8 illustrate a typical example of a method of using the degree-day for making heating comparisons for one building for two consecutive heating seasons. The heat quantity figures inserted are pounds of steam, but a similar comparison could be made using pounds of coal, gallons of oil, or cubic feet of gas.

For such a comparison, a two-year record, as shown in Table 8, is often used

TABLE 9. BUILDING LOAD FACTORS AND DEMANDS OF SOME DETROIT BUILDINGS

Building Classification	Load Factor	Lb of Demand per (Hour) (Sq Ft of Equivalent Installed Radiator Surface)
Clubs and Lodges	0.318	0.184
Hotels	0.316	0.207
Printing	0.287	0.217
Offices	0.263	0.209
Apartments	0.255	0.225
Retail Stores	0.238	0.182
Auto Sales and Service	0.223	0.248
Banks	0.203	0.158
Churches	0.158	0.152
Department Stores	0.138	0.145
Theaters	0.126	0.151

The year under consideration may then be compared, month by month, with the previous year. Column 3, Consumption for Heating, would be used if the same fuel is used for heating and process steam. Some reasonable figure must be assumed for the process requirement and should be deducted from the amount shown in column 2. This would leave in column 3 only the fuel chargeable to heating. The degree-day values in column 5 are obtainable from the local Weather Bureau. Values in column 6 are obtained by dividing corresponding values in column 3 by the degree-days in column 5. The heating index in column 6 is, then, a figure of heat consumption, corrected for outdoor temperature, and should be relatively constant month by month. Column 7 in Table 8 may be used if the heat consumption is to be compared on a building volume basis with average values shown in Table 7.

MAXIMUM DEMANDS AND LOAD FACTORS

In one form of district heating rates, a portion of the charge is based upon the maximum demand of the building. The maximum demand may be measured in several different ways. It may be taken as the instantaneous peak or as the rate of use during any specified interval. One method is to take the average of the three highest hours during the winter. These figures are shown for a number of buildings in Detroit in Table 9.[12]

These maximum demands were measured by an attachment on the condensation meter, and therefore represent the amounts of condensation passed through the meter in the highest hours, rather than the true rate at which steam is supplied. There might be slight differences in these two quantities due to time lag and to storage of condensate in the system, but wherever this has been investigated it has been found to be negligible.

The load factor of a building is the ratio of the average load to the maximum load and is an index of the utilization. Thus, in Table 9, the theaters, operating for short hours, have a load factor of 0.126 as compared with the figure of 0.318 for clubs and lodges.

REFERENCES

[1] Comfort Heating (*American Gas Association*, 1938).

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[4] Heat Losses and Efficiencies of Fuels in Residential Heating, by R. A. Sherman and R. C. Cross (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 185).

[5] The Stoker-Fired Warm-Air Furnace in the Research Residence, by S. Konzo (*University of Illinois, Engineering Experiment Station, Circular* 39, 1939, p. 98).

[6] Performance of a Hot Water Heating System in the $I = B = R$ Research Home at the *University of Illinois* (*University of Illinois, Engineering Experiment Station, Bulletin* No. 349, Jan. 4, 1944).

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[10] House Heating (Industrial Gas Series, *American Gas Association*, Third Edition).

[11] Report of Commercial Relations Committee (*Proceedings, National District Heating Association*, 1932).

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CHAPTER 18

GRAVITY WARM AIR SYSTEMS

Warm Air Leaders, Stacks, and Registers; Return Air Grilles, Ducts, and Connections; Outline of Design Procedure

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WARM air heating systems of the *gravity* type are described in this chapter.[1] In these systems the motive head producing flow depends upon the difference in weight between the heated air leaving the top of the casing and the cooled air entering the bottom of the casing, while in the forced air type a fan may supply all or part of the motive head.

A gravity warm-air furnace heating plant consists of a fuel-burning furnace or heater, enclosed in a casing of sheet metal, which is placed in the basement of the building. The heated air, taken from the top or sides near the top of the furnace casing, is distributed to the various rooms of the building through sheet metal warm-air pipes. The warm-air pipes in the basement are known as leaders, and the vertical warm-air pipes which are installed in the inside partitions of the building are called stacks. The heated air is discharged into the rooms through registers which are set in register boxes placed either in the floor or in the side wall, usually at or near the baseboard. A sectional view of a typical plant showing good installation practice is given in Fig. 1.

The air supply to the furnace is usually taken entirely from inside the building through one or more recirculating ducts, although in some cases an outside air supply duct is provided.

WARM AIR LEADERS, STACKS, AND REGISTERS

In a gravity circulating warm-air furnace system, the size of the leader pipe to a given room depends upon the length of the leader and the temperature of the warm air entering the room at the register. For most successful operation, the furnace should be centrally located with respect to register and stack positions so that the leaders will be of uniform length and as short as possible, in which case the frictional resistance to air flow and the temperature loss from the ducts will be about the same for all leaders and stacks.

Originally, the design was based on the heat carrying capacities per square inch of leader pipe area with register air temperatures of 175 F. Later, in the revision of the entire design procedure, as shown in the section entitled *Outline of Design Procedure*, the carrying capacities of leader pipes have been expressed directly in terms of Btu per hour.

In general, it is advisable to use two or more leader pipes to rooms requiring more than the capacity of a 12 in. round pipe. The tops of all sizes of leader pipes should be cut into the furnace bonnet at the same elevation, and from this point there should be a uniform upgrade of at least 1 in. per foot of run. Leaders over 12 ft in length, or having a large number of elbow fittings should be avoided if possible. In cases where such leaders are necessary, it is recommended that smooth transition fittings be used, and that duct insulation be applied. Asbestos paper, unless of the corrugated type, should not be considered as insulation. To assist in balancing the air distribution of the system, a damper should be placed in each leader pipe except one, this latter leader preferably being connected to a room heated at all times, such as a living room.

In a gravity circulating system, the ratio of stack to leader area is quite

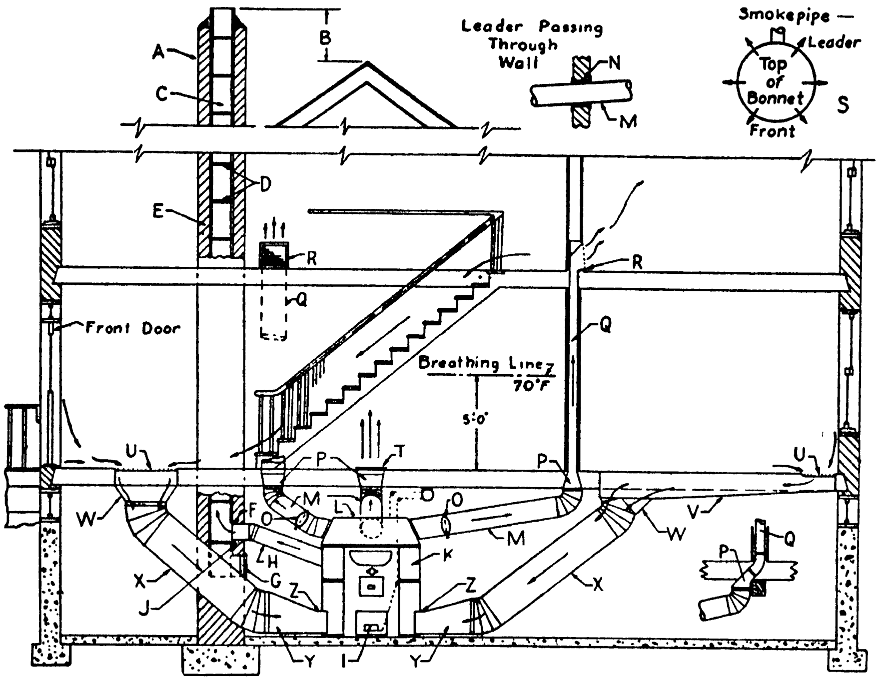


FIG. 1. A SECTIONAL VIEW OF A TYPICAL PLANT SHOWING GOOD INSTALLATION PRACTICE[a]

A. House chimney, no bends nor offsets.
B. Top of chimney at least 2 ft above ridge of roof.
C. Flue lining, fireclay.,
D. All joints air tight.
E. At least 8 in. brick.
F. No other connection beside that to furnace.
G. Cleanout frame and door, airtight.
H. Smoke pipe, end flush with inner surface of flue.
I. Draft door.
J. Use flue thimble.
K. Casing body.
L. Casing hood or bonnet, top of all leader collars on same level.
M. Round leader, pitch 1 in. per foot.
N. Sleeve with dead air space or 1 in. of non-combustible insulation around leader where passing through wall.
O. Dampers in all leaders, except one.
P. Transition fittings.
Q. Rectangular wall stack.
R. Baseboard register.
S. Distributes pipes equally around bonnet.
T. Floor register.
U. Return air face.
V. Panning under joist.
W. Transition collar.
X. Round return pipe.
Y. Transition shoe.
Z. Top of shoe at casing not above grate level.

[a] From *N.W.A.H.&A.C.A.* Manual 5: Gravity Code and Manual, Third Edition, 1947.

important, although little is gained by providing wall stacks with areas in excess of 75 percent of their connected leader pipe area. In most cases a $3\frac{1}{4}$ in. × 12 in. stack is the largest which can be installed in normal wall construction. Hence, any room having a heat loss much in excess of 9000 Btu per hr, will require two or more stacks, or one oversized stack built into a 6 in. studding space, providing the design register temperature is to be retained at the average value of 175 F, which has been recommended.

Registers used for discharging warm air into rooms should have a net area not less than the area of the leader pipe to which the register is attached. First story registers should be connected through boot and register box extensions having areas at least equal to leader areas. Upper story registers should be of the same width as the wall stack, and should be placed either in the baseboard or sidewall, preferably without offsets. First story registers may be of the baseboard or floor type, with the former location preferred. Floor registers are easy to install, but they may interfere with placement of carpets and may be dirt catchers.

RETURN AIR GRILLES, DUCTS, AND CONNECTIONS

The placement and number of return grilles will depend upon the size, details, and exposure of the house. Small compactly built houses may be adequately served by a single return grille effectively placed in the central hall. It is usually desirable to have two or more returns, provided that in two-story residences one return is placed to effectively receive the return air at the foot of the stairs. A return air connection must be carried to any room whose floor level is below that of adjacent rooms.

The return air grilles should have free areas at least equal to the ducts to which they connect, and should be installed in the floor, or in the baseboard with the top edge of the grille not more than about 14 in. above the floor line. Frictional resistance in the return air system is as detrimental as is resistance in the warm-air system, so that care should be exercised in locating return air grilles which require long return ducts.

Where a divided system of two or more returns is used, the grilles must be placed to serve the maximum area of cold wall or windows. Thus, in rooms having only small windows the grilles can be brought as close to the furnace as possible, but if the room has large window exposure the grille should be located near the exposure.

The frictional resistance of the long ducts used in parallel with short return ducts must be reduced to compensate for the length. This is accomplished by using design data given in the following section Outline of Design Procedure.

Return ducts from upstairs rooms may be necessary in spaces which are closed off from the rest of the house, or which have much outdoor exposure. Return grilles on different floor levels should not be connected to the same vertical return duct.

The ducts through which air is returned to the furnace should be designed to minimize resistance to air flow. They should be of ample area, with sizes selected according to capacity and construction, as specified in section Outline of Design Procedure, and should be streamlined. Horizontal ducts should pitch at least ½ in. per foot downward toward the furnace, avoiding fittings which would require lifting of the return air after the duct has passed under some obstacle.

Ducts returning air to the furnace should avoid heat sources which tend to reheat the return air. If the duct must be run over the top of the furnace, or above the vent pipe from the furnace, insulation should be interposed between the heat source and the duct.

The top of the return shoe should enter the casing below the level of the grate in the case of a coal furnace, and not more than 14 in. above the floor in the case of oil or gas furnaces. It may be wide to retain proper area.

OUTLINE OF DESIGN PROCEDURE

The data underlying the design procedure are given in detail in a circular[2] issued by the University of Illinois. In this procedure the design of the warm-air duct system is considered as an entire unit, so that for a given heat loss the sizes of leaders, stacks, boots, stackheads, and registers are all correlated. Similarly in the case of return ducts, the selection specifies a complete unit consisting of return grille, return duct, and shoe connection.

Recommended Standard Sizes

For the purpose of simplification and standardization, selected combinations of commercial sizes of warm air pipes, return air pipes, ducts, grilles, fittings, and registers are designated as Combination Numbers. The numbers assigned and the combinations selected as standard are listed in the following Tables 1 to 4 inclusive.[3]

TABLE 1. FIRST STORY WARM-AIR DUCTS[a]

Combination No.	Leader Pipe Diameter, In.	Register Size, In.		
		Floor	Baseboard	
			Size	Extension
1	8	8 x 10	10 x 8	2¼
2	9	9 x 12	12 x 8	2¼
3	10	10 x 12	12 x 9	3¼
4	12	12 x 14	13 x 11	5¼
5	14	14 x 16	....	...

[a] When the calculations indicate a requirement for a given room greater than Combination No. 4, two or more smaller units totalling the required capacity are recommended.

TABLE 2. SECOND STORY WARM-AIR DUCTS—SINGLE WALL STACKS AND FITTINGS

Combination No.	Leader Pipe Diameter, In.	Stack[b] Size In.	Register Size, In.			
			Floor	Baseboard		Sidewall
				Size	Extension	
11	8	10 x 3¼	8 x 10	10 x 8	2¼	10 x 8
12	9	12 x 3¼	9 x 12	12 x 8	2¼	12 x 8
14	10	14 x 3¼	10 x 12	12 x 8	2¼	12 x 8
15	12	12 x 5¼	...	12 x 9	3¼	..
16	12	14 x 5¼	...	13 x 11	5¼	..

[b] Recommended stack sizes. Tables may also be applied to 3 in. and 3 1/2 in. stack depths.

TABLE 3. SECOND STORY WARM-AIR DUCTS—DOUBLE WALL STACKS AND FITTINGS

Combination No.	Leader Pipe Diameter, In.	Stack Size, In.		Register Size, In.			
		Internal	External	Floor	Baseboard		Sidewall
					Size	Extension	
21	8	2½[c] x 10	3⅛[c] x 10⅝	8 x 10	10 x 8	2¼	10 x 8
22	8	3 x 10	3⅝ x 10⅝	8 x 10	10 x 8	2¼	10 x 8
23	9	2½[c] x 12	3⅛[c] x 12⅝	9 x 12	12 x 8	2¼	12 x 8
24	9	3 x 12	3⅝ x 12⅝	9 x 12	12 x 8	2¼	12 x 8

[c] Commercial sizes vary 1/8 in. from values shown.

TABLE 4. RETURN AIR DUCTS

Combination No.	Duct Dia. In.	Area at Shoe Connection, Sq In.	Metal Grille Sizes — Choose One: A	B	C	When Joist Lining is Used[d] — No. of Joists Lined	Minimum[e] Depth, In.	When Duct is Used — Choose One	
31	10			8 x 14	10 x 12	1	7	14 x 6	12 x 8
32	12		6 x 30	8 x 24	12 x 14	1	9	22 x 6	16 x 8
33	14	170	8 x 30	10 x 24	14 x 16	1	12	28 x 6	22 x 8
34	16	220	10 x 80	12 x 24		2	8	28 x 8	22 x 10
35	18	280	12 x 30	14 x 24		2	10	36 x 8	28 x 10
36	20	340	14 x 30	18 x 24		2	12.5	36 x 10	30 x 12
37	22	420	18 x 30			2	15.0	42 x 10	36 x 12
38	24	500	20 x 30			2	18.0	42 x 12	36 x 14

[d] Based on 14 in. space between joists.

[e] Use full depth of joist except when joist depth is less than minimum depth required, *when pan must be used.*

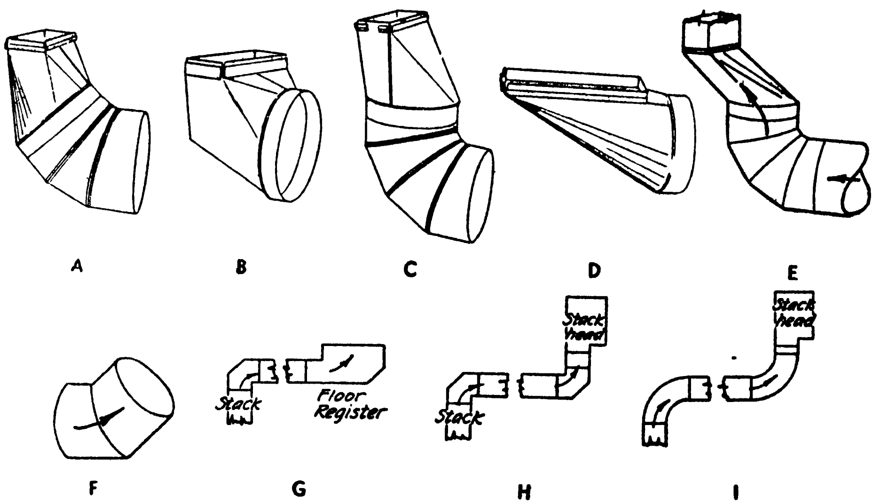


FIG. 2. TYPICAL WARM AIR BOOTS

TABLE 5. RESISTANCES OF WARM AIR BOOT COMBINATIONS EXPRESSED IN ELBOW EQUIVALENTS

Warm Air Boot	Name of Combination	Equivalent No. of 90-Deg Elbows
A	45-Deg Angle Boot and 45-Deg Elbow	1
B	90-Deg Angle Boot	1
C	Universal Boot and 90-Deg Elbow	1
D	End Boot	2
E	Offset Boot	2½
F	45-Deg Angle	½
G	Floor Register—Second Story	3
H	Offset	3
I	Offset	2

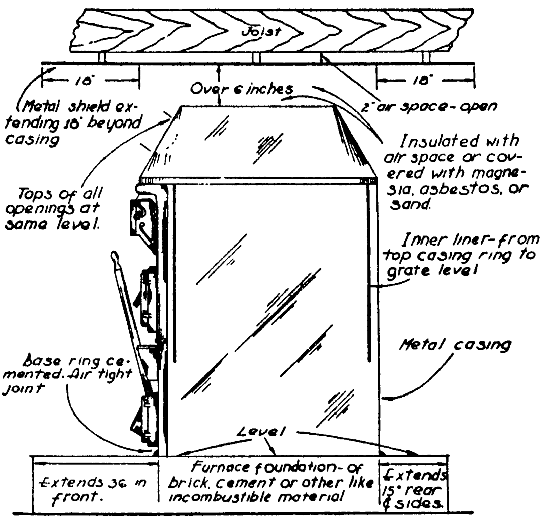


FIG. 3. DETAILS OF FURNACE BONNET, CASING, AND FOUNDATION (FROM N.W.A.H. & A.C.A. MANUAL 5, THIRD EDITION)

The selected types of boots are shown in Fig. 2, and their resistances expressed in equivalent elbows are shown in Table 5. It is essential that free areas be maintained throughout fittings.

Figs. 3 and 4 show recommended practice as given by the *N.W.A.H. & A.C.A.* For construction, design features, and ratings of gravity furnaces see Chapter 15.

Carrying Capacity

The Btu carrying capacities of the selected warm air and return air combinations are shown in Tables 6, 7 and 8.

The selected types of return air ducts and fittings are shown in Fig. 5.

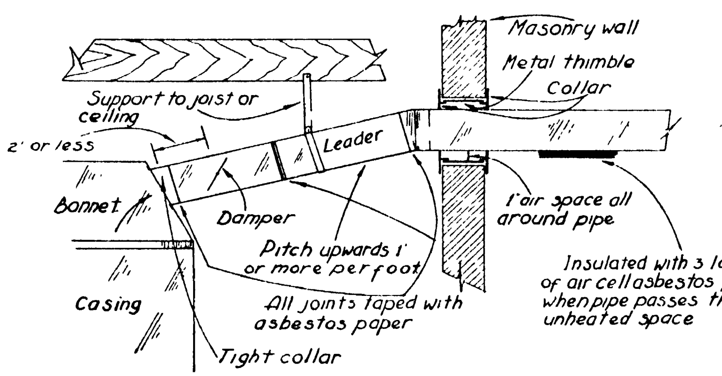


FIG. 4. DETAILS OF BONNET AND LEADER OF GRAVITY WARM-AIR FURNACE (FROM N.W.A.H. & A.C.A. MANUAL 5, THIRD EDITION)

TABLE 6. WARM AIR CARRYING CAPACITY, BTU DELIVERED, FIRST STORY REGISTERS[a]
Length of Leader Pipe—in Feet

Combination No.	No. of Elbows	4 Ft	6 Ft	8 Ft	10 Ft	12 Ft	14 Ft	16 Ft	18 Ft	20 Ft	22 Ft	24 Ft
1		6,020	5,850	5,680	5,510	5,340	5,170	5,000	4,830	4,660	4,490	4,320
2		7,620	7,400	7,180	6,970	6,760	6,540	6,320	6,110	5,890	5,680	5,460
3	1	9,400	9,140	8,870	8,600	8,340	8,070	7,810	7,540	7,270	7,010	6,740
4		13,350	12,970	12,590	12,210	11,830	11,450	11,080	10,700	10,320	9,950	9,560
5		17,520	17,020	16,530	16,040	15,550	15,050	14,550	14,050	13,560	13,060	12,560
1		5,850	5,660	5,490	5,330	5,160	5,000	4,840	4,670	4,510	4,340	4,180
2		7,360	7,150	6,940	6,730	6,520	6,320	6,110	5,910	5,700	5,500	5,290
3	2	9,090	8,840	8,580	8,320	8,060	7,800	7,550	7,290	7,040	6,780	6,520
4		12,910	12,540	12,170	11,800	11,430	11,060	10,690	10,320	9,950	9,580	9,210
5		16,940	16,450	15,990	15,500	15,040	14,550	14,080	13,600	13,120	12,650	12,150
1		5,620	5,460	5,310	5,150	4,990	4,830	4,670	4,510	4,350	4,190	4,030
2		7,120	6,910	6,710	6,510	6,310	6,110	5,900	5,700	5,500	5,300	5,100
3	3	8,780	8,530	8,280	8,030	7,780	7,530	7,290	7,040	6,800	6,550	6,300
4		12,450	12,100	11,750	11,400	11,050	10,700	10,350	10,000	9,650	9,300	8,950
5		16,360	15,900	15,440	14,970	14,510	14,050	13,600	13,130	12,660	12,200	11,750
1		5,420	5,260	5,110	4,960	4,800	4,650	4,500	4,350	4,190	4,040	3,890
2		6,860	6,660	6,460	6,270	6,080	5,890	5,690	5,500	5,300	5,110	4,910
3	4	8,460	8,200	7,980	7,740	7,500	7,260	7,020	6,780	6,550	6,310	6,070
4		12,010	11,670	11,330	10,990	10,650	10,310	9,970	9,630	9,290	8,950	8,610
5		15,770	15,320	14,880	14,420	13,990	13,540	13,100	12,650	12,200	11,750	11,310
1		5,240	5,090	4,940	4,790	4,640	4,500	4,350	4,200	4,050	3,910	3,760
2		6,630	6,440	6,250	6,060	5,880	5,690	5,500	5,320	5,130	4,940	4,750
3	5	8,180	7,950	7,720	7,490	7,260	7,030	6,800	6,560	6,330	6,100	5,860
4		11,610	11,290	10,950	10,620	10,300	9,970	9,640	9,320	8,990	8,660	8,320
5		15,250	14,800	14,380	13,950	13,520	13,090	12,650	12,230	11,800	11,370	10,940

[a] Additional values for 6 and 7 elbows are given in Manual 5, *N.W.A.H. & A.C.A.*

TABLE 7. WARM AIR CARRYING CAPACITY, BTU DELIVERED SECOND STORY REGISTERS[a]
Length of Leader Pipe—in Feet

Combination No.[b,c]	No. of Elbows	4 Ft	6 Ft	8 Ft	10 Ft	12 Ft	14 Ft	16 Ft	18 Ft	20 Ft	22 Ft	24 Ft
11-22		8,370	8,140	7,900	7,670	7,430	7,190	6,950	6,710	6,470	6,240	6,000
12-24		10,040	9,760	9,470	9,190	8,900	8,620	8,330	8,050	7,770	7,480	7,200
14	1	11,710	11,380	11,050	10,720	10,390	10,060	9,720	9,390	9,060	8,730	8,400
15		16,200	15,750	15,300	14,840	14,380	13,920	13,460	13,000	12,550	12,100	11,640
16		18,920	18,390	17,850	17,310	16,780	16,240	15,710	15,180	14,640	14,100	13,570
11-22		7,940	7,720	7,500	7,280	7,050	6,830	6,600	6,370	6,150	5,930	5,700
12-24		9,540	9,270	9,000	8,730	8,460	8,190	7,920	7,650	7,380	7,110	6,840
14	2	11,120	10,810	10,500	10,180	9,870	9,550	9,230	8,920	8,610	8,290	7,980
15		15,400	14,970	14,530	14,100	13,670	13,230	12,800	12,360	11,930	11,500	11,070
16		17,980	17,470	16,960	16,450	15,950	15,430	14,830	14,420	13,910	13,400	12,890
11-22		7,530	7,320	7,110	6,900	6,680	6,470	6,250	6,040	5,830	5,620	5,400
12-24		9,030	8,780	8,520	8,270	8,010	7,750	7,500	7,240	6,990	6,730	6,470
14	3	10,530	10,240	9,940	9,650	9,350	9,050	8,750	8,450	8,160	7,860	7,560
15		14,580	14,180	13,780	13,370	12,950	12,530	12,120	11,710	11,300	10,890	10,480
16		17,040	16,550	16,070	15,580	15,110	14,620	14,140	13,660	13,180	12,700	12,210
11-22		7,120	6,920	6,720	6,520	6,310	6,110	5,900	5,700	5,500	5,300	5,100
12-24		8,530	8,290	8,050	7,810	7,570	7,330	7,080	6,840	6,600	6,360	6,120
14	4	9,950	9,670	9,390	9,110	8,830	8,550	8,260	7,980	7,700	7,420	7,140
15		13,780	13,390	13,000	12,610	12,220	11,830	11,440	11,050	10,670	10,280	9,890
16		16,080	15,620	15,170	14,710	14,260	13,810	13,350	12,900	12,440	11,980	11,530
11-22		6,700	6,510	6,320	6,130	5,940	5,750	5,560	5,370	5,180	4,990	4,800
12-24		8,040	7,810	7,580	7,350	7,130	6,900	6,670	6,440	6,220	5,990	5,760
14	5	9,370	9,110	8,850	8,580	8,310	8,050	7,780	7,510	7,250	6,980	6,720
15		12,970	12,600	12,240	11,870	11,500	11,140	10,770	10,400	10,040	9,680	9,310
16		15,140	14,710	14,280	13,850	13,420	13,000	12,570	12,140	11,710	11,280	10,850

[a] When floor registers are used, see Fig. 2.
[b] No. 21 for Btu values multiply 11-22 values by 0.83.
[c] No. 23 for Btu values multiply 12-24 values by 0.83.

TABLE 8. RETURN AIR—CARRYING CAPACITY—BTU SERVICED

Return Air Combination No.	Duct Dia In.	Type A Btu per Hr	Types B and C Btu per Hr	Type D Btu per Hr	Type E Btu per Hr	Type F Btu per Hr	Return Air Combination No
31	10	11,300	9,500	7,800	5,000	7,800	31
32	12	16,300	13,700	11,300	7,200	11,300	32
33	14	22,200	18,700	15,300	9,800	15,300	33
34	16	29,000	24,400	20,000	12,800	20,000	34
35	18	36,700	30,800	25,300	16,200	25,300	35
36	20	45,300	38,000	31,300	20,000	31,300	36
37	22	54,800	46,000	37,800	24,100	37,800	37
38	24	65,200	54,800	45,000	28,700	45,000	38

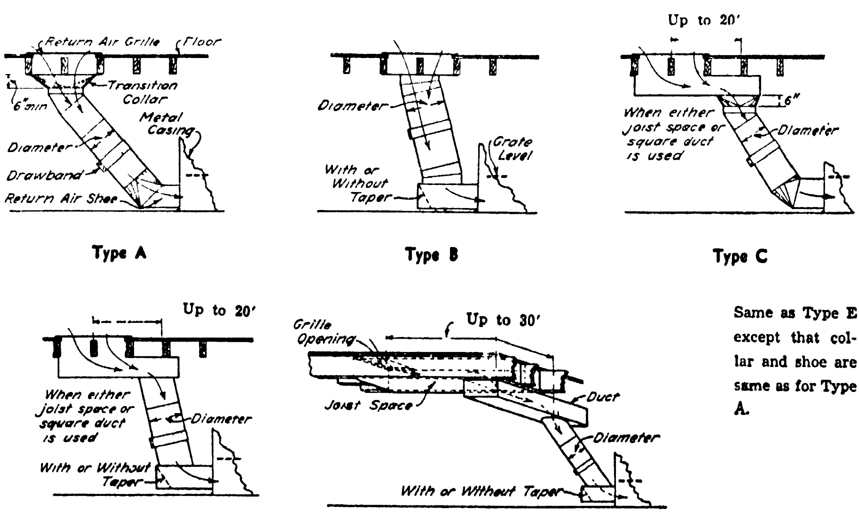
Design Procedure

The steps to be taken in designing a gravity warm-air duct system are:

1. Calculate the heat loss from each room as explained in Chapters 9, 10 and 11.

2. Prepare a layout showing (*a*) furnace, (*b*) chimney connection, (*c*) warm air registers (whether floor, baseboard or wall), (*d*) return air grilles.

3. Indicate on each warm-air leader (using symbols shown in Fig. 6): (*a*) whether the room to be heated is on first or second story; (*b*) the approximate length of leader pipe in the basement; (*c*) the number of right angle elbows and equivalent elbows (Table 5) required, including the elbow at the boot connection (see Fig. 2); (*d*) whether the register is to be located in the floor, in the baseboard, or in the wall.



Note: For Types C, D, E, and F return-air duct systems, reduce the carrying capacities shown in Table 8 by 1 percent for each 4 ft additional length in the horizontal run.

FIG. 5. TYPICAL ARRANGEMENTS OF RETURN-AIR DUCT SYSTEMS

4. Show the number and proposed locations of return-air grilles and the type of return-air system (see Fig. 5).

5. From Table 6, for first story, or from Table 7 for second story, select the combination number for the warm air system which will supply the heat required to each room, with the number of elbows and length of leader pipe previously determined. Then, using the combination number as found, read directly in Tables 1, 2, or 3 the leader, stack, and register sizes required.

6. From Table 8 select the combination number for the return-air system to correspond with the Btu serviced and the type of return-air system. Then from Table 4 select the duct and grille sizes, etc., corresponding to the same combination number.

7. Select a furnace having a register delivery, in Btu per hour, equal to the total heat loss from the structure.

Standard work sheets to facilitate design according to the above recommended procedure, are available from *N.W.A.H. & A.C.A.*

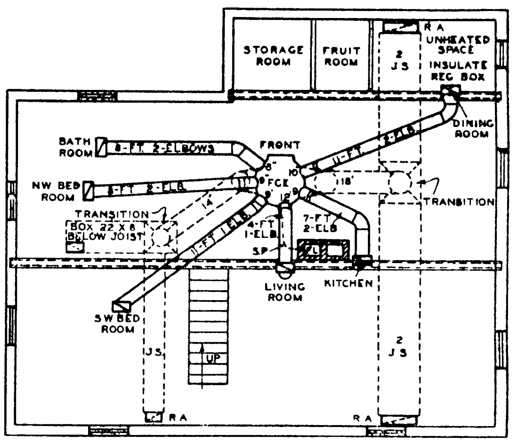


FIG. 6 TYPICAL BASEMENT LINE DRAWING

Design Examples

Examples 1 and 2 will illustrate the use of the tables in selecting warm-air and return-air system sizes.

Example 1: For a room which has a heat loss of 22,500 Btu per hr, select the size of first story warm-air system. There are three elbows, and the leader is approximately 10 ft long.

Solution: Since 22,500 Btu is beyond the capacities shown in Table 6, it is necessary to select two units of 11,250 each. From Table 6 in 10 ft leader column, and in section for three elbows, find 11,400 as nearest capacity which corresponds to Combination Number 4 in first column. Refer to Combination Number 4 in Table 1 and find that the leader should be 12 in. in diameter, and should be used with a 12 × 14 in. floor register or a 13 × 11 in. baseboard register with a 5½ in. extension.

Example 2: What is the size of a return-air system of Type D which is to service 35,000 Btu per hr?

Solution: From Table 8 find Combination Number 37 which will service 37,800 Btu per hr. Refer to Table 4 to find that Combination Number 37 will require a 22-in. diameter duct, a shoe area of 420 sq in., a metal grille 18 × 30 in., a duct 42 × 10 in. or 36 × 12 in. If joist lining is used the minimum depth should be 15 in. for two 2-joist spaces 14 in. wide, or 10 in. for three joist spaces.

REFERENCES

[1] The engineering data were obtained from *University of Illinois, Engineering Experiment Station Bulletins* Nos. 141, 188, 189 and 246; Warm Air Furnaces and Heating Systems, by A. C. Willard, A. P. Kratz, V. S. Day, and S. Konzo. See also Manual 5: Gravity Code and Manual for the Design and Installation of Gravity Warm Air Heating Systems, Third Edition, 1947, published by the *National Warm Air Heating and Air Conditioning Association.*

[2] Simplified Procedure for Selecting Capacities of Duct Systems for Gravity Warm Air Heating Plants, by A. P. Kratz and S. Konzo (*University of Illinois, Engineering Experiment Station Circular* 45, Dec., 1942).

[3] Gravity Code and Manual for the Design and Installation of Gravity Warm Air Heating Systems, Manual 5, Third Edition, 1947, *National Warm Air Heating and Air Conditioning Association.*

CHAPTER 19

FORCED WARM AIR SYSTEMS

Air Distribution, Standard Combinations of Parts, Simplified Method of Design, Design Procedure for Large Systems, Automatic Controls, Adjustment of System for Continuous Air Circulation, Ceiling Panel Systems, Perimeter Systems, Cooling Methods, Design of Cooling System

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IN forced warm air or fan furnace heating systems, the air circulations is effected by motor-driven centrifugal fans, commonly referred to as blowers. The advantages of forced air systems, as compared with gravity systems, are:

1. The furnace need not be centrally located but may be placed in any part of the basement.

2. Basement distribution ducts can be made smaller and can be so installed as to give full head room in all parts of the average basement, or be completely concealed from view where desired.

3. Circulation of air is positive, and in a properly designed system, can be balanced in such a way as to give a greater uniformity of temperature distribution.

4. Humidity control is more readily attained.

5. The air may be cleaned by sprays or filters, or both.

6. The fan and duct equipment may be utilized for a complete cooling and dehumidifying system for summer, using either ice, mechanical refrigeration, or low temperature water for cooling and dehumidifying, or adsorbers for dehumidifying.

7. The use of the fan increases the volume of air which can be handled, thereby increasing the rate of heat extraction from a given amount of heating surface and insuring sufficient air volume to obtain proper distribution in a large room.

8. Ventilation air may be positively introduced and heated.

The construction features of forced warm air furnace units and the function and selection of the various parts of a system are discussed in Chapter 15 and other publications.[1]

AIR DISTRIBUTION

The conditions of comfort obtained in a room are influenced greatly by the type of register used, and the locations of the supply registers and return grilles. In general it has been found that changes in the type, air velocity, and location of the supply register affect the room conditions much more than the changes in the location of the return grilles. One method is to locate the supply register near the floor, or high in the side wall, so that the warm air from the register *blankets* a cold wall, and mixes with the cold air descending from the exposed walls and glass. Another method is to locate the supply openings near the floor, or high in the side wall, on the inside wall, and the return openings near the greatest outside exposure. In any case, the warm air registers should be located so that the air stream never discharges directly against people at rest. Tests[2] in Warm Air Research Residence No. 1 at the *University of Illinois*, have indicated that continuous blower operation gave better results than intermittent operation.

Register and Grille Openings

Tests also conducted in Warm Air Research Residence No. 1 have indicated that comparable results are obtainable with either high side wall or baseboard registers, if proper registers and air velocities are selected. Baseboard registers should be of a deflecting-diffuser type which throw the air downward toward the floor and diffuse it at the same time. For baseboard registers, air temperatures under 125 F, and air velocities over 500 fpm, should be avoided as they may cause drafts.

High side wall registers must be of such type that the air is delivered horizontally or in a slightly downward direction, and must be so located as to avoid impingement of air on ceiling or wall. Directional flow diffusing

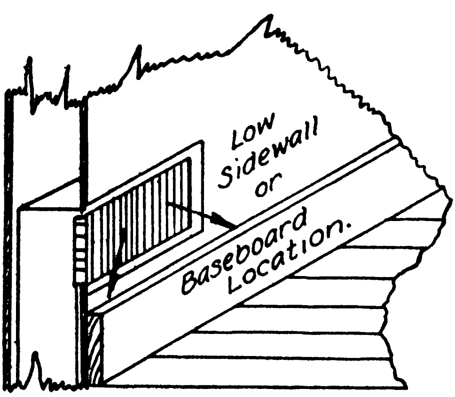


FIG. 1. RECOMMENDED TYPE OF BASEBOARD AND LOW SIDEWALL REGISTERS[a]

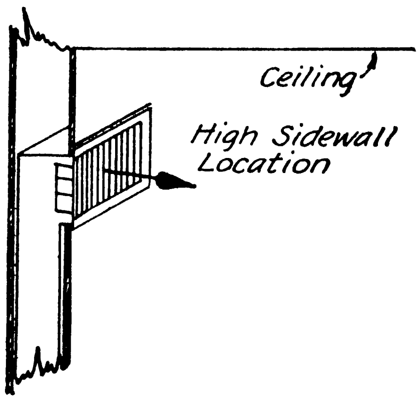


FIG. 2. RECOMMENDED TYPE OF HIGH SIDEWALL REGISTERS[b]

[a] Vertical bars with adjustable deflection, or fixed vertical bars with deflections to right and left not exceeding about 22 deg. For low sidewall location, the deflection for horizontal, multiple valve registers should not exceed 22 deg. For baseboard locations, the deflection for horizontal, multiple valve registers should not exceed about 10 deg.

[b] Horizontal valves, in back or front, to give downward deflections not to exceed from 15 to 22 deg.

type registers should be used to insure best results. Register air velocities should be such that the air velocity will be about 50 fpm three quarters of the distance from the register to the opposite wall.

Velocities through registers may be reduced by the use of registers larger than the connecting ducts. Diffusers should be used to spread the air uniformly over the register face. Basic rules for the location and selection of registers, together with explanations of factors affecting operation, are given[3] in Section B of Manual 7 of the *National Warm Air Heating and Air Conditioning Association*.

Registers should be well proportioned and decorated to harmonize with the trim. Air supply registers should be equipped with dampers, and all registers should be sealed against leakage around edges. The register types shown in Figs. 1 and 2 have been recommended as standard by the *National Warm Air Heating and Air Conditioning Association.*

Return air grilles may be located in hallways, near entrance doors, under windows, in exposed corners, or inside walls, depending on location of supply registers. Baseboard returns are preferable to floor grilles.

Table 1. Warm Air Duct System Combinations of Parts Selected as Standard

Combination No.	Stack Size In.	Branch Pipe Size, In.		Register Size, In. (See Figs. 1 and 2)		Required Increase in Width of Trunk Duct, In.
		Round	Rectangular	Base-board High or Low Sidewall	Floor Registers[a]	
1	2	3	4	5	6	7
41	10 x 3¼	6	4 x 8	10 x 6	8 x 10[a]	1
42	10 x 3¼	6	4 x 8	10 x 6	8 x 10[a]	2
43	12 x 3¼	7	5 x 8	12 x 6	9 x 12[a]	3
44	14 x 3¼	8	6 x 8	14 x 6	9 x 12[a] or longer	4
45	10 x 3¼ (2-Stacks)	9	8 x 8	(2) 10 x 6 or (1) 24 x 6	10 x 12[a]	5
46	12 x 3¼ (2-Stacks)	10	10 x 8	(2) 12 x 6 or (1) 30 x 6	12 x 14[a]	7

[a] Use these items only when the building construction or capacity requirements necessitate the use of floor registers. The sizes listed for floor registers correspond to the standard sizes for gravity warm air furnace systems, except for the sizes of the floor box collars. The use of standard blind boxes is suggested. A 12 x 5¼ in. stack may be used on Combination 45, and a 14 x 5¼ in. stack on Combination 46 with floor registers.

Dampers

Suitable dampers for air direction or volume control are essential to any duct system. Special care must be used, in the design of any system, to avoid turbulence and to minimize resistance. Sharp elbows, angles, and offsets should be avoided. Three types of dampers are commonly used. *Volume dampers* are used to completely cut off or reduce the flow through ducts. *Splitter dampers* are used where a branch is taken off from a main trunk. *Squeeze dampers* are used for adjusting the volume of air flow and resistance through a given duct. It is essential that a damper with positive locking device be provided for each main or duct branch. Labels placed on ducts should indicate the room being served. Damper positions should be marked for summer and winter operation, and to avoid tampering.

Ducts

The ducts may be either round or rectangular in cross section. The radii of elbows should preferably be not less than one and one-half times the pipe diameter for round pipes, or the equivalent round pipe size in the case of rectangular ducts. Warm air ducts passing through cold spaces, or where located in exposed walls, should have 1 to 2 in. of insulation.

Special attention should be given to the problem of noise elimination. The metal duct connection to and from the furnace casing and fan housing should be broken by strips of canvas. Motors and mountings must be carefully selected for quiet operation. Electrical conduit and water piping must not be fastened to, nor make contact with the fan housing. Installation of a fan directly under a return air grille is usually avoided.

Table 2. Return Air Duct System Combination of Parts Selected as Standard

Combination No.	Return-Air Intake Size, In.		Riser Size, In. Where Stack is Used in Stud Space	Branch Pipe Size, In.		When Joist Lining is Used[b] Number of Joist Spaces Lined and Minimum Depth of Space Required	Required Increase in Width of Trunk Duct (for 8 In. Depth of Duct), In.
	Baseboard	Floor[a]		Round	Rectangular		
1	2	3	4	5	6	7	8
51	10 x 6	6 x 10 or 4 x 14	10 x 3¼[c]	6	4 x 8	a space of 3 in. depth	1
52	10 x 6	6 x 10 or 4 x 14	10 x 3¼[c]	6	4 x 8	1 space of 3 in. depth	2
53	12 x 6	6 x 12 or 6 x 14	12 x 3¼[d]	7	5 x 8	1 space of 4 in. depth	3
54	14 x 6	6 x 14	14 x 3¼[d]	8	6 x 8	1 space of 5 in. depth	4
55	24 x 6 or 30 x 6	6 x 30	Two stacks each 10 x 3¼[c]	9	8 x 8	1 space of 6 in. depth or 2 spaces of 3 in. depth	5
56	30 x 6	6 x 30	Two stacks 12 x 3¼[d]	10	10 x 8	1 space of 7 in. depth or 2 spaces of 4 in. depth	7
57	—	8 x 30	—	12	15 x 8	1 space of 9 in. depth or 2 spaces of 5 in. depth	12

[a] Use these items only when building construction, or capacities, require the use of floor intakes. The sizes listed correspond to standard sizes for gravity installations, except floor box collars. The use of standard blind boxes is suggested.

[b] Based on 14 in. space between joists. Use full depth of joist, except when joist depth is less than minimum depth required, in which case a *drop pan must be used.* This may occur when two or more return ducts are connected to the same joist space.

[c] If it is desired to use 14 in. x 3⅝ in. stud space, it makes no difference whether this space has protruding keys or not.

[d] If it is desired to use 14 in. x 3⅝ in. stud space, the plaster base must be smooth, without any protruding plaster keys to interfere with the flow of air.

STANDARD COMBINATIONS OF PARTS

The combinations of parts selected as standard by the *National Warm Air Heating and Air Conditioning Association* are shown in Tables 1 and 2. A method for selecting these combinations is indicated in the following section *Simplified Method of Design.*

SIMPLIFIED METHOD OF DESIGN

A simplified method for selecting the combinations of branches, boots, stacks, and registers, is given[3] in Manual 7 of the *National Warm Air Heating and Air Conditioning Association.* In this method, the sizes of the branch ducts are obtained from two tables giving their Btu capacities. The proper combination of parts for each branch can be determined if the following information is available.

a. Location of room, that is, whether on first or second story.

b. Actual length of duct from bonnet to boot, in feet.

c. Btu loss from room to be heated.

d. Equivalent lengths in feet of all fittings and of the register. Fig. 3 shows the values of equivalent lengths of fittings commonly used for domestic systems.

This simplified method is applicable to structures having heat losses not in excess of approximately 120,000 Btu per hour. The capacities shown in Tables 3 and 4 are based upon the most reliable data pertaining to friction losses and temperature drops in ducts. They are also based upon a 100 deg temperature rise of the air, and a static pressure available for overcoming friction losses in the external duct system alone of 0.20 in. water gage. The use of this method assumes that the fan in the fan-furnace assembly will be capable not only of overcoming the resistance of the external duct system alone, but also the resistances imposed by the blower inlet, the filter, and the furnace casing. The combination numbers shown in the right hand column of Tables 3 and 4 correspond to those given in Tables 1 and 2. Tables 3 and 4 are also applicable for the selection of the return air branches. A depth of 8 in. has been adopted as the standard for the trunk ducts. The width of a trunk duct serving two branches is determined by adding to the width of the remote branch the value shown in column 7 of Table 1, or column 8 of Table 2.

DESIGN PROCEDURE FOR LARGE SYSTEMS[4]

For buildings having a heat loss in excess of 120,000 Btu per hour, the design procedure[5] given in Manual 9 of the *N.W.A.H. & A.C.A.*, may be used, except where ventilation air volume exceeds volume required to supply calculated heat loss. This procedure consists of:

1. Calculation of design heat losses from individual spaces in the structure. (See Chapter 11.)

2. Location of registers and return intakes on floor plan, showing types of registers, with distance from register to opposite wall and deflection of registers desired.

3. Laying out a proposed duct system for both warm air and return air sides of the system, and including details of types of fittings and the actual and equivalent lengths of each branch line from bonnet to register, without sizes. (See Fig. 3, Groups 1 through 6, for equivalent length of fittings.)

4. Determination of bonnet temperature.

If rating sheet for furnace-blower unit specifies a *fixed value of bonnet temperature*, enter table at this value. If not specified, use the following procedure: Use Table 5 for buildings having a heat loss between 120,000 and 350,000 Btu per hr or Table 6 for buildings having a heat loss greater than 350,000 Btu per hr. Select *shortest actual length* and read downward in nearest column in Tables 5 or 6 until lower heavy diagonal line is reached, but do not cross line. Run horizontally to first column of table and note *bonnet temperature*. Also select *longest actual length* and read downward in nearest column in Tables 5 or 6, until upper heavy diagonal line is just crossed. Run horizontally to left to obtain value for *bonnet temperature* in first column. Select as the design bonnet temperature any value between these two limits.

5. Determination of air volume to be delivered through each register and the respective register air temperatures.

Using Tables 5 or 6 and the design bonnet temperature selected, find the values of cfm per 1000 Btu for each duct length.

6. Selection of register sizes and pressure losses to produce necessary throw, for the air volumes handled.

Use Tables 7 or 8 to obtain required free area and pressure loss of register.

7. Design of duct system.

A. *Warm air branches*.

a. Use Table 9 to select maximum bonnet pressure *usually* required for the trunk carrying the maximum volume of air (cfm). If the maximum

Table 3. Capacity Tables for Warm Air and Return Air Branches[a, b]

FIRST STORY

For UNINSULATED Metal Ducts	Actual Length (from Bonnet to Boot) or, (from Return Plenum to Boot) in Feet							Warm Air Combination No.	Return Air Combination No.
	1 to 7 ft.	8 to 12 ft.	13 to 17 ft.	18 to 24 ft.	25 to 34 ft.	35 to 44 ft.	45 to 54 ft.		
	Col. A	Col. B	Col. C	Col. D	Col. E	Col. F	Col. G		
Section A.	7200	6700	6100	5600	4800	4100	3500	41	51
	12500	11700	10800	9900	8500	7400	6400	42	52
40 to 69	16000	15000	14000	13000	11300	9900	8700	43	53
Equivalent Ft for Fittings and Register	19100	18000	17000	16000	14200	12500	11000	44	54
	25000	23400	21600	19800	17000	14800	12800	45[c]	55
	32000	30000	28000	26000	22600	19800	17400	46[c]	56
	80000	75000	70000	65000	56500	49500	43500	—	57[c]
Section B.	5500	5100	4800	4500	3900	3400	3000	41	51
	9900	9200	8600	8100	7100	6200	5400	42	52
70 to 99	13100	12300	11600	10900	9700	8500	7500	43	53
Equivalent Feet	16300	15400	14500	13700	12200	10800	9500	44	54
	19800	18400	17200	16200	14200	12400	10800	45[c]	55
	26200	24600	23200	21800	19400	17000	15000	46[c]	56
	65500	61500	58000	54500	48500	42500	37500	—	57[c]
Section C.	4600	4300	4100	3800	3300	3000	2700	41	51
	8500	7900	7400	6900	6100	5300	4700	42	52
100 to 129	11300	10600	10000	9400	8400	7400	6500	43	53
Equivalent Feet	14300	13500	12700	11900	10500	9300	8300	44	54
	17000	15800	14800	13800	12200	10600	9400	45[c]	55
	22600	21200	20000	18800	16800	14800	13000	46[c]	56
	56200	52600	50200	47300	42000	37000	32400	—	57[c]
Section D.	4100	3800	3600	3400	2900	2600	2300	41	51
	7300	6900	6400	6000	5300	4700	4200	42	52
130 to 164	9800	9100	8600	8100	7200	6400	5600	43	53
Equivalent Feet	12300	11700	11000	10300	9100	8100	7100	44	54
	14600	13800	12800	12000	10600	9600	8400	45[c]	55
	19600	18200	17200	16200	14400	12800	11200	46[c]	56
	49200	45800	41900	40300	36000	31800	28100	—	57[c]
Section E.	3800	3500	3300	3100	2700	2400	2100	41	51
	6500	6100	5700	5400	4800	4300	3800	42	52
165 to 200	8800	8200	7700	7200	6400	5700	5000	43	53
Equivalent Feet	11000	10500	9900	9300	8300	7300	6400	44	54
	13000	12200	11400	10800	9600	8600	7600	45[c]	55
	17600	16400	15400	14400	12800	11400	10000	46[c]	56
	44100	41500	38800	36000	32000	28200	24700	—	57[c]

For INSULATED Ducts	Col. A	Col. B	Col. C	Col. D	Col. E	For Ducts that are COMPLETELY INSULATED with ½ In. Thick Insulation from Bonnet to Boot, use these Column Headings.
	1 to 9 ft.	10 to 17 ft.	18 to 24 ft.	25 to 34 ft.	35 to 54 ft.	

[a] These tables are for use in sizing both the warm air and the return air branches.

[b] Frictional resistances and temperature drops in ducts have both been accounted for in these tables.

[c] Use these items only when the building construction, or capacity requirements, necessitate the use of two adjoining stacks or floor registers.

TABLE 4. CAPACITY TABLES FOR WARM AIR AND RETURN AIR BRANCHES[a, b]

SECOND STORY

FOR UNINSULATED METAL DUCTS	ACTUAL LENGTH (FROM BONNET TO BOOT) OR, (FROM RETURN PLENUM TO BOOT) IN FEET							WARM AIR COMBINATION No.	RETURN AIR COMBINATION No.
	1 TO 7 FT.	8 TO 12 FT.	13 TO 17 FT.	18 TO 24 FT.	25 TO 34 FT.	35 TO 44 FT.	45 TO 54 FT.		
	COL. A	COL. B	COL. C	COL. D	COL. E	COL. F	COL. G		
Section A.	6300	5700	5200	4800	4100	3500	3100	41	51
	10900	10000	9200	8500	7300	6400	5600	42	52
40 to 69	14000	13000	12100	11400	10000	8800	7800	43	53
Equivalent Ft for	17000	15900	14900	13900	12400	11100	9900	44	54
Fittings	21800	20000	18400	17000	14600	12800	11200	45[c]	55
and	28000	26000	24200	22400	20000	17600	15600	46[c]	56
Register	70000	65000	60500	57000	50000	44000	39000	—	57[c]
Section B.	5000	4600	4300	4000	3400	3000	2700	41	51
	9000	8200	7600	7100	6200	5400	4700	42	52
70 to 99	11900	11000	10300	9600	8400	7500	6700	43	53
Equivalent Feet	14800	13900	13000	12200	10800	9600	8500	44	54
	18000	16400	15200	14200	12400	10800	9400	45[c]	55
	23900	22000	20600	19200	16800	15000	13400	46[c]	56
	59500	55000	51500	48000	42000	37500	33500	—	57[c]
Section C.	4200	3900	3700	3500	3000	2600	2400	41	51
	7700	7200	6700	6200	5400	4700	4100	42	52
100 to 129	10400	9700	9000	8400	7400	6500	5800	43	53
Equivalent Feet	13000	12100	11300	10600	9400	8300	7400	44	54
	15400	14400	13400	12400	10800	9400	8200	45[c]	55
	21700	19400	18000	16800	14800	13000	11600	46[c]	56
	52200	48400	45300	42300	37000	32500	29000	—	57[c]
Section D.	3800	3500	3200	3000	2700	2300	2100	41	51
	6800	6300	5800	5500	4800	4200	3700	42	52
130 to 164	9100	8400	7900	7400	6300	5700	5000	43	53
Equivalent Feet	11400	10500	9800	9200	8100	7200	6400	44	54
	13600	12600	11600	11000	9600	8400	7400	45[c]	55
	18200	16800	15800	14800	12600	11400	10800	46[c]	56
	45500	42000	39600	36800	32200	28200	25100	—	57[c]
Section E.	3500	3200	2900	2700	2500	2100	1900	41	51
	6100	5700	5300	5000	4400	3900	3400	42	52
165 to 200	8200	7600	7100	6700	5700	5100	4500	43	53
Equivalent Feet	10300	9500	8900	8300	7400	6500	5800	44	54
	12200	11400	10600	10000	8800	7800	6800	45[c]	55
	16400	15200	14200	13400	11400	10200	9000	46[c]	56
	41000	38300	35800	33400	29000	25800	22800	—	57[c]

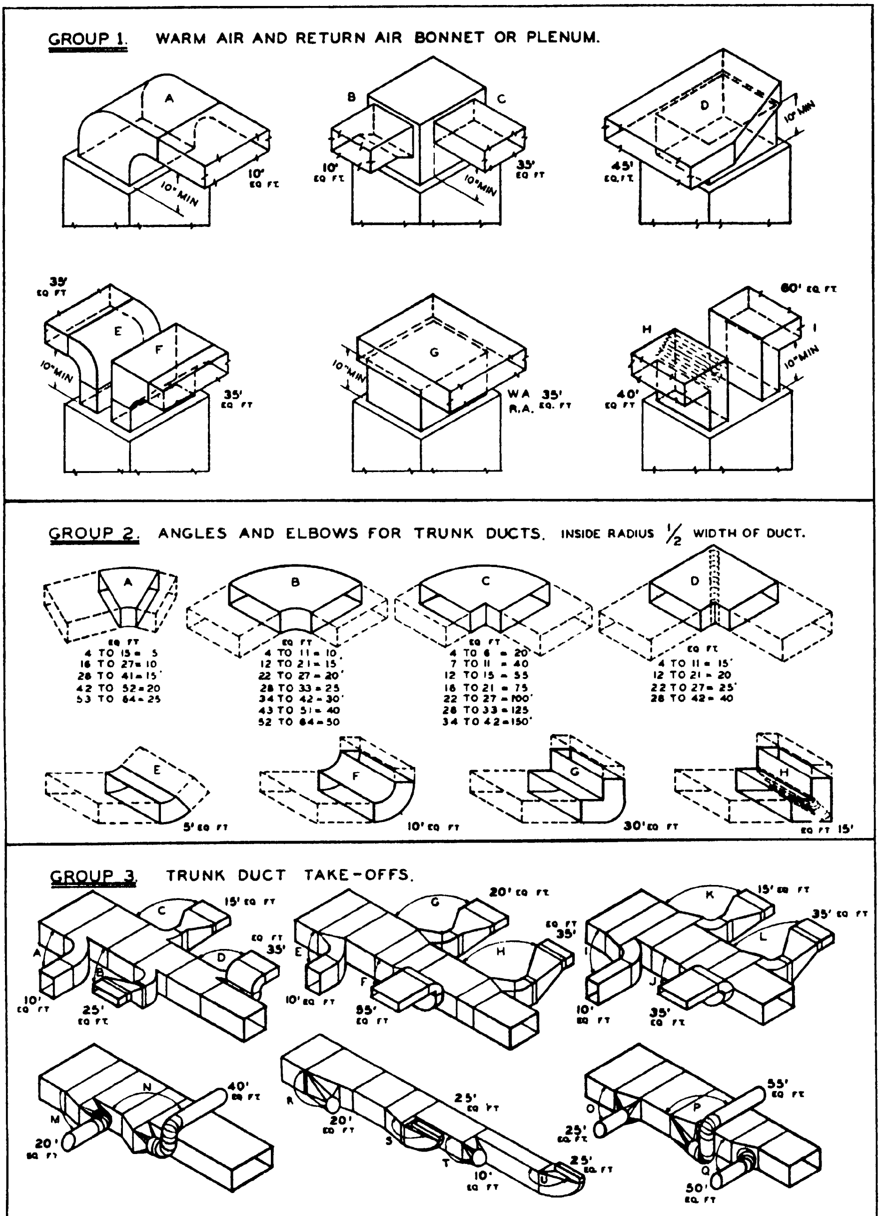
FOR INSULATED DUCTS	COL. A	COL. B	COL. C	COL. D	COL. E	COL. F	FOR DUCTS THAT ARE COMPLETELY INSULATED WITH ½ IN. THICK INSULATION FROM BONNET TO BOOT USE THESE COLUMN HEADINGS.
	1 TO 8 FT.	9 TO 14 FT.	15 TO 20 FT.	21 TO 27 FT.	28 TO 42 FT.	43 TO 54 FT.	

[a] These tables are for use in sizing both the warm air and the return air branches.

[b] Frictional resistance and temperature drops in ducts have both been accounted for in these tables.

[c] Use these items only when the building construction, or capacity requirements, necessitate the use of two adjoining stacks or floor registers.

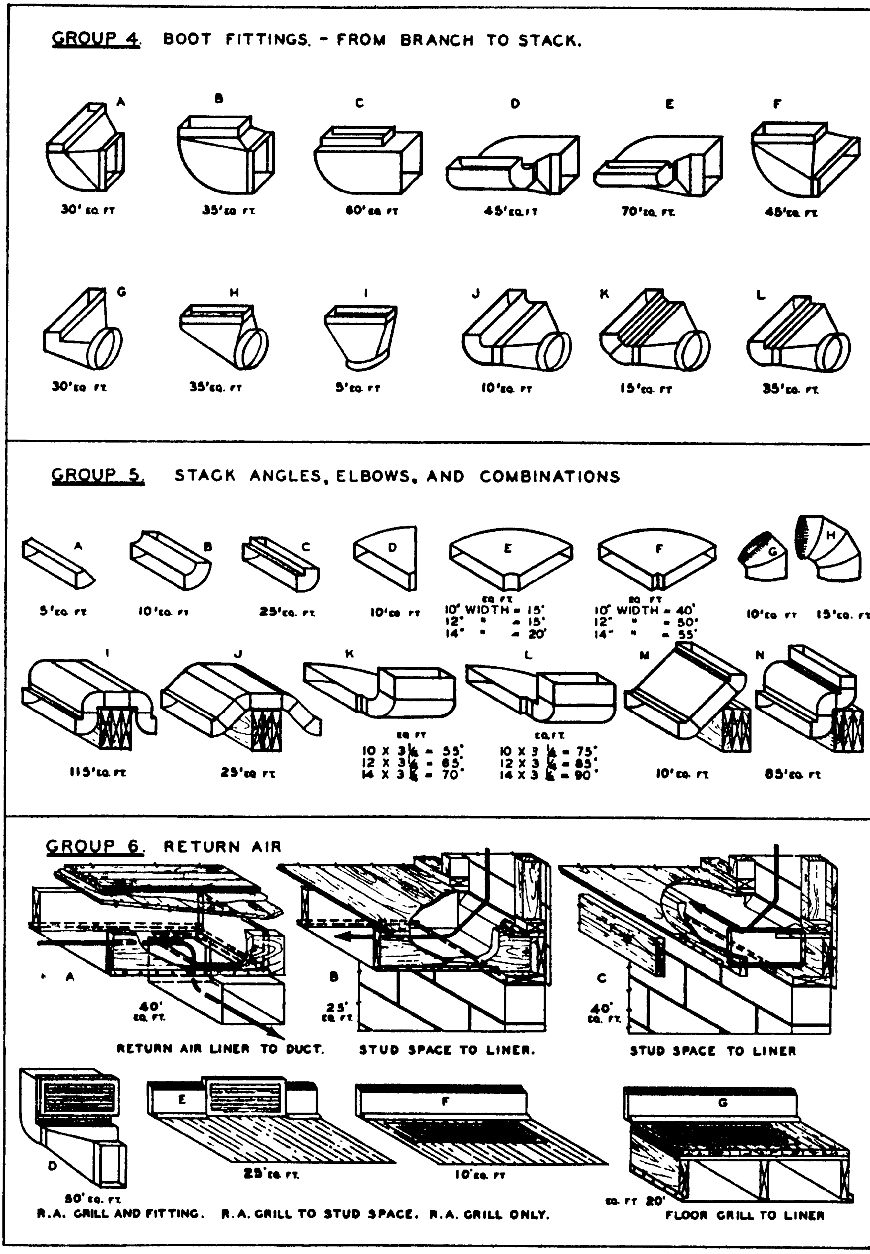
Fig. 3. Equivalent Length of Fittings



bonnet pressure is not high enough to accommodate the pressure loss through the registers, use a higher bonnet pressure. If the register presure is critically large, it may be necessary to reduce it by either using two registers in place of one, or using smaller deflection angles.

b. Obtain actual duct loss by subtracting register pressure loss from maximum bonnet pressure.

FIG. 3. EQUIVALENT LENGTH OF FITTINGS (Continued)



c. Obtain the pressure drop in each duct per 100 ft by use of Table 10.

d. Determine duct size by means of air friction chart (such as Fig. 1, Chapter 31), volume (cfm), and pressure drop per 100 ft of duct.

B. *Return air branches.*

a. Select a low value of actual duct loss obtained from step b under item A for the suction loss of return duct system.

FIG. 3. EQUIVALENT LENGTH OF FITTINGS (Concluded)

GROUP 7. REGISTERS (INCLUDING LOSSES IN STACKHEAD AND VELOCITY PRESSURE).

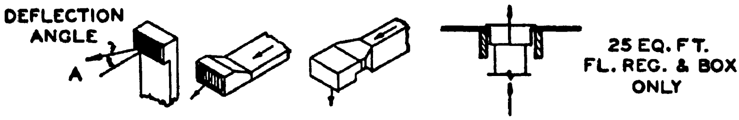


TABLE 4A. EQUIVALENT LENGTH FOR REGISTERS

Deflection Angle A	0°	15°	22°	30°	45°
Baseboard, High or Low Sidewall Registers	Eq. Ft. 35	40	45	60	115
Floor Registers with Box only	25				

For 2-way deflection registers, add the vertical and horizontal deflection angles together and multiply by 0.7. Select closest angle A in Table 4A.

b. Proceed in sizing return air branches by the same method described for the warm air branches.

C. *Trunk ducts for warm air and return air sides of system.*

a. Add air volumes of branches to be handled by each trunk duct.

b. The friction loss per 100 ft of trunk duct is determined by taking the smaller of the two values for friction loss for the two ducts meeting at the junction.

c. Determine trunk duct size by using an air friction chart, volume (cfm), and pressure drop per 100 ft of trunk duct.

8. Selection of Blower.

A. Determine total cfm air delivery (the sum of all branch cfm values).

B. Determine static pressure requirement.

a. For furnace-blower combination units it is the sum of bonnet pressure and suction pressure.

b. For blowers separately selected from the furnace it is the sum of bonnet pressure, filter loss, casing loss, losses through air washers, coils, and other devices.

9. Selection of Furnace.

A. Determine register delivery (the sum of room Btu losses).

B. Determine bonnet capacity.
Bonnet capacity = (total cfm) × (temperature rise) × 1.089.

C. Determine allowance for pick-up load. For buildings which are heated intermittently, such as churches and auditoriums, it is customary to add from 10 to 25 percent extra furnace capacity for warming of the structure.

AUTOMATIC CONTROLS

Air stratification, high bonnet temperatures, excessive flue gas temperatures, and heat overrun or lag in a properly designed system, can be largely eliminated through proper care in the planning and installation of the control system;[6] desirable controls usually employed are:

1. A *thermostat* located in a living room where maximum fluctuation in temperature can be expected, in order to secure frequent operation of fans, drafts, and burners. The thermostat location should not be on an outside wall, in a bedroom, bathroom, or sun room, or in a location where it will be affected by direct radiant heat from

Table 5. Selection of Bonnet Temperature, Register Temperature and Register Delivery (CFM)[a, b]

(For Buildings Having a Heat Loss between 120,000 and 350,000 BTU/Hr.)

Register Delivery Volume in CFM per 1000 BTU per Hr.

Temp. of Bonnet °F	Linear Distance from Bonnet to Register, in Feet																							
	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200	210	220	230	240
110 22.1[c]	108[a] 23.1[b]	107 23.6	106 24.2	104 25.4	103 26.0	102 26.6	101 27.3	100 28.0	99 28.8	98 29.6	97 30.5	96 31.5	95 32.3	94 33.3	93 34.6	92 35.9	91 37.3	90 38.6	90 38.6	89 39.9	89 39.9	88 41.3	88 41.3	87 42.7
120 18.4	118 19.0	116 19.6	114 20.3	112 21.1	111 21.6	109 22.5	108 23.1	106 24.2	105 24.8	103 26.0	102 26.6	101 27.3	100 28.0	99 28.8	98 29.6	97 30.5	96 31.5	95 32.3	94 33.3	93 34.6	92 35.9	91 37.3	90 38.6	90 38.6
130 15.9	128 16.4	126 16.9	123 17.6	121 18.1	119 18.7	117 19.3	115 20.0	113 20.7	111 21.6	110 22.1	108 23.1	107 23.6	105 24.8	104 25.4	103 26.0	102 26.6	101 27.3	99 28.8	98 29.6	97 30.5	96 31.5	95 32.3	94 33.3	93 34.6
140 14.0	137 14.6	134 15.1	131 15.7	129 16.2	126 16.9	124 17.3	122 17.9	120 18.4	118 19.0	116 19.6	114 20.3	112 21.1	110 22.1	109 22.5	107 23.6	106 24.2	105 24.8	103 26.0	102 26.6	101 27.3	100 28.0	99 28.8	98 29.6	97 30.5
150 12.6	147 12.9	143 13.5	140 14.0	137 14.6	134 15.1	131 15.7	129 16.2	127 16.6	124 17.3	122 17.9	120 18.5	118 18.4	116 19.6	114 20.3	112 21.1	110 22.1	109 22.5	107 23.6	106 24.2	105 24.8	103 26.0	102 26.6	101 27.3	100 28.0
160 11.4	156 11.8	152 12.3	149 12.7	145 13.2	142 13.7	139 14.2	136 14.8	133 15.3	130 15.9	128 16.4	126 16.9	123 17.6	121 18.1	119 18.7	117 19.3	115 20.0	113 20.7	112 21.1	110 22.1	108 23.1	107 23.6	106 24.2	104 25.4	103 26.0
170 10.5	166 10.8	161 11.3	157 11.7	153 12.2	150 12.6	146 13.1	143 13.5	140 14.0	137 14.6	134 15.1	131 15.7	128 16.4	126 16.9	124 17.3	122 17.9	120 18.4	118 19.0	116 19.6	114 20.3	112 21.1	110 22.1	109 22.5	107 23.6	106 24.2
180 9.7	175 10.0	170 10.5	166 10.8	161 11.3	157 11.7	153 12.2	150 12.6	146 13.1	143 13.5	140 14.0	137 14.6	135 14.9	131 15.7	128 16.4	126 16.9	124 17.3	122 17.9	120 18.4	118 19.0	116 19.6	115 20.0	113 20.7	111 21.6	110 22.1
190 9.1	184 9.4	179 9.8	174 10.1	169 10.6	165 10.9	160 11.4	156 11.8	153 12.2	149 12.7	145 13.2	142 13.7	139 14.2	136 14.7	133 15.3	130 15.9	128 16.4	126 16.9	124 17.3	121 18.1	119 18.7	117 19.3	115 20.0	113 20.7	112 21.1
200 8.6	193 8.9	187 9.2	182 9.6	176 9.9	171 10.4	167 10.7	163 11.1	158 11.6	154 12.1	151 12.4	147 12.9	144 13.4	140 14.0	138 14.4	135 14.9	132 15.5	129 16.2	127 16.6	125 17.2	122 17.9	120 18.4	118 19.0	116 19.6	114 20.3

[a] Register temperature, Fahrenheit. (Use upper value in each group).

[b] Register delivery volume in cubic feet per minute per 1000 Btu. (Use lower value in each group).

[c] Register delivery volumes in column 1 for zero length of duct.

TABLE 6. SELECTION OF BONNET TEMPERATURE, REGISTER TEMPERATURE AND REGISTER DELIVERY (CFM)[a, b]
(For Buildings Having a Heat Loss Greater than 350,000 BTU/Hr.)
Register Delivery Volume in CFM per 1000 BTU per Hr.

BONNET TEMP. °F.	LINEAR DISTANCE FROM BONNET TO REGISTER, IN FEET																							
	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200	210	220	230	240
140	138[a]	136	134	132	130	128	127	126	125	124	123	121	120	119	118	117	116	115	114	113	112	111	110	109
14.0[c]	14.4[b]	14.8	15.1	15.5	15.9	16.4	16.6	16.9	17.2	17.3	17.6	18.1	18.4	18.7	19.0	19.3	19.6	20.0	20.3	20.7	21.1	21.6	22.1	22.5
150	148	146	143	141	139	137	135	133	131	129	128	127	126	124	123	122	120	119	118	117	116	115	114	113
12.6	12.8	13.1	13.5	13.9	14.2	14.6	14.9	15.3	15.7	16.2	16.4	16.6	16.9	17.3	17.6	17.9	18.4	18.7	19.0	19.3	19.6	20.0	20.3	20.7
160	158	155	152	150	147	145	143	140	138	136	134	132	130	128	127	126	125	124	123	121	120	119	118	117
11.4	11.6	12.0	12.3	12.6	12.9	13.2	13.5	14.0	14.4	14.8	15.1	15.5	15.9	16.4	16.6	16.9	17.2	17.3	17.6	18.1	18.4	18.7	19.0	19.3
170	167	164	161	158	155	152	150	147	145	143	140	138	136	134	132	130	128	127	126	125	124	123	121	120
10.5	10.7	11.0	11.3	11.6	12.0	12.3	12.6	12.9	13.2	13.5	14.0	14.4	14.8	15.1	15.5	15.9	16.4	16.6	16.9	17.2	17.3	17.6	18.1	18.4
180	177	173	170	167	164	161	158	155	152	150	147	145	143	140	138	136	134	132	130	128	127	126	125	124
9.7	9.9	10.2	10.5	10.7	11.0	11.3	11.6	12.0	12.3	12.6	12.9	13.2	13.5	14.0	14.4	14.8	15.1	15.5	15.9	16.4	16.6	16.9	17.2	17.3
190	186	182	178	175	172	168	165	162	159	156	153	151	148	146	143	141	139	137	135	133	131	129	128	127
9.1	9.3	9.6	9.8	10.0	10.3	10.7	10.9	11.2	11.5	11.8	12.2	12.4	12.8	13.1	13.5	13.9	14.2	14.6	14.9	15.3	15.7	16.2	16.4	16.6
200	196	191	187	183	179	176	172	169	166	163	160	157	154	151	149	147	144	142	140	138	135	133	131	130
8.6	8.8	9.0	9.2	9.5	9.8	9.9	10.3	10.6	10.8	11.1	11.4	11.7	12.1	12.4	12.7	12.9	13.4	13.7	14.0	14.4	14.9	15.3	15.7	15.9

[a] Register temperature, Fahrenheit. (Use upper value in each group).
[b] Register delivery volume in cubic feet per minute per 1000 Btu. (Use lower value in each group).
[c] Register delivery volumes in column 1 are for zero length of duct.

TABLE 7. DETERMINATION OF FREE AREA AND PRESSURE LOSS OF REGISTER FOR 22 DEG DEFLECTION OF AIR[a, b, c, d, e]

Register Free Area in Square Inches (Upper value in each group)
Pressure Loss in Inches of Water (Lower value in each group)

CFM	Residential Use: Reg. Size or Equivalent	Residential Use: Pressure Loss	CFM	Residential Use: Reg. Size or Equivalent	Residential Use: Pressure Loss
Up to 59	10 x 4 10 x 6	.01 .01	100–119	12 x 6 14 x 4	.02 .02
60–69	10 x 6 12 x 4	.01 .02	120–129	12 x 6 14 x 6	.02 .02
70–99	10 x 6 12 x 6	.02 .02	130–169	14 x 6	.02
			170–189	14 x 8	.02

CFM	Distance From Register to Opposite Wall: Up to 18	19–21	22–24	25–27	28–30	31–34 (A)	35–39	40–49	50–59
190–209	68 0.02	49 0.03	38 0.05	29 0.08	24 0.11	19 0.16			
210–229	82 0.02	60 0.03	45 0.04	35 0.06	28 0.10	23 0.13			
230–249		71 0.02	54 0.04	42 0.06	34 0.08	28 0.11	22 0.17		
250–269		84 0.02	63 0.03	49 0.05	40 0.07	32 0.10	25 0.16		
270–299		100 0.02	76 0.03	60 0.04	48 0.06	38 0.09	30 0.13		
300–339			96 0.02	73 0.04	60 0.05	48 0.07	37 0.11		
340–379			122 0.02	95 0.03	78 0.04	61 0.06	47 0.09	33 0.18	
380–419				117 0.02	94 0.03	75 0.05	58 0.08	39 0.15	
420–459				142 0.02	113 0.03	91 0.04	70 0.06	47 0.13	
460–499				168 0.02	135 0.03	108 0.04	83 0.06	56 0.11	
500–539					159 0.02	126 0.03	97 0.05	66 0.10	
540–579					185 0.02	147 0.03	113 0.04	77 0.08	
580–619					212 0.02	169 0.03	130 0.04	88 0.08	51 0.18
620–659						192 0.02	147 0.03	100 0.07	59 0.15
660–699						217 0.02	166 0.03	113 0.06	75 0.13

B

A B

CFM	Distance From Register to Opposite Wall: 31–34	35–39	40–49 (A′)	50–59 (B′)	60–69	70–79	80–89
700–739	243 0.02	186 0.03	127 0.06	85 0.11			
740–779	271 0.02	208 0.03	141 0.05	94 0.10			
780–819		230 0.02	157 0.05	105 0.09			
820–859		254 0.02	173 0.04	115 0.08			
860–899		278 0.02	190 0.04	126 0.08			
900–930		304 0.02	207 0.04	138 0.07			
940–979		332 0.02	226 0.03	151 0.07	108 0.12		
980–1019		360 0.02	245 0.03	163 0.06	118 0.11		
1020–1059		390 0.02	265 0.03	177 0.06	127 0.11		
1060–1099			285 0.03	190 0.06	137 0.10		
1100–1139			307 0.03	204 0.05	147 0.09		
1140–1179			330 0.02	220 0.05	158 0.09		
1180–1219			353 0.02	235 0.04	169 0.08		
1220–1259			376 0.02	251 0.04	180 0.08		
1260–1299			401 0.02	267 0.04	192 0.07	136 0.13	
1300–1339			426 0.02	284 0.04	204 0.07	144 0.13	
1340–1379			452 0.02	301 0.04	217 0.06	154 0.12	109 0.22
1380–1419			479 0.02	319 0.04	230 0.06	163 0.12	116 0.21
1420–1459			508 0.02	338 0.03	244 0.06	172 0.11	122 0.20
1460–1500			536 0.02	356 0.03	256 0.06	182 0.10	129 0.19

A′ B′

[a] If register selected based on distance from register to opposite wall is unsatisfactory on account of size or pressure loss, it is permissible to shift one or more spaces left or right in the tables to obtain a more suitable register. If requirements fall in blank space, select two registers in place of one and divide CFM capacity between the two registers.

[b] Pressure loss is based on FLAT FACE ADJUSTABLE BAR TYPE and does NOT include stackhead.

[c] Values on the right of line A and A′ should not be used in applications such as churches, auditoriums, and concert halls.

[d] Values on right of line B and B′ should not be used in applications such as residential work, motion picture theaters, court rooms and schools.

[e] For floor and baseboard registers where a velocity of approximately 300 FPM is used, the free area $= \frac{\text{CFM} \times 144}{300}$ or approximately, $\frac{\text{CFM}}{2}$. Assume a pressure loss of .01.

Table 8. Determination of Free Area and Pressure Loss of Register for No Deflection of Air[a, b, c, d, e]

Register Free Air in Square Inches (Upper value in each group)
Pressure Loss in Inches of Water Column (Lower value in each group)

CFM	Distance from Register to Opposite Wall								
	19–21	22–24	25–27	28–30	31–34	35–39	40–49	50–59	60–69
190[f]–209	62 0.02	47 0.03	37 0.04	30 0.06	24 0.09	18 0.15			
210–229	76 0.02	58 0.02	45 0.04	36 0.05	29 0.08	22 0.12			
230–249		69 0.02	53 0.03	43 0.04	34 0.07	26 0.11			
250–269		81 0.02	63 0.03	50 0.04	40 0.06	31 0.09	21 0.18		
270–299		93 0.02	73 0.02	59 0.03	47 0.05	36 0.08	24 0.16		
300–339			95 0.02	77 0.03	61 0.04	46 0.06	32 0.12		
340–379			120 0.02	97 0.02	77 0.03	59 0.05	40 0.10		
380–419				119 0.02	95 0.03	73 0.04	50 0.08		
420–459				149 0.02	114 0.03	88 0.04	60 0.07		
460–499				171 0.02	136 0.02	105 0.03	71 0.06		
500–539					160 0.02	123 0.03	84 0.05		
540–579					186 0.02	143 0.02	97 0.05	65 0.09	47 0.17
580–619					213 0.02	164 0.02	111 0.04	74 0.08	54 0.15
620–659						180 0.02	127 0.04	85 0.07	61 0.14

A B

CFM	Distance from Register to Opposite Wall					
	35–39	40–49	50–59	60–69	70–79	80–89
660–699	210 0.02	143 0.03	95 0.07	69 0.12	49 0.22	
700–739	236 0.02	160 0.03	107 0.06	77 0.11	54 0.21	
740–779	262 0.02	179 0.03	119 0.06	86 0.10	61 0.18	
780–819	291 0.02	198 0.03	132 0.05	95 0.09	67 0.17	
820–859	320 0.02	218 0.03	145 0.05	105 0.08	74 0.15	
860–899	352 0.02	240 0.02	160 0.04	115 0.08	81 0.14	
900–939	385 0.01	262 0.02	175 0.04	126 0.07	89 0.13	
940–979	419 0.01	285 0.02	190 0.04	137 0.07	97 0.12	
980–1019	455 0.01	309 0.02	206 0.04	148 0.06	105 0.11	
1020–1059	493 0.01	335 0.02	223 0.03	161 0.06	113 0.11	
1060–1099	530 0.01	361 0.02	241 0.03	173 0.06	123 0.10	
1100–1139	571 0.01	388 0.02	258 0.03	186 0.05	132 0.09	93 0.17
1140–1179		416 0.02	277 0.03	199 0.05	141 0.08	100 0.16
1180–1219		446 0.02	297 0.03	214 0.04	151 0.08	107 0.15
1220–1259		476 0.02	317 0.03	228 0.04	161 0.07	116 0.14
1260–1299		507 0.01	338 0.02	243 0.04	172 0.07	122 0.13
1300–1339		539 0.01	359 0.02	258 0.04	183 0.07	130 0.12
1340–1379			382 0.02	274 0.04	195 0.07	138 0.12
1380–1419			403 0.02	290 0.03	206 0.06	146 0.12
1420–1459				308 0.03	218 0.06	155 0.10
1460–1500				324 0.03	230 0.06	163 0.10

A′ B′

[a] If register selected based on distance from register to opposite wall is unsatisfactory on account of size or pressure loss, it is permissible to shift one or more spaces left or right in the tables to obtain a more suitable register. If requirements fall in blank space, select two registers in place of one and divide CFM capacity between the two registers.

[b] Pressure loss is based on FLAT FACE ADJUSTABLE BAR TYPE and does NOT include stackhead.

[c] Values on the right of line A and A′ should not be used in applications such as churches, auditoriums, and concert halls.

[d] Values on right of line B and B′ should not be used in applications such as residential work, motion picture theaters, court rooms and schools.

[e] For floor and baseboard registers where a velocity of approximately 300 FPM is used, the free area $= \frac{CFM \times 144}{300}$ or approximately, $\frac{CFM.}{2}$ Assume a pressure loss of .01.

[f] For any volume under 190 CFM use Table 7.

TABLE 9. SUGGESTED BONNET PRESSURE

Inches of water

TOTAL CFM THROUGH ANY ONE DUCT	SUGGESTED BONNET PRESSURE IN. WATER	TOTAL CFM THROUGH ANY ONE DUCT	SUGGESTED BONNET PRESSURE IN. WATER
800–1000	0.10	3500 to 5000	0.15
1000–1200	0.10	5000 to 7500	0.25
1200–1800	0.10	7500 to 10,000	0.375
1800–2400	0.13	10,000 to 12,000	0.500
2400 to 3500	0.14	12,000 to 14,000	0.750

the sun or from a fireplace, or by direct heat from any warm air duct, register or chimney.

2. A *fan switch* control located in the bonnet to start blower operations at temperatures between 110 and 130 F, and to stop the blower at about 25 to 30 deg below the cut-in point. The lower settings are used for high sidewall register installations, and the higher settings for baseboard register installations. For most satisfactory results these settings should be as low as is feasible.

3. A protective *high limit switch* located in the bonnet to stop the system independently of the thermostat if the bonnet temperature exceeds 175 F.

4. On oil and gas burner installations, a protective control should be included to cut off the fuel supply if the fire is extinguished, or if there is a failure of the ignition system.

5. On automatic stoker installations, a control is usually included to operate the stoker regardless of thermostat settings whenever the bonnet temperature indicates that the fire is dying, or a time interval contactor is used to cause the stoker to run a few minutes out of each hour.

6. A *humidistat* to regulate the moisture supplied to the rooms, located either in one of the rooms or in the main return duct near the furnace.

ADJUSTMENT OF SYSTEM FOR CONTINUOUS AIR CIRCULATION

This procedure applies to the adjustment of an automatically-fired forced warm air heating system to provide continuous air circulation, when the control arrangement is of the type where the room thermostat controls the fire and the blower control (fan switch) in the furnace bonnet or warm air plenum controls the blower operation. This procedure as outlined in detail in Manual 6 of the *National Warm Air Heating and Air Conditioning Association*, is as follows:[7]

1. Adjust the fuel input in proper relation to the heat loss of the structure.
2. Determine the temperature rise through the furnace.
3. Adjust the air volume to produce a temperature rise through the furnace of about 100 deg.
4. Adjust the fan switch differential to a minimum of about 15 deg.
5. Adjust the fan switch cut-out point as low as practicable.
6. Adjust the room thermostat temperature differential to a minimum which will cause the burner to cycle frequently.
7. Balance the system by adjusting dampers to produce even temperature distribution between rooms.
8. Set the room thermostat at the desired room temperature.

WARM AIR CEILING PANEL SYSTEMS

Warm air ceiling panel heating systems utilize a false ceiling suspended $3\frac{1}{4}$ in. from the ceiling joists which have previously been covered on the bottom with sheets of plasterboard. Special hangers are used to suspend the false ceiling or heating panel from the joists. Steel supporting rods

TABLE 10. PRESSURE DROP IN DUCT

Inches of Water per 100 Feet of Duct Length

Equivalent Length of Duct (Ft)	Total Pressure Drop in Duct (In. of Water)																
	0.04	0.05	0.06	0.07	0.08	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.17	0.18	0.19	0.20
35–44	0.10	0.13	0.15	0.18	0.20	0.23	0.25	0.28	0.30	0.33	0.35	0.38	0.40	0.43	0.45	0.48	0.50
45–54	0.08	0.10	0.12	0.14	0.16	0.18	0.20	0.22	0.24	0.26	0.28	0.30	0.32	0.34	0.36	0.38	0.40
55–64	0.07	0.08	0.10	0.12	0.13	0.15	0.17	0.18	0.20	0.22	0.23	0.25	0.27	0.28	0.30	0.32	0.33
65–74	0.06	0.07	0.09	0.10	0.11	0.13	0.14	0.16	0.17	0.19	0.20	0.21	0.23	0.24	0.26	0.28	0.29
75–84	0.05	0.06	0.08	0.09	0.10	0.11	0.13	0.14	0.15	0.16	0.18	0.19	0.20	0.21	0.23	0.24	0.25
85–94	0.05	0.06	0.07	0.08	0.09	0.10	0.11	0.12	0.13	0.14	0.16	0.17	0.18	0.19	0.20	0.21	0.22
95–104	0.04	0.05	0.06	0.07	0.08	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.17	0.18	0.19	0.20
105–114	0.04	0.05	0.05	0.06	0.07	0.08	0.09	0.10	0.11	0.12	0.13	0.14	0.15	0.15	0.16	0.17	0.18
115–129	0.03	0.04	0.05	0.06	0.07	0.08	0.08	0.09	0.10	0.11	0.12	0.12	0.13	0.14	0.15	0.16	0.17
130–149	0.03	0.04	0.04	0.05	0.06	0.07	0.07	0.08	0.09	0.09	0.10	0.11	0.11	0.12	0.13	0.14	0.14
150–169	0.03	0.03	0.04	0.04	0.05	0.06	0.06	0.07	0.08	0.08	0.09	0.09	0.10	0.11	0.11	0.12	0.13
170–189	0.02	0.03	0.03	0.04	0.04	0.05	0.06	0.06	0.07	0.07	0.08	0.08	0.09	0.09	0.10	0.11	0.11
190–214	0.02	0.03	0.03	0.04	0.04	0.05	0.05	0.06	0.06	0.07	0.07	0.08	0.08	0.09	0.09	0.10	0.10
215–239	0.02	0.02	0.03	0.03	0.04	0.04	0.05	0.05	0.05	0.06	0.06	0.07	0.07	0.08	0.08	0.09	0.09
240–264	0.02	0.02	0.02	0.03	0.03	0.04	0.04	0.04	0.05	0.05	0.06	0.06	0.06	0.07	0.07	0.08	0.08
265–289	0.01	0.02	0.02	0.03	0.03	0.03	0.04	0.04	0.04	0.05	0.05	0.05	0.06	0.06	0.07	0.07	0.07
290–324	0.01	0.02	0.02	0.03	0.03	0.03	0.03	0.04	0.04	0.04	0.05	0.05	0.05	0.06	0.06	0.06	0.07
325–374	0.01	0.02	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.04	0.04	0.04	0.05	0.05	0.05	0.05	0.06
375–424	0.01	0.01	0.02	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.04	0.04	0.04	0.04	0.05	0.05	0.05
425–474	0.01	0.01	0.01	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.03	0.03	0.04	0.04	0.04	0.04	0.05
475–524	0.01	0.01	0.01	0.02	0.02	0.02	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.03	0.04	0.04	0.04
525–574	0.01	0.01	0.01	0.01	0.02	0.02	0.02	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.03	0.03	0.04
575–625	0.01	0.01	0.01	0.01	0.01	0.02	0.02	0.02	0.02	0.02	0.02	0.03	0.03	0.03	0.03	0.03	0.03

Equivalent Length of Duct (Ft)	Total Pressure Drop in Duct (In. of Water)																
	0.21	0.22	0.23	0.24	0.25	0.26	0.27	0.28	0.29	0.30	0.32	0.34	0.36	0.38	0.40	0.45	0.50
35–44	0.53	0.55	0.58	0.60	0.63	0.65	0.68	0.70	0.73	0.75	0.80	0.85	0.90	0.95	1.00	1.13	1.25
45–54	0.42	0.44	0.46	0.48	0.50	0.52	0.54	0.56	0.58	0.60	0.64	0.68	0.72	0.76	0.80	0.90	1.00
55–64	0.35	0.37	0.39	0.40	0.42	0.43	0.45	0.47	0.48	0.50	0.53	0.57	0.60	0.64	0.67	0.75	0.83
65–74	0.30	0.32	0.33	0.34	0.36	0.37	0.39	0.40	0.42	0.43	0.46	0.49	0.52	0.54	0.57	0.64	0.72
75–84	0.26	0.28	0.29	0.30	0.31	0.33	0.34	0.35	0.36	0.38	0.40	0.43	0.45	0.48	0.50	0.56	0.63
85–94	0.23	0.25	0.26	0.27	0.28	0.29	0.30	0.31	0.32	0.33	0.36	0.38	0.40	0.42	0.45	0.50	0.56
95–104	0.21	0.22	0.23	0.24	0.25	0.26	0.27	0.28	0.29	0.30	0.32	0.33	0.36	0.38	0.40	0.45	0.50
105–114	0.19	0.20	0.21	0.22	0.23	0.24	0.25	0.26	0.27	0.28	0.29	0.31	0.33	0.35	0.37	0.41	0.46
115–129	0.18	0.18	0.19	0.20	0.21	0.22	0.23	0.23	0.24	0.25	0.27	0.28	0.30	0.32	0.33	0.38	0.47
130–149	0.15	0.16	0.16	0.17	0.18	0.19	0.19	0.20	0.21	0.21	0.23	0.24	0.26	0.27	0.29	0.32	0.36
150–169	0.13	0.14	0.14	0.15	0.16	0.16	0.17	0.18	0.18	0.19	0.20	0.21	0.23	0.24	0.25	0.28	0.31
170–189	0.12	0.12	0.13	0.13	0.14	0.15	0.15	0.16	0.16	0.17	0.18	0.19	0.20	0.21	0.22	0.25	0.28
190–214	0.11	0.11	0.12	0.12	0.13	0.13	0.14	0.14	0.15	0.15	0.16	0.17	0.18	0.19	0.20	0.23	0.25
215–239	0.09	0.10	0.10	0.11	0.11	0.12	0.12	0.13	0.13	0.13	0.14	0.15	0.16	0.17	0.18	0.20	0.22
240–264	0.08	0.09	0.09	0.10	0.10	0.10	0.11	0.11	0.12	0.12	0.13	0.14	0.15	0.15	0.16	0.18	0.20
265–289	0.08	0.08	0.08	0.09	0.09	0.10	0.10	0.10	0.11	0.11	0.12	0.12	0.13	0.14	0.15	0.16	0.18
290–324	0.07	0.07	0.08	0.08	0.08	0.09	0.09	0.09	0.10	0.10	0.11	0.11	0.12	0.13	0.13	0.15	0.17
325–374	0.06	0.06	0.07	0.07	0.07	0.08	0.08	0.08	0.09	0.09	0.09	0.10	0.10	0.11	0.11	0.13	0.14
375–424	0.05	0.06	0.06	0.06	0.06	0.07	0.07	0.07	0.08	0.08	0.08	0.09	0.09	0.09	0.10	0.11	0.13
425–474	0.05	0.05	0.05	0.05	0.06	0.06	0.06	0.06	0.07	0.07	0.07	0.08	0.08	0.09	0.09	0.10	0.11
475–524	0.04	0.04	0.05	0.05	0.05	0.05	0.05	0.06	0.06	0.06	0.06	0.07	0.07	0.08	0.08	0.09	0.10
525–574	0.04	0.04	0.04	0.04	0.05	0.05	0.05	0.05	0.05	0.06	0.06	0.07	0.07	0.07	0.07	0.08	0.09
575–625	0.04	0.04	0.04	0.04	0.04	0.04	0.05	0.05	0.05	0.05	0.05	0.06	0.06	0.06	0.07	0.08	0.08

are installed through these hangers and metal lath is attached to the rods. The lath is then plastered to a thickness of $\frac{3}{4}$ in., making a completely sealed air space above the ceiling. Warm air is delivered to this sealed space through a standard warm air duct, installed in the usual manner. The air is then circulated over the entire ceiling being guided by sheet-metal baffles. After the air has passed over the ceiling, it is returned to the furnace through a return air duct for reheating. The system is closed, and no air is introduced into the heated space from the panel. (See also Chapter 23, Panel Heating.)

Because a warm air ceiling panel system involves a different type of ceiling construction, its installation is practically limited to new construction. Only automatically-fired and thermostatically-controlled furnace-blower units may be used with this system. The standard automatic heating controls consisting of a room thermostat, temperature limit control, blower control (fan switch), and primary control are all that are required. While warm air ceiling panel heating has particular advantages in one-story utility room houses, it can be just as effectively installed in one- and two-story houses with basements.

Complete design and installation procedure is given[8] in Manual 7-A of the *National Warm Air Heating and Air Conditioning Association.* This procedure is given in the following general outline:

1. Calculate the design heat losses from individual spaces in the structure. (See Chapter 11.)
   a. Determine furnace size from tables in Manual 7-A.
   b. Determine fan (blower) size from tables in Manual 7-A.

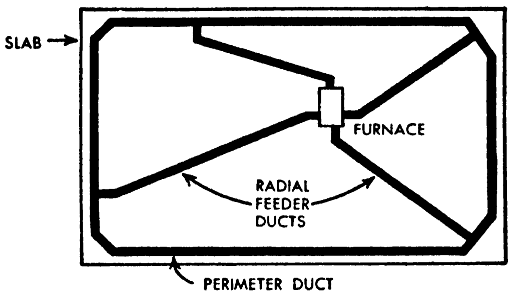


FIG. 4. SCHEMATIC LAYOUT OF PERIMETER SYSTEM[9]

2. Lay out the ceiling panel design, locating panel inlets and outlets, and make a tentative drawing of supply and return system.
3. Measure actual length of warm air travel from plenum to discharge into panel.
4. Determine equivalent length of fittings, including panel inlet connection to panel, from values given in Manual 7-A.
5. Select duct sizes required from capacity tables and check minimum allowable panel area.
6. Measure actual length of return air travel from warm air outlet into panel to connection at blower cabinet.
7. Determine equivalent length of fittings and turns from warm air outlet into panel to blower inlet.
8. Add actual lengths and equivalent lengths, determine width of space 3¼ in. deep that may be used, and find the size of the return air duct required.
9. Determine both warm air and return air trunk sizes.

WARM AIR PERIMETER SYSTEMS

Warm air perimeter heating systems are combination panel and convection systems that have been developed especially for structures having a concrete-slab floor laid on the ground. Warm air is delivered from the furnace-blower unit through round ducts imbedded in the concrete floor slab to a continuous duct along the outer perimeter of the house just inside the exterior walls. The warm air is introduced into the rooms through registers which are connected to the ducts, and are located under windows or at the points of greatest heat loss. The return air is collected through one or more grilles located high on the interior sidewalls or in the ceiling.

A suggested procedure[9] for the design and installation of a warm air perimeter heating system is given in Manual 4 of the *National Warm Air Heating and Air Conditioning Association.* This procedure follows the general outline:

1. Calculate design heat losses from individual spaces in the structure. (See Chapter 11.) Calculate design heat losses below grade using table in Form 5,[10] which accompanies Manual 4.
2. Lay out perimeter system design, locating ducts, registers, and return air grilles or intakes.
3. Determine length from furnace to each register using shortest distance.
4. Determine register sizes from tables in Form 5.
5. Determine size of return-air intake and ducts from tables in Form 5.

The furnace may be either the down-flow or the conventional up-flow type. In most cases, the down-flow type is preferred since it requires a minimum amount of floor area and ductwork, and eliminates the need for a duct to bring the warm air down from the top of the furnace, or a duct to bring the return air down to the blower inlet.

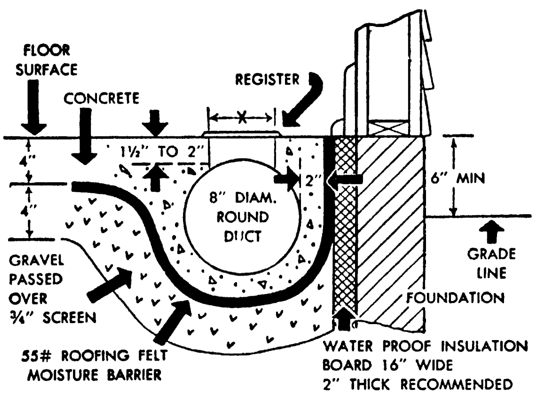


FIG. 5. CROSS-SECTION OF SLAB CONSTRUCTION CONTAINING PERIMETER DUCT[9]

Several arrangements of perimeter ducts may be used, but the most common is the type having a complete loop of a continuous duct around the edge of the slab, supplied by radial feeders extending from the furnace sub-floor plenum to the perimeter duct, as is shown in Fig. 4. A schematic cross-section of the slab construction, with the perimeter duct installed, is shown in Fig. 5.

The slab should be constructed on a well-drained site where drainage is away from the slab, and where there is no standing water at any time of the year. With slab construction, a suitable porous fill and a waterproof membrane as a moisture barrier beneath the slab are required by the *Federal Housing Administration.* They are highly necessary with warm air perimeter heating. Insulation must be placed between the edge of the slab and the foundation, and must extend completely around the slab to reduce the heat losses from the edge of the slab. Fig. 5 also shows many of the essential features of the slab and under-slab construction.

COOLING METHODS

A slight cooling effect may be obtained under certain conditions by the circulating basement air. A more positive cooling effect may be obtained by the use of an air washer where the temperature of the city or well water is sufficiently low (55 F or lower), and where a sufficient volume of water can be provided. Unless the temperature of the leaving water is below the dew-point temperature of the indoor air at the time the washer is started, both the relative and absolute humidities will be somewhat increased.

Coils of copper finned tubing through which cold water is pumped are available for cooling. They require less space than air washers, and have the advantage that no moisture is added to the air when the temperature of the water rises above the dew-point. Ample coil surface and fan capacity are necessary with this type of cooling.

It is thoroughly feasible to use ice or mechanical refrigeration in connection with a warm air system and to cool the building by this method, provided the building is reasonably well constructed and insulated. Windows and doors should be tight, and awnings should be supplied on the sunny side of the building. See also Chapters 29, 36 and 37.

Conclusions drawn from studies[11] conducted in the *University of Illinois* Research Residence, subject to the limitations of the test are:

1. An uninsulated building of ordinary residential type may require the equivalent of three tons of ice in 24 hr on days when the maximum outdoor temperature reaches 100 F, if an effective temperature of approximately 72 ET is maintained indoors.
2. The use of awnings at all windows in east, south, and west exposures may result in savings of from 20 to 30 percent in the required cooling load.
3. The cooling load per degree difference in temperature is not constant, but increases as the outdoor temperature increases.
4. The heat lag of the building complicates the estimation of the cooling load under any specified conditions and makes such estimates, based on the usual methods of computation, of doubtful value.
5. The seasonal cooling requirements are extremely variable from year to year, and the ratio between the degree-hours of any two seasons occurring within a 10-year period may be as high as 7.5 to 1. Hence, an average value of the degree-hours cooling per season is comparatively meaningless.
6. The duct system in a forced-air heating installation can be successfully converted to a system for conveying cool air for the purpose of cooling the structure. No condensation of moisture was observed when the duct temperatures were not less than 65 F.
7. Cooling by means of water at a temperature of 60 F is not satisfactory unless an indoor temperature of less than 80 F is maintained.
8. In the selection of cooling coils, the additional frictional resistance of the coil to flow of air must be given consideration.
9. Cooling the structure by introducing large quantities of outdoor air at night tended to reduce the amount of cooling required on the following day, and was a practical means of providing more comfortable conditions in those homes where cooling systems were not available.

DESIGN OF COOLING SYSTEM

The general procedure which may be used for the design of a summer cooling system in a forced-air installation is:

1. Calculate heat gain for each room or space to be conditioned. (See Chapters 9 and 12.) Allowance for addition of outside air must be included in this calculation.
2. Select a temperature of air leaving supply inlets. In *University of Illinois* Research Residence tests, a value of from 65 to 70 F was found satisfactory.
3. Determine indoor conditions to be maintained. In the Research Residence, 80 F dry-bulb and 45 percent relative humidity were found satisfactory.
4. Determine the quantity of air to be introduced into each room.
5. Estimate heat loss in duct system between cooling unit and supply registers.
6. Calculate the sensible and latent heat to be removed by the cooling unit.

7. Determine size of ducts in duct system and size of registers, as explained in this chapter.

8. Determine pressure loss in duct system and select fan having proper capacity.

9. Select cooling unit from manufacturer's data. Specify temperature and pressure of available cooling water, voltage and characteristics of electrical supply, and method of control of apparatus.

10. Select cooling coils from manufacturer's data to take care of latent heat load and to give required drop in air temperature with the weight of air flowing. (See Chapter 35.)

11. If system is to be used for both winter heating and summer cooling, duct sizes must be checked to insure that velocities and friction losses are reasonable for both conditions of operation. Adjustable dampers will be necessary to make changes in air distribution for the two seasons. Provision must also be made for changing fan speeds for summer and winter operation.

REFERENCES

[1] A Yardstick for the Evaluation of a Forced Warm Air Heating System (*National Warm Air Heating and Air Conditioning Association*, Manual 8, 1941).

[2] Performance of a Forced Warm-Air Heating System as Affected by Changes in Volume and Temperature of Air Recirculated, by A. P. Kratz and S. Konzo (A.S.H.V.E. Transactions, Vol. 48, 1942, p. 393).

[3] Code and Manual for the Design and Installation of Warm Air Winter Air Conditioning Systems (*National Warm Air Heating and Air Conditioning Association*, Manual 7, Second Edition, 1947).

[4] Proposed Design Procedure for Large Mechanical Warm Air Heating Systems, by S. Konzo, R. J. Martin, D. S. Levinson, and R. W. Roose (A.S.H.V.E. Transactions, Vol. 53, 1947, p. 177).

[5] Code and Manual for the Design and Installation of Large Warm Air Winter Air Conditioning Systems (*National Warm Air Heating and Air Conditioning Association*, Manual 9, Fourth Edition, 1950).

[6] Automatic Controls for Forced-Air Heating Systems, by S. Konzo and A. F. Hubbard (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 37).

[7] Service Manual for Continuous Air Circulation Technicians (*National Warm Air Heating and Air Conditioning Association*, Manual 6, First Edition, 1947).

[8] Code and Manual for the Design and Installation of Warm Air Ceiling Panel Systems (*National Warm Air Heating and Air Conditioning Association*, Manual 7-A, Second Edition, 1948).

[9] Warm-Air Perimeter Heating (*National Warm Air Heating and Air Conditioning Association*, Manual 4, First Edition, 1950).

[10] Work Sheet for Warm-Air Perimeter Systems (*National Warm Air Heating and Air Conditioning Association*, Form 5, Second Edition, 1950).

[11] Summer Cooling in the Research Residence, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick (*University of Illinois Engineering Experiment Station Bulletins* Nos. 290, 305 and 321). A.S.H.V.E. Research Report No. 1177—Summer Cooling in the Research Residence with a Gas-Fired Dehydration Cooling Unit, by A. P. Kratz, S. Konzo and E. L. Broderick (A.S.H.V.E. Transactions, Vol. 47, 1941, p. 203).

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CHAPTER 20

STEAM HEATING SYSTEMS

Classification of Steam Heating Systems by Types; One-pipe; Two-pipe, Sub-atmospheric and Orifice Systems; Sizing Piping for Steam Heating Systems; Pressure Reducing Valves; Boiler Connections; Condensate Return Pumps; Vacuum Heating Pumps; Traps; Drips; Connections to Heating Units; Control Valves

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STEAM heating systems may be classified according to any one of, or combination of, the following features: (1) piping arrangement, (2) pressure or vacuum conditions obtained in operation, (3) method of returning condensate to the boiler.

1. *By Piping Arrangement.* A steam heating system is known as a *one-pipe system* when a single main serves the dual purpose of supplying steam to the heating unit and conveying condensate from it. Ordinarily, to each heating unit there is but one connection which must serve as both the supply and the return, although separate supply and return connections may be used.

A steam heating system is known as a *two-pipe system* when each heating unit is provided with two piping connections, and when steam and condensate flow in separate mains and branches.

Heating systems may also be described as *up-flow* or *down-flow*, depending on the direction of steam flow in the risers; and as a *dry-return* or a *wet-return*, depending on whether the condensate mains are above or below the water line of the boiler or condensate receiver.

2. *By Pressure or Vacuum Conditions.* Steam heating systems may also be classified as *high pressure*, *low pressure*, *vapor*, and *vacuum* systems, depending on the pressure conditions under which the system is designed to operate.

A system is known as a *high pressure system* when the operating pressures employed are above 15 psig; as a *low pressure system* when pressures vary from 0 to 15 psig; as a *vapor system* when the system operates under both vacuum and low pressure conditions without the use of a vacuum pump; and as a *vacuum system* when the system operates under vacuum and low pressure conditions with the use of vacuum pump.

When automatic controls are employed to vary the pressure conditions in the system in accordance with outside weather conditions, the system may be known as a *sub-atmospheric*, *differential*, or *synchronized* system. These latter classifications are proprietary designations.

When orifices are employed on the inlets to the heating units the system may be known as an *orifice system.*

3. *By Method of Returning Condensate.* When condensate is returned to the boiler by gravity, the system is known as a *gravity return system.* In this system all heating units must be elevated sufficiently above the water line of the boiler, so that the condensate can flow freely to the boiler. Elevation of the heating units above the water line must therefore be sufficient to overcome pressure drops due to flow, as well as pressure differences due to operation.

Referring to Fig. 1 it will be noted that the boiler and wet-return form a U-shaped container, with the boiler steam pressure on the top of the water at one end, and the steam main pressure on the top of the water at the other end. The difference between these two pressures is the *pressure drop* in the system, *i.e.*, the friction and resistance to the flow of steam in passing from the boiler to the far end of the main, and the pressure reduction in consequence of the condensation occurring in the system. The water in the far end will rise sufficiently to overcome this difference in order to balance the pressures, and it will rise far enough to produce a flow through the return pipe and overcome the resistance of check valves, if installed.

If a one-pipe steam system is designed, for example, for a total pressure drop of $\frac{1}{8}$ psi, and utilizes a Hartford return connection instead of a check valve on the return, the rise in the water level at the far end of the return, due to the difference in steam pressure, would be ⅛ of 28 in. (28 in. head being equal to one pound per square inch), or 3½ in. Adding 3 in. to overcome the resistance of the return main, and 6 in. as a factor of safety for heating up, gives 12½ in. as the distance the bottom of the lowest part of the steam main and all heating units must be above the boiler water line. The same system, however, installed and sized for a total pressure drop of ½ psi, and with a check in the return, would require ½ of 28 in., or 14 in. for the difference in steam pressure, 3 in. for the flow through the return, 4 in. to operate the check, and 6 in. for a factor of safety, making a total of 27 in. as the required distance. Higher pressure drops would increase the distance accordingly.

When conditions are such that condensate cannot be returned to the boiler by the action of gravity, and either traps or pumps must be employed, the system is known as a *mechanical return system.* There are three general types of mechanical condensate return devices in common use: (*a*) the alternating return trap, (*b*) the condensate return pump, and (*c*) the vacuum return pump.

In systems where pressure conditions in the system vary between that of a gravity return and a forced return system, a boiler return trap or alternating receiver is employed and the system may be known as an *alternating return system.*

When condensate is pumped to the boiler under pressures of the atmosphere or above, the system is known as a *condensate pump return system.*

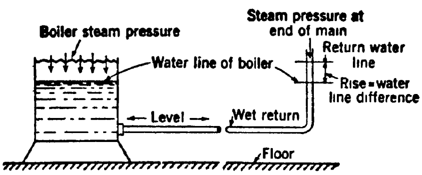

FIG. 1. DIFFERENCE IN STEAM PRESSURE ON WATER IN BOILER AND AT END OF STEAM MAIN

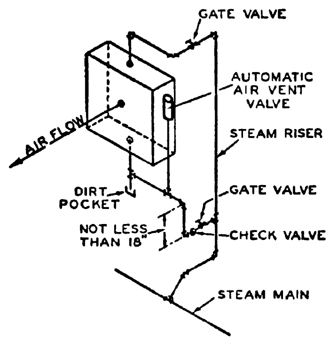

FIG. 2. TYPICAL TWO-PIPE CONNECTIONS TO UNIT HEATERS IN ONE-PIPE AIR VENT SYSTEMS

When condensate is pumped to the boiler under vacuum conditions, the system is known as a *vacuum pump return system.*

In either the *condensate* or *vacuum pump systems* it is highly desirable to arrange for gravity flow to a receiver and to the pump. The pump then forces condensate into the boiler against its pressure.

ONE-PIPE SYSTEMS

One-pipe systems, as previously defined, are systems in which steam and condensate flow in the same pipe. Radiators and other heating units, in general, have only one piping connection from main to unit, although it is possible to employ two connections to the same main as indicated in Fig. 2. Unit heaters in one-pipe systems may also have separate connections to the wet-return as shown in Fig. 5 Chapter 24.

There are several variations in the piping arrangement of a one-pipe system as follows:

1. Up-feed one-pipe systems where the radiators and other heating units are located above the supply mains. The mains in this instance convey both steam and condensate. Such a system is illustrated in Fig. 3. Typical connections to radiator or risers are illustrated in Fig. 4, and method of changing sizes of mains in Fig. 5.

2. Up-feed one-pipe systems where the radiators and other heating units are located above the mains, and the mains are dripped at each radiator connection to a wet-

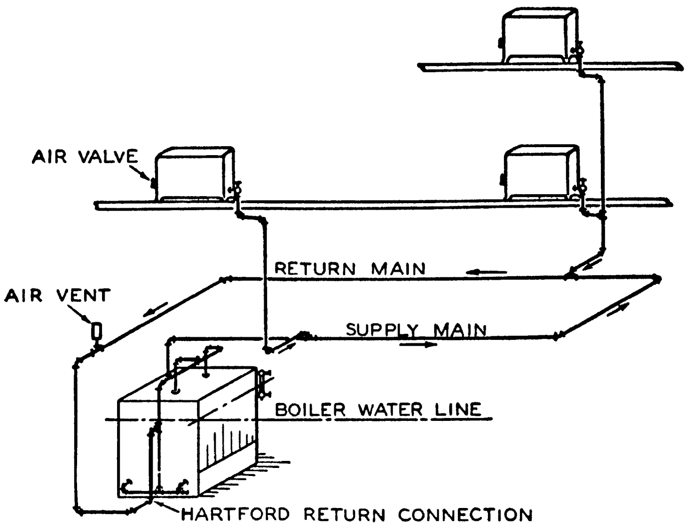

FIG. 3. TYPICAL UP-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

return, so that the steam main carries a minimum of the condensate. This system is illustrated in Fig. 6. Typical connections to radiators and risers are illustrated in Fig. 7. Up-feed systems are not recommended for systems higher than four stories

3. Down-feed one-pipe systems, where the radiators and other heating units are located below the supply main. In this arrangement only risers and connections to heating units convey both steam and condensate, and both are flowing in the same direction. The steam main is kept relatively free of condensate by dripping through the drop risers.

Each radiator or heating unit in a one-pipe system must be supplied with a thermostatic air valve which functions to relieve air from the heating unit under pressure, and to close when steam itself heats the thermostatic element of the valve.

To improve steam circulation in one-pipe systems quick-vent air valves should be provided at the ends and at intermediate points where the steam main is brought to a higher elevation, or where dropped below the water line. It is desirable to install the air-vent valves about a foot ahead of the drips, as indicated in Fig. 6, to prevent possible damage to their mechanisms by water.

Air valves are of two general types, the pressure and the vacuum types. The pressure type permits the inflow of atmospheric air to the system when the steam pressure in the system falls below atmospheric pressure. The vacuum type, which contains a small check valve, prevents the air from flowing back to the system and thereby maintains vacuum conditions in the system, and a consequent evaporation or generation of steam

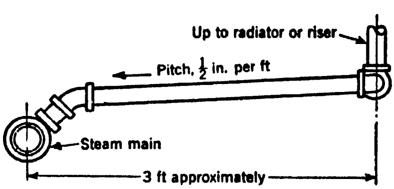

FIG. 4. TYPICAL STEAM RUNOUT WHERE RISERS ARE NOT DRIPPED

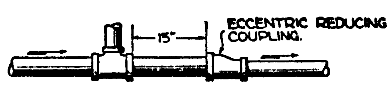

FIG. 5. METHOD OF CHANGING SIZE OF STEAM MAIN WHEN RUNOUTS ARE TAKEN FROM TOP

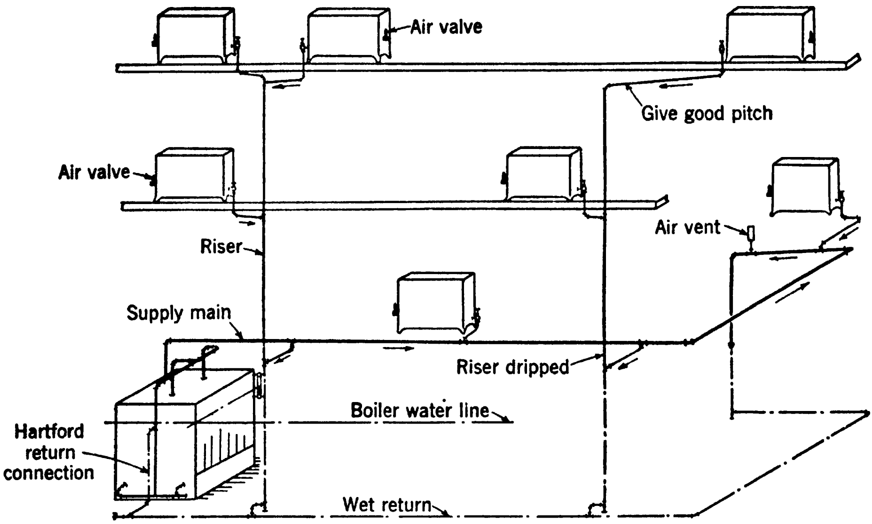

FIG. 6. TYPICAL UP-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

or vapor at sub-atmospheric pressures, and at consequent lower temperatures. Systems which use vacuum valves are known as *vapor* or *vacuum one-pipe systems.* The vapor or vacuum systems will maintain a more uniform temperature condition than the pressure systems.

Each heating unit in a one-pipe system may also be provided with a valve on the connection to the unit, although this is not essential except to shut the unit off in case it is not desired for heating. Valves on one-pipe systems must be either fully opened or fully closed. No throttling or modulating position can be maintained since, if a valve is partially closed, condensate will not drain from the unit. This condition is dangerous because it may create a low water condition in the boiler with consequent burning or cracking of the boiler, or create a hazard due to the freezing of the water-logged heating unit itself.

TWO-PIPE SYSTEMS

Two-pipe systems, as previously defined, are systems in which steam and condensate flow in separate pipes. Two-pipe systems may operate under high pressure, low pressure, vapor, or vacuum conditions. Either the up-flow or the down-flow arrangement of mains may be employed.

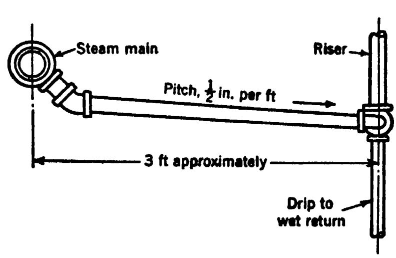

FIG. 7. TYPICAL STEAM RUNOUT WHERE RISERS ARE DRIPPED

Two-Pipe High Pressure Systems

Two-pipe high pressure systems operate at pressures above 15 psig, usually from 30 to 150 psig. They are usually used in large industrial type buildings, which are equipped with unit heaters or large built-up fan units, or in which high pressure steam is required for process work.

Fig. 8 illustrates a typical high pressure system. Because of the high pressures and the great differential between steam and return mains, it is possible to locate returns above the heating units and lift the condensate to these returns.

The condensate can be flashed into steam in low pressure mains if any are available, or passed through an economizer heater before being discharged to a vented receiver. It is, of course, necessary to provide for the elimination of air from high pressure systems, the same as in low pressure systems.

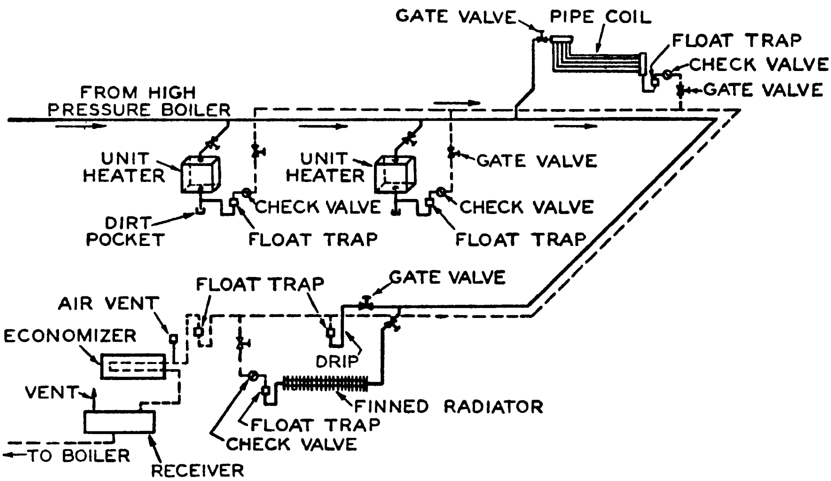

Fig. 8. Typical High Pressure Heating System

Return traps used on high pressure systems are usually of the bucket, inverted bucket, float or impulse type.

Two-Pipe Low Pressure Systems

Low pressure systems operate at pressures of 0 to 15 psig. The piping arrangement of both up-feed and down-feed low pressure systems is identical with those of two-pipe vapor systems described in the following section. The only difference between the two systems is in the type of air valve used. The air valves used in low pressure systems usually do not contain the check discs and hence, the system cannot operate under a vacuum. The low pressure systems are not as popular as the vapor systems, because they have the disadvantage of not holding heat when the rate of steam generation is diminishing. They also have the disadvantage of corroding to a greater extent than vapor systems, due to the continued presence of new air in the system.

Low pressure systems have the advantage, however, of returning condensate to the boiler readily and not retaining it in the piping, as may be possible in vapor systems when the system pressure exceeds the operating

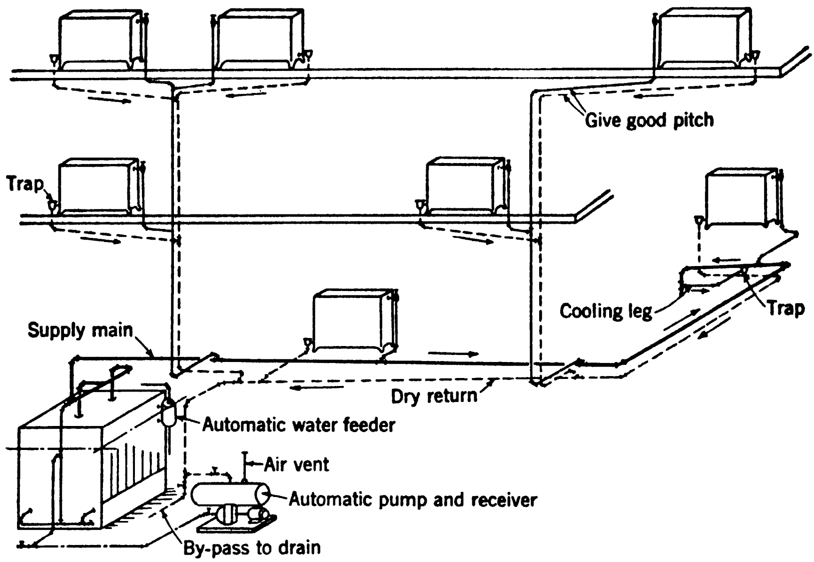

Fig. 9. Typical Installation Using Condensate Pump

range of the average vapor system. Fig. 9 illustrates a typical low pressure system with condensate pump.

Two-Pipe Vapor Systems

Two-pipe vapor systems operate at pressures varying from 20 in. vacuum or more (depending upon the tightness with which the system is assembled) to 15 psig without the use of a vacuum pump. A typical two-pipe up-feed vapor system is shown in Fig. 10, and a typical two-pipe down-feed system is illustrated in Fig. 11. The method of dripping drop risers in a down-feed system is illustrated in Fig. 12. Radiators discharge their condensate and air through thermostatic traps to the dry-return main. Air is eliminated,

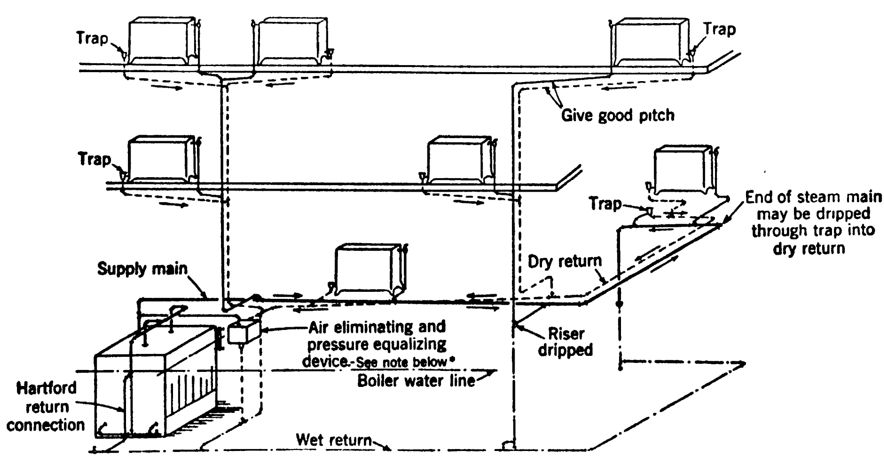

Fig. 10. Typical Up-Feed Two-Pipe System with Automatic Return Trap*

* Proper piping connections are essential with special appliances for pressure equalizing and air elimination.

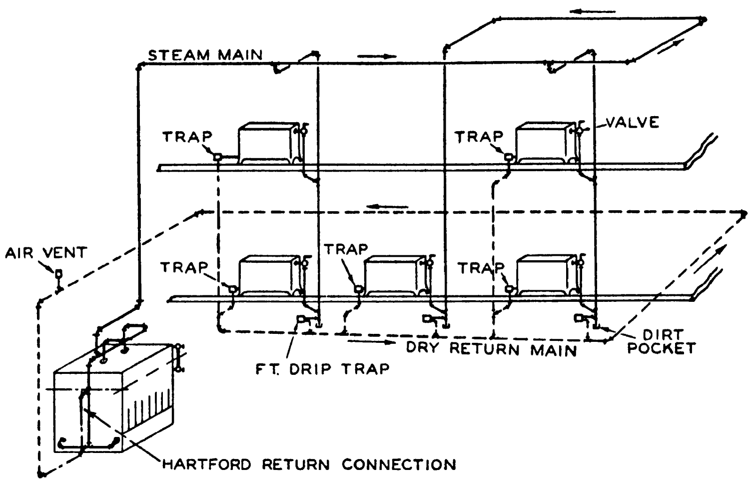

FIG. 11. TYPICAL DOWN-FEED TWO-PIPE SYSTEM

when the system is under pressure, at the ends of the supply and return mains just before they drop to the wet return. The vent valves are of the float and thermostatic type which opens when cool air contracts the thermostatic element, and closes when steam expands the element. The float element of the valve closes the valve when, due to pressure differences, water rises to the point of overflow in the main. The vent valves are also provided with a small check disc which closes to prevent the inflow of air to the system when the pressure drops below atmospheric pressure. This enables the system to operate under vacuum conditions at lower steam temperatures for a period of four to eight hours, depending on the tightness of the system.

Vapor systems may also be provided with an automatic return trap or alternating receiver which automatically returns condensate to the boiler when the boiler is steaming under pressure conditions which would prevent the return of condensate by gravity. The typical connections for an automatic return trap are illustrated in Fig. 13.

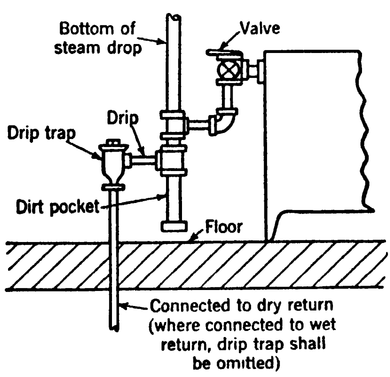

FIG. 12. DETAIL OF DRIP CONNECTIONS AT BOTTOM OF DOWN-FEED STEAM DROP

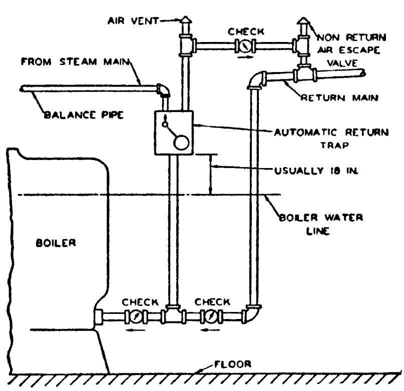

FIG. 13. TYPICAL CONNECTIONS FOR AUTOMATIC RETURN TRAP

Each heating unit in a vapor system, as in all two-pipe systems, is provided with a graduated or modulating valve which permits the control of heat in the radiator by varying the opening of the valve.

Two-Pipe Vacuum Systems

Vacuum systems operate under conditions of both low pressure and vacuum, but employ the use of a vacuum pump to insure maintenance of sub-atmospheric pressures in the return piping for all operating conditions. The system may operate transiently with sub-atmospheric pressure in the supply piping during the time the rate of steam generation is equivalent to or less than the total connected load.

A typical two-pipe up-feed vacuum system is illustrated in Fig. 14, and a down-feed arrangement in Fig. 15.

The return risers are connected in the basement into a common return main which slopes downward toward the vacuum pump. The vacuum

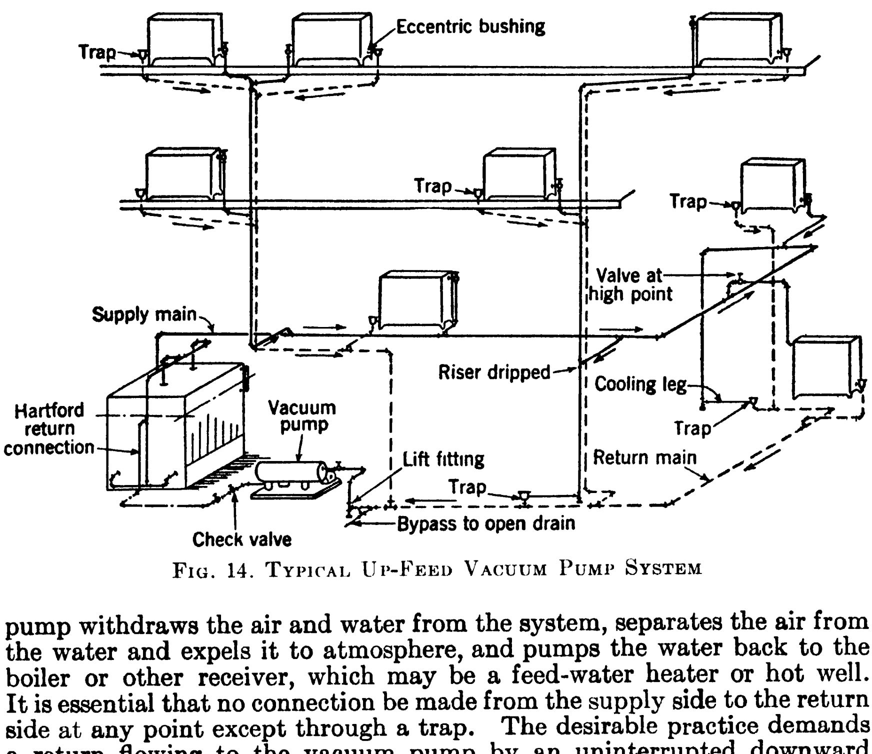

FIG. 14. TYPICAL UP-FEED VACUUM PUMP SYSTEM

pump withdraws the air and water from the system, separates the air from the water and expels it to atmosphere, and pumps the water back to the boiler or other receiver, which may be a feed-water heater or hot well. It is essential that no connection be made from the supply side to the return side at any point except through a trap. The desirable practice demands a return flowing to the vacuum pump by an uninterrupted downward slope. In some instances local conditions make it necessary to drop the return below the level of the vacuum pump inlet before the pump can be reached. In such an event one of the advantages of the vacuum system is the ability to raise the condensate to a considerable height, by the suction of the vacuum pump, by means of a lift connection or fitting inserted in the return. The height the condensate can be raised depends on the amount of vacuum maintained. It is preferable to limit lift connections to a single lift at the vacuum pump. A still more preferable arrangement is the use of an accumulator tank, or receiver tank, with a float control for the pump at the low point of the return main, located adjacent to the vacuum pump.

When the vertical lift is considerable, several lift fittings should be used in steps as shown in Fig. 16. This permits a given lift to be secured with a somewhat lower vacuum than where the vertical distance is served by a single lift. Where several lifts are present in a given system at different locations, the lifting cannot occur until the entire system is filled with steam. A lift connection for location close to the pump, where the size may be above the commercial stock sizes, is shown in Fig. 17. It is desirable that means be provided for manually draining the low point of the lift fittings to eliminate danger of freezing.

TWO-PIPE SUB-ATMOSPHERIC SYSTEMS

Sub-atmospheric systems are similar to vacuum systems but, in contrast, provide control of building temperature by variation of the heat

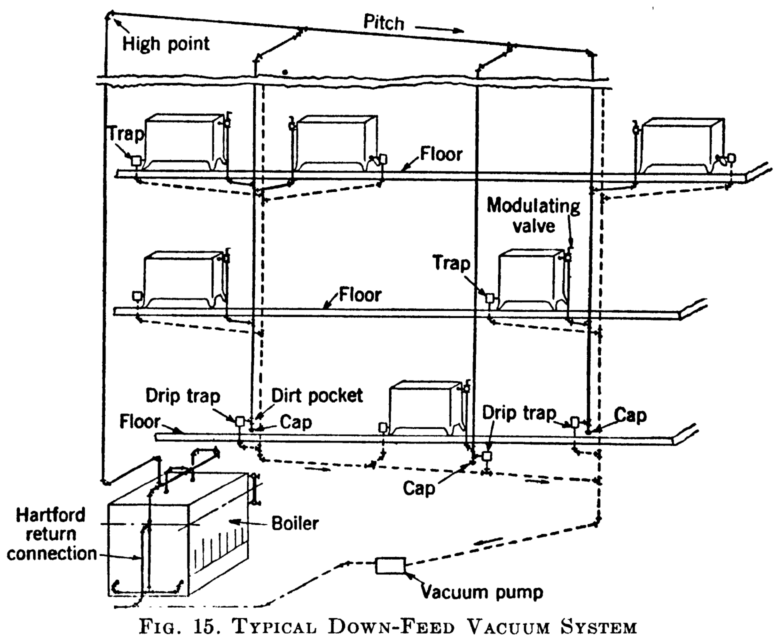

FIG. 15. TYPICAL DOWN-FEED VACUUM SYSTEM

output from the radiators. The radiator heat emission is controlled by varying the pressure, temperature and specific volume of steam in circulation. These systems differ from the ordinary vacuum system in that they maintain a controllable partial vacuum on both the supply and return sides of the system, instead of only on the return side. In the vacuum system, steam pressure above that of the atmosphere exists in the supply mains and radiators practically at all times. In the sub-atmospheric system, atmospheric pressure or higher exists in the steam supply piping and radiators only during severe weather. Under average winter temperature the steam is under partial vacuum which in mild weather may reach as high as 25 in. Hg, after which further reduction in heat output is obtained by restricting the quantity of steam.

The rate of steam supply is controlled by a valve in the steam main or by thermostatically controlling the rate of steam production in the boiler.

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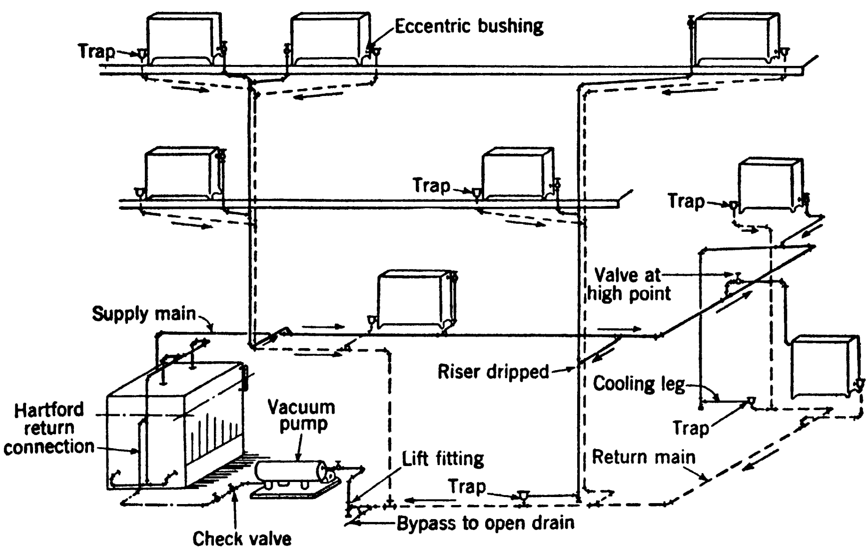

FIG. 14. TYPICAL UP-FEED VACUUM PUMP SYSTEM

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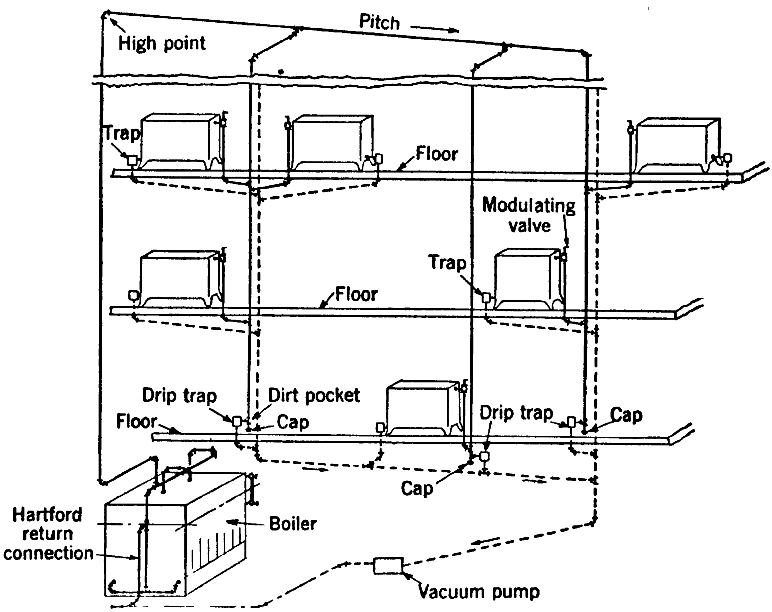


FIG. 15. TYPICAL DOWN-FEED VACUUM SYSTEM

output from the radiators. The radiator heat emission is controlled by varying the pressure, temperature and specific volume of steam in circulation. These systems differ from the ordinary vacuum system in that they maintain a controllable partial vacuum on both the supply and return sides of the system, instead of only on the return side. In the vacuum system, steam pressure above that of the atmosphere exists in the supply mains and radiators practically at all times. In the sub-atmospheric system, atmospheric pressure or higher exists in the steam supply piping and radiators only during severe weather. Under average winter temperature the steam is under partial vacuum which in mild weather may reach as high as 25 in. Hg, after which further reduction in heat output is obtained by restricting the quantity of steam.

The rate of steam supply is controlled by a valve in the steam main or by thermostatically controlling the rate of steam production in the boiler.

The control valve may be of the automatic *modulating* or *floating* type governed thermostatically from selected control points in the building, or it may be a special pressure reducing valve which will maintain the desired sub-atmospheric pressures by continuous flow into the heating main. In some systems radiator supply valves include adjustable orifices, or are equipped with regulating orifice plates. The sizes of orifices used are larger than for other types of orifice systems because, for equal radiator sizes, the volume flowing is larger. Orifices are omitted on some systems. Radiator traps and drips are designed to operate at any pressure from 15 psig to 26 in. Hg. A vacuum pump capable of operating at high vacuum is preferable to promote accuracy in the distribution of steam throughout the system, particularly in mild weather. This vacuum is partially self-induced by the condensation of the steam in the system under conditions of restricted supply used for reduction of the radiator heat emission.

The returns must grade downward constantly and uninterruptedly from the radiator return outlets to the inlet of the receiver of the vacuum pump.

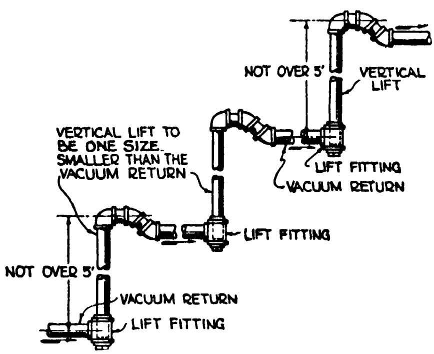


FIG. 16. METHOD OF MAKING LIFTS ON VACUUM SYSTEMS WHEN DISTANCE IS OVER 5 FT

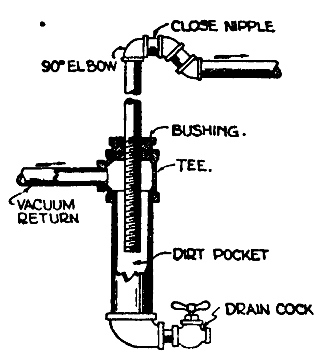


FIG. 17. DETAIL OF MAIN RETURN LIFT AT VACUUM PUMP

One radical difference between this and the ordinary vacuum system is that no lifts should be made in the return line, except at the vacuum pump. The receivers are placed at a lower level than the pump, and equipped with float control so that the pump may operate as a return pump under night conditions. The system may be operated in the same manner as the ordinary vacuum system when desired.

Steam for heating domestic hot water should be taken from the boiler header back of the control valve so that pressures sufficiently high for heating the water may be maintained on the heater. The sub-atmospheric method of heating can be used for the heating coils of ventilating and air conditioning systems. The flexible control of heat output secured by this method materially reduces the required size of by-pass around the heaters. Some applications of sub-atmospheric systems are proprietary.

TWO-PIPE ORIFICE SYSTEMS

Orifice steam heating systems may have piping arrangements identical with vacuum systems. Some of these omit the radiator thermostatic traps, but use thermostatic or combination float and thermostatic traps on all drip points. A return condensate pump with receiver vented to atmosphere, a return line vacuum pump, or a return trap, is generally used to return the condensate to the boiler or place of similar disposition, such as a feed-water heater or hot well. The heat emission from the radiators is controlled by varying the pressure differential maintained.

The principle on which these systems operate is based on the fact that the steam flow through an orifice will vary with square root of pressure drop when the ratio of the absolute pressures on the two sides of the orifice exceeds 58 percent. If the absolute pressure on the outlet side is less than 58 percent of the absolute pressure on the inlet side, no further increase in flow will be obtained as a result of the increased pressure difference. If an orifice is so designed in size as to exactly fill a radiator with 2 psig on one side and ¼ psig on the other, the absolute pressure relation is

$$\frac{14.7 + 0.25}{14.7 + 2.0} = 0.90 \text{ or } 90 \text{ percent.}$$

Should the steam pressure be dropped to ¼ psig on the supply pipe, the pressure on each side of the orifice would be balanced and no steam flow would take place. From this it will be apparent that if an orifice of a given diameter will fill a given radiator with steam when there is a given pressure on the main, reducing this steam main pressure will permit filling various desired portions of the radiator down to the point where the main pressure equals the back pressure in the radiator, provided the supply pipe pressures may be controlled sufficiently close. If orifices are designed on a similar basis for a given system and proportioned to the heating capacity of the radiators they serve, all radiators will heat proportionately to the steam pressure. The range of pressure variation is limited by the permissible noise level of the steam flowing under the pressure difference required for maximum heat output. The control of the steam supply is obtained by a valve placed in the steam main, which maintains a determined pressure, and by varying the vacuum in the return lines. The valves are frequently set manually from a remote location, guided by temperature indicating stations in the building; or thermostatically controlled from a thermostat on the roof, which automatically measures the differential of outside and inside temperatures. Since the range through which the pressures may be varied is usually from 0 to 4 psig, the control should be capable of maintaining close regulation to maintain the desired space temperatures, particularly in mild weather.

A recommended orifice schedule is shown in Table 1. Some systems use orifices not only in radiator inlets, but also at different points in the steam supply piping for the purpose of balancing the system to a greater extent. In this manner the difference between the initial and terminal pressure in the steam main may be compensated to a great extent. For example, if the initial pressure is 3 psig and the pressure at the end of the main is 2 psig, an orifice could be used in each branch for the purpose of obtaining a more uniform pressure throughout the system. Such a provision may be particularly useful in this system for branches close to the boiler where the drop in the main has not yet been produced. Some orifice systems are proprietary.

SIZING PIPING FOR STEAM HEATING SYSTEMS

The functions of the piping system are the distribution of the steam, the return of the condensate and, in systems where no local air vents are provided, the removal of the air. The distribution of the steam should be rapid, uniform and without noise, and the release of air should be facilitated as much as possible, as an air bound system will not heat readily nor properly. In designing the piping arrangement, it is desirable to maintain equivalent resistances in the supply and return piping to and from a radiator. Arranging the piping so the total distance from the boiler to the radiator is the same as the return piping distance from the heating unit back to the boiler, tends to obtain such a result. The

Table 1. Orifice Capacities for Low Pressure Steam Systems

This table is based on data from actual tests[a]

Orifice Diameter 64ths of an Inch	6 in. Hg Differential	5 in. Hg Differential	4 in. Hg Differential	2 in. Hg Differential	1 in. Hg Differential
	Capacity Expressed in Square Feet E D R				
7	18-23	16-21	15-19	10-13	
8	23-29	21-27	19-25	13-17	8-11
9	29-36	27-33	25-30	17-21	11-14
10	36-44	33-40	30-37	21-26	14-17
11	44-52	40-48	37-44	26-31	17-20
12	52-62	48-57	44-51	31-37	20-24
13	62-72	57-66	51-59	37-43	24-28
14	72-83	66-76	59-67	43-49	28-32
15	83-94	76-86	67-76	49-56	32-37
16	94-106	86-97	76-86	56-64	37-42
17	106-119	97-109	86-97	64-72	42-47
18	119-133	109-122	97-108	72-80	47-52
19	133-148	122-135	108-120	80-88	52-58
20	148-163	135-149	120-133	88-98	58-64
21	163-179	149-164	133-145	98-107	64-71
	Capacity Expressed in Pounds per Hour				
7	4.5-5.8	4.0-5.3	3.8-4.8	2.5-3.3	
8	5.8-7.3	5.3-6.8	4.8-6.3	3.3-4.3	2.0-2.8
9	7.3-9.0	6.8-8.3	6.3-7.5	4.3-5.3	2.8-3.5
10	9.0-11.0	8.3-10.0	7.5-9.3	5.3-6.5	3.5-4.3
11	11.0-13.0	10.0-12.0	9.3-11.0	6.5-7.8	4.3-5.0
12	13.0-15.5	12.0-14.3	11.0-12.8	7.8-9.3	5.0-6.0
13	15.5-18.0	14.3-16.5	12.8-14.8	9.3-10.8	6.0-7.0
14	18.0-20.8	16.5-19.0	14.8-16.8	10.8-12.3	7.0-8.0
15	20.8-23.5	19.0-21.5	16.8-19.0	12.3-14.0	8.0-9.3
16	23.5-26.5	21.5-24.3	19.0-21.5	14.0-16.0	9.3-10.5
17	26.5-29.8	24.3-27.3	21.5-24.3	16.0-18.0	10.5-11.8
18	29.8-33.3	27.3-30.5	24.3-27.0	18.0-20.0	11.8-13.0
19	33.3-37.0	30.5-33.8	27.0-30.0	20.0-22.0	13.0-14.5
20	37.0-40.8	33.8-37.3	30.0-33.3	22.0-24.5	14.5-16.0
21	40.8-44.8	37.3-41.0	33.3 36.3	24.5-26.8	16.0-17.8

Note.—The radiator orifice plates recommended in this table are made of brass stampings 0.023 in. thick cup-shaped to be inserted in radiator valve unions.

[a] Flow of Steam Through Orifices into Radiators, by S. S. Sanford and C. B. Sprenger (A.S.H.V.E. Transactions, Vol. 37, 1931, p. 371).

condensate which occurs in steam piping as well as in radiators must be drained to prevent impeding the ready flow of the steam and air. The effect of back pressure in the returns and excessive re-vaporization, such as occurs where condensate is released from pressures considerably higher than the vacuum or pressure in the return, must be avoided.

It is important that steam piping systems distribute steam not only at full *design* load, but during excess and partial loads. Usually the average winter steam demand is less than half of the demand at the *design* outside temperature. Moreover, in rapidly warming up a system even in moderate weather, the load on the steam main and returns may exceed the maximum operating load for severe weather, due to the necessity of raising the temperature of the metal in the system to the steam temperature, and the building to the *design* indoor temperature. Investigations of the return of condensate have revealed that as high as 143 percent of the design condensation rate may exist under conditions of actual operation.

The piping design of a heating system is greatly influenced by its operating characteristics. Heating systems do not operate under constant conditions, as conditions change continually, due to variation in load. As the system is being filled with steam, the pressures existing in various locations may be different from those which exist for appreciable periods at other locations, although at equilibrium conditions the pressures are approximately the same. In designing piping it is of especial importance to arrange the system to preclude trouble caused by such pressure differences. The systems which readily release the air, permit uniform

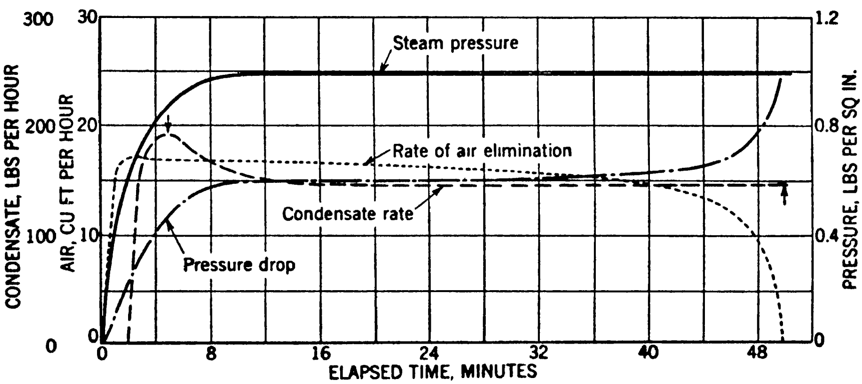


FIG. 18. RELATION BETWEEN ELAPSED TIME, STEAM PRESSURE, CONDENSATE AND AIR ELIMINATION RATES

pressures to be attained in much shorter time intervals than those which are sluggish. Results are given in Fig. 18 from investigations[1] to determine the rate of condensate and air return from a two-pipe gravity heating system. Variations in the steam pressure during the warming-up period, when the rate of air elimination and condensation is high, are clearly indicated in these curves.

It is evident that the condensate flow during the initial warming-up period reaches a peak, which is greater than the constant condensing rate eventually reached when the pressure becomes uniform. Moreover, the peak condensing rate is obtained when the system steam pressure is lower than that existing during a period of constant condensing rate. It will also be noted that the peak rate of air elimination does not coincide with the higher condensing rate.

Steam Flow

The rate of flow of dry steam or steam with a small amount of water flowing in the same direction, is in accordance with the general laws of gas flow, and is a function of the length and diameter of the pipe, the density of the steam, and the pressure drop through the pipe. This relationship

Table 2. Flow of Steam in Pipes

P = loss in pressure in pounds per square inch.
D = inside diameter of pipe in inches.
L = length of pipe in feet.
d = weight of 1 cu ft of steam.
W = pounds of steam per hour.

$$W = 5220\sqrt{\frac{PdD^5}{\left(1+\frac{3.6}{D}\right)L}}$$

$$P = 0.0000000367\left(1+\frac{3.6}{D}\right)\frac{W^2L}{dD^5}$$

Pressure Loss in Ounces	Col. 1	Pipe Size		Internal Area of Pipe Sq Inches	Col. 2	Avg Steam Press. psig	Col. 3	Length of Pipe in Feet	Col. 4
	$5220\sqrt{\frac{P}{100}}$	Nominal	Actual Internal Diameter		$\sqrt{\frac{D^5}{1+\frac{3.6}{D}}}$		$\sqrt{d}$		$\sqrt{\frac{100}{L}}$
0.25	65.28	1	1.049	0.864	0.536	−1.0[a]	0.187	20	2.240
0.50	92.28	1¼	1.380	1.496	1.178	−0.5[a]	0.190	40	1.580
1.00	130.5	1½	1.610	2.036	1.828	0.0	0.193	60	1.290
2	184.6	2	2.067	3.356	3.710	0.3	0.195	80	1.120
3	226.0	2½	2.469	4.788	6.109	1.3	0.201	100	1.000
4	261.0	3	3.068	7.393	11.183	2.3	0.207	120	0.912
5	291.8	3½	3.548	9.887	16.705	5.3	0.223	140	0.841
6	319.7	4	4.026	12.730	23.631	10.3	0.248	160	0.793
7	345.3	4½	4.506	15.947	32.134	15.3	0.270	180	0.741
8	369.1	5	5.047	20.006	43.719	20.3	0.290	200	0.710
10	412.7	6	6.065	28.886	71.762	30.3	0.326	250	0.632
12	452.0	7	7.023	38.743	106.278	40.3	0.358	300	0.578
14	488.3	8	7.981	50.027	149.382	50.3	0.388	350	0.538
16	522.0	9	8.941	62.786	201.833	60.3	0.415	400	0.500
20	583.6	10	10.020	78.854	272.592	75.3	0.452	450	0.477
24	639.3	12	12.000	113.098	437.503	100.3	0.507	500	0.447
28	690.5	14	13.250	137.880	566.693	125.3	0.557	600	0.407
32	738.2	16	15.250	182.655	816.872	150.3	0.603	700	0.378
40	825.4					175.3	0.645	800	0.354
48	904.1					200.3	0.685	900	0.333
80	1167.2							1000	0.316
160	1650.7							1200	0.289
320	2334.5							1500	0.258
480	2859.1							2000	0.224

Column 1 × 2 × 3 × 4 = lb of steam per hour that will flow through a straight pipe for a given condition.

Example 1: 1 oz drop — 2 in. pipe — 1.3 lb press. — 100 ft equivalent length:

130.5 × 3.710 × 0.201 × 1 = 97.2 lb per hour.
97.2 × 4[b] = 388.8 sq ft equivalent radiation.

Table 2 does not allow for entrained water in low-pressure steam, condensation in covered pipe and roughness in commercial pipe as found in practice.

[a] Pounds per square inch gage = 2.04 in. Vacuum, Mercury Column.
[b] The factor 4 is the approximate equivalent in square feet of steam radiation of 1 lb of steam per hour.

has been established by Babcock in the formula given at the top of Table 2. In Columns 1, 2, 3, and 4 of this table, the numerical values of the factors for different pressure losses, pipe diameters, steam densities and lengths of pipe have been worked out in convenient form so that the steam flowing in any pipe may be calculated by multiplying together the proper factors in each column, as shown in the example at the bottom of the table.

Pipe Sizes

The determination of pipe sizes for a given load in steam heating depends on the following principal factors:

1. The initial pressure and the total pressure drop which may be allowed between the source of supply and at the end of the return system.
2. The maximum velocity of steam allowable for quiet and dependable operation of the system, taking into consideration the direction of condensate flow.
3. The equivalent length of the run from the boiler or source of steam supply to the farthest heating unit.
4. The direction of flow of the condensate, whether against or with the steam.

Initial Pressure and Pressure Drop

Theoretically, there are several factors to be considered such as initial pressure and pressure required at the end of the line, but it is most important that: (1) the total pressure drop does not exceed the initial gage pressure of the system, and in actual practice it should never exceed one-half of the initial gage pressure; (2) the pressure drop is not so great as to cause excessive velocities; (3) there is a constant initial pressure, except on systems specially designed for varying initial pressures, such as the sub-atmospheric, which normally operate under controlled partial vacua and orifice and vapor systems, which at times operate under such partial vacua as may be obtained due to the condition of the fire; and (4) the rise in water due to pressure drop does not exceed the difference in level, for gravity return systems, between the lowest point on the steam main, the heating units, or the dry-return, and the boiler water line.

The present tendency in steam heating unmistakably points toward a constant lowering of initial pressures, even to those below atmospheric, and to the use of reasonably small pressure drops because a system designed in this manner will operate under higher pressures without difficulty. When a system designed for a relatively high initial pressure and a relatively high pressure drop is operated at a lower pressure, it is likely to be noisy and have poor circulation.

The total pressure drop should never exceed one-half of the initial gage pressure when condensate is flowing in the same direction as the steam. Where the condensate must flow counter to the steam, the governing factor is the velocity permissible without interfering with the condensate flow. A.S.H.V.E. Research Laboratory experiments limit this to the capacities given in Table 3 for horizontal pipes at varying grades.

Maximum Velocity

The capacity of a steam pipe in any part of a steam system depends upon the quantity of condensate present, the direction in which the condensate is flowing, and the pressure drop in the pipe. Where the quantity of condensate is limited and is flowing in the same direction as the steam, only the pressure drop need be considered. When the condensate must flow against the steam, even in limited quantity, the velocity of the steam **must not exceed limits above which the disturbance between the steam and** the counter-flowing water may produce objectionable sounds, such as water hammer, or may result in the retention of water in certain parts of the system until the steam flow is reduced sufficiently to permit the water to pass. The velocity at which such disturbances take place is a function of (1) the pipe size, whether the pipe runs horizontally or vertically; (2) the pitch of the pipe if it runs horizontally; (3) the quantity of condensate flowing against the steam; and (4) freedom of the piping from water pockets which under certain conditions act as a restriction in pipe size.

Reaming Important

Three factors of uncertainty always exist in determining the capacity of any steam pipe. The first is variation in manufacture, which apparently cannot be avoided. The second is the care used in reaming the ends of the pipe after cutting. The effect of both of these factors increases as the pipe size decreases. According to A.S.H.V.E. Research Laboratory tests, either of these factors may affect the capacity of a 1-in. pipe as much as 20 percent. The third factor is the uniformity in grading the pipe line. All of the capacity tables given in this chapter include a factor of safety. However, the factor of safety referred to does not cover abnormal defects or constrictions, *nor does it cover pipe not properly reamed.*

TABLE 3. COMPARATIVE CAPACITY OF STEAM LINES AT VARIOUS PITCHES FOR STEAM AND CONDENSATE FLOWING IN OPPOSITE DIRECTIONS[a]

Pitch of Pipe in Inches per 10 Ft. Velocity in Ft per Sec

PITCH OF PIPE	¼ IN.		½ IN.		1 IN.		1½ IN.		2 IN.		3 IN.		4 IN.		5 IN.	
Pipe Size Inches	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.	Capacity	Max. Vel.
	Capacity Expressed in Square Feet E D R															
¾	25.0	12	30.3	14	37.3	18	40.4	19	42.5	20	46.1	21	47.5	22	49.3	23
1	45.8	12	52.6	15	63.0	17	70.0	20	75.2	22	83.0	23	87.9	25	96.2	26
1¼	104.9	18	117.2	20	133.0	23	144.5	25	154.0	27	165.0	28	172.6	29	178.2	31
1½	142.6	18	159.0	21	181.0	23	196.5	25	209 3	27	224.0	28	234.8	30	242.6	31
2	236.0	19	263.5	20	299.5	23	325.5	25	346.5	27	371.5	28	388.4	29	401.1	30
	Capacity Expressed in Pounds per Hour															
¾	6.3	12	7.6	14	9.3	18	10.1	19	10.6	20	11.5	21	11.9	22	12.3	23
1	11.5	12	13.2	15	15.8	17	17.5	20	18.8	22	20.8	23	22.0	25	22.6	26
1¼	26.2	18	29.3	20	33.3	23	36.1	25	38.5	27	41.3	28	43.2	29	44.6	31
1½	35.7	18	39.8	21	45.3	23	49.1	25	52.3	27	56.0	28	58.7	30	60.7	31
2	59.0	19	65.9	20	74.9	23	81.4	25	86.6	27	92.4	28	97.1	29	100.3	30

[a] Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory.

Equivalent Length of Run

All tables for the flow of steam in pipes, based on pressure drop, must allow for the friction offered by the pipe, as well as for the additional resistance of the fittings and valves. These resistances generally are stated in terms of straight pipe; in other words, a certain fitting will produce a drop in pressure equivalent to so many feet of straight run of the same size of pipe. Table 4 gives the number of feet of straight pipe usually allowed for the more common types of fittings and valves. In all pipe sizing tables in this chapter the *length of run* refers to the *equivalent* *length of run* as distinguished from the *actual length* of pipe in feet. The length of run is not usually known at the outset; hence, it may be necessary to assume some pipe size at the start. Such an assumption frequently is considerably in error, and a more common and practical method is to assume the length of run and to check this assumption after the pipes are sized. For this purpose the length of run usually is taken as double the actual length of pipe.

TABLES FOR PIPE SIZING FOR LOW PRESSURE SYSTEMS[3]

Tables 5, 6, and 7 are based on the actual inside diameters of the pipe and the condensation of ¼ lb (4 oz) of steam per square foot of equivalent direct radiation (*abbreviated EDR*) per hour. The drops indicated are

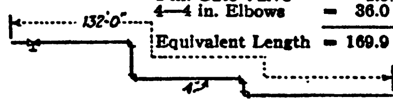
TABLE 4. LENGTH IN FEET OF PIPE TO BE ADDED TO ACTUAL LENGTH OF RUN—OWING TO FITTINGS—TO OBTAIN EQUIVALENT LENGTH

SIZE OF PIPE INCHES	LENGTH IN FEET TO BE ADDED TO RUN				
	Standard Elbow	Side Outlet Tee	Gate Valve[a]	Globe Valve[a]	Angle Valve[a]
½	1.3	3	0.3	14	7
¾	1.8	4	0.4	18	10
1	2.2	5	0.5	23	12
1¼	3.0	6	0.6	29	15
1½	3.5	7	0.8	34	18
2	4.3	8	1.0	46	22
2½	5.0	11	1.1	54	27
3	6.5	13	1.4	66	34
3½	8	15	1.6	80	40
4	9	18	1.9	92	45
5	11	22	2.2	112	56
6	13	27	2.8	136	67
8	17	35	3.7	180	92
10	21	45	4.6	230	112
12	27	53	5.5	270	132
14	30	63	6.4	310	152

[a]Valve in full open position.

Example of length in feet of pipe to be added to actual length of run.

Measured Length = 132.0 ft
4 in. Gate Valve = 1.9 ft
4—4 in. Elbows = 36.0 ft
Equivalent Length = 169.9 ft



drops in pressure per 100 ft of equivalent length of run. The pipe is assumed to be well reamed and without unusual or noticeable defects.

Table 5 may be used for sizing piping for steam heating systems by pre-determining the allowable or desired pressure drop per 100 equivalent feet of run, and reading from the column for that particular pressure drop. This applies to all steam mains on both one-pipe and two-pipe systems, vapor systems, and vacuum systems. Columns *B* to *G*, inclusive, are used where the steam and condensate flow in the same direction, while Columns *H* and *I* are for cases where the steam and condensate flow in opposite directions, as in risers and runouts that are not dripped. Columns *J*, *K*, and *L* are for one-pipe systems and cover riser, radiator valve and vertical connection sizes, and radiator and runout sizes, all of which are based on the

Table 5. Steam Pipe Capacities for Low Pressure Systems

(Reference to this table will be by column letter *A* through *L*)

This table is based on pipe size data developed through the research investigations of the American Society of Heating and Ventilating Engineers.

	Capacities of Steam Mains and Risers								Special Capacities for One-Pipe Systems Only		
	Direction of Condensate Flow in Pipe Line										
	With the Steam in One-Pipe and Two-Pipe Systems						Against the Steam Two-Pipe Only				
Pipe Size In.	1/32 psi or 1/2 Oz Drop	1/24 psi or 2/3 Oz Drop	1/16 psi or 1 Oz Drop	1/8 psi or 2 Oz Drop	1/4 psi or 4 Oz Drop	1/2 psi or 8 Oz Drop	Vertical	Horizontal	Supply Risers Up-Feed	Radiator Valves and Vertical Connections	Radiator and Riser Runouts
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*[a]	*I*[c]	*J*[b]	*K*	*L*[c]
	Capacity Expressed in Square Feet E D R										
3/4	—	—	30	—	—	—	30	—	25	—	—
1	39	46	56	79	111	157	56	34	45	28	28
1 1/4	87	100	122	173	245	346	122	75	98	62	62
1 1/2	134	155	190	269	380	538	190	108	152	93	93
2	273	315	386	546	771	1,091	386	195	288	169	169
2 1/2	449	518	635	898	1,270	1,800	635	395	464	—	260
3	822	948	1,160	1,650	2,330	3,290	1,130	700	800	—	475
3 1/2	1,230	1,420	1,740	2,460	3,470	4,910	1,550	1,150	1,140	—	745
4	1,740	2,010	2,460	3,480	4,910	6,950	2,040	1,700	1,520	—	1,110
5	3,210	3,710	4,550	6,430	9,090	12,900	4,200	3,150	—	—	2,180
6	5,280	6,100	7,460	10,550	14,900	21,100	7,200	5,600	—	—	—
8	11,000	12,700	15,500	21,970	31,070	43,900	15,000	12,000	—	—	—
10	20,000	23,100	28,300	40,100	56,700	80,200	28,000	23,000	—	—	—
12	32,000	37,100	45,500	64,300	91,000	129,000	46,000	38,000	—	—	—
16	61,000	69,700	84,800	121,000	170,000	242,000	88,000	76,000	—	—	—
	Capacity Expressed in Pounds per Hour										
3/4	—	—	8	—	—	—	8	—	6	—	7
1	10	12	14	20	28	40	14	9	11	7	7
1 1/4	22	25	31	43	61	87	31	19	20	16	16
1 1/2	34	39	48	67	95	135	48	27	38	23	23
2	68	79	97	137	193	273	97	49	72	42	42
2 1/2	112	130	159	225	318	449	159	99	116	—	65
3	206	237	291	411	581	822	282	175	200	—	119
3 1/2	307	355	434	614	869	1,230	387	288	286	—	186
4	435	503	614	869	1,230	1,740	511	425	380	—	278
5	806	928	1,140	1,610	2,270	3,210	1,050	788	—	—	545
6	1,320	1,520	1,870	2,640	3,730	5,280	1,800	1,400	—	—	—
8	2,750	3,170	3,880	5,490	7,770	11,000	3,750	3,000	—	—	—
10	5,010	5,790	7,090	10,000	14,200	20,000	7,000	5,700	—	—	—
12	8,040	9,290	11,400	16,100	22,700	32,200	11,500	9,500	—	—	—
16	15,100	17,400	21,200	30,300	42,400	60,500	22,000	19,000	—	—	—
	All Horizontal Mains and Down-Feed Risers						Up-Feed Risers	Mains and Un-dripped Run-outs	Up-Feed Risers	Radiator Connections	Run-outs Not Dripped

Note.—Steam at an average pressure of 1 psig is used as a basis for calculating capacities. All drops shown are in psi per 100 ft of equivalent run—based on pipe properly reamed.

[a] Do not use Column *H* for drops of 1/24 or 1/32 psi; substitute Column *C* or Column *B* as required.

[b] Do not use Column *J* for drop 1/32 psi except on sizes 3 in. and over; below 3 in. substitute Column *B*.

[c] Pitch of horizontal runouts to risers and radiators should be not less than 1/2 in. per ft. Where this pitch cannot be obtained, runouts over 8 ft in length should be one pipe size larger than called for in Table 5.

critical velocities of the steam to permit the counter flow of condensate without noise.

Return piping may be sized with the aid of Tables 6 and 7 where pipe capacities for wet, dry, and vacuum return lines are shown for the pressure drops per 100 ft corresponding to the drops in Table 5. *It is customary to use the same pressure drop on both the steam and return sides of a system.*

Example 2: What pressure drop should be used for the steam piping of a system if the measured length of the longest run is 500 ft, and the initial pressure is not to be over 2-psig?

Solution: It will be assumed, if the measured length of the longest run is 500 ft., that when the allowance for fittings is added, the equivalent length of run will not exceed 1,000 ft. Then, with the pressure drop not over one-half of the initial pressure, the drop could be 1 psi or less. With a pressure drop of 1 psi and a length of run of 1,000 ft, the drop per 100 ft would be $\frac{1}{10}$ psi, while if the total drop were $\frac{1}{2}$ psi, the drop per 100 ft would be $\frac{1}{20}$ psi. In the first instance the pipe could be sized according to Column *D* for $\frac{1}{16}$ psi per 100 ft, and in the second case, the pipe could be sized according to Column *C* for $\frac{1}{24}$ psi. On completion of the sizing, the drop could be checked by taking the longest line and actually calculating the equivalent length of run from the pipe sizes determined. If the calculated drop is less than that assumed, the pipe size is all right; if it is more, it is probable that there are an unusual number of fittings involved, and either the lines must be straightened or the column for the next lower drop must be used, and the lines resized. Ordinarily, resizing will be unnecessary.

TABLES FOR PIPE SIZING FOR HIGH PRESSURE SYSTEMS

Many of the recent installations of heating systems for large industrial type buildings have been designed for the use of high pressure steam, that is, without the use of pressure reducing valves. Such systems usually involve the use of unit heaters or large built-up fan units with blast heating coils. Pressures on these systems vary from 30 to 150 psi. Temperatures are controlled by a modulating or throttling type thermostatic valve controlled by the air temperature in the room, fan inlet or outlet.

Tables 8 to 11 may be used for the sizing of steam and return piping for systems of 30 and 150 psi pressure at various pressure drops. These tables are based on Babcock's formula, and have been used as the basis of design for a number of years.

SIZING PIPING FOR ONE-PIPE GRAVITY SYSTEMS

Gravity one-pipe air-vent systems, in which the equivalent length of run does not exceed 200 ft, should be sized by means of Tables 5, 6 and 7 as follows:

1. *For the steam main and dripped runouts to risers* where the steam and condensate flow in the same direction, use $\frac{1}{16}$-psi drop (Column *D*).
2. *Where the riser runouts are not dripped* and the steam and condensate flow in opposite directions, and *also in the radiator runouts* where the same condition occurs, use Column *L*.
3. *For up-feed steam risers* carrying condensate back from the radiators, use Column *J*.
4. *For down-feed systems, the main risers* of which do not carry any radiator condensate, use Column *H*.
5. *For the radiator valve size and the stub connection*, use Column *K*.
6. For the *dry-return main*, use Column *U*.
7. For the *wet-return main*, use Column *T*.

On systems exceeding an equivalent length of 200 ft, it is suggested that the total drop be not over $\frac{1}{4}$ psi. The return piping sizes should correspond with the drop used on the steam side of the system. Thus, where $\frac{1}{24}$-psi drop is being used, the steam main and dripped runouts would be

Table 6. Return Pipe Capacities for Low Pressure Systems

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter *M* through *EE*)

This table is based on pipe size data developed through the research investigations of the American Society of Heating and Ventilating Engineers.

Pipe Size Inches	Capacity of Return Mains and Risers																	
	1/32 Psi or ½ Oz Drop per 100 Ft			1/24 Psi or ⅔ Oz Drop per 100 Ft			1/16 Psi or 1 Oz Drop per 100 Ft			⅛ Psi or 2 Oz Drop per 100 Ft			¼ Psi or 4 Oz Drop per 100 Ft			½ Psi or 8 Oz Drop per 100 Ft		
	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*
	Mains																	
¾	—	—	—	—	—	326	—	—	400	—	—	568	—	—	800	—	—	1,130
1	500	248	—	580	285	570	700	320	700	1,000	412	994	1,400	460	1,400	—	—	1,980
1¼	850	520	—	990	595	976	1,200	670	1,200	1,700	868	1,700	2,400	962	2,400	—	—	3,390
1½	1,350	822	—	1,570	943	1,550	1,900	1,060	1,900	2,700	1,360	2,700	3,800	1,510	3,800	—	—	5,370
2	2,800	1,880	—	3,240	2,140	3,260	4,000	2,300	4,000	5,600	2,960	5,680	8,000	3,300	8,000	—	—	11,300
2½	4,700	3,040	—	5,300	3,470	5,450	6,700	3,800	6,700	9,400	4,900	9,510	13,400	5,450	13,400	—	—	18,900
3	7,500	5,840	—	8,500	6,250	8,710	10,700	7,000	10,700	15,000	9,000	15,200	21,400	10,000	21,400	—	—	30,200
3½	11,000	7,880	—	13,200	8,800	13,000	16,000	10,000	16,000	22,000	12,900	22,700	32,000	14,300	32,000	—	—	45,200
4	15,500	11,700	—	18,300	13,400	18,000	22,000	15,000	22,000	31,000	19,300	31,200	44,000	21,500	44,000	—	—	62,190
5	—	—	—	—	—	31,500	—	—	38,700	—	—	54,900	—	—	77,400	—	—	109,000
6	—	—	—	—	—	50,450	—	—	62,000	—	—	88,000	—	—	124,000	—	—	175,000
	Risers																	
¾	—	190	—	—	190	570	—	190	700	—	190	994	—	190	1,400	—	—	1,980
1	—	450	—	—	450	976	—	450	1,200	—	450	1,700	—	450	2,400	—	—	3,390
1¼	—	990	—	—	990	1,550	—	990	1,900	—	990	2,700	—	990	3,800	—	—	5,370
1½	—	1,500	—	—	1,500	3,260	—	1,500	4,000	—	1,500	5,680	—	1,500	8,000	—	—	11,300
2	—	3,000	—	—	3,000	5,450	—	3,000	6,700	—	3,000	9,510	—	3,000	13,400	—	—	18,900
2½	—	—	—	—	—	8,710	—	—	10,700	—	—	15,200	—	—	21,400	—	—	30,200
3	—	—	—	—	—	13,000	—	—	16,000	—	—	22,700	—	—	32,000	—	—	45,200
3½	—	—	—	—	—	17,900	—	—	22,000	—	—	31,200	—	—	44,000	—	—	62,200
4	—	—	—	—	—	31,500	—	—	38,700	—	—	54,900	—	—	77,400	—	—	109,000
5	—	—	—	—	—	50,500	—	—	62,000	—	—	88,000	—	—	124,000	—	—	175,000

TABLE 7. RETURN PIPE CAPACITIES FOR LOW PRESSURE SYSTEMS

Capacity Expressed in Pounds per Hour

(Reference to this table will be made by column letter *M* through *EE*)

This table is based on pipe size data developed through the research investigations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pipe Size Inches	CAPACITY OF RETURN MAINS AND RISERS																	
	1/32 Psi or 1/2 Oz Drop per 100 Ft			1/24 Psi or 2/3 Oz Drop per 100 Ft			1/16 Psi or 1 Oz Drop per 100 Ft			1/8 Psi or 2 Oz Drop per 100 Ft			1/4 Psi or 4 Oz Drop per 100 Ft			1/2 Psi or 8 Oz Drop per 100 Ft		
	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*
	MAINS																	
3/4	......	......	......	......	......	42	......	......	100	......	......	142	......	......	200	......	......	283
1	125	62	......	145	71	143	175	80	175	250	103	249	350	115	350	......	......	494
1 1/4	213	130	......	248	149	244	300	168	300	425	217	426	600	241	600	......	......	848
1 1/2	338	206	......	393	236	388	475	265	475	675	340	674	950	378	950	......	......	1,340
2	700	470	......	810	535	815	1,000	575	1,000	1,400	740	1,420	2,000	825	2,000	......	......	2,830
2 1/2	1,180	760	......	1,580	868	1,360	1,680	950	1,680	2,350	1,230	2,380	3,350	1,360	3,350	......	......	4,730
3	1,880	1,460	......	2,130	1,560	2,180	2,680	1,750	2,680	3,750	2,250	3,800	5,350	2,500	5,350	......	......	7,560
3 1/2	2,750	1,970	......	3,300	2,200	3,250	4,000	2,500	4,000	5,500	3,230	5,680	8,000	3,580	8,000	......	......	11,300
4	3,880	2,930	......	4,580	3,350	4,500	5,500	3,750	5,500	7,750	4,830	7,810	11,000	5,380	11,000	......	......	15,500
5	......	......	......	......	......	7,880	......	......	9,680	......	......	13,700	......	......	19,400	......	......	27,300
6	......	......	......	......	......	12,600	......	......	15,500	......	......	22,000	......	......	31,000	......	......	43,800
	RISERS																	
3/4	......	48	......	......	48	143	......	48	175	......	48	249	......	48	350	......	......	494
1	......	113	......	......	113	244	......	113	300	......	113	426	......	113	600	......	......	848
1 1/4	......	248	......	......	248	388	......	248	475	......	248	674	......	248	950	......	......	1,340
1 1/2	......	375	......	......	375	815	......	375	1,000	......	375	1,420	......	375	2,000	......	......	2,830
2	......	750	......	......	750	1,360	......	750	1,680	......	750	2,380	......	750	3,350	......	......	4,730
2 1/2	......	......	......	......	......	2,180	......	......	2,680	......	......	3,800	......	......	5,350	......	......	7,560
3	......	......	......	......	......	3,250	......	......	4,000	......	......	5,680	......	......	8,000	......	......	11,300
3 1/2	......	......	......	......	......	4,480	......	......	5,500	......	......	7,810	......	......	11,000	......	......	15,500
4	......	......	......	......	......	7,880	......	......	9,680	......	......	13,700	......	......	19,400	......	......	27,300
5	......	......	......	......	......	12,600	......	......	15,500	......	......	22,000	......	......	31,000	......	......	43,800

Table 8. Steam Pipe Capacities for 30 Psig Steam Systems[a]
Capacity Expressed in Pounds per Hour
(Steam and Condensate Flowing in Same Direction)

Pipe Size Inches	Drop in Pressure—Pounds per 100 Ft in Length					
	1/8	1/4	1/2	3/4	1	2
3/4	15	22	31	38	45	63
1	31	46	63	77	89	125
1 1/4	69	100	141	172	199	281
1 1/2	107	154	219	267	309	437
2	217	313	444	543	627	886
2 1/2	358	516	730	924	1,030	1,460
3	651	940	1,330	1,630	1,880	2,660
3 1/2	979	1,410	2,000	2,450	2,830	4,000
4	1,390	2,000	2,830	3,460	4,000	5,660
5	2,560	3,640	5,230	6,400	7,390	10,500
6	4,210	6,030	8,590	10,400	12,100	17,200
8	8,750	12,600	17,900	21,900	25,300	35,100
10	16,300	23,500	33,200	40,600	46,900	66,400
12	25,600	36,900	52,300	64,000	74,000	104,500

[a] Note: Steam at an average pressure of 30 psig is used as the basis for calculating the above table.

sized from Column *C*; radiator runouts and undripped riser runouts from Column *L*; up-feed risers from Column *J*; the main riser on a down-feed system from Column *C* (it will be noted that if Column *H* is used the drop would exceed the limit of $\frac{1}{24}$ psi); the dry return from Column *R*; and the wet-return from Column *Q*.

With a $\frac{1}{32}$-psi drop the sizing would be the same as for $\frac{1}{24}$ psi, except that the steam main and dripped runouts would be sized from Column *B*, the main riser on a down-feed system from Column *B*, the dry-return from Column *O*, and the wet-return from Column *N*.

Notes on Gravity One-Pipe Air-Vent Systems

1. Pitch of mains should not be less than ¼ in. in 10 ft.

2. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. per foot. Where this pitch cannot be obtained, runouts over 8 ft in length should be one size larger than called for in the table.

3. In general, it is not desirable to have a main less than 2 in. The diameter of the far end of the supply main should not be less than half its diameter at its largest part.

Table 9. Steam Pipe Capacities for 150 Psig Steam Systems[a]
Capacity Expressed in Pounds per Hour
(Steam and Condensate Flowing in Same Direction)

Pipe Size Inches	Drop in Pressure—Psi per 100 Ft in Length						
	1/8	1/4	1/2	3/4	1	2	5
3/4	29	41	58	71	82	116	184
1	58	82	117	143	165	233	369
1 1/4	130	185	262	320	370	523	827
1 1/2	203	287	407	497	575	813	1,290
2	412	583	825	1,010	1,170	1,650	2,600
2 1/2	683	959	1,360	1,650	1,920	2,710	4,290
3	1,240	1,750	2,480	3,020	3,500	4,940	7,820
3 1/2	1,860	2,630	3,720	4,550	5,250	7,420	11,700
4	2,630	3,720	5,260	6,430	7,430	10,500	16,600
5	4,860	6,880	9,730	11,900	13,800	19,500	30,800
6	7,960	11,300	16,000	19,500	22,600	31,900	50,400
8	16,600	23,500	33,200	40,600	47,000	66,400	105,000
10	30,800	43,400	61,700	75,600	87,300	123,000	195,000
12	48,600	68,800	97,300	119,000	138,000	194,000	307,500

[a] Note: Steam at an average pressure of 150 psig is used as the basis for calculating the above table.

TABLE 10. RETURN PIPE CAPACITIES FOR 30 PSIG STEAM SYSTEMS[a]
Capacity Expressed in Pounds per Hour

Pipe Size Inches	Drop in Pressure—Pounds per 100 Ft in Length				
	1/8	1/4	1/2	3/4	1
3/4	115	170	245	308	365
1	230	340	490	615	730
1 1/4	485	710	1,025	1,290	1,530
1 1/2	790	1,160	1,670	2,100	2,500
2	1,580	2,360	3,400	4,300	5,050
2 1/2	2,650	3,900	5,600	7,100	8,400
3	4,850	7,100	10,300	12,900	15,300
3 1/2	7,200	10,600	15,300	19,200	22,800
4	10,200	15,000	21,600	27,000	32,300
5	19,000	27,800	40,300	55,500	60,000
6	31,000	45,500	65,500	83,000	98,000

[a] Note: The above table is based on steam at pressures of 0 to 4 psig.

4. Supply mains, runouts to risers, or risers, should be dripped where necessary.
5. Where supply mains are decreased in size they should be dripped, or be provided with eccentric couplings, flush on bottom.

Example 3: Size the one-pipe gravity steam system shown in Fig. 19 assuming that this is all there is to the system, or that the riser and main shown involve the longest run on the system.

Solution: The total length of run actually shown is 215 ft. If the equivalent length of run is taken at double this, it will amount to 430 ft, and with a total drop of $\frac{1}{4}$ psi the drop per 100 ft will be slightly less than $\frac{1}{16}$ psi. It would be well in this case to use $\frac{1}{24}$ psi, and this would result in the theoretical sizes indicated in Table 12. These theoretical sizes, however, should be modified by not using a wet-return less than 2 in., while the main supply, *g-h*, if from the uptake of a boiler, should be made the full size of the main, or 3 in. Also the portion of the main *k-m* should be made 2 in. if the wet-return is made 2 in.

SIZING PIPING FOR ONE-PIPE VAPOR SYSTEMS

Piping for one-pipe vapor systems is sized so as to permit only a few ounces pressure drop in the system. Otherwise, the method follows that outlined for sizing one-pipe gravity systems.

TABLE 11. RETURN PIPE CAPACITIES FOR 150 PSIG STEAM SYSTEMS[a]
Capacity Expressed in Pounds per Hour

Pipe Size Inches	Drop in Pressure—Psi per 100 Ft in Length					
	1/8	1/4	1/2	3/4	1	2
3/4	156	232	360	465	560	890
1	313	462	690	910	1,120	1,780
1 1/4	650	960	1,500	1,950	2,330	3,700
1 1/2	1,070	1,580	2,460	3,160	3,800	6,100
2	2,160	3,300	4,950	6,400	7,700	12,300
2 1/2	3,600	5,350	8,200	10,700	12,800	20,400
3	6,500	9,600	15,000	19,500	23,300	37,200
3 1/2	9,600	14,400	22,300	28,700	34,500	55,000
4	13,700	20,500	31,600	40,500	49,200	78,500
5	25,600	38,100	58,500	76,000	91,500	146,000
6	42,000	62,500	96,000	125,000	150,000	238,000

[a] Note: The above table is based on steam at pressures of 1 to 20 psig.

TABLE 12. PIPE SIZES FOR ONE-PIPE UP-FEED SYSTEM SHOWN IN FIG. 19

PART OF SYSTEM	SECTION OF PIPE	RADIATION SUPPLIED EDR SQ FT	THEORETICAL PIPE SIZE (INCHES)	PRACTICAL PIPE SIZE (INCHES)
Branches to radiators . . .	. .	100	2	2
Branches to radiators . . .	. . .	50	1¼	1¼
Riser.	a to b	200	2	2
Riser.	b to c	300	2½	2½
Riser.	c to d	400	2½	2½
Riser.	d to e	500	3	3
Riser.	e to f	600	3	3
Runout to riser	f to g	600	3½	3½
Supply main	g to h	600	3	3
Branch to supply main	h to j	600	2½	3
Dry return main	f to k	600	1¼	2
Wet return main	k to m	600	1	2
Wet return main	m to n	600	1	2
Wet return main	n to p	600	1	2

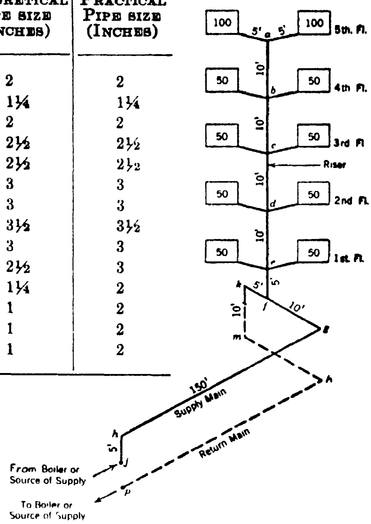


FIG. 19. RISER, SUPPLY MAIN AND RETURN MAIN OF ONE-PIPE SYSTEM

SIZING PIPING FOR TWO-PIPE HIGH PRESSURE SYSTEMS

Steam supply piping for two-pipe high pressure can be sized for greater pressure drops than that of the return piping. For a system using steam at 30 psig, the total pressure drop can be 5 to 10 psi, and for 150 psig systems, 25 to 30 psi.

It has been observed that the maximum total pressure in the returns of a 30-psig system is about 5 psig, and that of a 150-psig system is about 20 psig. The pressure in the return mains is, of course, caused by the discharge of traps and flashing of condensate into steam because the return line pressure is below that corresponding to the saturation temperature of the condensate. The usual practice in the sizing of high pressure returns has been to size on the basis of ½ psi per 100 ft of pipe for 30-psig systems, and 1 psi per 100 ft for 150-psig systems. This is an average figure which corresponds generally to several of the previously published tables for the design of high pressure return piping.

Notes on Two-Pipe High Pressure Systems

Pitch of mains should not be less than ¼ in. in 10 ft.

Pitch of horizontal runouts to risers and heating units should not be less than ½ in. per ft.

SIZING PIPING FOR TWO-PIPE LOW PRESSURE SYSTEMS

Piping for two-pipe low pressure systems is sized in the same manner as for two-pipe vapor systems, except that the pressure drop throughout the system can be based on ½ psi to 1 psi drop.

SIZING PIPING FOR TWO-PIPE VAPOR SYSTEMS

While many manufacturers of patented vapor heating accessories have their own schedules for pipe sizing, an inspection of these sizing tables indicates that in general as small a drop as possible is recommended. The reasons for this are: (1) to have the condensate return to the boiler by gravity; (2) to obtain a more uniform distribution of steam throughout the system, especially when it is desirable to carry a moderate or low fire; and (3) to prevent large variations in pressure which would nullify the value of graduated valves on radiators.

For small vapor systems where the equivalent length of run does not exceed 200 ft, it is recommended that the main and any runouts to risers that may be dripped, should be sized from Column *D*, Table 5, while riser runouts not dripped and radiator runouts should employ Column *I*. The up-feed steam risers should be taken from Column *H*. On the returns, the risers should be sized from Tables 6 and 7, Column *U* (lower portion), and the mains from Column *U* (upper portion). It should again be noted that the pressure drop in the steam side of the system is kept the same as on the return side, except where the flow in the riser is concerned.

On a down-feed system, the main vertical riser should be sized from Column *H*, but the down-feed risers can be taken from Column *D*, although it so happens that the values in Columns *D* and *H* for small systems correspond. This will not hold true in larger systems.

For vapor systems over 200 ft of equivalent length, the drop should not exceed $\frac{1}{8}$ psi to $\frac{1}{4}$ psi, if possible. Thus, for a 400 ft equivalent run the drop per 100 ft should be not over $\frac{1}{8}$ psi divided by 4, or $\frac{1}{32}$ psi. In this case the steam mains would be sized from Column *B*, the radiator and undripped riser runouts from Column *I*; the risers from Column *B*, because Column *H* gives a drop in excess of $\frac{1}{32}$ psi. On a down-feed system, Column *B* would have to be used for both the main riser and the smaller risers feeding the radiators in order not to increase the drop over $\frac{1}{32}$ psi. The return risers would be sized from the lower portion of Column *O* and the dry return main from the upper portion of the same column, while any wet returns would be sized from Column *N*. The same pressure drop is applied on both the steam and the return sides of the system.

Notes on Vapor Systems

1. Pitch of mains should not be less than $\frac{1}{4}$ in. in 10 ft.

2. Pitch of horizontal runouts to risers and radiators should not be less than $\frac{1}{2}$ in. per ft. Where this pitch cannot be obtained, runouts over 8 ft in length should be one size larger than called for in the table.

3. In general it is not desirable to have a supply main smaller than 2 in.

4. When necessary, supply main, supply risers, or runouts to supply risers should be dripped separately into a wet-return, or may be connected into the dry-return through a thermostatic drip trap.

SIZING PIPING FOR TWO-PIPE VACUUM SYSTEMS

Vacuum, atmospheric, sub-atmospheric and orifice systems are usually employed in large installations and have total drops varying from $\frac{1}{4}$ to $\frac{1}{2}$ psi. Systems in which the maximum equivalent length does not exceed 200 ft preferably employ the smaller pressure drop, while systems over 200 ft equivalent length of run, more frequently are designed for the higher drop, owing to the relatively greater saving in pipe sizes. For example, a system with 1200 ft longest equivalent length of run would employ a drop per 100 ft of $\frac{1}{2}$ psi divided by 12, or $\frac{1}{24}$ psi. In this case, the steam main would be sized from Column *C*, Table 5, and the risers also from Column *C* (Column *H* could be used as far as critical velocity is concerned, but the drop would exceed the limit of $\frac{1}{24}$ psi). Riser runouts, if dripped, would use Column *C* but, if undripped, would use Column *I*; radiator runouts, Column *I*; return risers, lower part of Column *S*, Tables 6 and 7; return runouts to radiators, one pipe size larger than the radiator trap connections.

Notes on Vacuum Systems

1. It is not generally considered good practice to exceed ⅛ psi drop per 100 ft of equivalent run, nor to exceed 1 psi total pressure drop in any system.
2. Pitch of mains should not be less than ¼ in. in 10 ft.
3. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. per ft. Where this pitch cannot be obtained, runouts over 8 ft in length should be one size larger than called for in the table.
4. In general, it is not considered desirable to have a supply main smaller than 2 in.
5. When necessary, the supply main, supply riser, or runout to a supply riser should be dripped separately through a trap into the vacuum return. A connection should not be made between the steam and return sides of a vacuum system without interposing a trap to prevent the steam from entering the return line.
6. Lifts should be avoided if possible, but when they cannot be eliminated they should be made in the manner described in this chapter.
7. No lifts can be used in orifice and atmospheric systems. In sub-atmospheric systems the lift must be at the vacuum pump.

SIZING PIPING FOR INDIRECT HEATING UNITS

Pipe connections and mains for indirect heating units are sized according to the quantity of steam condensed by each unit. The condensation per unit depends upon the entering temperature and the air velocity, and may be obtained from manufacturers' rating tables. Where two or more units are placed in series, the entering air temperature for any unit will be the leaving temperature for the preceding unit.

When the amount of condensation has been obtained for each unit, the pipe sizes should be based on the length of run and the pressure drop desired, as in the case of radiators. It is generally desirable to place the indirect heating units on a separate piping system rather than to connect them to the piping which supplies direct radiation. For type of connections see section on Connections to Heating Units.

PRESSURE REDUCING VALVES

While the illustrations given in Figs. 2 to 17 inclusive, indicate the various systems to be supplied by separate boiler plants, it is also possible to have steam supplied at high pressure by a boiler plant remotely located. In this case steam is supplied directly to the system or through a pressure reducing valve. Condensate can either be returned to the boiler plant or wasted to the sewer. The general arrangement of the systems fed through a pressure reducing valve will not vary from those illustrated with a boiler supply.

When high pressure steam is being supplied and lower steam pressures are required for heating, for domestic hot water, for utility services, etc., one or more pressure reducing valves (pressure regulators) are required.

These are used in two classes of service, one where the steam must be shut off tight to prevent the low pressure building up at time of no load, and the other where the low pressure lines will condense enough steam to offset normal leakage through the valve. In the latter case, double seated valves may be used in a manner that reduces the work required of the diaphragm in closing the valve and consequently, the size of the diaphragm. These valves also control the low pressures more closely under conditions of varying high pressures.

Valves that shut off all steam are called *dead end* type. They are single seated, and some of them have pilot operation that provides close control of the reduced pressure. If a thermostatically controlled valve is installed after, and near, a reducing valve in such a manner as to cut off the passage of steam, the dead end type should be used.

It is common practice, when the initial steam pressure is 100 psig or higher, to install two-stage reduction. If the radiation served is cast-iron, the *A.S.M.E.* code requires two reducing valves when the inlet pressure exceeds 50 psig. This makes a quieter condition of steam flow, as it is apparent that with one reduction, as for example from 150 to 2 psig, there is a smaller opening with greater velocity across the reducing valve and, consequently, more noise. A two-stage reduction also introduces a source of safety, since if one reducing valve were to build up its discharge pressure, this excess pressure would not be so great as the case might be in a one-stage reduction.

If an installation requires single seated valves and the pilot type cannot be used, it is necessary to use two-stage reduction, as single seated valves require sufficient diaphragm area to overcome the unbalanced pressure underneath the single valve. In many cases the large diameter of diaphragm required would make it impractical in construction. With a two-stage reduction, the diaphragm diameter required would be reduced. If a one-stage reduction is desired, it is necessary to use a pilot controlled pressure reducing valve, where low pressures are to be maintained closely.

In making a two-stage reduction, allowance for expansion of steam on the low pressure side of the valve should be made by increasing the pipe size. This also allows steam flow to be at a more nearly uniform velocity. Separating the valves by a distance up to 20 ft is recommended to reduce excessive hunting action of the first valve.

When the reduced pressure is approximately 15 psig or lower, the weight and lever diaphragm valve gives the best results with minimum maintenance. Above 15 psig, spring loaded diaphragm valves should be used, because of the extra weights required on weight and lever type. Pressure equalizing lines should not be connected too close to the valve. They should be connected into the bottom of the reduced pressure steam main, to allow maximum condensate to exist in the equalizing lines, or the connection can be made into the top of the main if a water accumulator is used to reduce the variation of the head of water on the diaphragm.

Care should be exercised in selecting the size of a reducing valve. The safest method is to consult the manufacturer. It is essential that sizes of piping to and from the reducing valve be such that they will pass the desired amount of steam with the maximum velocity desired. A common error is to make the size of the reducing valve the same size as that of the service, or outlet pipe size. Generally, this will make the reducing valve oversized, and bring about wire-drawing of valve and seat, due to small lift of the valve seat.

On installations where the steam requirements are relatively large and variable in mild weather or reduced demand periods, wire-drawing may occur. To overcome this condition, two reducing valves are installed in parallel, with the sizes selected on a 70 and 30 percent proportion of maximum flow. For example, if 50,000 lb of steam per hour are required, the size of one valve is on the basis of $0.7 \times 50{,}000$ lb, or 35,000 lb, and the other on the basis of $0.3 \times 50{,}000$ lb, or 15,000 lb. During the mild or reduced demand periods, steam will flow through the smaller valve only. During the remainder of the season, the larger valve is set to control at whatever low pressure is desired, and the smaller one at a somewhat lower pressure. Thus, when steam flow is not at its maximum, the smaller valve is closed, but it opens automatically when the maximum steam demand occurs, because this maximum demand creates a slight pressure drop in the service line.

The installation of reducing valves in pipe lines requires detailed planning. They should be installed to give ease of access for inspection and repair, and wherever possible, with diaphragm downward, except in cases of pilot operated valves.

There should be a by-pass around each reducing valve of size equal to one-half the size of reducing valve. The globe valve in by-pass line should be of a good type of construction, and must shut off absolutely tight. A steam pressure gage, graduated up to the initial pressure, should be installed on the low pressure side. Safety valves located on the low pressure side should be set 5 psi higher than the final pressure, but may be 10 psi higher than the reduced pressure if this reduced pressure is that of the first stage reduction of a double reduction. Strainers are sometimes installed on the inlet to the reducing valve, but are not required before a

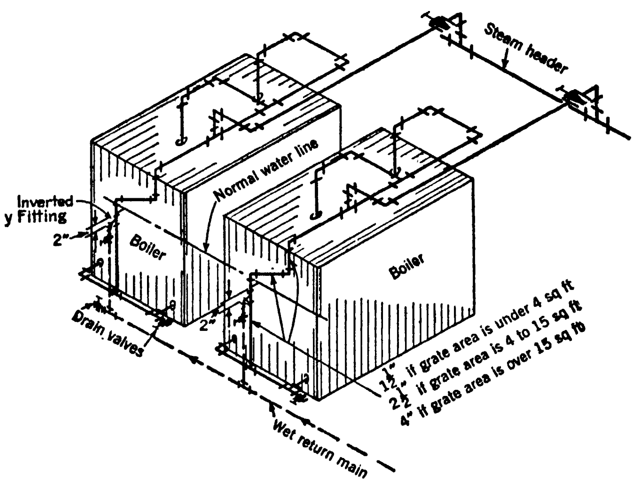


FIG. 20. THE HARTFORD RETURN CONNECTION

second-stage reduction. If a two-stage reduction is made, it is well to install a pressure gage immediately before the reducing valve of the second-stage reduction also. In sizes 3 in. and above, it is advisable to install a drip trap between the two reducing valves.

BOILER CONNECTIONS

Steam

Cast-iron, sectional heating boilers usually have several outlets in the top. Two or more outlets should be used whenever possible to reduce the velocity of the steam in the vertical uptakes from the boiler, and thus to prevent carrying of water into the steam main.

Return

Cast-iron boilers are generally provided with return tappings on both sides, while steel boilers are generally equipped with only one return tapping. Where two tappings are provided, both should be used to effect proper circulation through the boiler. The return connection should include either a Hartford return connection or a check valve to prevent the accidental loss of boiler water to the returns, with consequent danger of boiler damage. The Hartford return connection is to be preferred over the check valve, because the latter is apt to stick or not close tightly and, furthermore, because the check valve offers additional resistance to the condensate coming back to the boiler, which in gravity systems would raise the water line in the far end of the wet-return several inches.

In order to prevent the boiler from losing its water under any circumstances, the use of the Hartford return connection is recommended. This connection for a one- or two-boiler installation is shown in Fig. 20. The essential features of construction of a Hartford return connection are: (1) a direct connection (made without valves) between the steam side of the boiler and the return side of the boiler, and (2) a close nipple, or preferably an inverted Y-fitting connection about 2 in. below the normal boiler water line from the return main to the boiler steam and return pressure balance connection. Equalizing pipe connections between the steam and return are given in Fig. 20, based on grate areas, but in no case shall this pipe size be less than the main return piping from the system.

Sizing Boiler Connections

Little information is available on the sizing of boiler runouts and steam headers. Although some engineers prefer an enlarged steam header to serve as additional steam storage space, there ordinarily is no sudden demand for steam in a steam heating system, except during the heating-up period, at which time a large steam header is a disadvantage rather than an advantage. The boiler header may be sized by first computing the maximum load that must be carried by any portion of the header under any conceivable method of operation, and then applying the same schedule of pipe sizing to the header as is used on the steam mains for the building. The horizontal runouts from the boiler, or boilers, may be sized by calculating the heaviest load that will be placed on the boiler at any time, and sizing the runouts on the same basis as the building mains. The difference in size between the vertical uptakes from the boiler, which should be of same size as the boiler outlet tapping, and the horizontal main or runout, is compensated for by the use of reducing ells.

Return connections to boilers in gravity systems are made the same size as the return main itself. Where the return is split and connected to two tappings on the same boiler, both connections are made the full size of the return line. Where two or more boilers are in use, the return to each may be sized to carry the full amount of return for the maximum load which that boiler will be required to carry. Where two boilers are used, one of them being a spare, the full size of the return main would be carried to each boiler, but if three boilers are installed, with one spare, the return line to each boiler would require only half of the capacity of the entire system, or, if the boiler capacity were more than one-half the entire system load, the return would be sized on the basis of the maximum boiler capacity. As the return piping around the boiler is usually small and short, it should not be sized to the minimum.

With returns pumped from a vacuum or receiver return pump, the size of the line may be calculated from the water rate on the pump discharge when it is operated, and the line sized for a very small pressure drop. The relative boiler loads should be considered, as in the case of gravity return connections. Boiler header and piping sizes should be based on the total load.

CONDENSATE RETURN PUMPS

Condensate return pumps are used for gravity systems when the local conditions do not permit the condensate to return to the boiler under the existing static head. The return of the condensate permits the water to pass repeatedly through the cycle of vaporization, with subsequent condensation and return to the boiler. During such repeated cycles any incrustants or other substances in solution are precipitated and the water de-activated to a considerable extent, so that corrosion of a serious nature is seldom ever encountered where the condensate is repeatedly used. Serious corrosion is more frequently found in systems in which the condensate is wasted, and fresh make-up water is continually being introduced.

A generally accepted condensate pump unit for low pressure heating systems consists of a motor-driven centrifugal pump with receiver and automatic float control. Other types in use include rotary, screw, turbine and reciprocating pumps with steam turbine or motor drive, and direct-acting steam reciprocating pumps.

The receiver capacities of these automatic units should be sized so as not to cause too great a fluctuation of the boiler water line if fed directly to the boiler, and at the same time not so small as to cause too frequent operation of the unit. The usual unit provides storage capacity between stops in the receiver of approximately 1.5 times the amount of condensate returned per minute, and the pump generally has a delivery rate of 3 to 4 times the normal flow. This relation of receiver and pump size to heating system condensing capacity takes account of the peak condensation rate.

A typical installation of a motor driven automatic condensate unit is illustrated in Fig. 9.

VACUUM HEATING PUMPS

On vacuum systems, where the returns are under a vacuum, and sub-atmospheric systems, where the supply piping, radiation and the returns are under a vacuum, it is necessary to use a vacuum pump to discharge the air and non-condensable gases to atmosphere and to dispose of the condensate. Direct-acting steam-driven reciprocating vacuum pumps are sometimes used where high pressure steam is available, or where the exhaust steam from the pump can be utilized. In general, however, these have been replaced by the automatic motor-driven return line heating pump especially developed for this service. Steam turbine drive is also frequently used where steam at suitable pressures is available, the steam being used afterward for building heating. The usual vacuum pump unit consists of a compact assembly of exhausting unit for withdrawing the air-vapor mixture and discharging the air to atmosphere, and a water removal unit which discharges the condensate to the boiler. They are furnished complete with receiver, separating tank and automatic controls mounted as an integrated unit on one base. There are also special steam turbine driven units which are operated by passing the steam to be used in heating the building through the turbine with only a 2 to 3 psi drop across the turbine required for its operation. Under special conditions such as installations where it is necessary to return the condensate to a high pressure boiler, auxiliary water pumps may be supplied. In some instances separate air and water pumps may be used.

For rating purposes[3] vacuum pumps are classified as *low vacuum* and *high vacuum.* Low vacuum pumps are those rated for maintaining 5½ in. Hg vacuum on the system, and high vacuum pumps are those rated to maintain vacuums above 5½ in. Hg.

Manufacturers of vacuum pumps specify that the standard capacity of pumps shall be 0.3 to 0.5 cfm of air removal, and 0.5 gpm of water per 1000 *EDR* served. This capacity is at 5½ in. Hg of vacuum, and with condensate at 160 F. The larger air capacity is for smaller systems, and the smaller capacity for the larger systems.

Some manufacturers, however, specify more air capacity than standard where higher vacuums are desired and where air leakage is anticipated.

The vacuum that can be maintained on a system depends upon the relationship of the air leakage rate into the system to the operating air capacity of the hydraulic evacuator when operating at any given return line temperature. The hotter the returns, the lower will be the possible vacuum for a given air leakage rate into the system. It is particularly essential on high vacuum installations to see that the entire system is tight in order to reduce the amount of inward air leakage and, furthermore, to see that relatively higher temperature steam is prevented from entering the vacuum return lines through leaky traps, high pressure drips, etc. It is for this reason that the condensate from equipment using steam at high pressures should not be connected directly to a vacuum return line, but should drain to a flash tank or flash leg through a high pressure trap. The receiver should have an equalizing connection to a low pressure steam main and drain through a low pressure trap to the vacuum return main, as indicated later in this chapter in section on Drips.

Vacuum Pump Controls

In the ordinary vacuum system, the vacuum pump is controlled by a vacuum regulator which cuts in when the vacuum drops to the lowest point desired, and cuts out when it has been increased to the highest point, these points being varied to suit the particular system or operating conditions. In addition to this vacuum control, a float control is included which will start the pump whenever sufficient condensate accumulates in the receiver, regardless of the vacuum on the system. A selector switch is usually provided to allow operation at night as a condensate pump only, also to give manual or continuous operation when desired.

There are several variations in the control of the vacuum maintained on the system by the pump. In some sub-atmospheric systems where orifices are used, the vacuum pump control maintains a pressure difference between the supply and the return piping, which is held within relatively close limits. There are other sub-atmospheric systems which utilize special temperature-pressure actuated controls for maintaining the desired conditions in the return lines. Where various zones are connected to the same return main, the return vacuum must be controlled to meet the requirements of the zone operating at the lowest steam supply pressure.

Piston Displacement Vacuum Pumps

Piston displacement return vacuum heating pumps may be either electric or steam driven. Their piston speed in feet per minute should not exceed 20 times the square root of the number of inches in their stroke. They are usually supplied with an air separating tank, open to atmosphere, placed on the discharge side of the pump, and at an elevation sufficiently

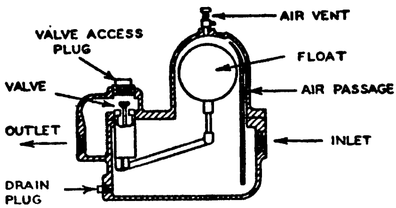


FIG. 21. SINGLE PORT FLOAT TRAP

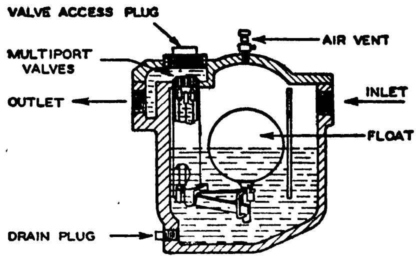


FIG. 22. MULTIPORT FLOAT TRAP

high to allow gravity flow of the condensate to the boiler. If the boiler pressure is too high for such gravity feed, then an additional steam pump for feeding the boiler is desirable. The extra pump is sometimes avoided by using a closed separating tank with a float controlled vent. In both arrangements, the air taken from the system must be discharged against the full discharge pressure of the vacuum pump. In the case of high or medium pressure boilers, it is better to use the atmospheric separator and the second pump.

In figuring the required displacement for such pumps, a value of from 6 to 10 times the volumetric flow of condensate is used for average vacuums and systems.

STEAM TRAPS

Steam traps, as the name implies, are automatic devices used to trap or hold steam in an apparatus or piping system until it has given up its latent heat, and to allow condensate and air to pass as soon as it accumulates. In general, traps consist of a vessel in which to accumulate the condensate, an orifice through which the condensate is discharged, a valve to close the orifice port, mechanisms to operate the valve, and inlet and outlet openings for the entrance and discharge of the condensate from the trap vessel.

Steam traps are classified according to the type of operating device by which they function. The traps which are available on the market today may be classified as (1) float, (2) thermostatic, (3) float and thermostatic, (4) upright bucket, (5) inverted bucket, (6) flash, (7) impulse, (8) tilting, (9) lifting, and (10) boiler return trap or alternating receiver.

Float Traps. Float traps operate by the rise and fall of a float due to a change of condensate level in the trap. When the trap is empty, the float is in its lowest position and the discharge valve is closed. As condensate accumulates in the trap

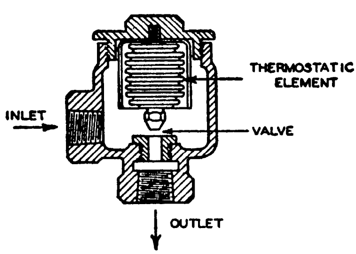


FIG. 23. THERMOSTATIC TRAP BELLOWS TYPE

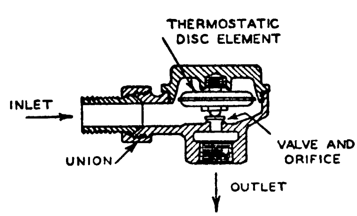


FIG. 24. THERMOSTATIC TRAP DISC TYPE

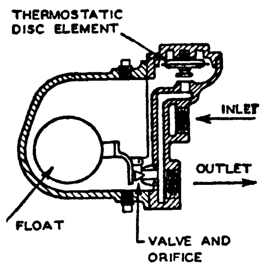


FIG. 25. TYPICAL FLOAT AND THERMOSTATIC TRAP

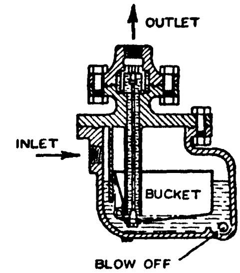


FIG. 26. UPRIGHT BUCKET TRAP

chamber, the float rises and gradually opens the valve, and the pressure of the steam pushes the condensate out of the valve. The discharge from a float trap is generally continuous, since the opening of the valve is proportioned to the flow of condensate through the trap. A gage glass may be used to indicate the height of condensate in the trap chamber. Unless float traps are well made and proportioned, there is danger of considerable steam leakage through the discharge valve due to unequal expansion of the valve and seat, and the sticking of moving parts. Float traps are made in sizes from ½ to 3 in., and for pressures varying from vacuum conditions to 200 psig. Float traps are used for draining condensate from steam separators, steam headers, blast coils, heating systems, steam water heaters, laundry equipment, sterilizers, and other equipment. When used for draining low pressure systems, float traps should be equipped with a thermostatic air vent (See Float and Thermostatic Traps). Figs. 21 and 22 illustrate types of float traps which are in use at the present time.

Thermostatic Traps. Thermostatic traps function by means of elements which expand and contract under the influence of heat and cold. Early types of thermostatic traps employed carbon posts and bi-metallic elements for expansion. In general, the modern type of thermostatic trap consists of thin corrugated metal bellows or discs enclosing a hollow chamber. The chamber is either filled with a liquid, or a small amount of a volatile liquid, such as alcohol, is introduced. The liquid expands or becomes a gas when steam comes in contact with the expansive element. The pressure created in either case expands the element and closes the valve of the trap against the escape of steam. When condensate or air comes in contact with the element, it cools and contracts, opening the valve and allowing them to escape.

The discharge from this type of trap is intermittent. Thermostatic traps find their use generally for the draining of condensate from radiators, convectors, pipe coils, drips, unit heaters, water heaters, cooking kettles, and other equipment. Except for radiators and convectors, it is recommended that a strainer be installed on the inlet connection to the trap to prevent dirt, pipe scale, and other foreign substance from entering the trap. A cooling leg of a length of pipe should also be provided ahead of the trap on unit heaters and similar apparatus to cool the condensate in order to help in the trap action. Thermostatic traps are made in sizes from ½

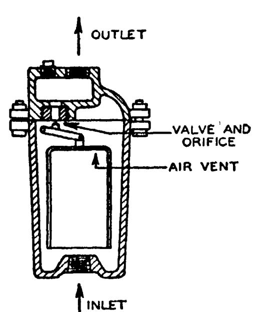


FIG. 27. INVERTED BUCKET TRAP

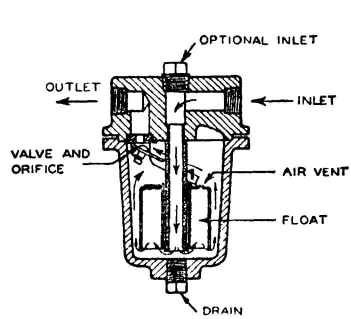


FIG. 28. INVERTED BUCKET TRAP WITH CENTRAL GUIDE

to 2 in., and for pressures ranging from vacuum conditions to 300 psig. Figs. 23 and 24 illustrate types of thermostatic traps which are available at the present time.

Float and Thermostatic Traps. This type of trap is a combination of the float trap and the thermostatic trap, and finds its use in the draining of condensate from unit heaters, blast heaters, and coil heaters for water, oil or other liquids where there is apt to be a large volume of condensate which would not permit successful operation of thermostatic traps alone. The function of the float element of this trap is to handle the condensate, and of the thermostatic element to permit the flow of air and to prevent the flow of steam around the float valve. Float and thermostatic traps are made in sizes from ½ to 2 in. and usually operate under pressures varying from vacuum conditions to 40 psig, although some are made to operate at a maximum pressure of 200 psig. Fig. 25 illustrates a typical float and thermostatic trap.

Upright Bucket Traps. In this type of trap, the condensate enters the trap chamber and fills the space between the bucket and the walls of the trap. This causes the bucket to float, and forces the valve against its seat, the valve and its stem usually being fastened to the bucket. When the condensate in the chamber rises above the edges of the bucket, it overflows into it and causes the bucket to sink, thereby withdrawing the valve from its seat. This permits the steam pressure acting on the surface of condensate in the bucket to force the water to the discharge opening. When the bucket is emptied, it rises and closes the valve and another cycle begins. The discharge from this type of trap is intermittent, and it requires a definite differential pressure (usually 1 psi at least) between the inlet and outlet of the trap in order to

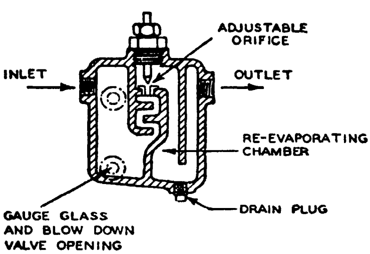


Fig. 29. Flash Trap

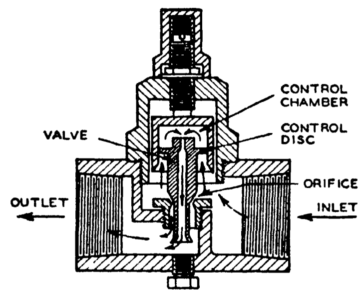


Fig. 30. Impulse Trap

lift the condensate out of the bucket to the return opening. Upright bucket traps are used for draining condensate and air from blast coils, unit heaters, steam mains, laundry equipment, sterilizers, water and oil heaters and other equipment. This type of trap is particularly suited for use where there are pulsating pressures, such as draining steam lines and separators to reciprocating pumps or engines. It is not influenced by pulsations or wide fluctuations of pressure. Upright bucket traps are obtainable in sizes varying from ½ to 2½ in., and for pressures varying from vacuum to 1200 psig. Fig. 26 illustrates an upright bucket trap.

Inverted Bucket Traps. In this type of trap, steam, condensate and air enter the trap under the bell or inverted bucket. Steam floats the inverted bucket and closes the valve. Condensate entering the trap enables the inverted bucket to fall, opening the valve. The condensate then discharges through the open valve until steam again enters and displaces the water contained in the bucket, thus restoring its buoyancy. The steam pressure entering through the open valve discharges the trap. Air is eliminated automatically by passing through the small vent hole located in the top of the inverted bucket. Inverted bucket traps for use on low pressure systems, particularly with blast coils or unit heaters, are usually furnished with a large capacity opening equipped with a bi-metallic thermostatic element which closes when heated by steam, and opens when cooled by air and condensate, allowing air to escape from the inverted bucket to the trap outlet. Inverted bucket traps are used for draining condensate and air from blast coils, unit heaters, steam drips, laundry equipment, sterilizers, steam water heaters and other equipment. They are particularly suited for draining condensate from steam lines or equipment where abnormal amounts of air must be discharged, and where there is also foreign matter such as dirt, sludge and oil draining to the trap. The discharge from inverted bucket traps, like that of the upright bucket traps, is intermittent and requires a definite differential pressure between the inlet and the outlet of the trap in order to lift the condensate from the bottom of the trap to the outlet of same. Inverted bucket traps are made in sizes from ½ to 3 in., and for pressures varying from vacuum to 2400 psig. Figs. 27 and 28 illustrate some of the types of inverted bucket traps which are available on the market at the present time.

Flash Traps. These traps depend on the property of condensate at a high pressure and temperature to flash into steam at a lower pressure. Condensate flows freely through the orifice of the trap due to the pressure difference from inlet to outlet of trap until steam enters the inlet chamber and mixes with the remaining condensate, heating the condensate and causing it to flash, thereby choking the flow through the orifice and allowing more condensate to accumulate in the trap. The discharge from flash type traps is intermittent. There are no moving parts in this type of trap. The orifice, however, is adjustable for the pressure differential required. A gage glass or float indicates whether the trap is operating. These traps can be used for draining condensate from steam water and oil heaters, blast heaters, unit heaters, dryers, vulcanizers, kitchen equipment, laundry equipment, evaporators, steam lines and other equipment, where the pressure differential between steam supply and condensate return does not drop below 5 psi. Flash type traps are made in sizes from ½ to 3 in., and for pressures varying from vacuum to 450 psig. Fig. 29 illustrates a trap of the flash type.

Impulse Traps. These traps are a modification of the flash trap, and depend on the same principle of flash for their operation. In the impulse trap the flashing action

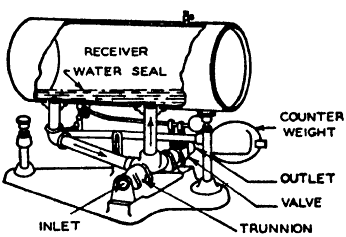


FIG. 31. TILTING TRAP

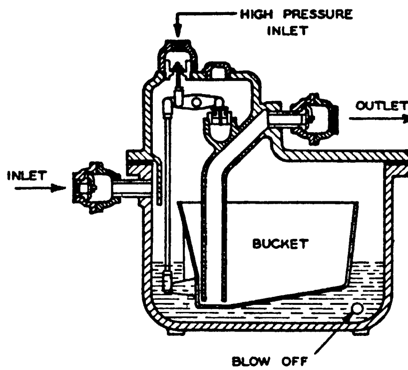


FIG. 32. LIFTING TRAP

is utilized to govern the movement of a valve by causing changes in pressure in a control chamber above the valve. A small portion of condensate, called control flow, by-passes continuously through the control chamber. At low and medium temperatures, the discharge through the center orifice reduces control chamber pressure, and the valve opens for free discharge of air and condensate.

When condensate reaches near-steam temperature, part of the control flow flashes into vapor, due to reduced pressure in control chamber. The increased volume of the condensate-vapor mixture restricts the discharge through the center orifice, and therefore the reduced pressure in control chamber builds up, closes the valve, and shuts off all discharge of hot condensate, except the small amount flowing through center orifice.

Under normal condensate loads, the valve opens and closes at short intervals. Under heavy loads, the valve opens wide and the discharge is heavy and continuous. Impulse traps can be used for draining condensate from steam mains, unit heaters, laundry equipment, kitchen equipment, oil and water heaters, sterilizers, and other equipment where the pressure at the trap outlet is 25 percent or less than that of the inlet pressure. Impulse traps are made in sizes from ½ to 2 in., and for pressures ranging from one to 600 psig. Fig. 30 illustrates a trap of the impulse type.

Tilting Traps. This type of trap as the name implies depends for its operation on the tilting of the trap receiver. When the receiver is in a horizontal position condensate accumulates until the weight of condensate overbalances that of a counterweight, when the receiver tilts. The tilting action opens the discharge valve, and steam pressure pushes the condensate out of the open discharge valve. When the receiver tank is emptied, except for a slight water seal, the receiver drops back to its

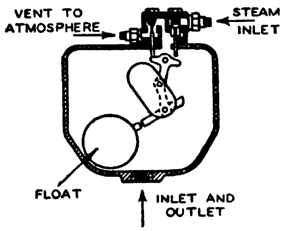


FIG. 33. BOILER RETURN TRAP OR ALTERNATING RECEIVER

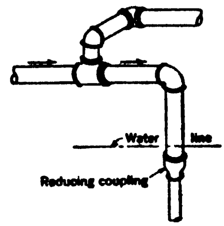


FIG. 34. DRIPPING MAIN WHERE IT RISES TO HIGHER LEVEL

horizontal position and closes the discharge valve, and is again in position to accumulate condensate.

Tilting traps are necessarily intermittent in operation, except that once the trap is in the discharge position, it will discharge condensate continuously as long as the flow of condensate is sufficient to overcome the balance of the counterweight.

This type of trap employs packing around the trunnion and valve stem in order to prevent the loss of steam and condensate. Tilting traps are used for draining laundry and dry cleaning equipment, steam cookers, drips from steam mains, steam separators and purifiers and other equipment. They are made in sizes from 1 to 3 in., and for pressures varying from 0 to 250 psig. Fig. 31 illustrates a type of tilting trap which is in use at the present time.

Lifting Traps. This type of trap is an adaptation of the upright bucket trap. It has the added feature of an auxiliary pressure inlet through which steam is introduced at a pressure higher than that of the trap inlet pressure. This high pressure steam forces the condensate to a point above the trap, and against a back pressure higher than that which is possible with normal steam pressure. Lifting traps are made in sizes from one to 3 in., and for pressures ranging from vacuum to 150 psig. Fig. 32 illustrates a trap of the lifting type.

Boiler Return Trap or Alternating Receiver. This device is not actually a steam trap in that it is not used to trap or hold steam, but is an adaptation of the lifting trap. It is used for returning condensate to a low pressure boiler, when due to excess pressure, the condensate cannot flow to the boiler by gravity without flooding the return mains, and endangering the boiler by permitting it to go dry. The boiler return trap is a vessel into which condensate alternately collects and is discharged into the boiler by boiler steam pressure. These traps are available in sizes from 1½ to 2½ in., and for pressures varying from 0 to 100 psig. A typical boiler return trap is shown in Fig. 33, and a typical connection to a low pressure heating system is indicated in Fig. 13.

Steam Trap Installations

The following general rules should govern the installation of traps of all types:

1. A vertical drip as long as possible and a strainer should be installed between the trap and the apparatus it drains. Exceptions to this rule are the installation of ther-

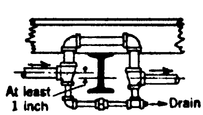


FIG. 35. LOOPING MAIN AROUND BEAM

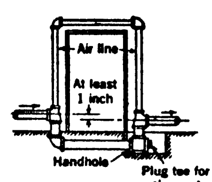


FIG. 36. LOOPING DRY RETURN MAIN AROUND OPENING

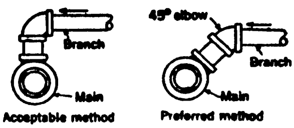
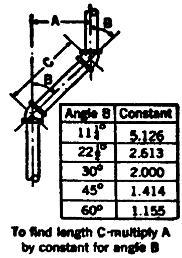


FIG. 37. METHODS OF TAKING BRANCH FROM MAIN



Angle B	Constant
11¼°	5.126
22½°	2.613
30°	2.000
45°	1.414
60°	1.155

FIG. 38. CONSTANTS FOR DETERMINING LENGTH OF OFFSET PIPE

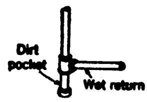


FIG. 39. DIRT POCKET CONNECTION

mostatic traps in radiators, convectors and pipe coils. These, in general, are attached directly to the units without strainers.

2. Whenever it is necessary to maintain in continuous service, apparatus which is to be drained, it is advisable to install a gate valve on each side of the trap, and a valved by-pass around the trap, so that the trap may be removed and repaired and condensate drained through the throttled by-pass valve.

3. Whenever it is necessary to install traps for lift service, as when the condensate must be discharged to a main located above the trap or where the trap must discharge against a definite back pressure, a check valve and a gate valve should be installed on the discharge side of the trap, the check valve to prevent continuous pressure on the discharge side of the valve, and the gate valve to shut off pressure in case the trap is removed for service or repair.

DRIPS

A steam main in any type of steam heating system may be dropped to a lower level without dripping if the pitch is downward with the direction of steam flow. Any steam main in any heating system can be elevated if dripped. Fig. 34 shows a connection where the steam main is raised and is drained to a wet-return. If the elevation of the low point is above a dry-return, it may be drained through a trap to the dry-return in two-pipe vapor, vacuum and sub-atmospheric systems. Horizontal steam pipes may also be run over obstructions without a change in level, if a small pipe is carried below the obstruction to care for the condensate (Fig. 35). Horizontal return pipes may be carried past doorways and other obstructions by using the scheme illustrated in Fig. 36. It will be noted that the large pipe, in this case, runs below the obstruction, and the smaller one over it.

Branches from steam mains in one-pipe gravity steam systems should use the *preferred connection* shown in Fig. 37, but where radiator condensate does not flow back into the main, the *acceptable* method shown in the same figure may be used. This acceptable method has the advantage of

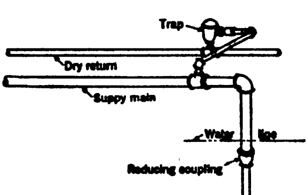


FIG. 40. DRIPPING END OF MAIN INTO WET RETURN

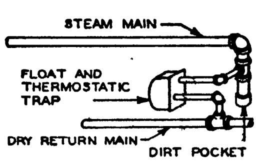


FIG. 41. DRIPPING END OF STEAM MAIN INTO DRY RETURN

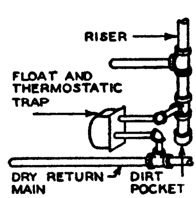


FIG. 42. DRIPPING HEEL OF RISER INTO DRY RETURN

giving a perfect swing joint when connected to the vertical riser or radiator connection, whereas the preferred connection does not give this swing without distorting the angle of the pipe. Runouts are usually made about 5 ft long to provide flexibility for movement in the main.

Offsets in steam and return piping should preferably be made with 90-deg ells, but occasionally fittings of other angles are used, and in such cases the length of the diagonal offset will be found as shown in Fig. 38.

Dirt pockets, desirable on all systems employing thermostatic traps, should be so located as to protect the traps from scale and sludge which

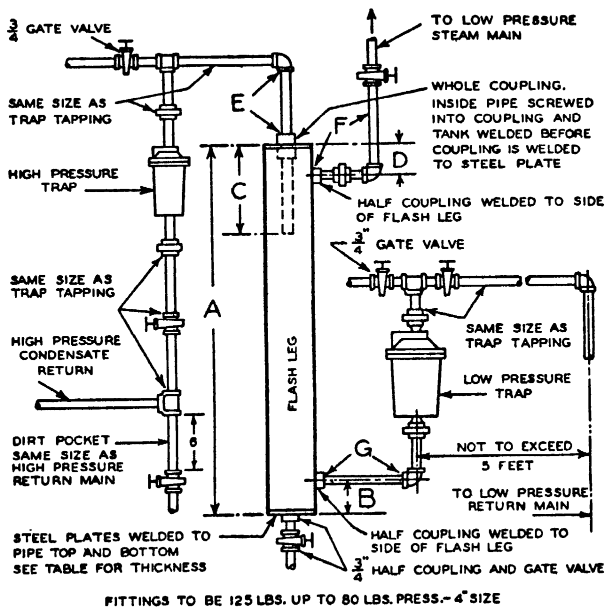


FIG. 43. FLASH LEG INSTALLATION FOR 80 PSI MAXIMUM STEAM WORKING PRESSURE

TABLE 13. DIMENSIONS APPLYING TO FIG. 43

Condensate per Hr., Lb at 70 Psi	Traps: HP Side, Pipe Size	Traps: HP Side, Orifice	Traps: HP Side, Pres. Range, psi	Traps: LP Side, Pipe Size	Traps: LP Side, Orifice	Traps: LP Side, Pres. Range, psi	Flash Leg, ips	A Ft.	B In.	C In.	D In.	E In.	F In.	G In.	Plate Thickness, In.
200	$\frac{3}{4}$	$\frac{1}{8}$	21–80	$\frac{3}{4}$	$\frac{3}{16}$	0–20	4	3	6	8	3	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{3}{8}$
300	$\frac{3}{4}$	$\frac{5}{32}$	31–70	$\frac{3}{4}$	$\frac{5}{16}$	0–15	4	3	6	8	3	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{3}{8}$
700	$\frac{3}{4}$	$\frac{3}{16}$	31–70	$\frac{3}{4}$	$\frac{3}{8}$	0–15	4	3	6	8	3	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{3}{8}$
1500	$\frac{3}{4}$	$\frac{9}{32}$	61–80	1	$\frac{1}{2}$	0–15	5	4	6	10	4	$\frac{3}{4}$	$1\frac{1}{2}$	1	$\frac{3}{8}$
2500	1	$\frac{9}{32}$	61–80	$1\frac{1}{4}$	$\frac{3}{4}$	0–15	8	5	6	10	4	1	2	$1\frac{1}{4}$	$\frac{1}{2}$
4000	$1\frac{1}{4}$	$\frac{11}{32}$	61–80	2	$1\frac{1}{16}$	0–15	10	6	6	12	5	$1\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{1}{2}$	$\frac{5}{8}$
8000	2	$\frac{9}{16}$	61–80	$2\frac{1}{2}$	$1\frac{1}{16}$	0–15	12	6	6	12	5	2	3	2	$\frac{3}{4}$
15000	$2\frac{1}{2}$	$\frac{9}{16}$	61–80	$2\frac{1}{2}$	$1\frac{1}{16}$	0–15	16	6	8	14	6	3	3	3	$\frac{7}{8}$

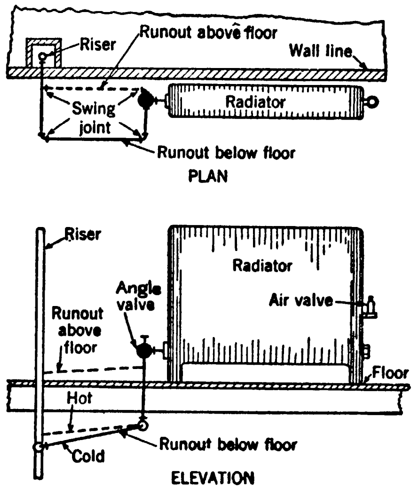


FIG. 44. ONE-PIPE RADIATOR CONNECTIONS

will interfere with their operation. Dirt pockets are usually made 8 in. to 12 in. deep, and serve as receivers for foreign matter which otherwise would be carried into the trap. They are constructed as shown in Fig. 39.

On vapor systems where the end of the steam main is dripped down into the wet-return, the air venting at the end of the main is accomplished by an air vent passing through a thermostatic trap into the dry-return line as shown in Fig. 40. On low pressure or vacuum systems, the ends of the steam mains are dripped and vented into the return through drip traps opening into the return line. A float and thermostatic type trap is recommended for dripping steam mains and risers as indicated in Figs. 41 and 42.

The dripping of high pressure mains, or of equipment using high pressure steam into low pressure or vacuum returns, is generally accomplished by the use of a flash tank or flash leg into which the high pressure trap is arranged to discharge. This tank provides the required space for the flashing from high temperature condensate to low pressure steam to take place. The low pressure steam therein generated is passed directly to the low pressure steam mains, and the condensate is discharged through a second trap to the

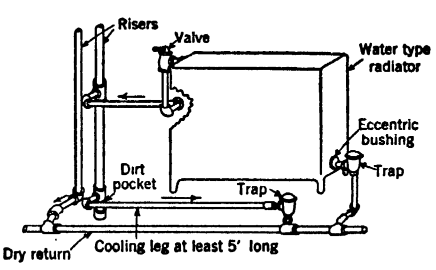


FIG. 45. TWO-PIPE TOP AND BOTTOM OPPOSITE END RADIATOR CONNECTIONS

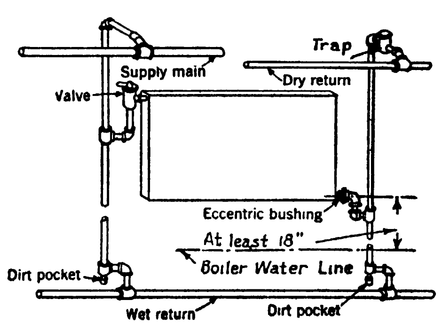


FIG. 46. TWO-PIPE CONNECTIONS TO RADIATOR HUNG ON WALL

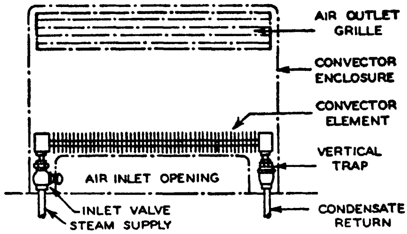


FIG. 47. TYPICAL CONVECTOR CONNECTIONS

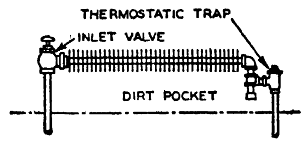


FIG. 48. TYPICAL CONNECTIONS TO FINNED PIPE CONVECTOR

low pressure or vacuum return. A typical arrangement of a flash leg, with sizes required for varying capacities, is given in Fig. 43.

CONNECTIONS TO HEATING UNITS

Riser, radiator and convector connections must not only be properly pitched at the time they are installed, but must be arranged so that the pitch will be maintained under the strains of expansion and contraction. These connections may be made by swing joints which permit the expansion or contraction to occur under heating and cooling without bending of pipes. To take care of expansion in long risers, either expansion joints of commercial construction or pipe swing joints are used. Anchoring of pipes between expansion joints is desirable.

Two satisfactory methods of making runouts for one-pipe systems for either the up-feed or the down-feed type are shown in Fig. 44. Where the vertical distance is limited and the runouts must run above the floor, the radiator may be set on pedestals or raised by means of high legs. Two methods of connecting a unit heater to a one-pipe steam heating system are illustrated in Fig. 2 (and also in Fig. 5 of Chapter 24).

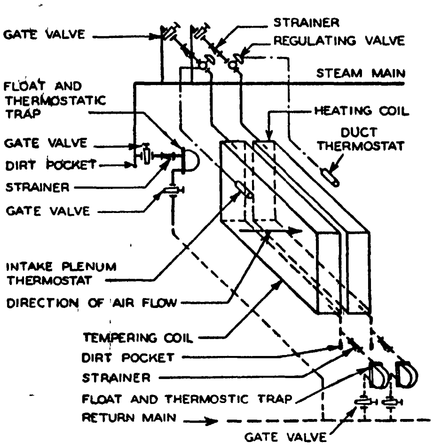


FIG. 49. TYPICAL CONNECTIONS TO FINNED TUBE BLAST HEATING COILS ARRANGED FOR SERIES FLOW OF AIR

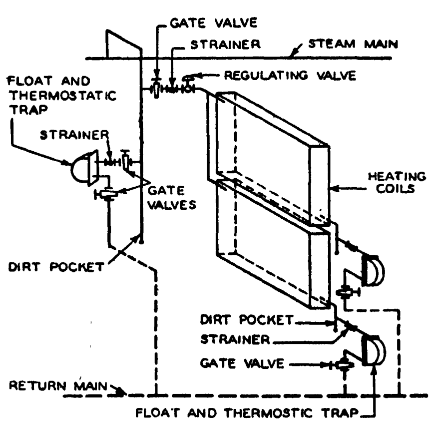


FIG. 50. TYPICAL CONNECTIONS TO FINNED TUBE BLAST HEATING COILS ARRANGED FOR PARALLEL FLOW OF AIR

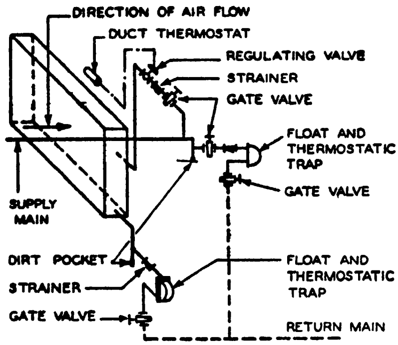


FIG. 51. TYPICAL CONNECTIONS TO FINNED TUBE BLAST HEATING COILS OF THE STEAM DISTRIBUTING OR NON-FREEZE TYPE

Typical two-pipe radiator connections are shown in Figs. 45 and 46. While these show top inlet supply connections which are preferred, it is also possible to connect the supply to the bottom of the radiator. Short radiators may be connected with top supply and bottom return on the same end.

A typical method of connecting convectors is shown in Fig. 47. Sometimes the supply valve is omitted on convector connections, and a damper is supplied in the outlet grille for heat control.

A typical connection for finned pipe convectors is shown in Fig. 48.

Typical connections to blast heaters are shown in Figs. 49, 50, and 51. Fig. 52 shows a typical return and connection for blast heaters connected to high pressure systems.

A typical two-pipe connection to a unit heater is indicated in Fig. 53.

CONTROL VALVES

Gate valves are recommended in all cases where service demands that the valve be either entirely open or entirely closed, but they should never

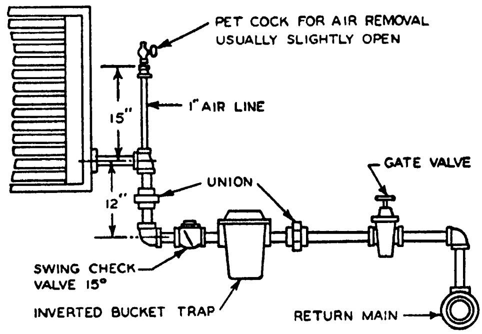


FIG. 52. TYPICAL RETURN CONNECTIONS TO FINNED TUBE BLAST HEATERS WITH HIGH PRESSURE STEAM

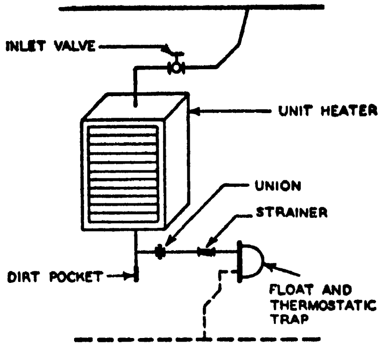


FIG. 53. TYPICAL UNIT HEATER CONNECTIONS FOR TWO-PIPE SYSTEM

be used for throttling. Angle globe valves and straight globe valves should be used for throttling in such cases as by-passes around pressure reducing valves or on by-passes around traps.

REFERENCES

[1] A.S.H.V.E. RESEARCH REPORT No. 954—Condensate and Air Return in Steam Heating Systems, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 199).

[2] Pipe size tables in this chapter have been compiled in simplified and condensed form for the convenience of the user; at the same time all of the information contained in previous editions of THE GUIDE has been retained.

[3] A.S.H.V.E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 33).

CHAPTER 21

HOT WATER HEATING SYSTEMS

Available Head; Friction Loss; Classification; System Design; Examples of Piping Design; One-Pipe Gravity, One-Pipe Forced Circulation, Two-Pipe Gravity, and Two-Pipe Forced Circulation Systems, Expansion Tanks, Installation Details, Zoning of Systems

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A HEATING system is called a hot water system if water is used to convey heat by flowing through pipes connecting a boiler or water heater with radiators, convectors or other suitable heat dispensing means. There are two types: the gravity system in which the water flows by virtue of thermo-syphon action, and the forced system in which a pump, usually driven by an electric motor, sometimes by a steam turbine or other means, maintains the necessary flow. Most panel heating systems (see Chapter 23) fall into the category of forced hot water systems, and the design procedures pertaining to pipe sizing and friction contained in this chapter are largely applicable to such systems.

Historically, the gravity system is much the older, and many such systems have been in satisfactory operation for several decades. Operation depends on the difference in density of the water due to difference in temperature in the flow and return pipes. The available head is therefore limited, and the pipes must be ample in size to permit adequate flow of water. In the forced system, the pipes, valves and fittings can be much smaller, with a resultant saving in the cost of installation, since the available head is limited only by consideration of economy in pumping the water. With the forced system, higher boiler temperatures and automatic control of the pump or circulator make possible the use of indirect water heaters with hot water systems when that is desirable. (See Chapter 48).

AVAILABLE CIRCULATION HEAD

The available head in a gravity circulation system may be found from the equation:

$$h_a = \frac{\rho_2 - \rho_1}{144} \times 2.31 \times 12{,}000 \qquad (1)$$

where

h_a = available head per foot of height, milinches (1 milinch = 1/1000 of 1 in. of water).

ρ_1 = average density of flow water, pounds per cubic foot.

ρ_2 = average density of return, pounds per cubic foot.

144 = square inches per square foot.

2.31 = height of water column equivalent to 1 psi, feet.

12,000 = milinches equivalent of 1 ft of water column.

The available head may also be found from Fig. 1. For example, at a flow temperature of 200 F and a 35 deg drop, and with the mains located 4 ft above the center line of the boiler, the available head is 600 milinches. This is found by following the 200 F flow riser line in Fig. 1 to its intersection with the 165 F return riser line, and then reading, horizontally, a head

Table 2. Heat-Carrying Capacity of Standard Black Pipes with Temperature Drop of 20 Deg[a]

Nominal Pipe Sizes 3/8 in. to 12 in., and Friction 4 to 800 milinches per foot (A = Capacity, Mbh. B = Velocity, inches per second) (One milinch equals 0.001 in.)

Milinch Friction Loss per Foot of Pipe		Nominal Pipe Size, Inches															
		3/8	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	5	6	8	10	12
4	*A*	0.75	1.35	2.85	5.4	11.3	17.0	33.0	53.1	95	141	197	363	596	1250	2320	3730
	B	1.5	1.7	2.1	2.4	2.9	3.2	3.8	4.3	5.0	5.5	6.0	7.0	7.9	9.6	11	12
6	*A*	0.9	1.7	3.6	6.75	14.0	21.2	41.3	66.4	119	176	248	456	748	1570	2920	4690
	B	1.8	2.1	2.6	3.0	3.6	4.0	4.7	5.3	6.2	6.9	7.5	8.8	10	12	14	16
8	*A*	1.05	2.0	4.2	7.9	16.4	24.8	48.4	77.9	140	207	291	535	879	1850	3440	5520
	B	2.1	2.5	3.0	3.5	4.2	4.7	5.6	6.3	7.3	8.0	8.8	10	12	14	17	19
10	*A*	1.2	2.2	4.7	8.9	18.6	28.0	54.7	88.1	158	234	329	605	997	2100	3910	6270
	B	2.4	2.8	3.4	4.0	4.8	5.3	6.3	7.1	8.2	9.1	9.9	12	13	16	19	27
12	*A*	1.35	2.45	5.2	9.8	20.5	31.0	60.4	97.4	175	259	364	671	1100	2320	4330	6950
	B	2.7	3.1	3.7	4.4	5.3	5.9	6.9	7.8	9.1	10	11	13	15	18	21	24
14	*A*	1.45	2.65	5.65	10.7	22.3	33.7	65.8	106	190	282	397	731	1200	2530	4730	7590
	B	2.9	3.4	4.1	4.8	5.7	6.4	7.6	8.5	9 9	11	12	14	16	20	23	26
16	*A*	1.55	2.85	6.05	11.5	24.0	36.3	70.8	114	205	303	428	787	1300	2730	5100	8190
	B	3.1	3.6	4.4	5.1	6.2	6.9	8.1	9.7	11	12	13	15	17	21	25	28
20	*A*	1.75	3.25	6.85	13.0	27.1	41.0	80.0	129	232	344	484	892	1470	3100	5790	9360
	B	3.5	4.1	4.9	5.8	7.0	7.7	9.2	10	12	13	15	17	20	24	28	32
25	*A*	2.0	3.65	7.75	14.7	30.6	46.3	90.5	146	263	389	548	1010	1670	3510	6570	10560
	B	4.0	4.6	5.6	6.5	7.9	8.8	10	12	14	15	17	19	22	27	32	36
30	*A*	2.2	4 0	8.55	16.2	33.8	51.2	100	162	290	430	607	1120	1850	3900	7280	11710
	B	4.4	5.1	6.1	7.2	8.7	9.7	11	13	15	17	18	22	25	30	35	40
35	*A*	2.35	4.4	9.3	17.6	36 8	55.7	109	176	316	469	661	1220	2010	4250	7940	12780
	B	4.7	5.5	6.7	7.9	9.5	11	13	14	16	18	20	23	27	33	39	44
40	*A*	2.55	4.7	10.0	18.9	39.6	59.9	117	189	341	505	712	1320	2170	4580	8570	13780
	B	5.1	5.9	7.2	8.4	10	11	13	15	18	20	22	25	29	35	42	47
50	*A*	2.85	5.3	11.3	21.4	44.7	67.7	133	214	386	572	807	1490	2460	5190	9720	15650
	B	5.7	6.7	8.1	9.5	12	13	15	17	20	22	24	29	33	40	47	54
60	*A*	3.15	5.85	12.4	23.6	49.4	74.9	147	238	427	633	893	1650	2730	5760	10780	17360
	B	6.3	7.4	8.9	11	13	14	17	19	22	25	27	32	36	44	52	60
70	*A*	3.45	6.35	13.5	25.7	53.8	81.4	160	258	465	690	973	1800	2970	6280	11760	18950
	B	6.9	8.0	9.7	11	14	15	18	21	24	27	29	35	40	48	57	65
80	*A*	3.7	6.8	14.5	27.6	57.9	87.6	172	278	500	743	1050	1940	3200	6770	12690	20440
	B	7.4	8.6	10	12	15	17	20	22	26	29	32	37	43	52	62	70
100	*A*	4.15	7.7	16.4	31.1	65.4	99.0	194	314	566	840	1190	2200	3630	7680	14400	23200
	B	8.3	9.7	12	14	17	19	22	25	30	33	36	42	48	59	70	80
150	*A*	5.2	9.6	20.4	38.8	81.6	124	243	393	709	1050	1490	2760	4560	9650	18120	29220
	B	10	12	15	17	21	23	28	32	37	41	45	53	61	74	88	101
200	*A*	6.05	11.2	23.9	45.4	95.5	145	285	461	832	1240	1750	3240	5360	11350	21320	34400
	B	12	14	17	20	25	27	33	37	43	48	53	62	71	87	104	118
300	*A*	7.5	13.9	29.7	56.6	119	181	356	577	1040	1550	2190	4060	6730	14270	26830	43300
	B	15	18	21	25	31	34	41	46	54	60	66	78	90	110	131	149
400	*A*	8.75	16.2	34.7	66.2	140	212	417	676	1220	1820	2570	4780	7910	16790	31580	51000
	B	18	21	26	30	36	40	48	54	64	71	78	92	105	129	154	175
500	*A*	9.85	18.3	39.2	74.8	158	239	471	765	1380	2060	2910	5410	8970	19040	35840	57880
	B	20	23	29	33	41	45	54	62	72	80	88	104	119	147	174	199
600	*A*	10.9	20.2	43.2	82.5	174	264	521	846	1530	2280	3220	5990	9930	21100	39740	64210
	B	22	26	32	37	45	50	60	68	80	89	97	115	132	162	193	221
800	*A*	12.7	23.6	50.5	96.5	204	310	610	992	1790	2670	3780	7030	11670	24820	46780	75620
	B	25	30	37	43	52	59	70	80	94	104	114	135	155	191	228	260

[a] For other temperature drops the pipe capacities may be changed correspondingly. For example, with a temperature drop of 30 deg. the capacities shown in this table are to be multiplied by 1.5.

alent number of elbows of the same pipe size which would have the same friction loss. An elbow is assumed to have the same friction loss as a straight pipe having a length equal to 25 diameters (nominal) of the elbow.

The resistance of various types of fittings expressed in equivalent elbow resistance is shown in Table 1.

The friction loss in pipes and tubing may be determined from Fig. 2 and Table 2 for black iron pipes, and from Fig. 3 and Table 3 for type L copper tubing. The scale at bottom of Figs. 2 and 3 is the heat carrying capacity in thousands of Btu per hour based on a 20 deg temperature difference between flow and return risers. In order to use the scale for other than 20 deg difference, refer to footnote under Figs. 2 and 3.

If the flow in a given pipe is calculated in *pounds per hour*, it may be

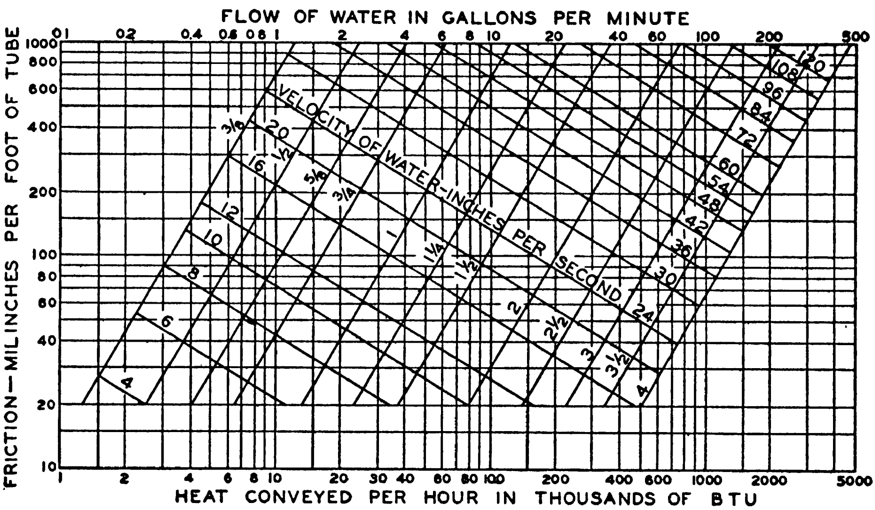

FIG. 3. FRICTION IN TYPE L COPPER TUBING

Lower scale of chart is based on 20 deg temperature difference between flow and return risers. To find friction when temperature drop is other than 20 deg, multiply the actual heat conveyed by $\left(\frac{20}{\text{actual temp. drop}}\right)$ and read the corresponding friction.

converted to corresponding *gallons per minute* by dividing the flow in pounds by 500, after which the friction may be found by entering Figs. 2 and 3 at the top scale reading which corresponds to the flow in gallons per minute as determined.

Orifices drilled in plates inserted in pipe unions are convenient means for introducing friction, where required to balance various circuits. The friction losses caused by various sizes of orifices are given in Table 4.

CLASSIFICATION OF SYSTEMS

Gravity or *forced* systems of piping may be classified according to piping arrangement and type of circulation as shown in Table 5. Flow and return main piping (gravity systems) for one-pipe, two-pipe direct return, and two-pipe reversed return systems are shown in Figs. 4, 5, and 6, re-

Table 3. Heat-carrying Capacity of Type L Copper Tubing with Temperature Drop of 20 Deg[a]

Nominal Tube Sizes ⅜ in. to 4 in., and Friction 60 to 720 milinches per foot. (A = Capacity, Mbh. B = Velocity, inches per second) (One milinch equals 0.001 in.)

Nominal Tube Size, In.		Milinch Friction Loss per Foot of Tube											
		720	600	480	360	300	240	180	150	120	90	75	60
⅜	A	10	9	8	6.8	6.2	5.4	4.6	4	3.6	3	2.8	2.4
	B	27	24	21	18	16.5	14	13	11	10	8.5	8	7
½	A	20	18	16	13.5	12	10.8	9	8	7	6	5.4	4.7
	B	35	30	25	21	19	17	15	13	12	10	9	8
⅝	A	36	30	26	22.1	20	17.8	15	13.1	11.8	9.9	9	7.9
	B	37	34	30	24	21	19	17	15	13	11	10	9
¾	A	51	46	40	34	31	28	23.2	20.5	18.1	15.3	13.9	12.1
	B	42	38	33	27	24	21	19	17	14	12	11.5	10
1	A	104	94	82	70	63	56	47	42	37	32	28	25
	B	48	45	39	34	30	25	22	19	17	14.5	13	12
1¼	A	185	169	149	125	112	100	84	75	66	56	50	44
	B	55	51	45	39	35	30	25	22	19	17	16	13
1½	A	300	270	235	200	180	160	134	120	105	90	81	71
	B	62	57	51	43	39	35	30	25	22	19	17	15
2	A	625	560	495	420	375	335	280	250	200	188	170	150
	B	76	68	59	51	47	42	36	32	27	22	20	18
2½	A	1130	1010	890	750	680	600	500	450	395	335	305	270
	B	90	80	69	58	49	47	42	37	33	26	23	21
3	A	1840	1650	1450	1210	1100	980	820	740	650	550	490	420
	B	98	90	80	66	59	52	47	42	36	30	27	23
3½	A	2750	2480	2170	1840	1650	1450	1210	1100	980	820	740	650
	B	110	100	89	75	66	57	51	45	40	35	30	26
4	A	3900	3505	3100	2600	2350	2090	1760	1580	1390	1180	1080	950
	B	120	108	96	83	73	63	55	49	44	37	34	29

[a] For other temperature drops the pipe capacities may be changed correspondingly. For example, with temperature drop of 30 deg the capacities shown in this table are to be multiplied by 1.5.

spectively. These figures would also illustrate forced circulation if a pump or circulator were shown in the return line at the boiler.

One-pipe gravity systems require very precise design owing to the small circulating head available. Also, circulation in them is slow, and temperature drop is large toward the end of the main, and consequently these systems are usually considered impractical.

One-pipe forced systems compared with gravity systems provide more rapid circulation, with consequent smaller temperature drop in mains and more uniform water temperature in all radiators, and are therefore preferred. Special flow and return fittings are available for improving the circulation to risers.

Two-pipe systems have separate flow and return mains. If the return main is *direct* as shown in Fig. 5 the radiator at the end of the system has

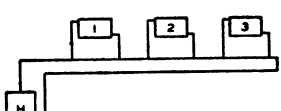

Fig. 4. One-Pipe System

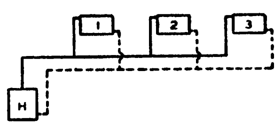

Fig. 5. A Two-Pipe Direct Return System

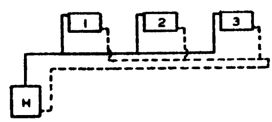

Fig. 6. A Two-Pipe Reversed Return System

TABLE 4. FRICTION (IN MILINCHES) OF CENTRAL CIRCULAR DIAPHRAGM ORIFICES IN UNIONS

(One milinch equals 0.001 in.)

Diameter of Orifices (Inches)	Velocity of Water in Pipe in Inches per Second: 2	3	4	6	8	10	12	18	24	36
					¾-in. Pipe					
0.25	1300	2900	5000	11,300	20,800	32,000	45,000			
0.30	650	1450	2500	5700	10,400	16,000	23,000	57,000		
0.35	330	740	1300	2900	5200	8000	12,000	26,000	47,000	
0.40	170	380	660	1500	2600	4000	6800	13,000	24,000	53,000
0.45		185	330	740	1300	2000	2900	6500	12,000	27,000
0.50			155	350	620	970	1400	3200	5700	13,000
0.55			75	170	300	480	700	1600	2800	6400
					1-in. Pipe					
0.35	900	2000	3500	7800	14,000	22,000	32,000			
0.40	460	1000	1800	4000	7200	12,000	17,000	37,000	65,000	
0.45	270	570	1000	2300	4100	6400	9300	21,000	37,000	
0.50	160	330	580	1400	2300	3700	5400	12,000	22,000	50,000
0.55		190	330	750	1300	2200	3000	7000	13,000	28,000
0.60			200	440	800	1300	1800	4200	7400	17,000
0.65			120	260	460	720	1100	2400	4300	10,000
					1¼-in. Pipe					
0.45	1000	2250	4000	8900	16,000	25,000	36,000			
0.50	660	1450	2600	5800	10,400	16,400	23,000	53,000		
0.55	430	950	1700	3800	6800	10,500	15,000	34,000	60,000	
0.60	280	630	1100	2500	4400	6900	10,000	22,000	40,000	
0.65	190	420	750	1700	3000	4700	6700	15,000	27,000	60,000
0.70		285	510	1150	2000	3100	4500	10,000	18,000	40,000
0.75		190	330	750	1300	2100	3000	6700	12,000	26,000
					1½-in. Pipe					
0.55	850	1900	3300	7400	13,000	21,000	30,000			
0.60	600	1300	2300	5400	8600	16,800	21,000	50,000		
0.65	400	850	1500	3600	7200	10,400	14,000	30,000	53,000	
0.70	260	600	1100	2600	4400	7000	10,000	21,000	39,000	
0.75	180	400	760	1800	3000	5000	7000	14,000	28,000	
0.80		300	540	1200	2200	3200	5000	10,200	19,000	45,000
0.85		200	380	860	1600	2300	3000	7800	13,000	30,000
					2-in. Pipe					
0.70	890	1850	3500	7400	14,000	22,300	33,000			
0.80	470	975	1800	3900	7400	11,700	17,000	37,000		
0.90	255	560	1000	2200	4200	6500	9500	20,500	38,000	
1.00	160	340	610	1320	2520	4000	5800	12,500	23,000	49,000
1.10		214	375	850	1600	2500	3700	7900	14,000	30,000
1.20			195	460	950	1360	1910	4200	8100	16,800
1.30				275	525	980	1375	3100	4400	8850

Note.—The losses of head for the orifices in the 1½-in. and 2-in. pipe were calculated from those in the smaller pipes, the calculations being based on the assumption that, for any given velocity, the loss of head is a function of the ratio of the diameter of the pipe to that of the orifice. This had been found to be practically true in the tests to determine the losses of head in orifices in ¾-in., 1-in., and 1¼-in. pipe, conducted by the Texas Engineering Experiment Station, and also in the tests to determine the losses of head in orifices in 4-in., 6-in., and 12-in. pipe, conducted by the Engineering Experiment Station of the University of Illinois. (*Bulletin* 109, Table 6, p. 38, Davis and Jordan).

the longest supply and longest return piping. The lengths of circuits to the various radiators may be equalized by using a *reversed return* main (see Fig. 6). In some cases *reversed return* mains require no more piping than *direct return* systems.

With *gravity* circulation and *direct* return piping it is necessary to design the longest circuit for the available circulating head, and to obtain the same resistance in all other circuits by proper selection of pipe sizes, by addition of fittings, or by use of orifices. When a *reversed return* system is used, it is usually found that very little adjustment is required to attain uniform distribution to all radiators.

Forced circulation in two-pipe systems, because of increased available circulating head, permits design for higher velocities with a consequent reduction in pipe sizes. The increased velocity also shortens the heating-up period and facilitates control of circulation. Reversed return mains are also advantageous in forced circulation systems, in equalizing piping resistance to all heating units.

TABLE 5. CLASSIFICATION OF HOT WATER HEATING SYSTEMS

Piping Arrangement	Type of Circulation	Expansion Tank	
One-Pipe	Gravity Forced	Open Open	Closed Closed
Two-Pipe Direct Return	Gravity Forced	Open Open	Closed Closed
Two-Pipe Reversed Return	Gravity Forced	Open Open	Closed Closed

PIPING SYSTEM DESIGN

In designing hot water heating systems certain assumptions are usually made for the purpose of simplification as follows:

1. Water temperature drop is assumed to be 30 to 35 deg for gravity systems and 20 deg for forced circulation systems. These values usually result in economical design but, particularly in large forced circulation systems, it is necessary to take into account the cost of pumping the water required at various velocities in relation to the annual charges in the capital cost of the system.

2. Water velocities in forced systems in excess of 4 fps are likely to cause disturbing noises in buildings other than factories.

3. Design outlet water temperatures in gravity systems are generally selected between 140 and 200 F (with the average approximately 180 F); while forced circulation design temperatures vary from 170 to 220 F, although higher temperatures can be used if the pressure in the system corresponds.

4. For forced circulation systems, the allowable friction loss, which is based upon the available circulating head, is determined partially by the characteristics of the pumps available.

5. Forced hot water system friction should usually be held between 600 and 250 milinches per foot. Above 600 milinches high velocities would be encountered, and below 250 milinches circulation would become too slow, so that much of the rapid response expected from forced circulation would be lost.

The water to be circulated is

$$W = H/(C \Delta t) \tag{2}$$

where

W = weight of water, pounds per hour [gallons per minute = $W/(8 \times 60)$].

H = heat required, Btu per hour.

C = specific heat of water (= 1).

Δt = drop in temperature between supply and return, Fahrenheit degrees.

The following graded series of examples of the design of hot water piping systems will illustrate the fundamental principles and methods. The differences between reversed return and direct return systems are shown, and the methods of balancing the several radiators or circuits are illustrated. A simple gravity system is shown in Fig. 7, and an elementary forced circulation system is diagrammed in Fig. 8.

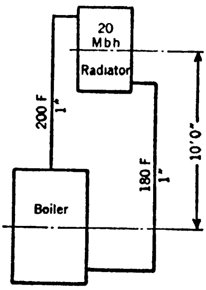

FIG. 7. GRAVITY SYSTEM

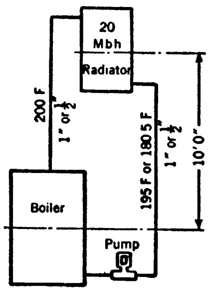

FIG. 8. FORCED CIRCULATION SYSTEM

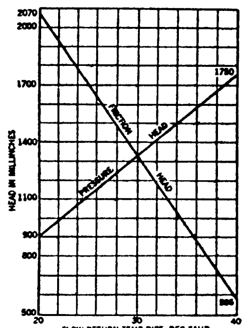

FIG. 9. DETERMINATION OF REQUIRED TEMPERATURE DIFFERENCE

Elementary Gravity System

Example 1: A simple gravity-circulation system is illustrated in Fig. 7 with one radiator that is giving off heat at the rate of 20,000 Btu per hr or 20 Mbh. The boiler imparts heat to the water at the same rate, and the water circulates at a uniform velocity. This uniform velocity is such that the friction of the circuit is equal to the head developed by the difference in density between the supply and return water and the height of the system. The circuit consists of 1 boiler, 1 radiator, 2 ells, 1 radiator valve and a total of 24 ft of pipe.

Solution: With the average water temperatures of 200 and 180 F in the supply and return risers, respectively, the head will be 90 milinches per foot of water column. This head may be found from Fig. 1. Since the center of the radiator is 10 ft above the center of the boiler, the total head of the circuit is 10 x 90, or 900 milinches, or 0.9 in. of 190 F water. The friction of the circuit must then also be 900 milinches. The friction of 1 ft of 1 in. pipe is found from Fig. 2 to be about 46 milinches at 20 Mbh, and the corresponding velocity 9 in. per second. Note that all values in Fig. 2 are based on a temperature difference of 20 deg.

Similarly, if a 1¼ in. pipe were to be used, the friction head would be about 12 milinches per foot, and the corresponding velocity about 5 in. per second, from Fig. 2.

To find the friction in the elbows, boiler, radiator, and valve, Table 1 is used, and the entire circuit is found to be equal to 10 elbow-equivalents plus 24 ft of pipe. Each elbow-equivalent is equal to a pipe length of 25 times the nominal diameter. Then the equivalent lengths of straight pipe are 45 ft of 1 in. pipe or 50 ft of 1¼ in. pipe. In many cases, it is sufficiently accurate to add 50 percent to the total pipe length to correct for resistance of fittings.

Hence, if 1 in. pipe is used, the friction of the circuit will be 45 × 46, or 2070 milinches, and if 1¼ in. pipe is used, the friction will be 50 × 12, or 600 milinches. A 1 in. pipe would, therefore, be too small and a 1¼ in. pipe too large to permit the desired circulation with a flow-return temperature difference of 20 deg.

If the circuit is of 1 in. pipe, the circulation will take place with a temperature difference greater than 20 deg, and if the circuit is of 1¼ in. pipe, the circulation will take place with a temperature difference smaller than 20 deg. To find, for example, the temperature difference at which a circuit of 1 in. pipe would transmit the required 20 Mbh, assume the difference to be 40 deg.

From Fig. 1, the head available for producing circulation would be 175 milinches per foot, or 1750 for the system, for a temperature drop from 200 to 160 F. The friction of the system may be found from Fig. 2; the chart of this figure is based on a temperature difference of 20 deg; if the temperature difference were 40 deg, the heat conveyed would be twice that shown in the chart. Hence, find 10 Mbh on the lower scale, proceed vertically upward to the intersection with the 1 in. line, and from there to the left scale read 13 milinches per foot. Note that the velocity would then be only about 5 in. per second. The total friction would then be 45 × 13 or 585 milinches. Since the head would be 1750, circulation would take place with a temperature difference less than 40 deg. The required temperature difference may be determined by constructing the diagram of Fig. 9, from which it appears that the temperature difference with which the 1 in. pipe circuit would function is about 30 deg. Hence, if the

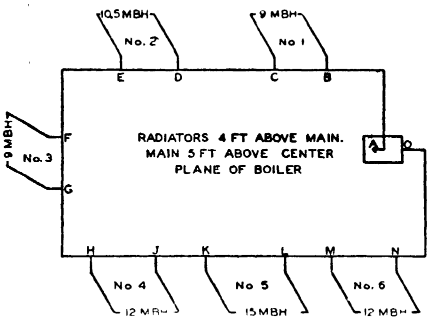

FIG. 10. ONE-PIPE GRAVITY CIRCULATION SYSTEM (EXAMPLE 3)

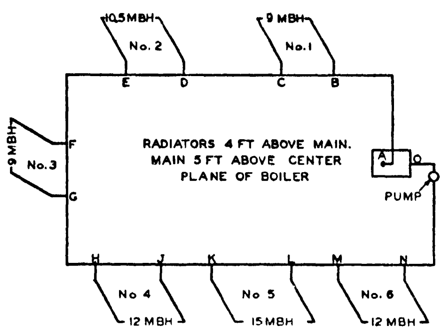

FIG. 11 ONE-PIPE FORCED CIRCULATION SYSTEM (EXAMPLE 4)

flow riser temperature is 200, the return riser temperature will be 170, and the average water temperature in the radiator about 185 F.

Elementary Forced Circulation System

Example 2: Design a system for the piping arrangement shown in Fig. 8, accordint to one of the outlined procedures. The procedure may be as follows: Assume the head developed by the circulating pump and the pipe size and, find the flow-return temperature difference; or, assume the head developed by the pump and the flow-return temperature difference, and find the pipe size; or, assume the pipe size and the flow-return temperature difference, and find the head which the circulating pump must develop.

Solution: Assume that the circulating pump will develop a head of 2 ft or 24,000 milinches and that a 1 in. pipe is to be used. The equivalent length of the circuit will then be 45 ft, as in Fig. 7, and the available head will be 24,000/45, or 533 milinches per foot. In Fig. 2, find 533 on the left scale, move horizontally to the intersection with the 1 in. pipe line, and read about 77 Mbh delivered by the pipe (with a velocity of about 35 in. per second) for a temperature difference of 20 deg. Since the circuit is to deliver only 20 Mbh, the temperature difference will be 20 divided by 77 and multiplied by 20, or 5.2 deg. Hence, if the flow riser temperature is 200, the return riser temperature will be about 195, and the average water temperature in the radiator about 197.5 F.

If a ½ in. pipe were used instead of a 1 in., the equivalent length of circuit would be 35 ft instead of 45; the unit head, 686 milinches instead of 533; the velocity, 27 in. per second instead of 35; the temperature difference, 19.5 instead of 5.2; and the average water temperature in the radiator, about 190.5 instead of 197.5 F.

If the 1 in. pipe is used for the circuit, the gravity head will be 22 milinches per foot, or 220 for the circuit (Fig. 1, 200 to 195). Since this is only 1 percent of the pump head (24,000 milinches), it may be neglected in the calculation, as was done previously. However, there are cases in which the gravity head is so large compared with the pump head that it should be included in the calculation.

The methods just described for the design of the two elementary systems are fundamental, and apply to the design of all hot water heating systems. In every system, however large and complicated, the pipe system must be such that the head forcing the water from the boiler to any one radiator is equal to the friction in that radiator's circuit when the radiator is receiving its proper quantity of hot water, and the system is functioning at a steady rate.

Other examples illustrating design of various systems follow.

One-Pipe Gravity Circulation System

Example 3: Select pipe sizes for the one-pipe gravity system having a total load of 67,500 Btu, shown in Fig. 10. Assume: flow temperature 190 F, return temperature 160 F, mains 5 ft above datum plane of boiler, center plane of radiators 4 ft above the mains, length of main 100 ft.

Solution: From Fig. 1 the available circulating head for 190 F flow and 160 F return temperature is 126 milinches per foot of height. The available circulating head for design of the main is therefore 5 × 126 = 630 milinches. The measured length of main, plus 50 percent added for resistance of fittings, equals 150 ft equivalent length

The main can then be designed for a friction loss of 630 ÷ 150 = 4 milinches per foot. From Table 2 at 4 milinch friction loss, a 2 in. pipe will supply 33 Mbh and a 2½ in. pipe will supply 53.1 Mbh at 20 deg drop. This is equivalent at 30 deg drop to 49.5 Mbh for 2 in., and 79.6 Mbh for 2½ in. pipe. A 2½ in. main will therefore be selected, and the pressure drop will be somewhat less than 4 milinches per foot.

The piping from main to radiators is sized in a similar manner. Assume that water reaches point B, Fig. 10, at 190 F and has a 30 deg drop in the radiator circuit. From Fig. 1 the available head is 126 milinches per foot of height or a total of 4 × 126 = 504 milinches for the circuit (with the radiator 4 ft above the main).

The measured length of piping is 11 ft and the fittings add 14 elbow equivalents (which would be equivalent to 22 ft if the pipe size is assumed to be ¾ in.); the equivalent length is therefore 33 ft. The circuit can therefore be designed for a friction loss of 504 ÷ 33 = 15 milinches per foot.

From Table 2 by interpolation a ¾ in. pipe would supply 5.85 Mbh at 20 deg drop or 8.78 Mbh at 30 deg drop. Since the load is 9 Mbh, the ¾ in. size will be satisfactory.

The remaining radiator circuits may be sized in a similar manner. Allowance should be made in one-pipe gravity systems for the drop in temperature which occurs in the supply main as the cooler water returns from the radiators. The drop will be in the same proportion to the total drop of 30 deg as the load supplied to any point in the main bears to the total system load, *e.g.*, the temperature at D will be $190 - \left(\frac{9000}{67{,}500} \times 30\right) = 186$ F. At point F the temperature will be $190 - \left(\frac{9000 + 10{,}500}{67{,}400} \times 30\right) = 181$ F.

One-Pipe Forced Circulation System

Example 4: Select pipe sizes for the one-pipe forced circulation system having a load of 67,500 Btu shown in Fig. 11. Assume a water temperature drop of 20 deg. The water temperature does not affect the size of piping, but does affect the radiator sizes required.

Solution: The water to be circulated at 20 deg drop will be 67,500 ÷ 20 = 3375 lb per hour or $\frac{3375}{8 \times 60} = 7$ gpm.

By reference to manufacturers' pump capacity charts (typical example, Fig. 12), it will be found that a 1 in. pump will deliver 7 gpm against a head of 4½ ft (54,000) milinches).

Since the main from A to O has an equivalent length of 150 ft (100 ft actual length plus 50 percent added for friction loss in fittings), the main may be sized for 54,000/150 = 360 milinches per foot.

From Table 2 by interpolation at 360 milinches friction loss and at 20 deg drop, a 1 in. pipe would supply 62,600 Btu per hour, and a 1¼ in. pipe would supply 131,600. Since the 1 in. pipe is too small, a 1¼ in. pipe will be used.

Since the 1¼ in. pipe offers less than 360 milinches resistance per foot, the velocity of water will increase until the output of the pump and the friction loss are in equilibrium at some point on the pump performance curve, for instance, at 10 gpm and a head of 4 ft or 48,000 ÷ 150 = 320 milinches per foot of pipe. The friction loss in the main between flow and return connections to radiators will be assumed to be 320 milinches per foot.

In determining sizes for the piping from the main to any radiator, the resistance in the radiator circuit such as B-C (which has a load of 9 Mbh) is made equal to the

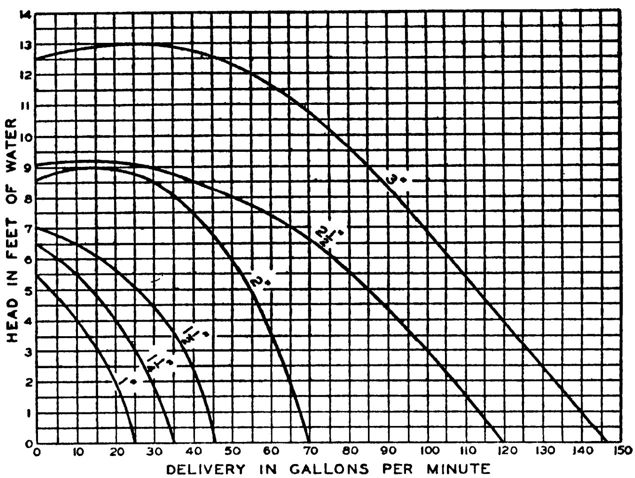

Fig. 12. Performance Chart for Circulating Pump

resistance in the main from B to C which, if there are 3 ft of main between connections, is 3 × 320 = 960 milinches. If the total equivalent length of the radiator circuit determined by use of Table 1 is 32 ft, the radiator circuit B-C will be sized for a friction loss of 960 ÷ 32 = 30 milinches per foot, for which in Table 2 a ¾ in. pipe is found to supply 8550 Btu per hr, and will be considered ample.

Other radiator circuits such as D-E, F-G, etc., can be sized in a similar manner.

Two-Pipe Gravity System (with Reversed Return)

Example 5: Select pipe sizes for the two-pipe gravity system shown in Fig. 13. The center plane of the highest radiator is 8 ft above the center plane of the boiler. Assume a 180 F flow temperature and a 150 F return temperature.

Solution: The piping should be sized so that the frictional resistance at the desired rate of flow is equal to the available circulating head.

From Fig. 1 at 180 F flow and 150 F return temperature, the available head is 118 milinches per foot of height or 8 × 118 = 944 milinches total for the highest radiator. The longest circuit from boiler to radiator and back to boiler must therefore have a resistance of 944 milinches. The longest circuit (see Fig. 13) is A-D + D-H + H-N containing 38 ft of pipe and, if 50 percent is added for equivalent length of fittings, the equivalent length is 57 ft.

The circuit should then be designed for a friction loss of 944 ÷ 57 = 16 milinches per foot (approximately). The pipe size may be found from Table 2 at 16 milinches per foot, but since the temperature drop is 30 deg, find pipe size corresponding to 20/30 of actual load for each section as follows:

Section	Load Mbh	Size of Pipe for 16 Milinches per Foot	Section	Load Mbh	Size of Pipe for 16 Milinches per Foot
A-B	58	1½	G-H	11	1
B-C	31	1¼	H-K	19	1¼
C-D	20	1¼	K-L	25	1¼
D-E	16	1	L-M	31	1¼
E-F	6	¾	M-N	58	1½

Piping to the radiators may be sized from Table 2 for the same resistance, 16 milinches per foot, but since the temperature drop is 30 deg, find pipe size corresponding to 20/30 of actual load for each section as follows:

Radiator	#1	#2	#3	#4	#5	#6
Load, Mbh	11	8	6	6	24	3
Pipe size, In	1	¾	¾	¾	1¼	½

A hot water heating system will adjust its rate of flow until the friction loss balances the available head. It is therefore self-correcting in regard to small errors made in selection of pipe sizes.

Two-Pipe Forced Circulation System

Example 6: Select pipe sizes for the two-pipe forced circulation reversed return system having a total load of 159 Mbh shown in Fig. 14. Assume a difference of 20 deg in supply and return water temperature. The total equivalent length of the longest circuit is 180 ft. The gravity circulating head due to difference in temperature may be disregarded in design.

Solution: The water to be circulated is 159,000 ÷ 20 = 7950 lb per hr or $\frac{7950}{60 \times 8}$ = 16.5 gpm. From a pump performance chart such as Fig. 12 it is found that 16.5 gpm will be delivered by a 1 in. pump against a 3 ft head (36,000 milinches) or a 1¼ in. pump against a 4.5 head (54,000 milinches).

The longest circuit, including the supply and return main and the longest radiator circuit, is 120 ft and, if 50 percent is added for friction loss in fittings, the equivalent length is 180 ft. If the 1 in. pump is used, the piping will be sized for 36,000/180 = 200 milinches per foot, resulting in selection from Table 2 of a 2 in. main for the Section A-B which supplies 159 Mbh. The large difference in pump and main size, as well as the low velocity resulting from the 200 milinch per foot friction loss, indicates that the 1¼ in. pump should be considered. The design friction loss, if the 1¼ in. pump is used, can be 54,000/180 = 300 milinches per foot and, at this friction loss, Table 2 will indicate the pipe sizes for the various sections in Fig. 14 as follows:

Supply			Return		
Section	Mbh	Pipe Size, In.	Section	Mbh	Pipe Size, In.
A-B	159	1½	J-K	16	¾
B-C	91	1¼	K-L	28	¾
C-D	75	1¼	L-M	42	1
D-E	63	1¼	M-N	54	1
E-F	49	1	N-O	75	1¼
F-G	37	1	O-P	91	1¼
G-H	16	¾	P-Q	159	1½

The radiator circuits may also be sized for the same friction loss, 300 milinches per foot, using Table 2 as follows:

Radiator	#1	#2	#3	#4	#5 and #6	#7
Load, Mbh..	16	12	14	12	21	16
Pipe size, In . .	¾	½	½	½	¾[a]	¾

[a] Where circuit divides, use ¾ in. branch to #5 and ½ in. to #6 radiator.

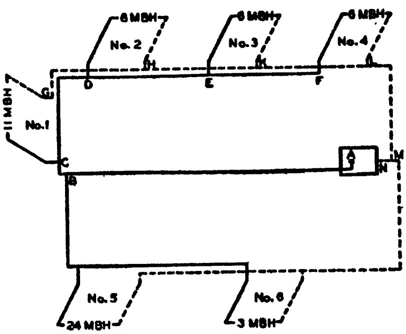

FIG. 13. TWO-PIPE REVERSED RETURN GRAVITY SYSTEM (EXAMPLE 5)

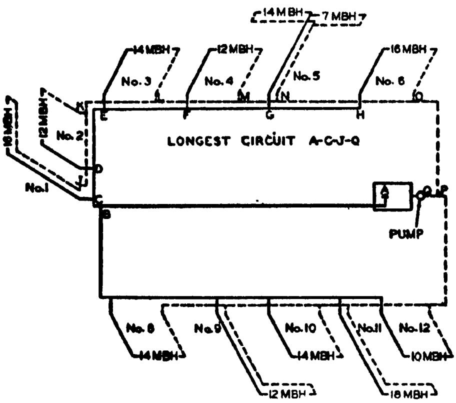

FIG. 14. FORCED CIRCULATION TWO-PIPE REVERSED RETURN SYSTEM

EXPANSION TANKS

Water heated from 40 F to 200 F expands about 0.04 of the original volume. The expansion tank permits the change in volume of the water in the heating system to take place without producing undesirable stresses due to pressure in any part of the system. Expansion tanks may be open, as illustrated in Fig. 15, or closed as shown in Fig. 16. An open expansion tank has free vent to the atmosphere and consequently, the pressure on the surface of the water is always that of one atmosphere. The minimum contents of an open tank should be 0.06 of the volume of the water in the system including that in the boiler, heat transmitters, pipes, etc. This capacity is 50 percent in excess of the actual increase in volume of water

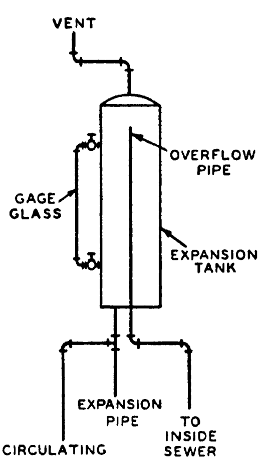

FIG. 15. AN OPEN EXPANSION TANK

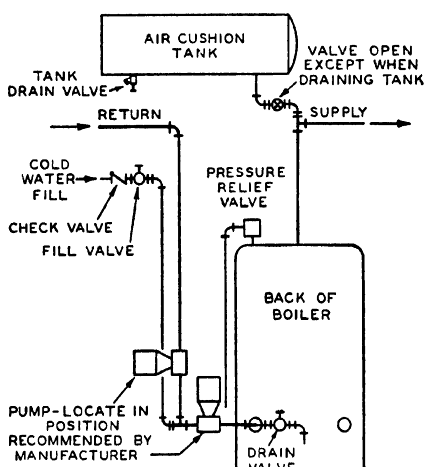

FIG. 16. A CLOSED EXPANSION TANK

due to increase in temperature from 40 F to 200 F. The tank should be located at least 3 ft above the highest radiator. Provision must be made to prevent freezing of the water in the tank as well as in the pipe leading to the tank.

In accordance with Paragraph H-92 of the *A.S.M.E.* Low Pressure Heating Boiler Code 1946: "All hot water heating systems shall be so installed that there will be no opportunity for the fluid relief column to freeze or to be accidentally shut off. If the system is equipped with an *open expansion tank*, an *internal over-flow* from the upper portion of the expansion tank must be provided in addition to an open vent, the internal over-flow to be carried within the building to a suitable plumbing fixture or to the basement." (See Fig. 15.)

When the vent from an open expansion tank is extended through the roof, it should be not less than 4 in. in diameter from a point below the roof, through and beyond the roof line. This will prevent vapor, which sometimes rises from an expansion tank, from closing the vent during outside freezing temperatures.

In a gravity circulation system, the pipe to the open expansion tank should be connected to the supply riser from the boiler, so that the air liberated from the water in the boiler will enter the expansion tank.

In a forced circulation system, the pipe to an open expansion tank should be connected on the suction side of the circulating pump.

A *closed expansion tank* is sealed against free venting to the atmosphere. The tank may be above the highest radiator or heat transmitter, or may be below the lowest one. The minimum contents of a closed expansion tank must be such that the expansion of the water due to increase in temperature will be cushioned against a reservoir of compressed air above the water level in the expansion tank. The tank must provide space not only for the change in water volume, but also for variations in air volume within the tank due to changes in air pressure. If the closed expansion tank is below the heat transmitters, the tank should be larger than if it is above them, and the higher the building, under such circumstances, the larger should be the air capacity in excess of that required for increase in water volume due to temperature rise.

The size of an expansion tank for installation in a closed system may be determined by the following formula:

$$V = \frac{E}{\dfrac{P_1}{P_1 + 0.434H} - \dfrac{P_1}{P_3}} \tag{3}$$

where

V = required tank capacity, gallons.

E = expansion of water from cold system to flow riser temperature, gallons.

P_1 = atmospheric pressure, psia.

P_3 = maximum tank pressure specified for heated system, psia.

H = height of top of filled system above tank, feet. (NOTE: Top of system open to atmosphere when system is filled.)

Example 7: Select a closed expansion tank for basement installation on a system containing 5000 gal and operating at 200 F flow temperature. The static head, due to the height of the system, is 70 ft. The maximum pressure should not exceed 100 psig.

Solution: Assume that the system is filled at 40 F. Then

$$E = 0.04 \times 5000 = 200 \text{ gal.}$$
$$P_1 = 14.7 \text{ psia}$$
$$P_3 = 100 + 14.7 = 114.7 \text{ psia.}$$

Substituting these values in Equation 3,

$$V = \frac{200}{\dfrac{14.7}{14.7 + 0.434\,(70)} - \dfrac{14.7}{114.7}} = \frac{200}{0.327 - 0.128} = 1000 \text{ gal.}$$

The size of a basement-located closed expansion tank should be at least equal to the following:

One story buildings: $x = 0.10\ V$ Three story buildings: $x = 0.17\ V$

Two story buildings: $x = 0.13\ V$ Four story buildings: $x = 0.23\ V$

where

x = expansion tank size in gallons.

V = water volume in heating system in gallons.

This condition favors, especially in tall buildings, the placing of the closed expansion tank above the highest heat transmitter.

TABLE 6. REQUIRED A.S.M.E. SIZE OF CLOSED EXPANSION TANK

Sq Ft of Equivalent Direct Radiation Installed	Gallon Tank	Sq Ft of Equivalent Direct Radiation Installed	Gallon Tank
Up to 350	18	Up to 1400	40
Up to 450	21	Up to 1600	2—30
Up to 650	24	Up to 1800	2—30
Up to 900	30	Up to 2000	2—35
Up to 1100	35	Up to 2400	2—40

For systems with more than 2400 sq ft of installed equivalent direct water radiation, the required capacity of the cushion tank shall be increased on the basis of one gallon tank capacity per 33 sq ft of additional equivalent direct radiation.

It is common practice to use multiple tank installations on large systems in lieu of one tank, the required capacity of which would be beyond commercially available sizes.

Any closed expansion tank located above the heat transmitters of a hot water heating system should be connected by a direct pipe with the flow main leaving the boiler, in order to enable the air to pass easily to the expansion tank. In a closed hot water heating system the water under pressure tends to absorb air at a rate increasing with pressure increase and decreasing with temperature increase.

Means must be provided to adjust and to observe the proportion of air within any closed expansion tank. This involves the provision of an air inlet valve, a water gage, and a relief valve. A source of supply of compressed air for renewing the air cushion is highly desirable, especially in large, high pressure, hot water heating systems where it is inconvenient, if not impracticable, to drain down the water in the system so as to permit introduction of atmospheric pressure air.

In smaller installations, gage fittings on the tank are usually omitted in order to prevent air loss from the tank. Occasional inspection of the tank or the necessity of adding water to the system will then indicate whether air should be added to the tank.

The *A.S.M.E.* Low Pressure Heating Boiler Code 1946, in paragraph H-92, specifies that provision must be made for draining a closed expansion tank without emptying the system. The Code also specifies the minimum sizes of closed expansion tanks based upon the equivalent direct radiation installed. (See Table 6.)

For every hot water heating system, the designer should calculate the volume of water contained in the radiators, piping system, boiler, etc., in order to select the proper size of expansion tank. The water content of the piping can be obtained from Table 7. For a rough selection of size, however, it is sometimes assumed that 50 percent of the volume of water is contained in the radiators, and that the water content per square foot of radiator heating surface is 0.2 gal for column radiators, and 0.13 gal for tube type radiators.

Another rough method for determining the size of an expansion tank to be located above the highest radiator, is to divide the square feet of radiation by the factor 40 to obtain the required capacity in gallons of the tank.

INSTALLATION DETAILS

Items that should be considered in the design of piping for a hot water system are:

All piping must be so pitched that all air in the system can be vented either through an open expansion tank, radiators or automatic relief valves. When piping must be run around an obstacle such as a beam, it is advisable to drop the piping below the beam. If looped over the beam, it becomes necessary to provide for venting of air from the high point of the pipe.

TABLE 7. VOLUME OF WATER IN STANDARD PIPE

PIPE SIZE, IN.	LINEAL FT OF PIPE CONTAINING 1 GAL	PIPE SIZE, IN.	LINEAL FT OF PIPE CONTAINING 1 GAL
1/2	63.1	2	5.75
3/4	36.1	2 1/2	4.02
1	22.2	3	2.60
1 1/4	12.8	4	1.52
1 1/2	9.47	5	0.96
		6	0.67

When changing the size of horizontal runs of pipe, eccentric fittings should be used to keep the tops of the pipes in line to permit free passage of air along the pipe.

All piping must be arranged so that the entire system can be drained. Sections of piping individually valved shall have corresponding drain valves.

In large buildings, the piping may be zoned according to exposure of building, usage of building, or method of control.

All piping must be installed so that it is free to expand and contract with changes of temperature without producing undue stresses in the pipes or connections. For this purpose it is generally sufficient to allow for a variation in length of 1 in. for 100 ft of pipe.

The pipe system should be designed so that each circuit has its correct friction for balanced water distribution. This may be done by change of pipe size or change in piping detail.

The connections from the boiler to the mains should be short and direct, to reduce the friction, and should allow for expansion.

The mains and branches should pitch up and away from the heater, generally not less than 1 in. in 10 ft.

The connections from mains to branches and to risers should be such that circulation through the risers will start in the right direction. Hence, in a one-pipe system the flow connection must be nearer the heater than the return connection. In a correctly-designed two-pipe system, the pressure in the flow main is higher than that in the return main, and a slight variation in the distances of the flow and return connections from the heater is not material; but it is generally best to have the two connections about equally distant from the heater.

Generally, connections to risers or radiators are taken out of the top of mains at either 45 or 90 deg from the horizontal plane.

Supply connections are usually made at the bottom of radiators so that circulation will not be stopped by accumulation of air, as would be the case with a top supply

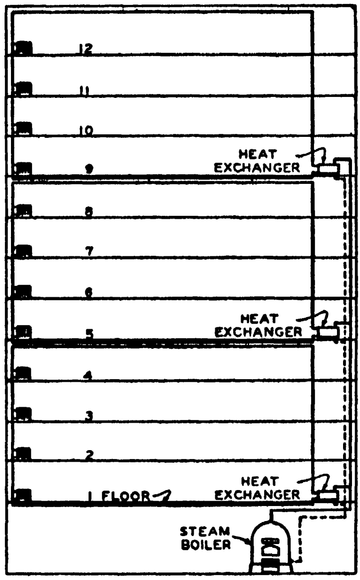

FIG. 17. VERTICAL ZONING OF HOT WATER HEATING SYSTEM IN A 12-STORY BUILDING

connection. Short radiators are sometimes connected for top supply and bottom return on the same end. When so connected, attention must be given to venting of air from the top of the radiator somewhat oftener than when bottom connections are used.

Unless used as heating surface, all piping, both flow and return, should be insulated.

All large systems should be provided with extra stop and drain valves, suitably located so that parts of the system may be isolated for repairs without making it necessary to drain the water from the entire system.

Relief Value. The *A.S.M.E.* Low Pressure Heating Boiler Code requires that all hot water heating boilers shall be equipped with a tested and rated relief valve having adequate capacity to match the gross output of the boiler. In order to comply with the requirement, relief valves must be connected to the top of the boiler and a discharge pipe must be connected to the relief valve and so arranged that there will be no danger of scalding attendants. The proper location of the relief valve is shown in Fig. 16.

ZONING

In large hot water systems, improved control and economy can be achieved by separating the systems into sections or zones (vertical or horizontal) which can be operated independently of each other. Variations in heat requirement of the different zones, as influenced by the exposure of the building, solar heat, weather conditions, heat from processes, type of occupancy, building chimney effect, etc., can readily be compensated for when heat can be supplied only where needed.

In tall buildings, vertical zoning such as shown in Fig. 17 not only provides the advantages of control and economy, but also reduces the water pressure in the system to that caused only by the number of floors served by each section. As shown in Fig. 17 a steam boiler can conveniently be used to supply steam to the heat exchangers supplying heated water to each zone.

CHAPTER 22

RADIATORS AND CONVECTORS

Heat Emission of Radiators and Convectors, Radiators, Convectors, Ratings of Radiators and Convectors, Effect of Operating Conditions, Heating Effect, Heating Radiators and Convectors, Enclosed Radiators

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RADIATORS and convectors are heat emitting units used in steam and hot water heating systems for supplying heat by radiation and convection to a room. The function of any radiator or similar device is the maintenance of occupancy comfort through the control of the mean radiant and air temperatures in the area. Since heat losses through the various parts of the structure constantly tend to lower these temperatures below the comfort point, the radiator or convector should be so placed and regulated that its output will replace the losses when and where they occur. If 80 percent of the room heat loss occurs through a cold wall or window area, then 80 percent of the input should be introduced in or directed toward that area.

The term *radiator* refers to a unit which emits a large part of its heat by radiation and includes cast-iron radiators, baseboard radiation and pipe coils. Cast-iron radiator types may be column, large-tube, small-tube or wall. Baseboard radiators may be of the radiant cast-iron type, radiant-convector cast-iron type, or finned-tube type. The last type, however, is actually a convector.

The term *convector* refers to a unit which emits the greater portion of its heat by convection. It includes such units as conventional convectors in which a heat emitting element of either cast-iron or of the finned-tube type is enclosed in a cabinet, but may also be of the unenclosed finned-tube type.

HEAT EMISSION OF RADIATORS AND CONVECTORS

Most heating units emit heat by *radiation* and *convection*. An exposed *radiator* emits roughly half of its heat by radiation, the amount depending upon the size and number of sections. In general, a thin radiator, such as a wall radiator, emits a larger proportion of its heat by radiation than does a thick radiator. When a radiator is enclosed or shielded, the proportion of heat emitted by radiation is reduced. The balance of the emission occurs by conduction to the air in contact with the heating surface, and this heated air rises by circulation due to convection, and transmits this warm air to the space which is to be heated.

Convectors transfer the smaller proportion of their heat to the room by radiation. Since most of their heat is transferred by convection, the heat emission is dependent upon the vertical distance between the heating element and the outlet grille at the top of the convector cabinet.

The output of a radiator or convector can be measured only by the heat it emits and is generally expressed in units of: Btu per hr; *Mbh* (1000 Btu per hr); or in equivalent direct radiation (*e.g.*, 240 Btu per hr for steam).

Table 1. Column-Type Cast-Iron Radiator

Height Inches	Generally Accepted Rating per Section[a]					
	One Column		Two Column		Three Column	
	Sq Ft	Btu/hr	Sq Ft	Btu/hr	Sq Ft	Btu/hr
15			1½	360		
18					2¼	540
20	1½	360	2	480		
22			2¼	540	3	720
23	1⅔	400	2⅓	560		
26	2	480	2⅔	640	3¾	900
32	2½	600	3⅓	800	4½	1080
38	3	720	4	960	5	1200
45			5	1200	6	1440
	Four Column		Five Column		Six column	
	Sq Ft	Btu/hr	Sq Ft	Btu/hr	Sq Ft	Btu/hr
13					3	720
16					3¾	900
18	3	720	4⅔	1120	4½	1080
20					5	1200
22	4	960				
26	5	1200	7	1680		
32	6½	1560				
38	8	1920	10	2400		
45	10	2400				

[a] These ratings are based on steam at 215 F and air at 70 F. They apply only to installed radiators exposed in a normal manner; not to radiators installed behind enclosures, grilles, or under shelves. For Btu per hour ratings at other temperatures, divide table values by factors found in Table 6.

Table 2. Large-Tube Cast-Iron Radiators

Sectional, cast-iron, tubular-type radiators of the large-tube pattern, that is, having tubes approximately 1¾ in. in diameter, 2½ in. on centers.

Number of Tubes per Section	Catalog Rating per Section[a]		Height	Width	Section Center Spacing[b]	Leg Height[c] To Tapping
	Sq Ft	Btu/hr	In.	In	In.	In.
	1¾	420	20		2½	4½
	2	480	23		2½	4½
3	2⅓	560	26	4⅝	2½	4½
	3	720	32		2½	4½
	3½	840	38		2½	4½
	2¼	540	20			
	2½	600	23		2½	4½
4	2¾	660	26	6¼–6¹³⁄₁₆	2½	4½
	3½	840	32		2½	4½
	4¼	1020	38		2½	4½
	2⅔	640	20		2½[d]	4½
	3	720	23		2½[d]	4½
5	3½	840	26	8–8⁹⁄₁₆	2½[d]	4½
	4⅓	1040	32		2½[d]	4½
	5	1200	38		2½[d]	4½
	3	720	20		2½	4½
	3½	840	23		2½	4½
6	4	960	26	9–10⅜	2½	4½
	5	1200	32		2½	4½
	6	1440	38		2½	4½
	2½	600	14		2½	3
7	3	720	17	11⅜–12¹³⁄₁₆	2½	3
	3⅔	880	20		2½	3 or 4½

[a] These ratings are based on steam at 215 F and air at 70 F. They apply only to installed radiators exposed in a normal manner; not to radiators installed behind enclosures, grilles, or under shelves. For Btu per hour ratings at other temperatures, divide table values by factors found in Table 6.

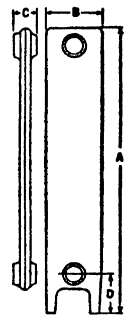
[b] Maximum assembly 60 sections. Length equals number of sections times 2½ in.

[c] Where greater than standard leg heights are required, this dimension shall be 6 in., except for 7-tube sections, in heights from 13 to 20 in., inclusive, for which this dimension shall be 4½ in. Radiators may be furnished without legs.

[d] For 5-tube hospital-type radiation, this dimension is 3 in.

TABLE 3. SMALL-TUBE CAST-IRON RADIATORS

Number of Tubes per Section	Catalog Rating per Section[a]		Section Dimensions				
			A Height[c]	B Width Minimum	B Width Maximum	C Spacing[b]	D Leg Height[c]
	Sq Ft	Btu/hr	In.	In.	In.	In.	In.
3[d]	1.6	384	25	3¼	3½	1¾	2½
4[d]	1.6	384	19	4 7/16	4 13/16	1¾	2½
	1.8	432	22	4 7/16	4 13/16	1¾	2½
	2.0	480	25	4 7/16	4 13/16	1¾	2½
5[d]	2.1	504	22	5⅝	6 5/16	1¾	2½
	2.4	576	25	5⅝	6 5/16	1¾	2½
6[d]	2.3	552	19	6 13/16	8	1¾	2½
	3.0	720	25	6 13/16	8	1¾	2½
	3.7	888	32	6 13/16	8	1¾	2½



[a] These ratings are based on steam at 215 F and air at 70 F. They apply only to installed radiators exposed in a normal manner; not to radiators installed behind enclosures, grilles, or under shelves. For Btu per hour ratings at other temperatures, divide table values by factors found in Table 6.
[b] Length equals number of sections times 1¾ in.
[c] Overall height and leg height, as produced by some manufacturers, are one inch (1 in.) greater than shown in Columns *A* and *D*. Radiators may be furnished without legs. Where greater than standard leg heights are required this dimension shall be 4½ in.
[d] Or equal.

RADIATORS

Column and large-tube radiators are no longer manufactured, but since many of these units are still in use, Tables 1 and 2 are included to provide principal dimensions and average ratings of them.

The small-tube type radiators, with a spacing of 1¾ in. per section, are about the only available cast-iron radiating surface for homes and office buildings. Small-tube radiators occupy less space than the older column and large-tube radiators, and are particularly suited for installation in recesses.

After a study of the demand for various sizes of radiators, the *Institute of Boiler and Radiator Manufacturers*, in cooperation with the Division of Simplified Practice, *National Bureau of Standards*, established Simplified Practice Recommendation R174-47 for small-tube cast-iron radiators. Table 3 shows the size and dimensions now being manufactured.

Wall radiators are hung from wall brackets and are well adapted to use

TABLE 4. CAST-IRON WALL RADIATORS

Approximate Dimensions—Inches			Heat Output[a]	
Height	Length or Width	Thickness	Sq Ft	Btu/hr
13¼	16½	3	6½	1560
13¼	22	3	8	1920
22	13¼	3	8	1920
13¼	29	3	11	2640
29	13¼	3	11	2640

[a] These ratings are based on steam at 215 F and air at 70 F. They apply only to installed radiators exposed in a normal manner, not to radiators installed behind enclosures, grilles, or under shelves. For Btu per hour ratings at other temperatures divide table values by factors found in Table 6.

in factory buildings. Tests have shown that the heat emitted from a wall-type radiator may be reduced from 5 to 10 percent if the radiator is placed near the ceiling with the bars horizontal and in an air temperature exceeding 70 F. When radiators are placed near the ceiling, there is usually such a large difference in the temperature between the floor level and the ceiling that it becomes difficult to heat the living zone of the rooms satisfactorily. Dimensions and heat emission rates for wall radiators are given in Table 4.

Baseboard radiation consists of long, low units which are made to resemble conventional baseboards, and are installed along the outside walls of rooms in place of the usual wooden baseboard. Units are made either of hollow cast-iron panels (with, or without fins on the back) or of ferrous or non-ferrous finned tubing installed behind a metal enclosure. They are primarily used in hot water systems, but may also be used in two-pipe steam systems.

There are various kinds of baseboard radiation available,[1] the *radiant* type and the *convector* type. Radiant baseboards have a substantial portion of the front face water backed, and do not depend upon an enclosure for their heat output. Cast-iron radiant baseboards may be either (1) a full radiant, called Type R, or (2) a radiant convector, called Type RC. The Type RC unit, in addition to the radiant front face, has extended convection heating surface on the rear face to increase its output. The convector type of baseboards includes finned tube units with which enclosures are used. The front of the enclosure supplies radiant heat.

TABLE 5. HEAT EMISSION OF PIPE COILS PLACED VERTICALLY ON A WALL (PIPES HORIZONTAL) CONTAINING STEAM AT 215 F AND SURROUNDED WITH AIR AT 70 F

Btu per linear foot of coil per hour (not linear feet of pipe)

SIZE OF PIPE	1 IN.	1¼ IN.	1½ IN.
Single row	132	162	185
Two	252	312	348
Four	440	545	616
Six	567	702	793
Eight	651	796	907
Ten	732	907	1020
Twelve	812	1005	1135

The basic advantage of the baseboard radiator or of the long low narrow radiator is that its normal placement is along the cold walls and under areas where the greatest heat loss occurs in a room. Other advantages claimed for the baseboard radiator are: it is inconspicuous; it is clean in operation; it offers a minimum of interference with furniture placement, and, it distributes the heat near the floor. This last characteristic reduces the floor to ceiling temperature gradient to about 2 to 4 F deg, and tends to produce uniform temperatures throughout the room. It also makes baseboard radiators especially adaptable to basementless homes, where cold floors are prevalent.[2]

Heat loss calculations for baseboard heating systems are the same as those used for other types of radiation. The procedure for designing baseboard heating systems is given in I = B = R Installation Guide No. 5.[1] Ratings for baseboard radiation are expressed in Btu per linear foot.

Pipe coils are assemblies of standard pipe or tubing (1 in. to 2 in.) which are used as radiators. In older practice these coils were commonly used in factory buildings, but are not often found in this service today. When coils are used, the miter type assembly is preferable, as it readily permits expansion in the pipe.

The heat emission of pipe coils placed vertically on a wall, with the pipes horizontal, is given in Table 5, which has been developed from available data and does not represent definite results of tests. For such coils the heat emission varies as the height of the coil. The heat emission of each pipe in *ceiling* coils, placed horizontally, is about 126 Btu, 156 Btu, and 175 Btu per linear foot of pipe, respectively, for 1-in., $1\frac{1}{4}$-in., and $1\frac{1}{2}$-in. coils.

CONVECTORS

Convectors are space heating devices composed of a casing with outlet grille, and an extended surface heating element of fin-tube or cast-iron fin surface. The casing usually contains a damper. The air enters the enclosure near the floor line below the heating element, is heated in passing

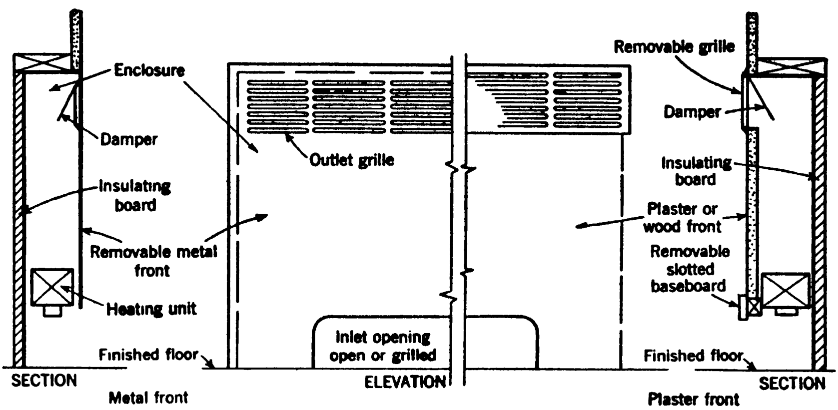


FIG. 1. TYPICAL RECESSED CONVECTOR

through the element, and delivered to the room through the outlet grille located near the top of the enclosure. The room air movement thus established accomplishes a reduction in floor to ceiling temperature differential, and tends to assure comfort in the living zone. A typical recessed convector is shown in Fig. 1. Factory-assembled units comprised of a heating element, casing and outlet grille with damper, are widely used. Grilles may be used over the air inlets.

In cases where enclosures are to be used but are not furnished by the heater manufacturer, it is important that the proportions of the cabinet and the grilles be so designed that they will not impair the performance of the assembled convector. It is desirable that the enclosure or housing for the convector fit as snugly as possible so that the air to be heated cannot by-pass the heating element in passing through the enclosure.

Cast-iron heating units may be concealed in a cabinet or enclosure for appearance. In such cases a greater percentage of heat is conveyed to the room by convection thereby resulting in a form of gravity convector.

The output of a convector, for any given length and depth, is a function of the height of the discharge grille above the heating element. The published ratings are generally given in terms of Btu per hour or square feet of *Equivalent Direct Radiation*, EDR. For steam convectors, as for radiators, 240 Btu per hr may be taken as an equivalent square foot of radiation. When more than one heating unit is used, one mounted above the other in the same cabinet, the output of the upper unit or units will be materially less than that of the bottom unit.

RADIATOR AND CONVECTOR RATINGS

A standard method of testing radiators was adopted by the A.S.H.V.E. in 1927.[3] This Code provides for a standard test room, the temperature of which is to be maintained at 70 F, measured in the center of the room at

TABLE 6. CORRECTION FACTORS FOR DIRECT CAST-IRON RADIATORS AND CONVECTORS[a]

Steam Press. (Approx.)		Heating Medium Temp F Steam or Water	Factors for Direct Cast-Iron Radiators							Factors for Convectors						
			Room Temperature F							Inlet Air Temperature F						
Gage Vacuum In. Hg.	Abs Lb per Sq In.		80	75	70	65	60	55	50	80	75	70	65	60	55	50
22.4	3.7	150	2.58	2.36	2.17	2.00	1.86	1.73	1.62	3.14	2.83	2.57	2.35	2.15	1.98	1.84
20.3	4.7	160	2.17	2.00	1.86	1.73	1.62	1.52	1.44	2.57	2.35	2.15	1.98	1.84	1.71	1.59
17.7	6.0	170	1.86	1.73	1.62	1.52	1.44	1.35	1.28	2.15	1.98	1.84	1.71	1.59	1.49	1.40
14.6	7.5	180	1.62	1.52	1.44	1.35	1.28	1.21	1.15	1.84	1.71	1.59	1.49	1.40	1.32	1.24
10.9	9.3	190	1.44	1.35	1.28	1.21	1.15	1.10	1.05	1.59	1.49	1.40	1.32	1.24	1 17	1.11
6.5	11.5	200	1.28	1.21	1.15	1.10	1.05	1.00	0.96	1.40	1.32	1.24	1.17	1.11	1.05	1.00
Lb per Sq In.																
1	15.6	215	1.10	1.05	1.00	0.96	0.92	0.88	0 85	1.17	1.11	1.05	1 00	0 95	0 91	0.87
6	21	230	0.96	0.92	0.88	0.85	0.81	0.78	0 76	1.00	0.95	0.91	0.87	0.83	0 79	0 76
15	30	250	0 81	0.78	0.76	0.73	0 70	0.68	0.66	0.83	0.79	0.76	0.73	0.70	0.68	0.65
27	42	270	0.70	0.68	0 66	0.64	0.62	0.60	0.58	0.70	0.68	0.65	0.63	0.60	0.58	0.56
52	67	300	0.58	0.57	0.55	0.53	0.52	0.51	0.49	0 56	0.54	0.53	0.51	0.49	0.48	0.47

[a] To determine the size of a radiator or a convector for a given space, multiply the heat loss of the space in Btu per hour by the proper factor from the above table and select radiator or convector having an equivalent Btu per hour rating.

An alternate method is to divide the heat loss in Btu per hour by 240 and multiply the result by the proper factor from the above table and select radiator or convector having an equivalent square foot rating.

To determine the heating capacity of a radiator or a convector under conditions other than the basic ones with the heating medium at a temperature of 215 F, and the room temperature at 70 F in the case of a radiator, and the inlet air temperature at 65 F in the case of a convector, divide the heating capacity at the basic rating conditions by the proper factor from the above table.

an elevation of 5 ft above the floor. The steam temperature in the radiator is to be 215 F, which corresponds to 15.6 lb per sq in. absolute. The weight of condensate per hour, under these standard conditions, multiplied by the difference in the enthalpy of the steam entering the radiator and that of the condensate leaving the radiator, gives the radiator output in Btu per hour. This output divided by 240 gives the steam rating of the radiator in equivalent square feet, EDR.

Similar test methods for convectors are the A.S.H.V.E. Codes for Testing and Rating Concealed Gravity Type Radiation,[4] (Steam Code 1932 and Hot Water Code 1933). These Codes recognize a different type of test booth, and the air temperature used is that of the air entering the convector casing instead of the temperature in the center of the room. The entering air temperature for standard test conditions is 65 F. For hot water the standard test conditions call for a mean temperature of the water in the convector of 170 F.

The method of testing and rating both ferrous and non-ferrous convectors, which is now generally accepted, is given in Commercial Standard CS140–47, Testing and Rating Convectors, which has been developed cooperatively by the *Convector Manufacturers Association*, the *Institute of Boiler and Radiator Manufacturers*, other members of the trade, and the *National Bureau of Standards*.

The rating of a top outlet convector is established at a value not in excess of the *condensation capacity* (which is the heat extracted from the steam or water in the convector, under standard test conditions). The rating of a front outlet convector includes the *condensation capacity* plus an allowance for heating effect in the occupied zone, based on convector enclosure height from bottom of the enclosure to top of the outlet. A table of heights and heating effect allowances is given in the Commercial Standard CS140–47, and lists allowances from zero percent for a 38-in. height to 15 percent for a 20-in. height, or less.

For an inclined outlet convector the rating includes the *condensation capacity*, plus a heating effect allowance obtained by multiplying the allowance for a front outlet convector by a factor (angle of outlet to horizontal ÷90).

Approval of convector ratings may be obtained by the manufacturer by submitting test data to a *Convector Rating Committee* appointed by the Division of Trade Standards of the *National Bureau of Standards*. Requests should be addressed to the Division of Trade Standards.

A Testing and Rating Code for Baseboard Type of Radiation[5] was adopted by the *Institute of Boiler and Radiator Manufacturers* in 1950. This code contains test procedures for determining steam ratings which are obtained from the condensation capacity (converted to standard conditions) by adding a maximum of 15 percent. The ratings are expressed in Btu per hour per linear foot, and may also be expressed in square feet of steam radiation per linear foot.

Water ratings are determined by applying the factors shown in Table 7 to the steam ratings, and are expressed in Btu per hour per linear foot for each average water temperature listed. A method for testing baseboard radiation with water for the purpose of determining water ratings is being considered.

Manufacturers who wish to publish baseboard radiation ratings as $I = B = R$ ratings may submit test data to the $I = B = R$ Baseboard Rating Committee and receive approval of test procedure and ratings. The following catalog information must be given for $I = B = R$ Steam Ratings: (1) rating in Btu per hour per linear foot, (2) percentage added to capacity in determining ratings, and (3) name and other type of designation.

The following information must be given for $I = B = R$ Water Ratings: (1) rating in Btu per hour per linear foot for each average water temperature listed, (2) percentage added to capacity in determining ratings, (3) name or other type of designation, and (4) a statement that the water ratings have been determined by applying to the $I = B = R$ Steam Ratings the factors (see Table 7) approved by the *Institute of Boiler and Radiator Manufacturers*.

Effect of Operating Conditions

The heat output of a radiator is proportional to the 1.3 power of the temperature difference between the air in the room at the 60 in. level and the heating medium in the radiator. The heat output of a convector is proportional to the 1.5 power of the temperature difference between the air entering the convector and the heating medium, steam or hot water, within the convector.[6] For hot water the arithmetical average between

Table 7. Factors to Convert $I = B = R$ Steam Ratings to Hot Water Ratings at Temperatures Indicated

Average Radiator Temperature	Factor	Average Radiator Temperature	Factor	Average Radiator Temperature	Factor
150	0.45	175	0.65	200	0.86
155	0.49	180	0.69	205	0.91
160	0.53	185	0.73	210	0.95
165	0.57	190	0.78	215	1.00
170	0.61	195	0.82	220	1.05

entering and leaving water temperatures is used. These laws may be expressed as correction factors to change from output under standard rating-test conditions, to output under other operating conditions. Such factors are given in Table 6.

When it is desired to change the output under any test conditions to the corresponding output under standard code test conditions, the reciprocal form of correction factor may be derived. The equations for steam units are:

For radiators

$$C_s = \left(\frac{215 - 70}{t_s - t_r}\right)^{1.3} \qquad (1)$$

For convectors

$$C_s = \left(\frac{215 - 65}{t_s - t_i}\right)^{1.5} \qquad (2)$$

The output under standard conditions will be:

$$H_s = C_s H_t \qquad (3)$$

where

C_s = correction factor.

t_s = steam temperature during test, Fahrenheit degrees.

t_r = room temperature during test, Fahrenheit degrees.

t_i = inlet air temperature during test, Fahrenheit degrees.

H_s = heat emission rating under standard conditions, Btu per hour.

H_t = heat output under test conditions, Btu per hour.

The relation between the size of the radiator or convector and the size of the test room will affect the results obtained in a capacity-rating test.[7] The height and location of the radiator and the insulation of the test room are other important factors that are not specifically regulated by the codes.

For a radiator, the finish coat of paint affects the heat output. Oil paints of any color will give about the same results as unpainted black or rusty surfaces, but an aluminum or a bronze paint will reduce the heat emitted by radiation. The net effect may be a reduction of 10 percent or more in the total heat output of the radiator.[8,9,10]

Radiator enclosures and convector cabinets of proper design may improve the heat distribution within the room as compared to the heat distribution obtained with an unenclosed radiator.[11]

Heating Effect

For several years the term *heating effect* has been used to designate the relation between the *useful output* of a radiator, in the comfort zone of a room, and the total input as measured by steam condensation or water temperatures.[12, 13] The application of such a heating effect factor implies that some radiators and convectors use less steam than others for producing equal comfort heating results in the room.

All authorities do not agree that the use of heating effect factors are justified. No standard method for evaluating the heating effect of radiators and convectors and correlating it with comfort has yet been accepted. One method, with test data[14] on radiators and convectors, and making use of the eupatheoscope for evaluating the environment produced, has been suggested by the *University of Illinois*. The principle underlying the eupatheoscope involves the measurement of the heat loss from a sizable body by radiation and convection, when the surface is maintained at some constant temperature. Through the use of this instrument and its calibration curve, non-uniform environments may be referred to uniform environments

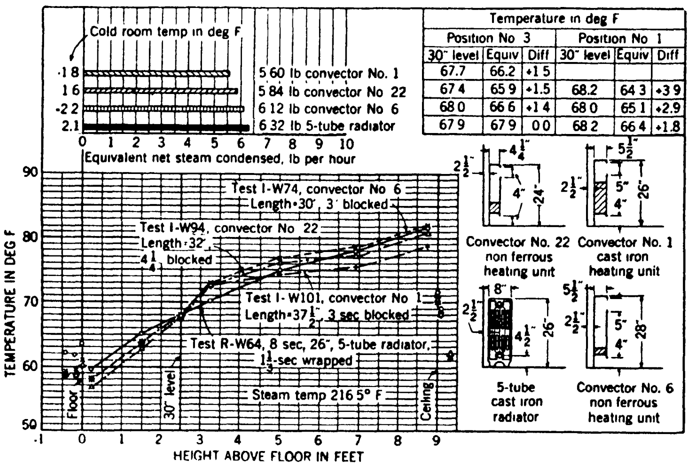


FIG. 2. TEMPERATURE GRADIENTS AND EQUIVALENT TEMPERATURES FOR RADIATOR AND CONVECTORS WITH COMMON 30 IN. LEVEL TEMPERATURE

in which the air and all surrounding surfaces are at the same temperature. The temperatures of the uniform environments are referred to as equivalent temperatures.

The Kata thermometer,[15] the thermo-integrator,[16, 17] and the globe [18] thermometer are other instruments which have been used to measure the influence of air temperature, air movement and radiation in an environment.

Data given in Fig. 2 show that while the air temperature at the 30-in. level is the same for the three convectors and the one large-tube cast-iron radiator, in position No. 3 in the test room, the equivalent temperature is 1.5 deg lower than the air temperature in the case of the three convectors, and the same as the air temperature in the case of the radiator. The difference between the minimum and the maximum amount of heat required to maintain the common air temperature at the 30-in. level is of the order of 13 percent.

In Fig. 3 are shown the results of tests made with the same three convectors and the one large-tube cast-iron radiator, so adjusted in size that each gave approximately the same equivalent temperature in the No. 3 position in the test room. The difference between the minimum and the maximum amount of heat required to maintain the common equivalent temperature is of the order of 7 percent.

Figs. 2 and 3 show results obtained in cold room tests in which the radiators were continuously filled with steam at 215 F. Under these conditions of operation, air temperature gradients are likely to be exaggerated as compared with those encountered with the intermittent operation usually obtained in actual practice.

The following statements applying to the use of radiators are based on experience and test results:[13]

1. The heating effect of a radiator cannot be judged solely by the amount of steam condensed within the radiator.

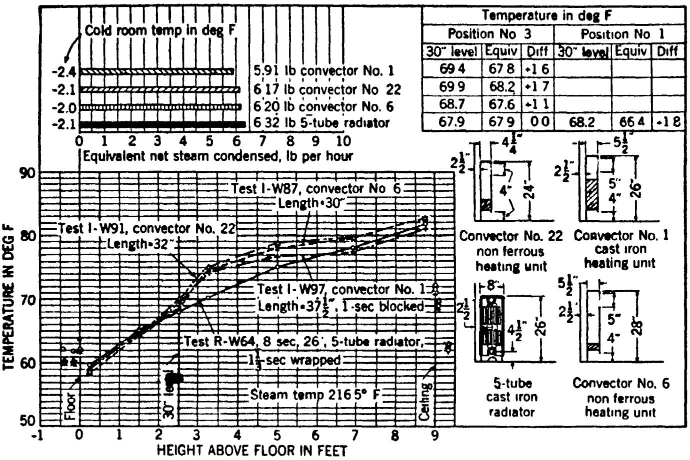


Fig. 3. Temperature Gradients and Equivalent Temperatures for Radiator and Convectors with Common Equivalent Temperature

2. Smaller floor-to-ceiling temperature differentials can be maintained with long, low, thin, direct radiators, than can be maintained with high, direct radiators.

3. The larger portion of the floor-to-ceiling temperature differential in a room of average ceiling height heated with direct radiators occurs between the floor and the breathing level.

4. The comfort level (approximately 2 ft-6 in. above floor) is below the breathing line level (approximately 5 ft-0 in. above floor), and temperatures taken at the breathing line may not be indicative of the actual heating effect of a radiator in the room. The comfort-indicating temperature should be taken below the breathing line level.

5. High column radiators placed at the sides of window openings do not produce as comfortable heating effects as long, low, direct radiators placed beneath windows.

HEATING THE RADIATOR AND CONVECTOR

The maximum condensation occurs in a heating unit when the steam is first turned on. Tests[19] on an old-style column-type cast-iron radiator indicated that in the first 10 min the condensation rate reached a peak of 0.95 lb per sq ft of radiation per hour and 10 to 15 min later dropped to a rate of 0.24 lb. In one-pipe gravity systems the rate of steam supply to the heating unit, while heating up, is frequently retarded by controlled elimination of air through air valves or traps. In two-pipe systems automatic control valves may also retard the supply of steam. Vacuum types

of air venting valves may be used to reduce the length of the venting periods.

ENCLOSED RADIATORS

The general effect of an enclosure placed about a direct radiator is to restrict the air flow, diminish the radiation and, when properly designed, improve the heat distribution within the heated space.

Investigations[13] indicate that in the design of the enclosure three things should be considered:

1. There should be better distribution of the heat below the breathing line level to produce greater heating comfort and lowered ceiling temperatures.
2. The lessened steam consumption may not materially change the radiator heating performance.
3. The enclosed radiator may inadequately heat the space.

A comparison between a bare or exposed radiator (A) and the same radi-

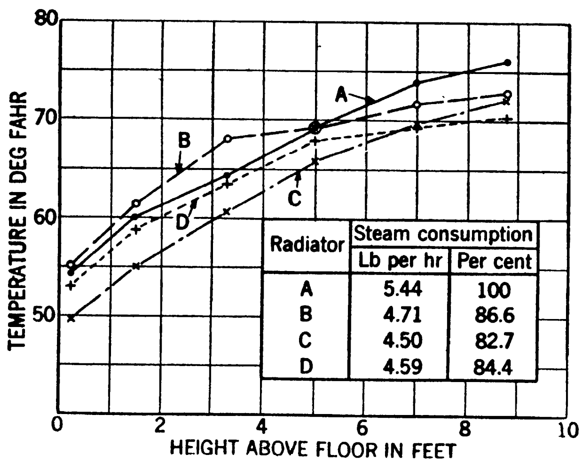


FIG. 4. STEAM CONSUMPTION OF EXPOSED AND CONCEALED RADIATORS

ator with a well-designed enclosure (B), with a poorly-designed enclosure (C), and with a cloth cover (D) will illustrate the relative heating characteristics. In Fig. 4 the curve (B) reveals that the enclosed radiator used less steam than the exposed radiator, but gave a satisfactory heating performance. A well-designed shield placed over a radiator gives about the same result. Curve (C) shows the unsatisfactory effects produced by improperly-designed enclosures. Curve (D) shows that the effect of a cloth cover extending downward 6 in. from the top of the radiator was to make the performance unsatisfactory and inadequate.

Some commercial enclosures and shields for use on direct radiators are equipped with water pans for the purpose of adding moisture to the air in the room. Tests[20] show that an average evaporative rate of about 0.235 lb per square foot of water surface per hour may be obtained from such pans, when a radiator is steam heated and the relative humidity in the room is between 25 and 40 percent. This source of supply of moisture alone is not adequate to maintain a relative humidity above 25 percent on a zero day.

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CHAPTER 23

PANEL HEATING

Definitions; Application Methods: Imbedded Piping for Ceilings, Walls, or Floors; Warm Air and Electrically Heated Ceilings, Walls, or Floors; Output from Panel Surfaces: Radiation, Convection and Combined Heat Transfer; Floor Panel Design: Design Conditions, Calculation of Room Heat Loss, Determination of UMRT, Determination of Panel Output, Graphical Determination of Panel Input and Water Temperature; Ceiling and Wall Panel Design; Hot Water Piping, Installation and Control

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IN this chapter the term, Panel Heating, is used to describe a method of space heating which employs large heated areas of interior room surfaces operating at relatively low surface temperatures (80 to 125 F). The heating elements usually consist of warm water piping, warm air ducts or low temperature electrical resistance elements imbedded in, or located behind, ceiling, wall, or floor surfaces.

Panel heating may be considered as another method of convenient and effective space heating. The heat loss requirements are calculated in the conventional manner, the heat release from the heated surfaces is expressed in terms of heat output per square foot of surface per hour, and the room air temperatures to be maintained are approximately the same as those maintained by heating systems employing cast-iron radiators, convectors, or warm air ducts.

This chapter does not include a separate discussion of such topics as the influence of radiation on human comfort, the mechanisms by which human beings release heat, and other similar topics that apply to all methods of heating interior spaces for human comfort. The reader is referred to Chapter 6 for a detailed discussion of this subject.

APPLICATION METHODS

The great majority of panel installations of the past 40 years (which is the period of the modern utilization of this method of heating) have used warm water as the heating medium which is circulated in imbedded piping. More recently, the use of warm air ducts, and imbedded electrical heating elements, has come into favor, especially where specific local factors have influenced such use. Steam has been used only occasionally because of the problems which result from its higher temperature.

When the heating medium is warm water, both ferrous (steel or wrought iron) or non-ferrous (generally copper or aluminum) pipe or tubing are used widely in ceiling, wall, or floor panel construction. Tubing sizes used are $\frac{3}{8}$, $\frac{1}{2}$, $\frac{5}{8}$ and $\frac{7}{8}$ in. O.D., while piping is generally $\frac{1}{2}$, $\frac{3}{4}$ or 1 in. I.P.S. Where coils are imbedded in concrete or plaster, no screw threads should be used for ferrous pipe coils. The construction should be of all-welded type. Changes in direction should be made by bending the pipe itself, rather than by use of fittings.

For non-ferrous tubing, solder-sweated couplings are used. It is recommended that a medium temperature solder of 95 percent tin—5 percent antimony, or its equivalent, be used. All piping or tubing is, generally, subjected to a hydrostatic test of at least three times the working pressure, but not less than 150 psig.

The more common forms of application of panel heating fall into the following general categories: (1) imbedded piping for ceilings; (2) imbedded piping for walls; (3) imbedded piping for floors; (4) air heated ceilings, walls or floors; (5) electrically heated ceilings, walls or floors.

Imbedded Piping for Ceilings

When piping is imbedded in ceilings, the construction used is generally one of the following:

a. Pipe or tubing is imbedded in the lower portion of a concrete slab, generally very close to its lower surface. If plaster is to be applied to the concrete, the piping

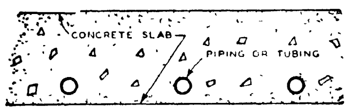
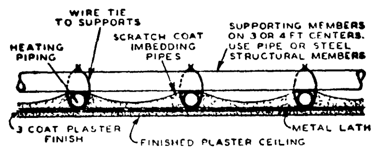


Fig. 1. Coils in Structural Concrete Slab



Suspended Plaster Ceiling

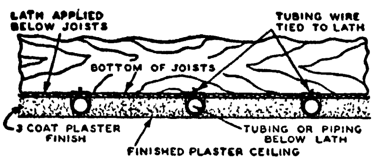
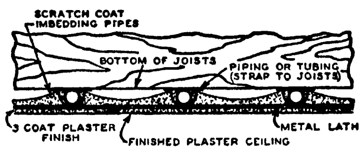


Fig. 3. Coils in Plaster Below Lath



Plaster Ceiling Below Joists

Fig. 2. Coils in Plaster Above Lath

may be placed directly on the wood forms. If the slab is to be used without plaster finish, then the piping is installed about ½ in. above the undersurface of the slab. Fig. 1 shows this method of construction. It is important that any local construction codes which may affect the position of the piping be consulted.

b. Pipe or tubing is imbedded in a metal lath and plaster ceiling. If the lath is suspended to form a hung ceiling, the piping may be suspended as a prefabricated pipe coil so that the lath may be held by wire to the underside of the pipe coil. Plaster is then applied to the metal lath, care being taken to imbed the coil, as shown in Fig. 2.

c. Copper tubes of the smaller diameters are attached to the underside of wire lath or gypsum lath. Plaster is then applied to the lath to imbed the tubing, as shown in Fig. 3.

d. Other forms of ceiling construction utilize prefabricated panels of metal, composition board, wood paneling, etc., having water warm piping, tubing or channels built into the panel sections.

Coils are usually of the sinuous type, although some header or grid type coils have been used in ceilings. Coils may be of either ferrous or non-ferrous piping or tubing, with coil pipes spaced from 4½ to 9 in. on centers, depending on the required output, pipe or tubing size, and other factors.

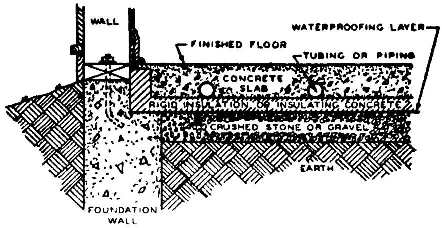


FIG. 4. COILS IN FLOOR SLAB ON GRADE

Where plastering is applied to pipe coils, a standard three-coat gypsum plastering specification[1] is followed, with a minimum of $\frac{3}{8}$ in. of cover below the tubes when the tubes are installed below the lath. Generally, the surface temperature of plaster panels should not exceed 120 F, and this is usually met by limiting the water temperature in the pipes or tubes in contact with the plaster to a maximum temperature of 140 F. Insulation should be placed above the coils to reduce the *reverse loss* which is the difference between the heat supplied to the coil and the net useful output to the heated room.

Imbedded Piping for Walls

Although not so universally used as ceiling panels, wall panels may be constructed by any of the methods outlined for ceilings.

Imbedded Piping for Floors

The construction for piping imbedded in floors will depend upon whether (a) the floor is laid on grade, or (b) the floor is above grade.

a. Both ferrous and non-ferrous pipe and tubing are used in floor slabs which rest on grade. The coils are constructed as either sinuous-continuous pipe coils, or arranged as header coils with the pipes spaced from 6 to 18 in. on centers. The coils are generally installed with $1\frac{1}{2}$ to 4 in. of cover above the coils. It is recommended that insulation be used to reduce the perimeter and reverse losses. Fig. 4 shows the application of pipe coils in slabs resting on grade. Generally, a waterproofing layer is desirable to protect insulation and piping.

b. Where the coils are imbedded in structural load supporting slabs above grade, construction codes may affect their position. Otherwise, the coil piping is installed in the same manner as described for slabs resting on grade.

Air Heated Ceilings, Walls, and Floors

Several methods have been devised to warm the interior room surfaces by circulating heated air through passages behind these surfaces. In some

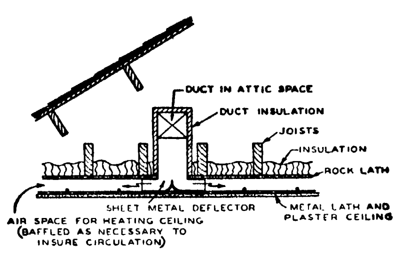


FIG. 5. WARM AIR PLASTER CEILING CONSTRUCTION

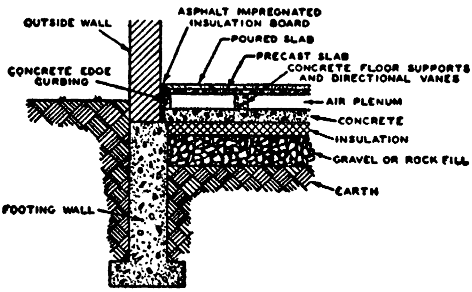


FIG. 6. WARM AIR FLOOR PANEL CONSTRUCTION

cases, the heated air is recirculated in a closed system. In others, all or a part of the air is passed through the room on its way back to the furnace to provide supplementary heating and ventilation.[2] Figs. 5 and 6 indicate two common types of construction. Care must be exercised to assure compliance with any building codes that might apply. (See also section on Warm Air Ceiling Panels in Chapter 19).

Electrically Heated Ceilings, Walls, or Floors

Several different forms of electric resistance units are available for heating the interior room surfaces. These include: (1) resistance cables that may be imbedded in a manner similar to hot water piping in concrete or plaster; (2) prefabricated electric heating panels to be attached to room surfaces; and (3) electrically heated fabrics or other materials for application to, or incorporation into, finished room surfaces. Again, the problem of compliance with applicable codes must be considered. Figs. 7 and 8 indicate two methods that have been used.

OUTPUT FROM PANEL SURFACES

The heat transfer from a panel is accomplished by two basic heat transfer processes: radiation and convection, which are considered in following paragraphs.

Radiation Transfer

The radiation transfer can be evaluated by means of the relationship set up by Stefan and Botzmann:

$$q_r = 0.173\, F_a F_e \left[\left(\frac{T_s}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4 \right] \qquad (1)$$

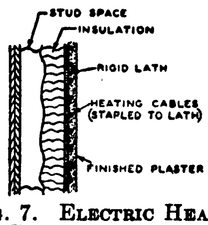


FIG. 7. ELECTRIC HEATING CABLES IN PLASTER

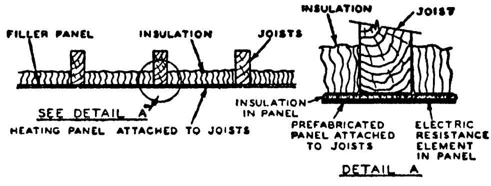


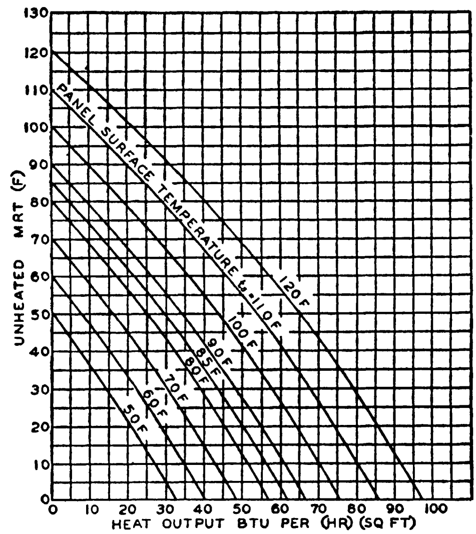
FIG. 8. PREFABRICATED ELECTRIC PANEL

where

q_r = heat transfer by radiation, Btu per (square foot)(hour).
T_s = absolute temperature of panel heated surface, Fahrenheit.
T_w = absolute mean radiant temperature of all unheated surfaces, Fahrenheit.
F_a = the configuration factor (dimensionless).
F_e = the emissivity factor (dimensionless).

For large parallel planes or large enclosed surfaces as ordinarily encountered in panel heating practice:

$$F_e = \frac{1}{\frac{1}{e_1} + \frac{1}{e_2} - 1} \tag{2}$$



t_s = panel surface temperature

FIG. 9. HEAT OUTPUT BY RADIATION

where

e_1 and e_2 = emissivities of the respective surfaces.

In heating practice e_1 and e_2 are usually equal to 0.9 and F_e to 0.82. Also, the configuration factor F_a is equal to 1 for large parallel planes, long concentric cylinders, or smaller bodies in large enclosures. Therefore, for ordinary rooms with parallel walls, regular floors and ceilings, with an emissivity factor of 0.82, Equation 1 can be simplified to:

$$q_r = 0.142\left[\left(\frac{T_s}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4\right] \tag{3}$$

Irregularities in room surfaces and materials may introduce some error in the application of this radiation relationship. However, most authorities are in agreement that the heat emission by radiation, as calculated in this manner, can be considered accurate within 10 percent. Radiation values are shown in Fig. 9 for various surface temperatures. These values are applicable for either floor or ceiling panel radiation outputs.

Convection Transfer

Convection values of heat transfer are not easily established. Convection in a panel heated space is usually considered to be of the *natural* type, that is, air motion is generated by the warming (by conduction) of the boundary layer of air which starts moving as soon as its temperature exceeds that of the surrounding air. In practice, however, there are many factors interfering with *natural* convection. Infiltration, localized drafts, ventilation, and movement of persons are all likely to disturb this process so that it becomes difficult to determine the exact convection effect. Until results from current research become available, an interim evaluation of convection heat exchange can be determined for ordinary panel heating applications from available theoretical relationships.

As investigated[3, 4, 5] by Nusselt and Henky, Griffiths and Davis, Wilkes and Peterson, McAdams and others, natural convection is found to be affected by only two factors:

a. Temperature difference between the heat emitting surface and the surrounding air.
b. The position of the surface.

The basic general equation for natural convection from a flat surface is of the following form:

$$q_c = f_c(t_s - t_a)^n \tag{4}$$

where

q_c = heat transfer by convection, Btu per (square foot) (hour).
f_c = a coefficient (surface conductance) representing the heat transfer from a unit area per unit difference in temperature, Btu per (square foot) (hour) (Fahrenheit degree temperature difference between surface and air).
n = an exponent depending on surface position and temperature difference between the surface and the surrounding air.
t_s = temperature of the surface, F.
t_a = temperature of the air, F.

The value of n is usually taken as 1.25 regardless of the magnitude of the temperature difference and the position of the surface. However, Wilkes and Peterson state that a value of 1.12 for n is more appropriate for *low temperature differences*, with heat flow upward from horizontal surfaces, and that a value of 1.00 is best in the case of heat flow downward from horizontal surfaces. The various investigators mentioned also indicate that values of f_c vary from 0.2 to 0.38 for downward heat flow from ceilings, and from 0.38 to 0.81 for upward heat flow from floors. The variation in the value of f_c, as reported by the various investigators, is dependent upon the temperature difference and the value of the exponent n.

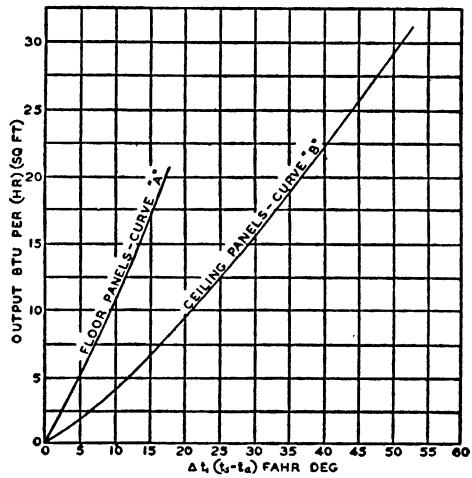
More recent experimental work indicates a provisional correlation between measured convection outputs in actual floor panel heated rooms and the laboratory values obtained by the previous investigators. This correlation indicates that the values obtained from Wilkes and Peterson's equation

$$q_c = 0.81\ (t_s - t_a)^{1.12} \tag{5}$$

for heat flow upward from horizontal surfaces and small temperature differences, give results that check within reasonable precision the convection output from floor panels, with temperature differences as recommended. Curve A in Fig. 10 is based on the equation of Wilkes and Peterson, and gives values which may be used for convection outputs from floor panels.

More experimental data are required on convection outputs of ceiling panels. The warming of the air by a ceiling panel does not in itself create convection currents. Instead, it is the action of cooling along the outside wall or walls which causes the air movement. In lieu of actual test data which would take this factor into account, it is recommended that the values obtained from Curve B in Fig. 10, based on the Nusselt and Henky equation

$$q_c = 0.22\ (t_s - t_a)^{1.25} \tag{6}$$



t_s = panel surface temperature
t_a = inside air temperature

FIG. 10. HEAT OUTPUT BY CONVECTION FROM FLOOR AND CEILING PANELS

should be used for determining convection outputs from ceiling panels for various surface temperatures.

Combined Heat Transfer

The *sum* of the *radiant* heat transfer from Fig. 9 and the *convective* transfer from curve A or B of Fig. 10, gives the combined useful heat transfer to the room for any combination of panel surface temperature, room air temperature, unheated mean radiant temperature (UMRT), and panel location.

Example 1: Find the combined total useful heat transfer from a square foot of floor panel having a surface temperature of 85 F, when the average room air temperature is 70 F and the mean radiant temperature UMRT of the unheated room surfaces is 60 F.

Solution:

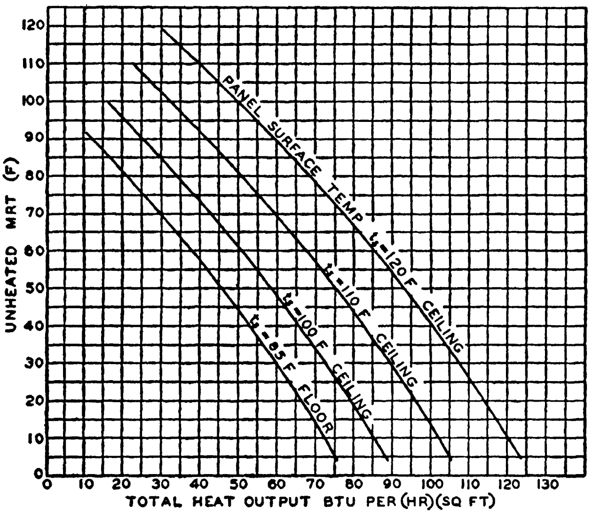
	Btu/(hr)(sq ft)
Radiation	
(Fig. 9 for UMRT of 60 F and t_s of 85 F)	21.4
Convection	
(Fig. 10, Curve A for a temperature difference of 15 deg F)	16.9
Total useful heat transfer	**38.3**

Example 2: Find the combined total useful heat transfer from a square foot of ceiling panel having a surface temperature of 100 F, when the average room air temperature is 70 F and the mean radiant temperature UMRT of the unheated room surfaces is 60 F.

Solution:

	Btu/(hr)(sq ft)
Radiation	
(Fig. 9 for UMRT of 60 F and t_s of 100 F)	35.3
Convection	
(Fig. 10, Curve B for a Δt of 30 F)	15.5
Total useful heat transfer	50.8

Since the greater number of panel installations are designed for an average room air temperature of 70 F, it is possible to present in chart form (see Fig. 11) the relationship between panel output (radiation plus convection) and the unheated mean radiant temperature for various surface temperatures and panel positions based on 70 F inside air temperature.



Output based on 70 F inside air temperature

FIG. 11. TOTAL PANEL OUTPUT RADIATION PLUS CONVECTION

From Fig. 11 the panel output for *Examples* 1 and 2 can be read directly, 38.3 and 50.8 Btu, respectively. Similarly, combined transfer rates for floor or ceiling panels operating under other conditions may be determined from Fig. 11, or from Figs. 9 and 10.

Before Figs. 9 and 11 can be used, the unheated mean radiant temperature UMRT of the room surface must be known. For the purpose of this work, the UMRT may be defined as the average temperature of the unheated surfaces of the space, weighted according to the areas of the surfaces.

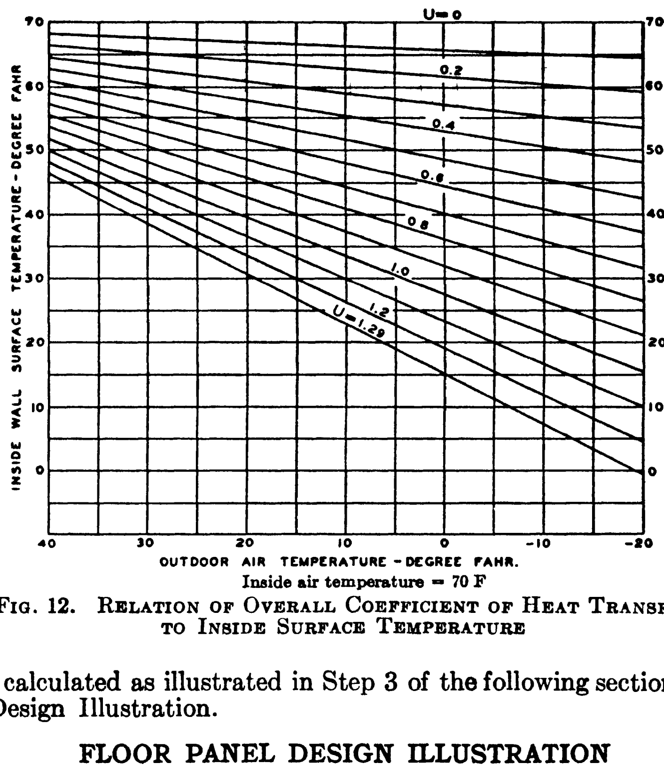
The surface temperature of the inside walls may be assumed to be the same as the inside air temperature. The surface temperatures of outside walls or exposed ceilings, for any combination of outside design temperature and transmission coefficient, may be obtained from Fig. 12. If the inside design temperature is other than 70 F, the values determined from

Fig. 12 must be corrected by values obtained from Fig. 13. The following example illustrates the use of Figs. 12 and 13.

Example 3: The outside walls of a room have a coefficient of heat transfer $U = 0.15$. Inside and outside design air conditions are 68 F and -20 F, respectively. Determine the inside surface temperature of the wall.

Solution: From Fig. 12 the inside surface temperature for 70 F and -20 F, and a U value of 0.15 is found to be 61.5 F. Then from Fig. 13, for a $\Delta t = -2$ deg and a U value of 0.15, the correction is -1.8 deg. As the correction is negative, the inside surface temperature will then be $61.5 - 1.8 = 59.7$ F. The total panel output can then be determined from Figs. 9, 10, and 11.

After the various wall surface temperatures have been determined, as described, the unheated mean radiant temperature UMRT of the space



Inside air temperature = 70 F

FIG. 12. RELATION OF OVERALL COEFFICIENT OF HEAT TRANSFER TO INSIDE SURFACE TEMPERATURE

may be calculated as illustrated in Step 3 of the following section on Floor Panel Design Illustration.

FLOOR PANEL DESIGN ILLUSTRATION

The several steps in the design of a panel heating system are listed and illustrated in the following example of the design of a floor panel for the room shown in Fig. 14.

1. *Selection of Design Conditions*

   a. Outside design temperature, -10 F.
   b. Inside design room air temperature, 70 F.
   c. In this problem a floor panel heating system will be used. No basement. Insulation will be used between slab and fill.
   d. Heating medium, hot water.
   e. Floor surface temperature, 85 F (Maximum).

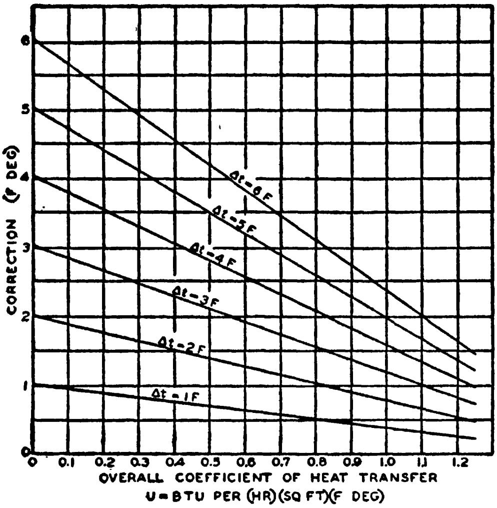


FIG. 13. INSIDE WALL SURFACE TEMPERATURE CORRECTION FOR INSIDE AIR TEMPERATURES OTHER THAN 70 F

$\Delta t = t_a - 70$
If Δt is positive: $t_w' = t_w +$ correction
If Δt is negative: $t_w' = t_w -$ correction

t_a = inside air temperature
t_w = inside wall surface temperature based on t_a = 70 F
t_w' = actual inside wall surface temperature

2. *Calculation of Room Heat Loss*

The heat transmission coefficients of the room surfaces are given in Fig. 14. They would usually be obtained from Chapter 9. The room heat loss can be calculated in the conventional manner, as outlined in Chapter 11. The calculations are shown in Table 1. The heat loss through the surface areas containing the heating medium is not included in the calculation.

The calculated rate of heat loss will determine the amount of heat which must be supplied by the heating panels to the room.

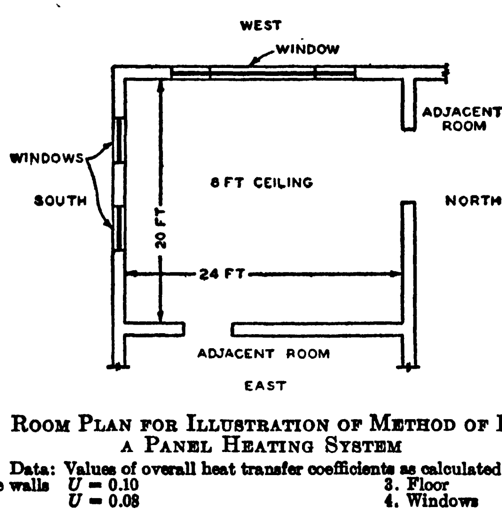


FIG. 14. ROOM PLAN FOR ILLUSTRATION OF METHOD OF DESIGNING A PANEL HEATING SYSTEM

Data: Values of overall heat transfer coefficients as calculated:

1. Outside walls	$U = 0.10$	3. Floor	$U = 0.36$
2. Ceiling	$U = 0.08$	4. Windows	$U = 0.55$

TABLE 1. CALCULATED HEAT LOSS OF ROOM (Fig. 14)

SURFACES	AREA SQ FT	U	CALCULATION	HEAT LOSS BTU/HR
Outside walls	272	0.10	272 x 0.10 x 80	2,176
Glass	80	0.55	80 x 0.55 x 80	3,520
Inside walls	352	—	No heat loss	0
Ceiling	480	0.08	480 x 0.08 x 80	3,072
Floor	480	—	Heating panel	0
Infiltration	—	—	3,840 cu ft x 1.50 x 80 x 0.018	8,295
Total heat loss				17,063

3. *Determination of UMRT*

The calculations for determining UMRT are shown in Table 2. The inside surface temperatures shown in the fourth column were determined from Fig. 12.

TABLE 2. CALCULATION OF UMRT

SURFACE		AREA SQ FT	INSIDE SURFACE[a] TEMPERATURE F	PRODUCT (AREA x TEMPERATURE)
Outside walls	0.10	272	65	17,680
Adjacent inside walls	—	352	70	24,640
Ceiling	0.08	480	66	31,680
Glass	0.55	80	43	3,440
Total		1,184		77,440

$$\text{UMRT} = \frac{\text{Total of products}}{\text{Total area}} = \frac{77{,}440}{1{,}184} = 65.4 \text{ F}$$

[a] Values of inside wall surface temperature for various U values and outside design temperatures found in Fig. 12.

4. *Determination of Panel Output*

Values of total panel output for various panel positions, panel surface temperatures and unheated MRT are given in Fig. 11. From Fig. 11, with a floor panel surface temperature of 85 F and a UMRT of 65.4 F, the total panel output is 34 Btu per (hr) (sq ft of panel surface).

Panel area = 24 x 20 = 480 sq ft
Panel output = 480 x 34 = 16,320 Btu per hr

This panel output of 16,320 Btu per hr is reasonably close to the calculated room loss of 17,063, and is satisfactory. If this value of total heat output from panel to the room were much less, *i.e.*, 10 or more percent less, than the calculated heat loss from the room, some method of supplementary heating would be needed. Floor panel surface temperatures exceeding 85 F are not recommended.

5. *Determination of Panel Input and Water Temperature*

Fig. 15 presents a graphical method of determining the required water temperature and total heat input to a panel for various cover depths, tube or pipe sizes and spacings, and rate of heat output to the room.

The panel output for this example has previously been found to be 34 Btu per (hr) (sq ft). Since some of the heat input to the panel is lost to the ground, it is necessary to supply more than 34 Btu per (hr) (sq ft) to the panel by means of the hot water heating medium. The amount of heat which must be supplied to the panel to give the required output may be determined graphically from Fig. 15. Assume that the panel has ¾ in. pipes, spaced on 12 in. centers, and that the depth of cover from top of panel surface to top of pipes is 2.5 in. The four sections of Fig.

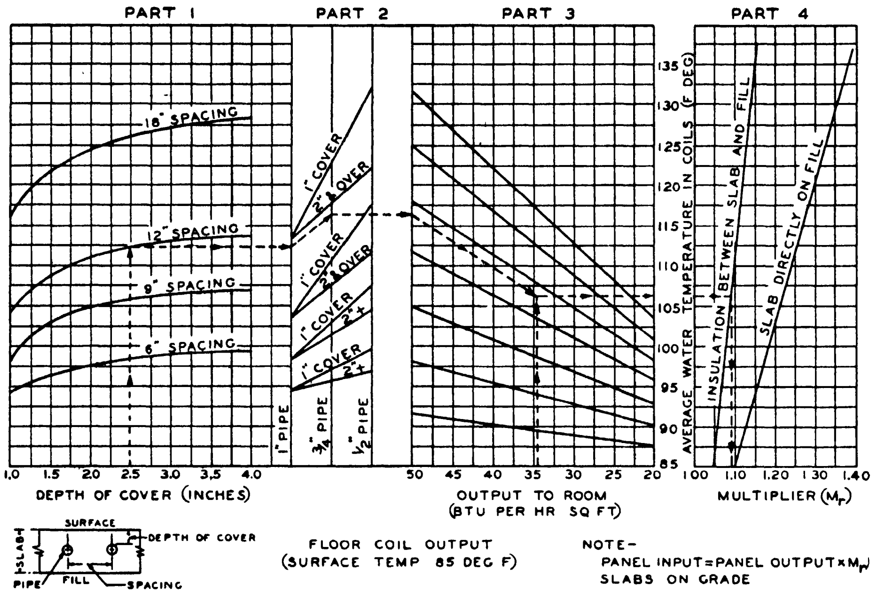


Fig. 15. Relation of Water Temperatures and Coil Outputs to Coil Spacing and Depth of Bury for Floor Coils in Slabs on Grade

15 are marked Part 1 to Part 4 and, as will be evident by following the dashed line on the chart, are used as follows:

Part 1. Starting from the depth of cover (2.5 in.), proceed vertically to the line representing pipe spacing on centers (12 in.), and then horizontally to the first ordinate of Part 2.

Part 2. Move parallel to the nearest upward sloping line, indicating "2 in. & over" cover, to intersect the ordinate representing ¾ in. pipe, and then proceed horizontally to the first ordinate of Part 3.

Part 3. Proceed parallel to and along the nearest downward sloping line to an intersection with the ordinate representing the panel output (34 Btu per sq ft), and then move horizontally to the right hand scale of Part 3 and read a required average water temperature in the coil of 106 F.

Part 4. From the water temperature 106 F just found in Part 3, proceed horizontally into Part 4 to intersect the line representing insulation between the slab and fill. Directly below this intersection read 1.09 (on the bottom scale) as the multiplier to be used.

The required panel input is, therefore, 1.09 x panel output, or 1.09 x 34 = 37.1 Btu per (sq ft) (hr).

The total required panel input is 37.1 x panel area = 37.1 x 480 = 17800 Btuh.

CEILING AND WALL PANEL DESIGN

Where coils are imbedded in plaster on ceilings or walls, design procedure is simplified considerably by the physical limitations of the space available. For tubing fastened to the underside of lath, the largest practical size is ⅝ in. O.D., while for ferrous pipes above the lath, it is 1 in. I.P.S. Therefore, the actual tube or pipe size selected is usually determined by the length of coil circuit and its flow resistance in consideration of the available circulating head.

In order to obtain a reasonably even heat distribution over the finished

plaster heating surface, pipes or tubes should be spaced on about 6 in. centers, and not over 9 in. centers. Within these limitations it is found in practice that heat output rates do not vary too seriously with variations in pipe and tube size and tube spacing.

In general, for plaster ceiling panels with tubes or pipes spaced on 4½ to 9 in. centers, the temperature of the circulating water is about 10 to 25 deg above the desired surface temperature.

The hourly heat output per square foot of panel surface may be found by means of Figs. 11, 12, and 13, and the calculations for room heat loss and UMRT, as previously illustrated in the example of floor panel design.

HOT WATER PIPING

When water is used as the heating medium, the piping layout and arrangement should be based on the design principles outlined in Chapter 21 for Two-Pipe Forced Circulation Systems. The pressure drops through the coils should be carefully calculated, and it is recommended that all branch circuits and coils be balanced to provide for uniform distribution by means of regulating valves or tees. Generally, a 15 to 20 deg total temperature drop is used in determining water flow rates, and a total pump head of more than 30 ft is undesirable due to noise caused by high water velocity.

Panel systems involving several rooms and panels comprising a single zone, require that all coils be selected for the same inlet water temperatures. Panel areas and pipe spacing must be selected to make this possible.

INSTALLATION DETAILS, ACCESSORIES, AND CONTROLS

Installation details, as given in Chapter 21, Hot Water Systems, also apply to piping systems for panel heating. Control problems of panel heating systems are discussed in Chapter 34, Automatic Control.

Efficient venting of air from the coils may be obtained by arranging the circulation so that the air moves in the direction of water flow to a high point in the system where an automatic float-type vent or connection to expansion tank should be provided for its release. Coils should be installed as nearly level as possible or, if they are on sloping surfaces due to structural conditions, they should be arranged to vent air at their high points. Arrangements of pumps, expansion tanks, drainage, and flow and return mains may be generally the same as for conventional hot water heating systems.

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CHAPTER 24

UNIT HEATERS AND UNIT VENTILATORS

Definitions; Unit Heaters: Classification, Application, Ratings, Location, Space Temperatures, Control, Piping, Boiler Capacity; Unit Ventilators: Ratings, Capacity Requirements, Application, Selection, Control, Location, Exhaust Flues; Window Ventilators

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DESCRIPTIONS of heating, cooling, ventilating, humidifying, and dehumidifying systems are given in other chapters. This chapter deals with unit heaters and unit ventilators. Unit air conditioners and unit coolers are discussed in Chapter 25.

Definitions

The generally accepted meaning of the word *unit* in the terms unit heaters, unit ventilators, and unit humidifiers, is that of a factory-made encased assembly of the functional elements indicated by its name. Such units can be shipped complete or in sections, so that the only field work necessary is the assembling of the sections, providing proper supports, and connecting the unit to sources of heat (or fuel), power and water supply and, if necessary, to vent pipes for combustion gases.

The term *unit heater* denotes an assembly of elements, the principal function of which is heating. The essential elements of a unit heater are a fan and motor, a heating element, a housing, and outlet vanes or diffusers.

The term *unit ventilator* denotes an assembly, the principal function of which is to ventilate. It may serve to circulate air within the space, or to introduce air from without the space, or may accomplish both purposes. The essential elements of a unit ventilator are a fan and motor, a heating element, a set of dampers, a housing, and outlet vanes or diffusers.

UNIT HEATERS

Classification

The various types of unit heaters which are at present available can usually be classified according to one of the three following methods:

1. *By type of heater.* Under this classification there are three types of heating elements to be considered: (*a*) the *steam or hot water type*, (*b*) the *electric type*, and (*c*) the *direct fired type* which may be *gas*, *oil*, or *coal fired.*

2. *By type of fan.* Under this classification there are two types of fans to be considered: (*a*) the *propeller type*[1] and (*b*) the *centrifugal type*. Either type may be arranged for horizontal or vertical delivery of air.

3. *By arrangement of elements.* Under this classification there are two types of heaters to be considered: (*a*) the *draw-through type*, in which the fan draws air through, and (*b*) the *blow-through type*, in which the fan blows air through the heater.

Unit heaters are available in any combination of the three preceding general classifications. For example, the steam or hot water type may be

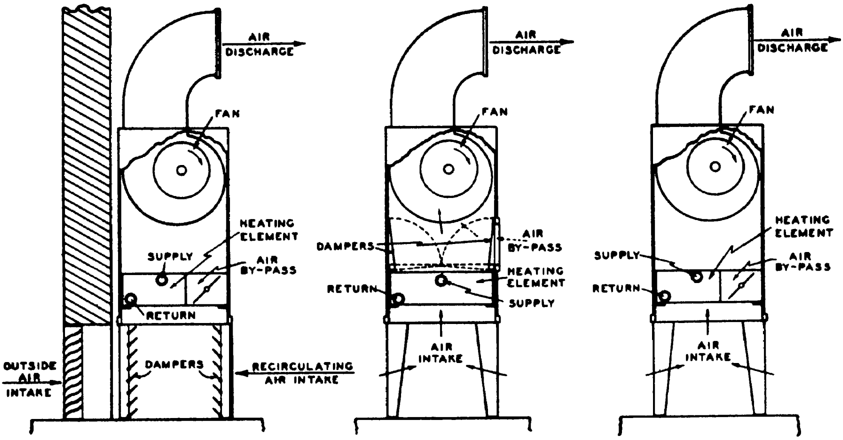

Fig. 1. Centrifugal Fan Type Unit Heater—Floor Mounted

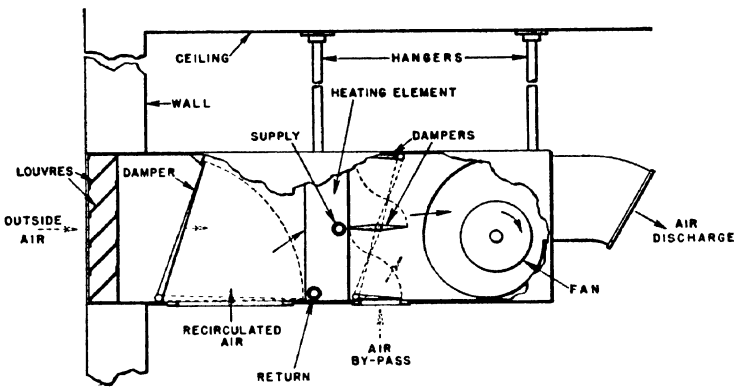

Fig. 2. Centrifugal Fan Type Unit Heater—Suspended

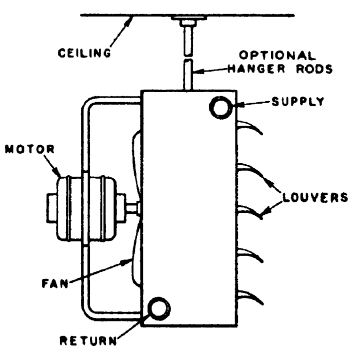

Fig. 3. Propeller Fan Type Unit Heater—Horizontal Blow

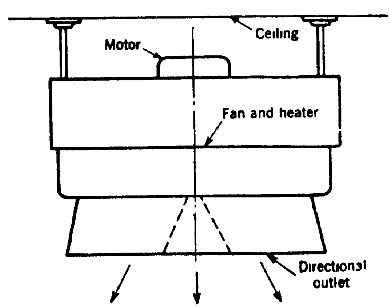

Fig. 4. Propeller Fan Type Unit Heater—Vertical Blow

secured with either the propeller or centrifugal type of fans, and in either the draw-through or blow-through type.

Unit heaters also vary in other minor respects. For example, steam and return inlets and outlets may be located on top and bottom, respectively, or on the same side of the unit. Some units are supported by the piping and some have independent supports. The heating surface of steam or water type units is generally made up of a non-ferrous tube-and-fin assembly; or it may be fabricated of steel, or cast in steel or iron.

Application of Unit Heaters

Steam or hot water unit heaters are used principally for heating commercial and industrial structures such as garages, factories, laboratories, and stores. They may also be used for heating finished rooms, if properly applied and concealed, and if some consideration is given to the problem of noise.

Unit heaters may also be adapted to a number of industrial processes, such as drying and curing, in which the use of heated air in rapid circulation with uniform distribution is of particular advantage. They may be used for moisture absorption, such as fog removal in dye houses, or for the prevention of condensation on ceilings or other cold surfaces of buildings in which process moisture is released. When such conditions are severe, it is necessary that the unit heaters draw air from outside in enough volume to provide a rapid air change, and that they operate in conjunction with ventilators or fans for exhausting the moisture-laden air. (See discussion of condensation in Chapter 9.)

There are a variety of applications which are favorable to the use of *electric unit heaters*. For supplemental heat in residence bathrooms, for the heating of ticket booths, watchmen's offices, factory offices, locker rooms and other isolated rooms scattered over large areas, their use is peculiarly adaptable. They are particularly useful in isolated and untended pumping stations or pits where they may be thermostatically controlled to prevent freezing temperatures.

Gas-fired unit heaters find application in industrial plants, offices, stores, garages; in fact, in almost every location where steam-type units are used. The installation cost of gas-fired units is usually less than that of a type requiring that a new boiler be installed, unless the number to be installed would justify the cost of a boiler and steam or hot-water unit heater system.

Oil-fired unit heaters are used in industrial plants, garages and commercial buildings.

Coal-fired unit heaters are of finned, welded steel, or cast-iron construction, and equipped with centrifugal blowers. They are usually stoker-fired to insure proper firing of fuel. They are used principally in large industrial plants such as foundries or assembly plants, and provide a convenient source of heat, are readily installed and economical in operation, since all heat given off by the surface of the heater remains in the heated space.

There are three major factors to consider in the application of unit heaters, namely: (1) location of unit, (2) air distribution, and (3) heating medium.

Outlet Velocities

Outlet velocities of unit heaters vary from about 400 to 2500 fpm, depending upon the type of unit and the distance to which the air is to be projected. Noise and drafts must be considered in the choice of air velocities, since both increase with increase of air velocity. Velocities and blow distances for the various types of unit heaters illustrated in Figs. 1, 2, 3 and 4 are given in Table 1.

In the selection of unit heaters it is important to ascertain that the blow is sufficient. The blow is dependent to a marked degree on the temperature of air leaving the heater, as well as upon its velocity. (See discussion under heading of Inlet, Outlet, and Space Temperatures with Unit Heaters.)

Air Outlets

In order to direct the air to points desired and to diffuse the air to avoid drafts, unit heaters are commonly equipped with directional outlets, adjustable louvers, or fixed types of diffusers.

Ratings of Unit Heaters

It is standard practice to rate unit heaters on the basis of the amount of heat delivered by the air in Btu per hour above an entering air temperature of 60 F. This applies to all types of unit heaters, the steam or hot water type, the electric type and the direct fired type. There are, however, other factors which must be taken into account, especially when an attempt is made to compare one type of heater with another. These are the temperature of the heating element and the velocity of air through it. Consideration is given to these factors in the discussion of ratings for each type of unit heater in the following paragraphs.

TABLE 1. OUTLET VELOCITIES AND DISTANCE OF BLOW FOR VARIOUS TYPES OF UNIT HEATERS

Type of Unit Heater	Outlet Velocities FPM	Distances of Blow—Ft[a]
Centrifugal Fan	1500–2500	20–200
Horizontal Propeller Fan	400–1000	30–100
Vertical Propeller Fan	1200–2200	70

[a] Refer to manufacturers' tables for limits of blow.

Steam. Rating of steam unit heaters has been standardized by a code[2] in which the following items are the basis of rating: dry saturated steam at 2 psig pressure at the heater coil; air at 60 F (29.92 in. Hg barometric pressure) entering the heater; and heater operating free of external resistance to air flow.

The capacity of a heater increases as the steam pressure increases, and decreases as the entering air temperature increases. The heating capacity for any condition of steam pressure and entering air temperature other than standard may be calculated approximately from any given rating by the use of factors in Table 2 for the blow-through or draw-through types.

Hot Water. A standard for the rating of hot water type unit heaters has also been established by code[3] in which the following items are the basis of rating: entering water at 200 F; entering air at 60 F (29.92 in. Hg barometric pressure); and heater operating free of external resistance to air flow. This code also prescribes a method of translating the output in Btu and the temperature rise as obtained under test conditions to standard conditions of air and water temperature.

Electric. Electric type unit heaters are available in sizes up to at least 60 kw capacity. They consist of resistance type heating elements combined with fan and motor, together with a suitable casing. Electric unit heaters

TABLE 2. CONSTANTS FOR DETERMINING THE CAPACITY OF UNIT HEATERS FOR VARIOUS STEAM PRESSURES AND TEMPERATURES OF ENTERING AIR

(Based on Steam Pressure of 2 psig and Entering Air Temperature of 60 F)

	Steam Pressure psig	Temperature of Entering Air											
		−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
Blow-Through Type	0	1.54	1.45	1.37	1.27	1.19	1.11	1.03	0.96	0.88	0.81	0.74	0.67
	2	1.59	1.50	1.41	1.32	1.24	1.16	1.08	1.00	0.93	0.85	0.78	0.71
	5	1.64	1.55	1.46	1.37	1.29	1.21	1.13	1.05	0.97	0.90	0.83	0.76
	10	1.73	1.64	1.55	1.46	1.38	1.29	1.21	1.13	1.06	0.98	0.91	0.84
	15	1.80	1.71	1.61	1.53	1.44	1.34	1.28	1.19	1.12	1.04	0.97	0.90
	20	1.86	1.77	1.68	1.58	1.50	1.42	1.33	1.25	1.17	1.10	1.02	0.95
	30	1.97	1.87	1.78	1.68	1.60	1.51	1.43	1.35	1.27	1.19	1.12	1.04
	40	2.06	1.96	1.86	1.77	1.68	1.60	1.51	1.43	1.35	1.27	1.19	1.12
	50	2.13	2.04	1.94	1.85	1.76	1.67	1.58	1.50	1.42	1.34	1.26	1.19
	60	2.20	2.09	2.00	1.90	1.81	1.73	1.64	1.56	1.47	1.39	1.31	1.24
	70	2.26	2.16	2.06	1.96	1.87	1.78	1.70	1.61	1.53	1.45	1.37	1.29
	75	2.28	2.18	2.09	1.99	1.90	1.81	1.72	1.64	1.55	1.47	1.40	1.32
	80	2.31	2.21	2.11	2.02	1.93	1.84	1.75	1.66	1.58	1.50	1.42	1.34
	90	2.36	2.26	2.16	2.06	1.97	1.88	1.79	1.71	1.62	1.54	1.46	1.38
	100	2.41	2.31	2.20	2.11	2.02	1.93	1.84	1.75	1.66	1.58	1.50	1.42
Draw-Through Type	0	1.48	1.41	1.33	1.25	1.18	1.11	1.03	0.96	0.89	0.82	0.75	0.69
	2	1.52	1.44	1.36	1.29	1.22	1.14	1.07	1.00	0.93	0.86	0.79	0.73
	5	1.57	1.49	1.41	1.33	1.26	1.19	1.11	1.05	0.98	0.91	0.84	0.77
	10	1.64	1.56	1.48	1.40	1.33	1.25	1.18	1.11	1.04	0.97	0.90	0.84
	15	1.69	1.61	1.53	1.46	1.38	1.31	1.24	1.17	1.10	1.03	0.96	0.90
	20	1.73	1.65	1.57	1.50	1.42	1.35	1.28	1.21	1.14	1.07	1.00	0.94
	30	1.80	1.73	1.65	1.57	1.50	1.42	1.35	1.28	1.21	1.15	1.08	1.01
	40	1.86	1.79	1.71	1.64	1.56	1.49	1.42	1.35	1.28	1.22	1.15	1.08
	50	1.93	1.85	1.77	1.70	1.63	1.55	1.48	1.42	1.35	1.28	1.21	1.15
	60	1.97	1.90	1.82	1.75	1.67	1.60	1.53	1.46	1.40	1.33	1.26	1.19
	70	2.02	1.94	1.87	1.80	1.72	1.65	1.58	1.51	1.44	1.38	1.31	1.24
	75	2.04	1.97	1.90	1.82	1.75	1.68	1.61	1.54	1.47	1.40	1.33	1.27
	80	2.06	1.99	1.91	1.84	1.77	1.70	1.63	1.56	1.49	1.42	1.35	1.29
	90	2.10	2.03	1.95	1.88	1.80	1.73	1.66	1.59	1.52	1.46	1.39	1.32
	100	2.15	2.07	1.99	1.92	1.85	1.77	1.70	1.63	1.56	1.49	1.43	1.36

Note: To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering air and 2 psig.

When increasing steam pressure it is important to determine whether the heater is suitable for the increased pressure application, and whether the resulting increased outlet temperature is satisfactory.

are made in the built-in-wall model, suspension model, and free-standing or portable model.

Electric unit heaters are rated on the energy input to the heater, expressed in terms of kilowatts, Btu or EDR. Quite often all three ratings are given in parallel columns in the catalogs.

Gas-Fired. Gas-fired unit heaters are built in both suspended and floor models, with either propeller or centrifugal type fans. They are available in a wide range of sizes from about 24,000 to over 4,000,000 Btu per hr capacity, and are usually rated in terms of both input and output according to the approval requirements of the *American Standards Association.* Any gas-fired unit which is thermostatically controlled and has a pilot must have an element in the pilot flame which will automatically close the gas valve on pilot failure.

Oil-Fired. The oil-fired unit heater is usually equipped with a centrifugal fan, and can be obtained in sizes ranging from 125,000 to 1,650,000 Btu per hr output capacity in standard units. It is furnished in either the floor-mounted or in smaller sizes in the suspended type.

Stoker-Fired. The stoker-fired type of unit heater can be obtained in ranges of from 300,000 to 6,000,000 (or more) Btu per hr output capacity. Ratings are based upon delivered output at heater outlet.

Effect of Resistance Upon Capacity

Unit heaters are customarily rated as free delivery type units. If outside air intakes, air filters, or ducts on the discharge side are used with the unit, a reduction in air and heating capacity will result because of this added resistance. The percentage of this reduction in capacity will depend upon the characteristics of the heater, and on the type, design and speed of the fans, so that no specific percentage reduction can be assigned for all heaters at a given added resistance. In general, however, propeller fan type units will experience a larger reduction in capacity than housed centrifugal fan units for a given added resistance, and a given heater will have a larger reduction in capacity as the fan speed is lowered. The heat output to be expected under other than free delivery conditions should be secured from the manufacturer.

Determining Unit Heater Requirements

The formulas given in the section on Unit Ventilators may be used to determine unit heater capacity requirements.

Location of Unit Heaters

Care should be taken in the location of unit heaters to insure free air circulation to the intake. The best arrangement is to locate units so that they discharge air nearly parallel to exterior walls, and in a direction which will produce a rotational circulation around the room. This is preferable to directing the discharge against the outside walls.

Various types and makes of unit heaters are illustrated in the *Catalog Data Section* of this edition. As hot blasts of air in working zones are usually objectionable, heaters mounted on the floor should have their discharge outlets above the head line, and suspended heaters should be placed in such manner and turned in such direction that the heated air stream will not be objectionable in the working zone. In the interest of economy, however, the elevation of the heater outlet and the direction of discharge should be so arranged that the heated air is brought as close above the head line as possible, yet not into the working zone.

In connection with the use of vertical type unit heaters, care must be exercised in the selection of the heater. It has been found that the higher the unit is placed above the floor, the lower must be the outlet temperature of the air leaving the heater in order that the heated air may be forced into the occupied zone.

Inlet, Outlet and Space Temperatures with Unit Heaters

In the selection of unit heaters for any particular design, consideration should be given to the temperature of air entering the heaters, as well as the temperature to be maintained in the working zone of the space. In general, the temperature differences per foot of elevation, when using unit heaters, are less than corresponding variations when using direct radiation.[4] High velocity units will maintain slightly lower temperature differences than low discharge velocity units. Correspondingly, units with lower discharge air temperature will maintain lower temperature differences than units with higher discharge temperatures. Directional control of the discharged air from a unit heater can be an important factor, added to qualities of reasonably good outlet velocity and outlet temperature, in effecting satisfactory distribution of heat and reducing floor-to-ceiling temperature difference.

Since the outlet temperature of air from a unit heater increases with the temperature of the heating medium, such as high pressure steam, heaters

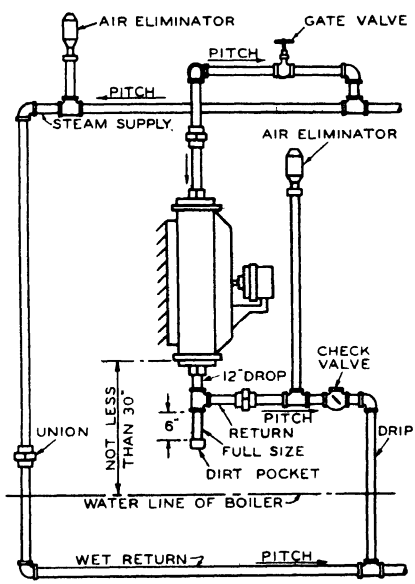

FIG. 5. UNIT HEATER CONNECTION TO ONE-PIPE GRAVITY STEAM SYSTEM

can be obtained with heating elements having less than the regular amount of heating surface in order to obtain, with the high temperature heating medium, approximately the same leaving air temperature as would be obtained from a lower temperature heating medium.

When some *outside air* is introduced, the temperature of the mixture of outside and recirculating air must be calculated and used as the entering air temperature at the heater. Unit heaters connected in this manner perform the function of unit ventilators. For a discussion of this function see the section of this chapter entitled *Unit Ventilators.*

For recirculating heaters located *at the floor* or with *intakes at the floor*, the temperature of air entering the heater should be assumed to be the same as that to be maintained in the room itself.

Automatic Control of Unit Heaters

Thermostatic control of unit heaters may be accomplished either by starting and stopping the fan, or by controlling the flow of the heating medium to the heating element. If the fan is controlled, it is advisable to provide a temperature-operated switch to prevent the fan from starting until the heating element is heated throughout. Unit heaters may be used in summer as a means of circulating air to give some measure of comfort due to air motion. In such cases the heating element should be shut off from the source of heat. The thermostat which prevents the fan from starting until the heating element is heated, should be provided with a bypass switch, which, upon being closed, will permit the fan to be operated independently of the heating element.

Piping Connections For Steam Unit Heaters

Piping connections for steam unit heaters are similar to those for other types of fan blast heaters. The piping of unit heaters must conform strictly to the system requirements while at the same time permitting the heaters

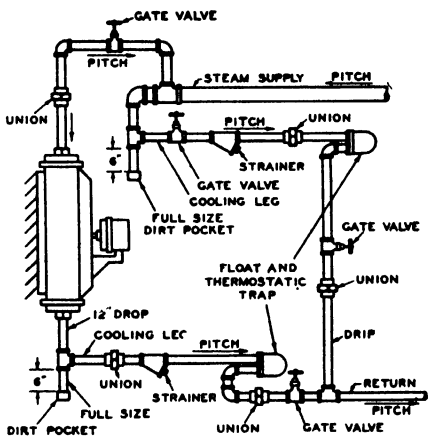

FIG. 6. UNIT HEATER CONNECTION FOR VACUUM OR VAPOR SYSTEM DISCHARGING CONDENSATION INTO DRY RETURN

themselves to function as intended. The basic piping principles for steam systems are discussed in Chapter 20.

Rapid condensation of steam, especially during heating-up periods, is characteristic of this type of equipment. The return piping must be planned to keep the heating coil free of rapid condensation, while the steam piping must be ample to carry a full supply of steam to the unit to take the place of that condensed. Adequate sizes of piping are especially important where a unit heater fan is operated under start-and-stop control, and where all or part of the air is taken from the outside. In such installations the condensation rate may vary rapidly, and the necessity for ample pipe capacity is particularly important.

A method of connecting a unit heater to a one-pipe gravity system is illustrated in Fig. 5. When the return main is located above the boiler water line, an artificial water line must be created by providing an equalizing loop to prevent steam passing into the return, and thus into other units.

A piping arrangement where both the air and condensate pass through a common return to a boiler, with vent trap or condensate pump and receiver, is shown in Fig. 6. The traps must pass air and condensate rapidly to keep the return piping partially full of water.

Since unit heaters are often constructed with sufficient strength, the use of high pressure steam in them is a common practice. As shown in Fig. 7, the condensate and air reach the overhead return through traps, and check valves are located in the return piping. It is, however, preferable to locate the high pressure return below the heater.

For two-pipe closed gravity return systems, the return from each unit should be fitted with a heavy-duty or blast trap, and an automatic air valve should be connected into the return header of each unit heater. Provision must be made to compensate for the pressure drop by elevating the unit heater above the water line of the boiler or of the receiver.

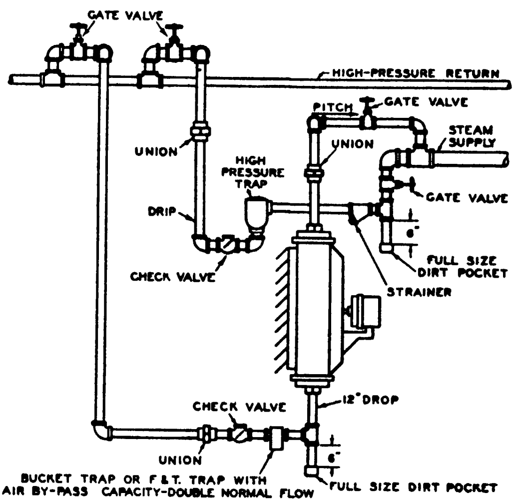

FIG. 7. METHOD OF CONNECTING UNIT HEATER TO HIGH PRESSURE SYSTEM

In pump and receiver systems, the air may be eliminated by individual air valves on the heaters, or it may be carried into the returns as in vacuum systems, and the entire return system be free-vented to the atmosphere, provided all units, drip points, and radiation are properly trapped to prevent steam entering the returns.

On vacuum or open vent systems, the return from each unit should be fitted with a large capacity trap to discharge the water of condensation, and with a thermostatic air valve for eliminating the air, or with a heavy-duty trap for handling both the condensation and the air, provided the air finally can be elininated at some other point in the return system.

For high pressure systems the same kind of traps may be used as with vacuum systems, except that they must be constructed for the pressure used. If the air is to be eliminated at the return header of the unit, a high pressure air valve can be used; otherwise the air may be passed with the condensate through the high-pressure return trap, and then eliminated at some other point in the system.

Fig. 8 represents the connections to a hot water heating system. The air vent is not required if the main is above the heater in which case air can be eliminated through the piping system.

BOILER CAPACITY FOR STEAM UNIT HEATERS

The capacity of the boiler should be based on the rated capacity of the unit heaters at the lowest entering air temperature and highest fan speed that will occur, plus an allowance for pipe line losses. It is unwise to install a single unit heater as the sole load on any boiler, particularly if the unit heater motor is started and stopped by thermostatic control. The wide and sudden fluctuations of load that occur under such conditions would require closer attention to the boiler than is usually possible in a small installation. Where oil or gas fuel is used in the boiler, it is possible by means of a pressure operated switch to control the boiler, in response to this

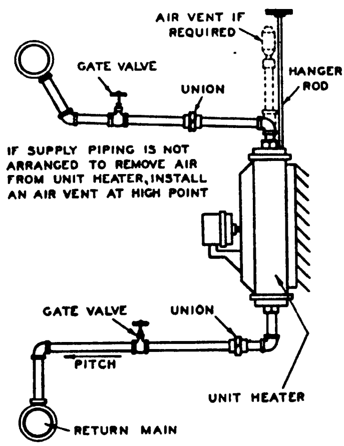

Fig. 8. Method of Connecting Unit Heater to Hot Water System

rapid fluctuation. In most cases, and particularly where the boiler is coal-fired, it is advisable to use two or more smaller units instead of one large unit heater.

Steam pressures below 5 lb can be used with safety for recirculating unit heaters when their heating surfaces are designed for those pressures, and when proper provision is made for returning the condensate. If units receive air that may be at a temperature below freezing, a steam pressure of not less than 5 lb should be maintained in the heating element, or a corresponding differential in pressure between the supply and return piping should be maintained by means of a vacuum.

UNIT VENTILATORS

A unit ventilator is essentially a modified unit heater. In addition to heating, it has the additional function of introducing outdoor air for ventilation and cooling. The typical unit is arranged to introduce outdoor air and recirculated air to the room in varying quantities and is equipped with a system of control that permits both the heating and cooling effect to be varied while the fans are operating continuously. Either steam or forced hot water may be employed as the heating medium. Unit ventilators are intended primarily for use in schools, meeting rooms, offices, or other applications where the density of occupancy indicates the need for ventilation. In normal operation, the discharge air temperature from a unit ventilator is varied in accordance with the room demands. Where a heating effect is required, the air delivered is above room temperature. Where the heat generated within the room by occupants, sun, etc., is sufficient to cause overheating, the air delivery temperature must be below that of the rooms. It is customary to equip unit ventilators with control devices that prevent the delivery of air at a temperature that will cause cold drafts. Unit ventilators may be of the heating element or damper controlled type, constructed on the blow-through (Fig. 9) or draw-through principle as illustrated in Fig. 10.

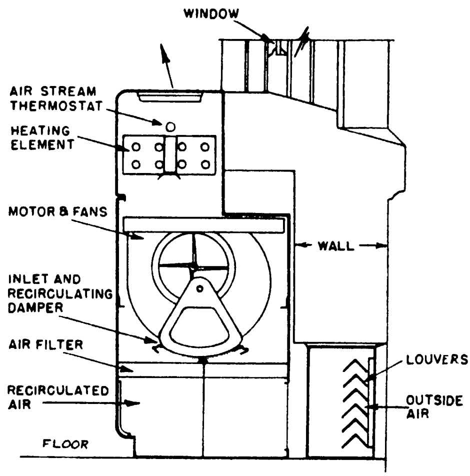

FIG. 9. TYPICAL BLOW-THROUGH TYPE UNIT VENTILATOR SHOWING ONE OF MANY ARRANGEMENTS OF DAMPERS AND HEATING ELEMENTS

Ratings of Unit Ventilators

Unit ventilators are customarily cataloged with two ratings: the *anemometer rating* and the *standard air rating*. The anemometer air rating is peculiar to school house ventilation and was for many years the standard by which school officials checked and specified unit ventilator capacities. It originated as a convenient field measurement for checking air quantities, and is the basis of rating for ventilation requirements under many state and local building codes. The anemometer rating (air capacity) is obtained by averaging the air velocities obtained by an anemometer held over equal subdivisions of the outlet grille at a distance of 2 in. from the grille, and multiplying the average velocity by the gross area of the grille. (The average velocity is sometimes obtained by moving the anemometer over the outlet 2 in. from its face.) The anemometer rating is based on the final temperature of the air leaving the grille while the unit is delivering outdoor air and room air in the proper proportion and the heating element is supplied with steam or hot water as specified.

The standard air rating is obtained in accordance with the A.S.H.V.E.

Standard Code for Testing and Rating Steam Unit Ventilators.[5] This code requires that the following rating information be supplied:

Rating Factors to be Specified. The rating of the unit ventilator shall specify:

a. Final temperature at different entering air temperatures.

b. Total EDR at different entering air temperatures.

c. Air delivered by the unit in cubic feet per minute at the *standard basis of rating* with the fans operated at rated speed, with all air being blown through the heating unit, and with the standard louver and grille on the outlet.

The Standard Basis of Rating shall be as follows:

a. Dry saturated steam at a temperature at the unit corresponding to an absolute pressure of 16.7 psi (218.5 F).

b. Entering air temperature of zero Fahrenheit degrees.

c. Volume delivered in cubic feet per minute converted to standard air at 70 F.

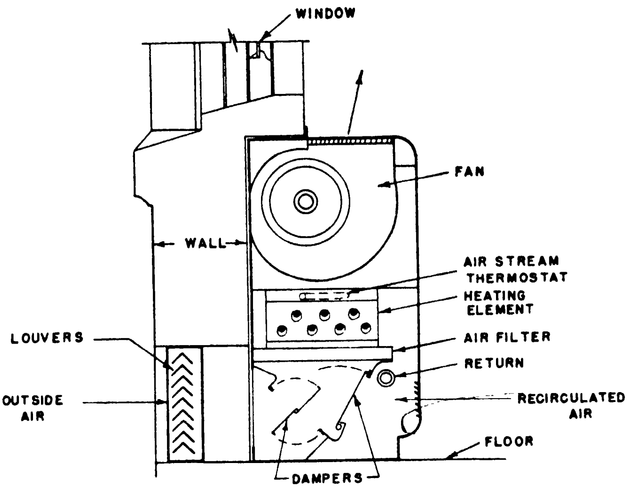

Fig. 10. Typical Draw-Through Type Unit Ventilator Showing One of Many Arrangements of Dampers and Heating Elements

Rating Tables for unit ventilators shall contain the following data in addition to the standard rating, for entering air temperatures from −30 F to +60 F:

a. Inlet temperature, Fahrenheit degrees.

b. Final temperature, Fahrenheit degrees.

c. Total EDR at the specified entering temperature.

d. Surplus or heating EDR at the specified entering temperature.

Surplus or Heating Equivalent Direct Radiation for the purposes of this code shall be construed to mean difference between the total EDR at a specified inlet temperature and the EDR required to heat the air from that temperature to 70 F.

Table 3 shows the air handling capacities by the two methods of rating and the approximate room heating equivalent in EDR of an intermediate size of heating element. Heating elements are available for either a higher or lower capacity.

Heating Capacity Requirements for Unit Ventilators

Since a unit ventilator has the dual function of introducing outdoor air for ventilation and maintaining a specified room temperature, the heat required by the unit may be similarly divided as (1) ***heat required for ventilation*** (H_v) and (2) ***surplus heat*** (H_s). The *surplus heat* is available for maintaining room temperatures. If auxiliary radiation is installed, the *surplus heat* requirement may be reduced by a corresponding amount. The sum of H_v and H_s is the total heat (H_t) to be supplied by the unit ventilator.

These quantities of heat are related by the following equations:

$$H_v = 0.24\ W\ (t - t_o) \quad (1)$$
$$H_t = 0.24\ W\ (t_f - t_o) \quad (2)$$
$$H_s = H_t - H_v = 0.24\ W\ (t_f - t) \quad (3)$$
$$W = d\ 60\ Q \quad (4)$$
$$H_t = H_s + 0.24\ d\ 60\ Q\ (t - t_o). \quad (5)$$

where

d = density of air, pounds per cubic foot (0.075 lb per cu ft for Standard Air by definition).
H_s = surplus heat, Btu per hour.
H_v = heat required to warm air for ventilation, Btu per hour.
H_t = total heat requirements for both heating and ventilation, Btu per hour.

TABLE 3. TYPICAL CAPACITIES OF UNIT VENTILATORS FOR AN ENTERING AIR TEMPERATURE OF ZERO

Cubic Feet of Air per Minute		Total Capacity in Square Feet, Equivalent Direct Radiation	Capacity Available for Heating the Room, Square Feet Equivalent Direct Radiation	Final Air Temperature F Deg
Anemometer Rating	Standard Air Rating			
750	500	214	56	95
1000	750	320	84	95
1260	1000	427	112	95
1560	1250	534	141	95

Q = volume of air handled by the ventilating equipment, cubic feet per minute.
t = temperature to be maintained in the room, Fahrenheit degrees.
t_o = outside temperature, Fahrenheit degrees.
t_f = temperature of the air leaving the unit, Fahrenheit degrees.
W = weight of air circulated, pounds per hour.
0.24 = specific heat of air at constant pressure (approximate value).

Example 1: The heat loss of a certain room is 24,000 Btu per hour, and the ventilating requirements are 1000 cfm. If the room temperature is to be 70 F and all air is taken from the outside at zero, what will be the total heat demand on the unit if it is required to provide for both the heating and ventilating requirements (combined system)?

Solution: Since the surplus heat is available to replace the heat loss of the room,

$$H_s = 24{,}000 \text{ Btu per hour.}$$

Substituting in Equation 5:

$$H_t = 24{,}000 + 0.24 \times 0.075 \times 60 \times 1000\ (70 - 0) = 99{,}600 \text{ Btu per hour}$$

$$t_f = \frac{24{,}000}{0.24 \times 0.075 \times 60 \times 1000} + 70 = 92.2 \text{ F}$$

If in *Example 1* a 1000 cfm (Standard Air) unit were required, but only

25 percent of the air introduced were outdoor air, the solution would be as follows:

$$H_t = 24{,}000 + 0.24 \times 0.075 \times 60 \times 0.25 \times 1000\ (70 - 0) = 42{,}900$$

$$t_f = \frac{24{,}000}{0.24 \times 0.075 \times 60 \times 1000} + 70 = 92.2$$

The only difference from *Example 1* is that, in the latter case, the ventilation load has been reduced.

Applications of Unit Ventilators

Items to be considered in the application of unit ventilators are: (1) combination with other means of heating, (2) selection of unit ventilator size, (3) cycle of control, (4) location of units, and (5) method of venting and exhausting.

In a *split* system the unit ventilator heat output is supplemented by that of additional radiators or convectors, and consequently a corresponding reduction in required unit ventilator heating capacity may be made.

The *combined* system employs a unit ventilator with sufficient heating capacity for both ventilation and normal heat losses. In such a case no direct radiation is required. The cost of installation of a combined system is usually less than that of a split system. The unit ventilator is normally arranged so that a fixed minimum of outdoor air is introduced at all times. The amount of outdoor air may be governed by state or local codes, or may be calculated by the engineer to meet the need of the application. In some cases, a variable amount of outdoor air is introduced depending on outside temperatures. This is done in order to conserve fuel, with some sacrifice of ventilation.

Selection of Unit Ventilator Size

The primary consideration in the selection of the size of unit ventilator is the number of occupants in the space. Other factors to be considered are state and local code requirements, volume of the room, density of occupancy, and the usage of the room. A safe rule for determining air capacity is to allow a total air quantity of 30 cfm per person, or six to nine room air changes through the unit, whichever is greater. If the ventilation requirements are less than this value, some air may be recirculated from the room. With this quantity of air handled, it is possible to obtain satisfactory cooling in mild weather. Since cooling is an important function, the unit ventilator must be selected to supply adequate air quantities.

After selecting the basic size of unit, the coil capacity to meet the heating requirement can be determined from the manufacturer's tables.

Control of Unit Ventilators

Three cycles of control are available for use with unit ventilators. These cycles of control determine the sequence of operation of the dampers and heating element as follows:

Cycle X—All Outdoor Air. During the heating-up period, the damper remains closed to outdoor air and the unit ventilator recirculates room air. Just before the desired room temperature is reached, the room thermostat operates the damper to admit only outdoor air. No recirculation takes place during the periods of room occupancy.

Cycle Y—A Variable Quantity of Outdoor Air with a Fixed Minimum. During the heating-up period, the damper remains closed to outdoor air and the unit ventilator recirculates room air. Just before the desired room temperature is reached, the room thermostat causes the damper to open to admit the desired minimum quantity of outdoor air, the balance of the air being taken from the room. As long as the minimum quantity of outdoor air is sufficient to prevent overheating, the damper remains in this minimum position. If more outdoor air is needed for cooling, the damper, under control of the room thermostat and the air stream thermostat, is operated to increase the proportion of outdoor air as needed, up to the maximum.

Cycle Z—A Variable Quantity of Outdoor Air without a Fixed Minimum. During the heating-up period, the damper remains closed to outdoor air and the unit ventilator recirculates room air. Just before the desired room temperature is reached, the air stream thermostat, placed ahead of the heating element to control the mixture of room air and outdoor air, assumes full control over the damper. Thereafter, this instrument positions the damper to maintain a predetermined constant-temperature mixture of indoor and outdoor air during all occupied periods. Meanwhile, the heating element, under the control of the room thermostat, adds just sufficient heat to the air mixture to maintain the desired room temperature.

Location of Unit Ventilator

The location of the unit ventilator in a room is important. Wherever possible it should be placed against an outside wall and on the center line

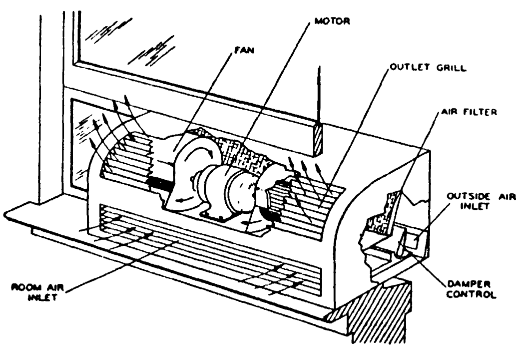

FIG. 11. TYPICAL WINDOW VENTILATOR

of the room. It is difficult to obtain proper air distribution if the unit is installed either on an inside wall or in a corner of the room. Standard units discharge the air stream upward, but for special cases units may be installed to discharge air horizontally. Units may be set against the wall or partially recessed into the wall to save space without materially affecting the results.

Air Exhaust Vents and Flues

The size and location of the air exhaust vent[6] outlet are important and in many cases, are regulated by laws for public buildings. Where no codes govern, the location and size of vents are left to the discretion of the engineer.

Best results have been obtained with a velocity through the vent openings nearly equal to that at which the air is introduced into the room, thus maintaining a slight pressure in the room. Calculated velocities at the vent openings of from 600 to 800 fpm produce the best diffusion results from this system. Many states, however, have regulations that will not permit velocities as high as 800 fpm. If a vent opening at or near the floor is near a desk or place where a person is seated, a velocity of 800 fpm in the vent opening will produce an objectionable draft. In such a case the velocity in the vent opening should not exceed 400 to 450 fpm, although duct velocities may be maintained at 600 to 800 fpm if codes permit.

In school buildings provided with wardrobes or cloakrooms, the vents may be so located that the air passes through these spaces, ventilating them with air which otherwise would be passed to the outside without being used to the best advantage. Many state and local codes for ventilation of public buildings make this arrangement mandatory.

WINDOW VENTILATORS

A window ventilator illustrated in Fig. 11 consists of filter and switch controlled motor-driven fans enclosed in a cabinet to be mounted on the window sill. Such units accomplish ventilation, air cleaning, and air circulation, but have no means of heating the air. The direction of air discharge is manually adjustable for seasonal operation.

REFERENCES

[1] See *National Association of Fan Manufacturers* standard definitions in Chapter 32.

[2] Standard Code for Testing and Rating Steam Unit Heaters (A.S.H.V.E. Transactions, Vol. 36, 1930, p. 165), prepared by a Joint Code Committee of the American Society of Heating and Ventilating Engineers and the *Industrial Unit Heater Association* and adopted 1930. Code revised 1950.

[3] Standard Code for Testing Hot Water Unit Heaters prepared by Engineering Committee of *Industrial Unit Heater Association*. Adopted by *Industrial Unit Heater Association* August 1942 and published September 1942.

[4] A.S.H.V.E. Research Report No. 958—Temperature Gradient Observations in a Large Heated Space, by G. L. Larson, D. W. Nelson and O. C. Cromer (A.S.H.V.E. Transactions, Vol. 39, 1933, p. 243). A.S.H.V.E. Research Report No. 1011—Tests of Three Heating Systems in an Industrial Type of Building, by G. L. Larson, D. W. Nelson and John James (A.S.H.V.E. Transactions, Vol. 41, 1935, p. 185).

[5] A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators (A.S.H.V.E. Transactions, Vol. 38, 1932, p. 25).

[6] A.S.H.V.E. Research Report No. 936—Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson and R. W. Kubasta (A.S.H.V.E. Transactions, Vol. 38, 1932, p. 463). A.S.H.V.E. Research Report No. 1017—Air Supply to Classrooms in Relation to Vent Flue Openings, by F. C. Houghten, Carl Gutberlet and M. F. Lichtenfels (A.S.H.V.E. Transactions, Vol. 41, 1935, p. 279).

CHAPTER 25

UNIT AIR CONDITIONERS AND UNIT AIR COOLERS

Definitions, Classification of Unit Type Equipment, Component Parts of Unit Type Equipment, Sound Isolation, Modifications of Remote Units, Ratings of Unit Air Conditioners, Application of Unitary Equipment, Unit Air Coolers

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THIS chapter presents the physical characteristics of air cooling units and air conditioning units; a suggested procedure for selection of units; and some of the factors involved in the application of unitary equipment. In general, factory produced unit equipment can be obtained to accomplish all of the functions possible from field assemblies, but the advantages of unit equipment are most apparent in small and moderate capacities. Above 12,000 cfm capacity, or approximately 40 tons of refrigeration capacity, handling and assembly costs generally favor the use of field assembled units. Multiple application of unitary equipment is frequently justified for large gross tonnage installations where zoning or a minimum amount of air distributing ducts is desirable.

DEFINITIONS

The term *air conditioning unit* has been loosely used as a name for all types of factory produced air handling, cooling, or heating units. A joint committee[1, 2, 3] on Rating Refrigerating Equipment has defined the various types of unitary equipment:

1. A *Cooling Unit* is a specific air treating combination consisting of means for air circulation and cooling within prescribed temperature limits.[1]
2. An *Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning, and heat transfer, with control means for maintaining temperature and humidity within prescribed limits.[1]
3. A *Cooling Air Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning, and heat transfer, with control means for cooling and maintaining temperature and humidity within prescribed limits.[1]
4. A *Self-Contained Air Conditioning or Cooling Unit* is one in which a condensing unit is combined in the same cabinet with the other functional elements. Self-contained air conditioning units are classified[2] according to the method of rejecting condenser heat (water cooled, air cooled, and evaporatively cooled), method of introducing ventilation air (no ventilation, ventilation by drawing air from outside, ventilation by exhausting room air to the outside, or ventilation by a combination of the last two methods), and method of discharging air to the room (free delivery or pressure type).
5. A *Free Delivery Type Unit* takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.[1]
6. A *Pressure Type Unit* is for use with one or more external elements which impose air resistance.[1]
7. A *Forced-Circulation Air Cooler* is a factory encased assembly of elements by which heat is transferred from air to refrigerants.[3]

CLASSIFICATION OF UNIT TYPE EQUIPMENT

Field assembled apparatus as described in Chapter 29 can be designed in shape, size, and capacity for any application, with the refrigeration and heating system exactly balanced to load conditions. To obtain the economies of mass production, factory built units must be standardized in a few models per manufacturer. Each model covers a range of capacities within the capacity of its fan to deliver air against the resistance of the unit and against the system resistance. For this reason, the unit performance will usually represent a compromise between actual load requirements and the rated capacity. Within the range of accuracy of most load calculations, this compromise is not objectionable.

If the condensing unit and cooling and heating coil surfaces are carefully selected, and if proper consideration is given to reduction of piping losses, the performance of the combined system will compare favorably with field assembled apparatus. A system, in which the air handling unit is separated from the condensing unit, is called a remote system, and the conditioning unit is designated as a remote unit. The economical capacities of remote units usually range from 10 to 40 tons.

For applications where load calculations are subject to considerable variance, and where close control is not considered essential, further economies of factory assembly can be obtained by combining the air handling and condensing equipment in one unit. This effects another compromise between load calculations and equipment selection, since the capacity of the combined unit is then dependent on the predetermined balance between a particular coil and condensing unit. These combination units are called self-contained units and, under the aptly descriptive name *store conditioners*, find economical application in 3, 5, 7½, 10, and 15 ton refrigeration capacities. These capacities, or limited multiples thereof, meet the load requirements of the majority of small and medium sized commercial establishments. With some modifications, these units can also be adapted to light industrial work.

To meet the requirements of individual comfort in small rooms and offices, where load calculations are subject to the indefinite design condition of *feeling cool*, self-contained units, called room coolers, find extensive and economical application. These units are usually restricted to summer and intermediate season operation, and range from ⅓ to 1½ tons of refrigeration capacity.

A special application of remote units is found in the unit air cooler which is used extensively in refrigeration work. Its primary function is to reduce temperatures in insulated and sealed storage spaces, and humidity control is a secondary consideration. Because of the small temperature differences between the coil and room temperatures, unit coolers handle three to five times as much air per ton as remote units used in air conditioning.

The attic fan or exhaust fan is sometimes referred to as a cooling unit, but since it contains no element of heat transfer, it is treated in Chapter 32, Fans.

COMPONENT PARTS OF UNIT TYPE EQUIPMENT

Units can be obtained for producing any of the required effects on air. As they function most satisfactorily when doing the work for which they were designed, field modifications are usually inadvisable because of expense involved, as well as the possibility of causing unexpected difficulties in operation. The basic design considerations of unitary equipment are discussed in the next following paragraphs.

Remote Units. Remote units can be obtained in two general classes, horizontal as shown in Fig. 1 and vertical as in Fig. 2. Their construction is essentially the same, except for the drain pan and filter locations.

Casings. Casings are generally constructed of sheet metal with angle iron frames and with removable panels for access to coil connections, blower bearings, filters, and drain. Casings should be air tight. Panels should be tight fitting with cam or similar fastenings for easy opening. Panel openings at coils for heavy units should be large enough to receive coils after the casing is suspended. Frames should be fitted with lugs strong enough to suspend horizontal units. Non-metallic casings are of advantage in small remote units for reducing sound, particularly when propeller type fans are used.

Insulation. Remote units are available with waterproof and vermin-proof, sound and heat absorbing insulation on the inside of the casing. They are also available with flanges and flanged access doors to permit insulation after installation.

Drain Pans. Because of the corrosive effect of mild picric, carbonic, and sulfurous acids absorbed by condensate, drain pans are usually made of 14 gage or heavier metal. They should be hot dipped galvanized after fabrication, or otherwise treated to resist corrosion. Some manufacturers extend the drain pan under the entire unit, but in any case it should extend far enough to catch any condensate carried over from the coils. The drain connection should be readily accessible for cleaning and, in air conditioning work, should be generously sized and trapped. Some municipal codes require a minimum size of $1\frac{1}{4}$ I.P.S.

Blowers. The usual practice among manufacturers is to use light construction in the blowers in remote units, although a few are available with heavy duty blowers in the larger sizes. These blowers work under almost constant conditions without overload or shock, and will usually last as long as the unit with reasonable maintenance. As lubrication of bearings is very important, it is good practice to locate the oil cups conveniently outside of the unit. In any application where considerable dehumidification or humidification is involved, such as in a system where the unit is handling 100 percent outside air, it is important that the blowers be painted with asphaltum or other corrosion resistant paint to prevent excessive oxidation and corrosion of the blowers.

Some of the smaller suspended type units, Fig. 3, use propeller fans with a trailing edge blade in order to obtain required pressure characteristics with quiet operation. Since most of the motors driving these fans are direct-connected and use brushes for starting, adequate access should be provided for maintenance and inspection.

Cooling Coils. The cooling and dehumidifying coils used in unit air conditioners are essentially the same as those used in central station units The face area of the coil is usually fixed, and the number of rows deep in the direction of air flow is the variable that determines capacity. It should be remembered that adequate coil surface is important for efficient performance of any system, and that there is little economy in reducing coil depth to less than four rows.

Where multiple circuits are used in the larger coils, equal distribution of the cooling medium to the various circuits is vital in order to develop full capacity of the coil. The cooling coils also perform the function of dehumidification. To prevent carryover of condensate, eliminator plates

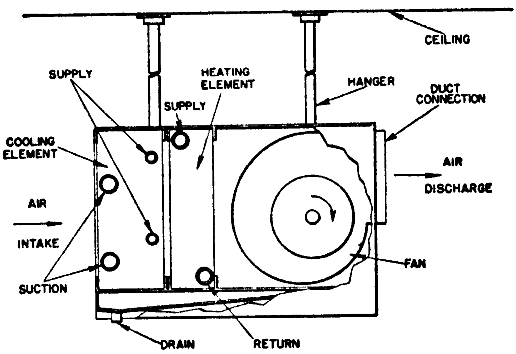

FIG. 1. HORIZONTAL REMOTE TYPE UNIT AIR CONDITIONER

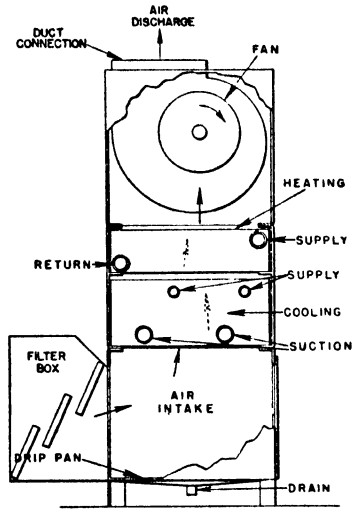

FIG. 2. VERTICAL REMOTE TYPE UNIT AIR CONDITIONER

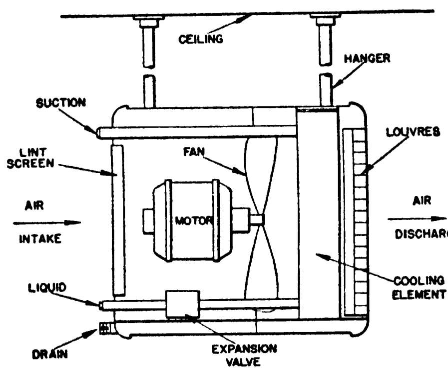

FIG. 3. SUSPENDED PROPELLER FAN TYPE UNIT AIR CONDITIONER

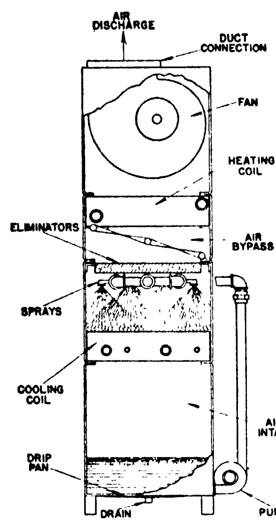

FIG. 4. SPRAY TYPE REMOTE UNIT AIR CONDITIONER

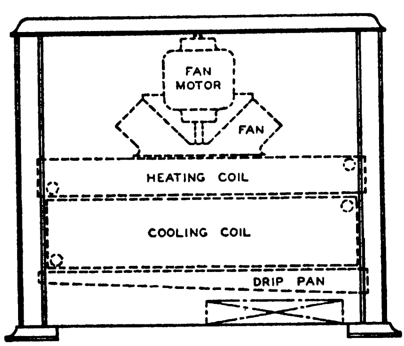

FIG. 5. REMOTE FLOOR TYPE ROOM UNIT AIR CONDITIONER

should be used if face velocities exceed 500 to 530 fpm, unless adequate means of catching the droplets are provided.

When air is drawn upward through dehumidifying coils, as in some vertical units, water is entrained within the fins and held in suspension. This increases the resistance pressure against which the blower operates, and results in wide variation in air volumes handled between dry and wet coil conditions. Some unit manufacturers have so designed vertical units that the air passes through the coils horizontally in order to overcome this difficulty.

Heating Coils. Heating coils of unit air conditioners are usually conventional blast coils, and can be obtained with or without non-freeze steam distribution features. They usually match the cooling coils in face area, and are one or two rows deep depending on the heating requirements. Where coils are selected for hot water and have more than two rows of tubes, the air resistance and the space requirements of the total number of rows of cooling and heating coils should be carefully checked.

Humidification. Spray type humidifiers are usually used in remote systems, but in some cases pan type humidifiers or steam humidifiers are also used. Some condensation on the inside of the unit casing may occur with possible water damage if the unit is located in a cold space without adequate insulation. Spray type humidifiers should be located so that no carryover of moisture occurs.

Filters. It is almost axiomatic that all units should have filters. Some small suspended units of 1 ton capacity or less, with low coil face velocities and propeller fans, are equipped only with lint screens or operate without filters, but in them the coils must be periodically cleaned and there is constant danger of clogging of the drain, with the possibility of water damage. Filters used are usually of the throw-away type, although cleanable filters are available for most of the larger units. Care should be taken to insure adequate filter surface, since the cross-sectional area of the unit is seldom adequate for filter area. V-shaped or staggered filter arrangements are quite commonly used to increase filter area.

Motors. In some units where the motor is mounted inside, adequate access for maintenance and clearance for tightening belts are imperative. When motors are mounted in this manner, the heat equivalent of the motor input must be added to the heat load to be absorbed by the system, and this requires a lower exit air temperature at the coils. Usually, however, the motor is located outside of the casing where it is readily accessible for service. In this case, only the brake horsepower required by the fan is transformed into heat to be included in the load calculations.

In either application, motors should be selected with adequate horsepower to handle the design volume of air against the resistance of the system when the coils are wet, and then checked against the possible horsepower requirements for the increased volume of air obtained when the coils are dry.

SOUND ISOLATION

Both suspended and vertical floor mounted units can transmit vibration through the supports. Wherever such transmission of sound might be objectionable, the supports should be isolated through rubber-in-shear or other sound deadeners. (For design of suitable sound deadeners see section Controlling Vibration from Machine Mountings in Chapter 40).

MODIFICATIONS OF REMOTE UNITS

Features of various modifications of remote air conditioners are given in the following paragraphs.

Spray Type Unit

Fig. 4 shows a spray type unit used by designers who prefer air washing and coil wetting features. These units are equipped with a pump that sprays water or brine over the coils. Due to the direct mixing of the condensate and the spray, provision must be made for overflow in summer and replacement of water evaporated in winter.

Dehumidifying Units

In a further modification of spray type units, absorbent brine solutions such as lithium chloride are used to remove moisture from the air. As explained in Chapter 37, the latent heat of the moisture removed is changed to sensible heat, so that coils must be used as after-coolers to obtain the right dry-bulb temperatures. Factory produced units are also available for use with solid adsorbents such as silica gel.

Remote Room Units

For individual rooms, with cooling load requirements of ½ to 1½ tons, remote units are available in attractive casings for installation within the room. A suspended type is shown in Fig. 3 and a floor type, such as is usually installed in place of an existing radiator, is shown in Fig. 5. Furnished with chilled water from a central plant, these units offer a satisfactory method of conditioning existing offices, hotel, and apartment rooms. These units may be obtained with filters and outside air connections, but for most satisfactory application are used as supplements to central systems that supply properly conditioned and filtered air to the areas served.

Induction type units, using primary conditioned air under pressure to induce local circulation, are described in Chapter 29.

Self-Contained Units

A typical large self-contained unit is shown in Fig. 6. It is essentially a remote vertical type conditioner mounted on top of a sound insulated enclosure containing the condensing unit. Air distribution is obtained by means of grilles mounted in the discharge plenum, when the unit is located in the conditioned area. Duct distribution of conditioned air can be obtained by removing the plenum, connecting directly to the blower discharge, and safing the top of the unit.

The heat generated by the compression of refrigerant gases and that given off by the electric motor are removed from the compressor compartment in four ways: by the use of a water coil in the compressor compartment; by utilizing the cold suction gases; by drawing part of the return air through the compressor compartment, and finally by circulating room air through the compressor compartment by means of a fan attached to the motor shaft.

The 7½, 10, and 15 ton self-contained units usually have horizontal type conditioners. The condensing unit enclosures are not completely sound insulated, since they are not usually installed in the conditioned area. Most units can be divided into two or three sections for ease of handling and installation. Although most large self-contained units have water-cooled condensers, the 7½, 10, and 15 ton units can be obtained for operation with evaporative condensers.

Self-Contained Room Cooling Units

Small self-contained units can be obtained with water-cooled condensers, but they are generally air cooled.

The air-cooled types are small in capacity, ranging from ⅓ to 1½ hp. Their principal application is for conditioning such spaces as hotel rooms, offices and residential living quarters. A duct connection between the unit and an outside window or ventilated air shaft is required to permit

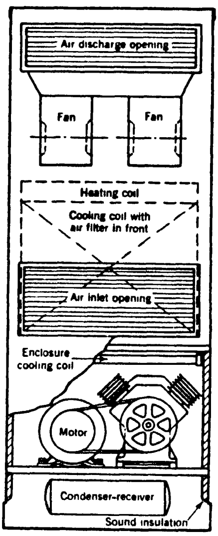

Fig. 6. Self-Contained Water-Cooled Air Conditioner

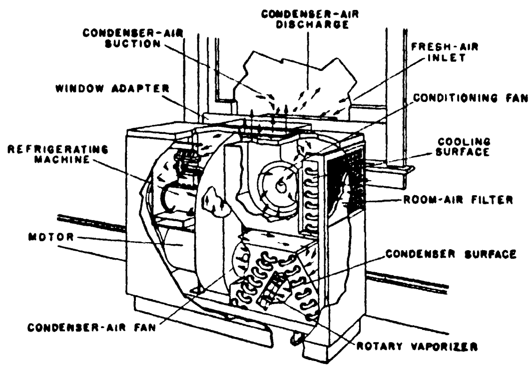

Fig. 7. Self-Contained Air-Cooled Unit Air Conditioner

disposal of the heat extracted from the conditioned area. The unit may stand in front of the window or be mounted on the window sill. Various styles and types of windows encountered tend to increase the difficulty of making the window connections. The evaporation of condensate on the condenser coils, as a means of disposing of moisture, will tend to increase the condensing capacity and reduce the operating head pressure. Some units add supplementary water so that increased capacity may be obtained from constantly wetted condenser coil surface. Connections to an electrical outlet may be made through a conventional cord and plug or a permanent electrical connection, depending on local code rulings pertaining to the installation of small motors. The exterior finish of the unit in metal, wood or fabric is decorated to harmonize with office or bedroom furnishings.

A unit of the air-cooled condenser type for floor mounting is shown in Fig. 7. Of the two fans shown, the lower one acts as condenser air fan, and in some units this fan is arranged with slingers for discharging condensate on the condenser coil, while the upper fan discharges air into the conditioned area. A feature of the design shown in Fig. 7 is that the condensate from the cooling coil is sprayed over the condenser surface and vaporized, thus eliminating the need for drain connections. A simple dampering arrangement is generally provided for exhausting some air from the room, in addition to introducing outside air and recirculating required amounts of air. It is possible to remove the equipment for winter storage or utilize the ventilating features for winter operation.

Controls for Room Cooling Units

Control devices provided for self-contained cooling units generally will include all necessary means for automatic operation. Provision is also made for adding auxiliary external controls, when desired. Remote units are not generally equipped with controls. Control systems for remote units, and auxiliary controls for self-contained units, are covered in the general treatment of Controls in Chapter 38.

RATINGS OF UNIT AIR CONDITIONERS

Two standards have been used for rating and testing of unit air conditioners: (1) Standard Method of Rating and Testing Air Conditioning Equipment,[1] covering all types of air conditioning units except the self-contained type and (2) Standard Method of Rating and Testing Self-Contained Air Conditioning Units for Comfort Cooling[2] covering the self-contained type. (*ASRE* Standard Methods of Rating and Testing Air Conditioners, *ASRE* Standard 16-R, covers rating and testing of all types which operate non-frosting when cooling and dehumidifying at standard rating conditions.)

The standard rating of a self-contained unit for the conditions specified in Table 1, includes all items which apply to the function of a unit as: (1) name of unit, (2) functions which unit performs, (3) data on cooling, (4) data on heating, (5) data on air flow, and (6) data on humidification.

The standard rating conditions for unit air conditioners, other than the self-contained type, are identical with those in Table 1, except that entering wet-bulb temperature for cooling is expressed as 50 percent relative humidity (66.7 F wet-bulb) instead of 67 F wet-bulb temperature. In addition, the saturated suction refrigerant temperature for comfort cooling is specified at 40 F. This condition is omitted from Table 1 for self-contained units because it is immaterial in the rating of a unit that includes the evaporator and condensing unit.

APPLICATION OF UNITARY EQUIPMENT

One of the chief advantages resulting from use of factory produced units is the saving in installation and field assembly labor, a factor that should always be kept in mind when selecting a location for these units. Because of their compactness, the tendency exists to put them in closets, storage rooms, and other inaccessible places, where installation is so difficult that much of this cost advantage is lost.

Access panels are provided on units for the proper servicing and maintenance of the equipment. Adequate outside clearance at these panels is essential. Wherever there is danger of freezing of coils or clogging with dirt, sufficient clearance should be available for replacing them without removing parts of the building.

The outstanding source of difficulties in unitary systems is usually dirty filters. The characteristics of the light weight fans used are such that air volume drops off rapidly with increase in static resistance. Since changing of filters is an unpleasant duty likely to be neglected unless it can be done easily, the operator should be at the same level as the filters, rather than under them, when they are being removed.

Fan speeds should be selected accurately for the system resistance. Variable pitch motor pulleys are often provided in a unit for minor field

TABLE 1. STANDARD RATING BASIS FOR SELF-CONTAINED AIR CONDITIONING UNITS

FUNCTIONS	TYPES OF UNITS	RATING CONDITION		
		Item	Description	Value
All	All	a	Barometric Pressure	29.92 in. Hg.
Cooling	Water-Cooled, Air-Cooled and Evaporatively-Cooled Condensers	b	Unit Ambient and Air Entering Room—Air Inlet (1) Dry-Bulb (2) Wet-Bulb	 80 F 67 F
		c	Ventilation Air	See Note
	Water-Cooled Condensers	d	Water Temperature Entering Unit	75 F
		e	Water Temperature Leaving Unit	95 F
	Air-Cooled and Evapor-Cooled Condensers	f	Air Entering Outside Air Inlet (1) Dry-Bulb (2) Wet-Bulb	 95 F 75 F
Heating	All Types Provided with Heating Function	g	Unit Ambient and Total Air Entering Unit	70 F
		h	Heating Medium, Pressure or Temperature (1) Dry Saturated Steam (2) Water In (3) Water Out	 16.7 lb per sq in. abs 180 F 160 F
Humidifying	All Types Provided with Humidifying Function	i	Unit Ambient	70 F
		j	Total Air Entering Unit (1) Dry-Bulb (2) Wet-Bulb	 70 F 53 F
Air Circulation	All	k	Filters	New and Clean

Note: Rating shall be based on both ventilation and recirculated room air entering at 80 F dry-bulb and 67 F wet-bulb temperature. (The *Note* as given in the code has been condensed in order to remove material not pertinent to this chapter).

adjustment of air volume. Any such field adjustment should be made when filters are dirty, to simulate average operating conditions.

Because varying sizes of coils are used within the same casing, it is very important that coil safing be carefully installed to prevent by-passing of unconditioned air. If coils are not equipped with individual casings, additional safing may be required on top to prevent short circuiting air down through the upper edges of the fins. In this same category is the need for careful installation of the various sections of sectionalized units, using a sealing compound if necessary to prevent air leakage into the fan section of the unit.

Since the drain connection is usually made on the exit side of the coil, it is important that the drain line be properly sealed. This seal should be at least twice as deep as the suction on the fan in inches of water, to prevent gurgling sounds, and to insure a positive seal against infiltration of odors and moisture laden air. Drain pans should not be used to support the coils, unless they are designed to hold this weight without sagging. As the movement of air draws the condensate or excess humidification water toward the fan, drain connections are usually placed on the exit side of the coil. The advantages of quick drainage are lost if improperly supported coils distort drain pans and cause water to accumulate in the center or back of the pan.

When the fans of vertical units are stopped, condensate that has been held up in the coils by fan suction, drops into the drain pan and splashes against the casing. If water damage is to be avoided, flashings should be provided to prevent this water from running out of the unit at the seams.

When locating unitary equipment, floor and beam loadings should be carefully checked. Suspended horizontal units can add 50 to 100 lb per square foot to the loading on the floor above. Should this floor be already heavily loaded, or be a roof structure designed for a 40 lb per square foot snow load, excess beam deflection may occur and cause cracking of plaster or concrete fire-proofing. A small fire, normally of little consequence, may cause a rupture of a heavily loaded structure and permit the equipment to drop with extensive property damage. Self-contained units should be carefully installed since their weights run as high as 200 lb per square foot. When they are installed in street floor shops, the extra precaution of placing a column beneath them in the basement is an inexpensive method of reducing vibration, as well as providing insurance against overloaded floor beams.

The services required for operation of unitary equipment should conform to the many restrictive, but necessary, local municipal codes. Existing buildings seldom are wired adequately for the electrical load imposed by the starting of an air conditioning compressor on any branch circuit. Even the smallest room cooler can draw enough current to reduce the voltage of a lighting circuit to the point where it is visibly apparent. This voltage drop may even affect the life of the unit due to the relatively slow starting. The cost of a separate electrical circuit of adequate capacity from the main panel is more than justified; it is a necessary expense in the majority of installations.

A water supply of adequate capacity and pressure is necessary to prevent overloading of electrical equipment by high head pressures. The average city water supply pressure is adequate for installations up to the third floor. Since most water cooled units require about 20 lb pressure, including control valve losses, it is important that any units served by gravity from roof tanks be checked carefully if located less than 40 ft below the tank.

Drain connections from condensers should flow to an open and properly trapped sink as required by most city codes. This prevents back pressures on the city water system in the event of condenser failure. A check valve should also be installed in the water supply as a further precaution against contamination.

When installing small remote or self-contained units with outside air connections in buildings more than 6 stories high, the effect of wintertime stack action in elevator and stairwells requires special attention. This stack action is the cause of negative pressures on the lower floors, tending to draw cold air through the units, and positive pressures on the upper floors preventing adequate ventilation and disrupting air distribution. It can also cause annoying whistling at door openings and lead to serious complaints in hotels and offices. Wherever the removal of such units is impracticable, it is important that carefully fitted, felt-edged dampers be installed in the outside air intakes with adequate locking devices.

One further consideration when installing self-contained units in conditioned areas is that any maintenance or repairs to be required in future years must be carried on in occupied space.

UNIT AIR COOLERS

Unit air coolers are intended principally for product cooling, but are often used for cooling spaces to low temperatures. They differ from normal air conditioning units only in features required to produce lower temperatures. In using such units, relative humidity and dry-bulb temperatures should be considered as carefully as in comfort air conditioning systems. Where the product being cooled is sealed in containers impermeable to water vapor, relative humidity becomes a secondary consideration.

In recent years the unit cooler has almost entirely supplanted the gravity type prime-surface and finned-tube coil. It provides more positive control of air and better air distribution, is more efficient in performance, requires less space, and is less expensive. In the past, applications of unit coolers were often limited to cold storage warehouses and to retail and wholesale markets for meat, fruit, and vegetables. In such applications, control of relative humidity was considered unimportant because of the low storage temperature or the temporary nature of the storage. In most of these cases, experience had indicated that approximately correct relative humidities would be obtained if the temperature differential between the room and refrigerant were maintained within a desirable range.

Unit coolers, however, can perform satisfactorily in installations requiring accurate control of relative humidity, air motion, and dry-bulb temperature, and thereby prevent excessive weight loss, mold and slime growth, and moisture absorption by hygroscopic materials such as dried fruits.

Unit coolers, especially in the smaller sizes, are very similar in appearance to unit heaters. Copper or steel prime or finned surface tubes are arranged in single or multiple circuits, depending on the loading. Propeller or centrifugal fans either blow or draw room air over the tubes. The fan and coil are generally enclosed in a casing provided with a drip pan. The motor horsepower requirements are a function of the air quantity and coil depth, that is, the number of rows of tubes. Finned coils are generally four to six rows in depth while prime-surface coils range from six to ten rows in depth. Fin spacing is based principally on operating temperature and the ratio between latent and total load. For operation below 32 F, fin spacing may vary from 2 to 6 fins per inch, while above 32 F it may vary from 5 to 8 fins per inch. Both direct expansion refrigerants and brine are used successfully as cooling mediums.

Unit coolers may be arranged for either free or duct delivery. Face velocities vary from 200 to 800 fpm, depending principally upon the intended application of the unit. In the larger sizes particularly, speed adjustment of the fan is generally provided to permit variation of the air delivery, and thus obtain a closer control of the relative humidity. While unit coolers are usually installed in the storage space, remote installation combined with appropriate duct work may be required by space or other considerations. Units are available for floor, wall, or ceiling mounting, thus providing an upward, downward, or horizontal discharge. Power, refrigerant, and drip pan connections are required, plus additional connections for defrosting, if necessary.

Various methods of defrosting are used. In one method, hot gas is supplied to the interior of the tubes uniformly throughout the coil. The entire refrigerant circuit is thus contacted to obtain complete defrosting of all frosted surfaces. Electric defrosting generally involves the incorporation of heating elements within the construction of the coil, or the use of strip heaters in a dampered closed air circuit. Warm water may be sprayed over the coil surface for defrosting. With storage temperatures above 32 F, defrosting may be accomplished by shutting down the refrigeration system and circulating the room air over the coil. In every case, defrosting requires a cessation of refrigeration and fan shut down. Where continuous operation of the system is desired, a brine spray over the coil may be used unless it might damage the product in storage.

Ratings

As various means of expressing unit cooler capacity are utilized in the industry, different manufacturers suggest different methods of selection. The engineer should be aware of the conditions and factors which affect rating, selection, and performance of a cooler. These items are discussed in following paragraphs.

The refrigerating capacity of the unit may be either gross or net, the latter being less than the gross by an amount equal to the heat equivalent of the input to the unit cooler motor. In either case, the capacity should be given for a particular air volume.

Dry or flooded rating conditions should be stated, as well as temperature level. The temperature level determines whether the coil surface is wetted or frosted, and it will also establish the refrigerant side pressure drop for any given load applied to a specific unit cooler. The refrigerant side pressure drop increases as the evaporating temperature decreases, and thus temperature level exercises a significant effect on the average coil surface temperature and the consequent condensing unit selection.

Coil capacity rating is usually expressed as *simplified rating* or *sensible heat ratio rating*: the latter derives its name from the ratio of the sensible heat load to the total load. The *simplified rating* expresses the capacity in terms of Btu per (hour) (Fahrenheit degree temperature differential *between the refrigerant and the air*). The *sensible heat ratio rating* expresses the unit cooler capacity in terms of Btu per (hour) (Fahrenheit degree temperature differential *between the refrigerant and the coil surface temperature*). The *simplified rating* method does not require knowledge of the sensible heat ratio. When the total load has been obtained from the load calculations, it is necessary only to assume a temperature differential between the air and the refrigerant in order to select a unit cooler. Since the extent of dehumidification will be a function of this assumed temperature differential, it is apparent that the relative humidity in the storage space is dependent upon the correct assumption of this temperature differential. For many applications, the correct temperature differential has been established between certain maximum and minimum values. In such applications, the simplified rating offers a quick method of selection. Where past experience is lacking, and in any case where close control of relative humidity is desired, the simplified rating may be used for tentative selection of the unit, but the sensible heat ratio method should be used in the final selection.

In the sensible heat ratio selection method, the total heat is used in conjunction with the air distribution requirements for making a tentative selection of a unit cooler. Then, for the specific air volume and surface area of the unit cooler selected, it is necessary to determine the refrigerant temperature required to maintain the dry-bulb temperature and relative humidity desired in the storage space. In order to apply air conditioning psychrometric techniques for accurate control of the room conditions a knowledge of the sensible and latent loads is required. The relationship of these loads is evident in the term, sensible heat ratio. From this ratio and the unit cooler air volume, the supply air conditions to maintain the storage room design conditions may be calculated.

The extent to which these supply air conditions differ from the average coil surface temperature is a function of the fin spacing, fin style, coil depth, and other factors inherent in the design of the coil. From a knowledge of the efficiency of his specific coils, a manufacturer can use the sensible heat ratio to determine the average coil surface temperature necessary to maintain desired storage room conditions, without actually evaluating the supply air conditions.

For a unit cooler adjusted to deliver a specific air volume, there exists a specific differential between the storage room conditions and the average coil surface temperature, and, therefore, it is possible to rate coolers on the basis of this room-to-surface temperature differential.

Refrigerant side pressure drop and the characteristics of the heat transfer surface, introduce a differential between the average coil surface temperature and the refrigerant temperature. Thus, the manufacturer must also present data from which the engineer can determine the overall room-to-refrigerant temperature differential necessary with the unit cooler selected. Some manufacturers rate cooling equipment on the basis of the overall room-to-refrigerant temperature differential instead of using the intermediate average coil surface temperature. In either case the significant consideration is that this overall temperature differential must be determined from a knowledge of the sensible heat ratio and the air volume of a unit cooler under consideration. If the tentatively selected unit cooler does not posses the proper capacity, it may often be possible to adjust the fan speed to a new air volume. At the new air volume and new overall differential, the cooler may be able to deliver the necessary cooling capacity. If not, the procedure must be repeated with another size of unit cooler, the final selection being based on the economical balance between unit cooler, the compressor, and the condenser.

Procedures for rating and testing room coolers are given in an ASRE Standard[8] which establishes four groups of conditions (numbered I to IV) under which units may be rated. Many manufacturers establish and publish their ratings in accordance with this standard.

In this standard, forced circulation air coolers are classified according to air side surface conditions as (1) dry coil, (2) sprayed coil, and (3) spray—no coil; or according to type of air delivery to room as (1) free delivery fan, or (2) pressure fan. Natural convection air coolers are classified according to type as (1) external baffle, (2) built-in baffle, and (3) without baffle.

Arrangement and Operation

The refrigerant is usually supplied to the evaporator through a thermostatic expansion valve, thus obtaining dry expansion in the evaporator.

Many prime surface coils utilize flooded expansion as obtained with a float valve to improve the heat transfer coefficient. Where dry expansion is used, many coil manufacturers recommend the use of a liquid-vapor heat exchanger to increase the coil efficiency by obtaining the control of superheat in the heat exchanger rather than in the evaporator coil. Liquid subcooling thus obtained contributes to an increase in the overall efficiency of the refrigeration system.

Where two or more evaporator coils are to be attached to a single condensing unit, and different evaporator temperatures are desired, a back pressure valve may be installed to limit the minimum evaporating temperature of the warmer coils. This valve also finds application where fluctuation in the evaporator temperature prevents accurate control of air temperature and humidity.

The importance of securing uniform air distribution to every part of the product zone, and the use of a permissible velocity of air over the product, must be recognized when selecting the unit coolers and their outlets. For each product there are certain maximum and minimum permissible velocities. If the average velocity falls below the maximum allowable velocity, it is of no consequence so long as the air distribution throughout the storage space is uniform and every portion of the room is reached by cooled air. It is important to note that the quantity of air in motion in the refrigerated space is not only that passing through the coils. A quantity of air many times in excess of the air handled by the units is always set in motion by the induction effect of the moving cooled air. Therefore, in evaluating the velocity for any given area, the total air set in motion must be considered. It is a function of the type of outlet, the discharge air velocity, and the location of the unit relative to restrictive walls and product.

Unit location is also important from the standpoint of occupant comfort, low velocity outlets being preferred for floor type units. High velocity outlets are acceptable in fur storage vaults, ice cream hardening rooms and other spaces where air motion is not an important factor.

In general, unit air coolers should not be suspended in front of door openings, or close to them where moist warm air will be drawn directly into the unit each time the door is opened, thereby causing excessive frosting and loss of capacity. Better performance will be obtained by placing the unit air cooler in a location such that the air will be discharged toward the door. If the shape of the space is such that this location would result in excessive air velocity over the product, then the location of the unit air cooler should be changed so that the air is discharged parallel to the wall in which the door is located.

REFERENCES

[1] Prepared by a Joint Committee of the *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electrical Manufacturers' Association*, and *Air Conditioning Manufacturers' Association* (A.S.R.E. *Circular* No. 13–42).*

[2] Standard Method of Rating and Testing Self-Contained Air Conditioning Units for Comfort Cooling prepared by a Joint Committee of the *American Society of Refrigerating Engineers*, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, *Refrigerating Machinery Association*, *National Electrical Manufacturers' Association*, and *Air Conditioning Manufacturers' Association* (A.S.R.E. *Circular* No. 16).*

[3] Proposed A.S.R.E. Standard Methods of Rating and Testing Forced-Circulation and Natural Convection Air Coolers for Refrigeration (A.S.R.E. *Circular* No. 25–44).

* *ASRE* has combined *Circulars* No. 13–42 and No. 16 in *ASRE Standard* No. 16–R, Methods of Rating and Testing Air Conditioners.

CHAPTER 26

PIPE, FITTINGS, WELDING

Pipe Materials, Types of Pipe, Commercial Pipe Dimensions, Expansion and Flexibility of Pipe, Hangers and Supports, Threading Practice, Types of Fittings, Flange Facings and Gaskets, Welding in Erection of Piping, Valves

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IMPORTANT considerations in the selection and installation of pipe and fittings for heating, ventilating, and air conditioning are dealt with in this chapter.

PIPE MATERIALS

Use of corrosion-resistant materials for pipe, including special alloy steels and irons, wrought-iron, copper, and brass, has increased considerably during the past few years. The recent development of copper, brass, and bronze fittings which can be assembled by soldering or sweating, permits the use of thin-wall pipe and thereby has reduced the initial cost of such installations. The following brief discussion indicates the variety of pipe materials and the types of pipe available.

Wrought-Steel Pipe. Because of its low price, the great bulk of wrought pipe used for heating and ventilating work at the present time is of wrought steel. The material used for steel pipe is a mild steel made by the acid-bessemer, the open-hearth, or the electric-furnace process. Ordinary wrought-steel pipe is made either by shaping sheets of metal into cylindrical form and welding the edges together, or by forming or drawing from a solid billet. The former is known as *welded pipe*, the latter as *seamless pipe*.

Many types of welded pipe are available, although the smaller sizes most frequently used in heating and ventilating work are made by the lap-weld, resistance-weld, or butt-weld process. While the lap-weld and resistance-weld processes produce a better weld than the butt type, lap-weld and resistance-weld pipe are seldom manufactured in nominal pipe sizes less than 2 in. Seamless pipe can be obtained in the small sizes at a somewhat higher cost.

Seamless steel pipe is frequently used for high pressure work or where pipe is desired for close coiling, cold bending, or other forming operations. Its advantages are its somewhat greater strength which permits use of a thinner wall and, in the small sizes, its freedom from the occasional tendency of welded pipe to split at the weld when bent.

Wrought-Iron Pipe. Wrought-iron pipe is claimed to be more corrosion-resisting than ordinary steel pipe, and therefore its somewhat higher first cost is said to be justified on the basis of longer life expectancy. Wrought-iron pipe may be identified by the spiral line marked into each length, either knurled into the metal or painted on it in red or other bright color. Otherwise, there is little difference in the appearance of wrought iron and steel pipe, although microscopic examination of polished and etched specimens will readily disclose the difference.

Cast-Ferrous Pipe. There are now available several types of cast-ferrous metal pipe made of a good grade of cast-iron with or without additions of nickel, chromium, or other alloy. This pipe is available in sizes from 1½ in. to 6 in., and in standard lengths of 5 or 6 ft, with external and internal diameters closely approximating those of extra strong wrought pipe. Cast-ferrous pipe may be obtained coupled, beveled for welding, or with ends plain or grooved for the several types of couplings. It is easily cut and threaded as well as welded. The fact that it is readily welded enables the manufacturers to supply the pipe in any lengths practicable for handling.

Alloy Metal Pipe. Steel pipe bearing a small alloy of copper or other alloying element, and iron pipe bearing a small amount of copper and molyb-

TABLE 1. DIMENSIONS OF SCHEDULES 30 AND 40 AND STANDARD WEIGHT PIPE[a]

Size	Diameter In.		Thickness[b], In.	Weight per Ft, Lb		Threads per In.	Circumference, In.		Transverse Area, Sq In.			Length of Pipe, Ft per Sq Ft		Length of Pipe, Ft Containing 1 Cu Ft	Weight of Water, Lb per Ft
	External	Internal		Plain Ends	Threads and Couplings		External	Internal	External	Internal	Metal	External Surface	Internal Surface		
⅛	0.405	0.269	**0.068**	0.244	0.245	27	1 272	0.845	0.129	0.057	0 072	9.431	14.199	2533.775	0.025
¼	0.540	0 364	**0.088**	0.424	0.425	18	1.696	1 144	0.229	0.104	0 125	7.073	10.493	1383 789	0 045
⅜	0.675	0 493	**0.091**	0.567	0.568	18	2.121	1.549	0.358	0 191	0 167	5.658	7.748	754 360	0 083
½	0.840	0.622	**0.109**	0.850	0.852	14	2.639	1.954	0.554	0.304	0.250	4.547	6.141	473.906	0.132
¾	1 050	0.824	**0.113**	1.130	1.134	14	3.299	2.589	0.866	0.533	0.333	3.637	4.635	270.034	0.231
1	1 315	1.049	**0.133**	1.678	1 684	11½	4.131	3.296	1.358	0.864	0 494	2.904	3 641	166 618	0 375
1¼	1.660	1.380	**0.140**	2.272	2.281	11½	5.215	4 335	2.164	1.495	0.669	2.301	2.768	96.275	0 65
1½	1.900	1.610	**0.145**	2.717	2.731	11½	5 969	5.058	2.835	2.036	0.799	2.010	2.372	70.733	0.88
2	2.375	2.067	**0.154**	3.652	3.678	11½	7.461	6.494	4.430	3.355	1.075	1.608	1.847	42.913	1 45
2½	2.875	2.469	**0.203**	5 793	5.819	8	9.032	7.757	6 492	4.788	1 704	1 328	1 547	30 077	2 07
3	3.500	3.068	**0.216**	7.575	7 616	8	10.996	9 638	9 621	7.393	2.228	1 091	1.245	19 479	3 20
3½	4.000	3.548	**0.226**	9.109	9.202	8	12.566	11 146	12.566	9.886	2.680	0 954	1.076	14.565	4.29
4	4.500	4.026	**0.237**	10.790	10 889	8	14 137	12 648	15.904	12 730	3.174	0 848	0.948	11.312	5 50
5	5.563	5.047	**0.258**	14.617	14 810	8	17.477	15 856	24 306	20 006	4 300	0 686	0.756	7 198	8.67
6	6.625	6.065	**0.280**	18.974	19.185	8	20 813	19 054	34.472	28 891	5 581	0 576	0.629	4.984	12.51
8[c]	8.625	8.071	0 277	24.696	25.000	8	27 096	25 356	58.426	51 161	7 265	0.443	0.473	2 815	22 18
8	8.625	7.981	**0.322**	28.554	28.809	8	27.096	25 073	58.426	50.027	8.399	0 443	0.478	2.878	21.70
10[c]	10.750	10.136	0.307	34.240	35 000	8	33.772	31 843	90.763	80.691	10 072	0.355	0.376	1.785	34 95
10	10.750	10 020	**0.365**	40 483	41.132	8	33.772	31.479	90.763	78.855	11 908	0.355	0.381	1.826	34.20
12[c]	12.750	12.090	0.330	43 773	45.000	8	40 055	37.982	127.676	114.800	12 876	0.299	0.315	1.254	49.70
12	12 750	12.000	0.375	49.562	50.706	8	40.055	37.699	127.676	113.097	14.579	0.299	0.318	1 273	49.00

[a] Standard-weight wrought-iron pipe has approximately the same wall thicknesses and weights as contained herein for steel pipe. For exact dimensions, see *American Standard for Wrought-Iron and Wrought-Steel Pipe, ASA* B36.10.

[b] Thicknesses shown in bold face type are identical with thicknesses for Schedule 40 pipe of *ASA* B36.10.

[c] Same as Schedule 30, *ASA* B36.10.

denum, have been claimed to possess more resistance to corrosion than plain steel pipe and they are advertised and sold under various trade names.

Copper Pipe and Fittings. Owing to inherent resistance to corrosion, copper and brass pipe have always been used in heating, ventilating, and water supply installations, but the cost with standard dimensions for threaded connections has been high. The recent introduction of fittings which permit erection by soldering or sweating, allows the use of pipe with thinner walls than would be possible with threaded connections, thereby reducing the cost of installations.

The initial cost of brass and copper pipe installations generally runs higher than the corresponding job with steel pipe and screwed connections in spite of the use of thin wall pipe, but the corrosive nature of the fluid conveyed or the inaccessibility of some of the piping may warrant use of a more expensive material than plain steel. The advantages of corrosion-resisting pipe and fittings should be weighed against the correspondingly higher initial cost.

COMMERCIAL PIPE DIMENSIONS

The two weights of steel and wrought-iron pipe commonly used are known as *standard weight* and *extra strong*, which correspond to Schedules 40 and 80, respectively, of the *American Standard for Wrought-Iron and Wrought-Steel Pipe, ASA* B36.10. The same external diameter is used for both

TABLE 2. STANDARD WEIGHTS AND DIMENSIONS OF WELDED AND SEAMLESS STEEL PIPE[a]

Size	Outside Diameter, In.	No. of Threads per In.	Standard-Weight Pipe				Extra-Strong Pipe				Double Extra-Strong Pipe[b]	
			Schedule 30		Schedule 40		Schedule 60		Schedule 80			
			Wall Thickness, In.	Weight per Ft, Lb T & C	Wall Thickness, In.	Weight per Ft, Lb T & C	Wall Thickness, In.	Weight per Ft, Lb Plain Ends	Wall Thickness, In.	Weight per Ft, Lb Plain Ends	Wall Thickness In.	Weight per Ft, Lb Plain Ends
1/8	0 405	27	...	...	0.068	0.25	...	...	0.095	0.31	...	...
1/4	0 540	18	...	...	0 088	0.43	...	...	0.119	0 54	...	...
3/8	0.675	18		...	0 091	0.57	...	...	0.126	0.74	...	...
1/2	0.840	14	.	...	0.109	0.85	...	...	0.147	1.09	0.294	1.71
3/4	1.050	14			0 113	1.13	...	...	0 154	1 47	0.308	2.44
1	1 315	11½	.	.	0 133	1.68	...	...	0.179	2.17	0.358	3.66
1¼	1.660	11½	.		0 140	2.28	...	...	0.191	3.00	0 382	5.21
1½	1 900	11½			0 145	2.73	...	...	0 200	3 63	0.400	6.41
2	2.375	11½	...	...	0 154	3 68	...	...	0.218	5 02	0 436	9 03
2½	2.875	8	...		0.203	5.82	...	...	0.276	7.66	0.552	13.70
3	3.500	8	...	...	0.216	7.62	...	...	0.300	10.25	0.600	18.58
3½	4.000	8	...		0.226	9.20	...	...	0 318	12.51	0 636	22.85
4	4.500	8	...	...	0.237	10.89	.	...	0.337	14 98	0.674	27.54
5	5.563	8	...	...	0.258	14.81	.	...	0.375	20.78	0.750	38.55
6	6.625	8			0.280	19.19	...	...	0.432	28.57	0.864	53 16
8	8 625	8	0 277	25.00	0.322	28.81	...	...	0.500	43 39	0.875	72.42
10[c]	10 750	8	0.307	35 00	0.365	41.13	0.500	54.74	...	...	...	...
12[d]	12.750	8	0.330	45.00	0.375	50.71	0 500[d]	65.41	...	.	...	...

From Standard Specifications for Welded and Seamless Steel Pipe of the *American Society for Testing Materials, A.S.T.M.* Designation A120.

[a] Sizes larger than those shown in the table are measured by their outside diameter, such as 14 in. outside diameter, etc. These larger sizes will be furnished with plain ends, unless otherwise specified. The weights will correspond to the manufacturers' published standards although it is possible to calculate the theoretical weights for any given size and wall thickness on the basis of 1 cu in. of steel weighing 0.2833 lb.

[b] The American Standard for Wrought-Iron and Wrought-Steel Pipe *ASA* B36.10-1939 has assigned no schedule number to *Double Extra-Strong* pipe.

[c] A 10 in. Standard Weight pipe is also available with 0.279 in. wall thickness, but this wall is not covered by a Schedule Number.

[d] Owing to a departure from the *Standard-Weight* and *Extra-Strong* wall thicknesses for the 12 in. nominal size, Schedules 40 and 60, Table 2 of the *ASA* B36.10-1939, Standard for Wrought-Iron and Wrought-Steel Pipe, the regular *Standard* and *Extra-Strong* wall thicknesses (0.375 in. and 0.500 in.) have been substituted.

weights of each nominal size for manufacturing reasons, as well as to afford interchangeability in threading and other elements associated with fabrication and erection. Hence, the difference in wall thickness is accompanied by a corresponding change in inside diameter. In sizes up to 14 in., pipe is designated by its nominal size which corresponds roughly to the inside diameter of Schedule 40 pipe. In sizes 14 in. and upward, pipe is designated by its outside diameter (O.D.), and the wall thickness is specified.

While the demands for pipe for the heating and ventilating industry are reasonably well served by Schedule 40 (standard weight) pipe, the erection of pipe by welding sometimes warrants using lighter wall thicknesses. The considerations governing pipe wall thickness and its relation to joint design are covered in the *American Standard Code for Pressure Piping, ASA* B31.1-1942, see Section 122. Standard schedules of pipe thicknesses are contained in the *American Standard for Wrought-Iron and Wrought-Steel Pipe, ASA* B36.10, which includes standard-weight and

Table 3. Standard Dimensions and Weights, and Tolerances in Diameter and Wall Thickness for Copper Water Tubes[a]

(All tolerances in this table are plus and minus except as otherwise indicated)

Standard Water Tube Size, In.	Actual Outside Diameter In.	Average Outside Diameter: Tolerance, In.		Wall Thickness In.: Type K		Type L		Type M		Theoretical Weight, Lb per Ft		
		Annealed	Drawn Temper	Nominal	Tolerance	Nominal	Tolerance	Nominal	Tolerance	Type K	Type L	Type M
1/8	0.250	0.002	0.001	0.032	0.003	0.025	0.0025	0.025	0.0025	0.085	0.068	0.068
1/4	0.375	0.002	0.001	0 032	0.004	0.030	0.0035	0.025	0.0025	0.134	0.126	0.107
3/8	0.500	0.0025	0.001	0.049	0.004	0.035	0.0035	0.025	0.0025	0.269	0.198	0.145
1/2	0.625	0.0025	0.001	0.049	0.004	0.040	0.0035	0.028	0.0025	0.344	0.285	0.204
5/8	0.750	0.0025	0.001	0.049	0.004	0.042	0.0035	0.030	0.0025	0 418	0 362	0 263
3/4	0.875	0.003	0.001	0.065	0.0045	0.045	0.004	0.032	0.003	0.641	0.455	0.328
1	1.125	0.0035	0.0015	0.065	0.0045	0 050	0.004	0.035	0.0035	0.839	0.655	0 465
1 1/4	1.375	0.004	0.0015	0.065	0.0045	0.055	0.0045	0.042	0.0035	1.04	0 884	0.682
1 1/2	1.625	0.0045	0.002	0.072	0.005	0.060	0.0045	0.049	0.004	1.36	1.14	0.940
2	2.125	0.005	0.002	0.083	0.007	0 070	0.006	0.058	0.006	2.06	1.75	1.46
2 1/2	2.625	0.005	0.002	0 095	0.007	0.080	0.006	0.065	0.006	2.93	2.48	2 03
3	3.125	0.005	0.002	0.109	0.007	0.090	0.007	0.072	0.006	4.00	3.33	2.68
3 1/2	3.625	0.005	0.002	0.120	0.008	0 100	0.007	0.083	0.007	5.12	4.29	3.58
4	4.125	0.005	0.002	0.134	0 010	0.110	0.009	0.095	0.009	6.51	5.38	4.66
5	5.125	0.005	0.002	0.160	0.010	0 125	0.010	0.109	0.009	9.67	7.61	6.66
6	6.125	0.005	0.002	0.192	0.012	0.140	0.010	0.122	0.010	13.9	10.2	8.92
8	8.125	0.006	+0.002 −0.004	0.271	0.016	0 200	0.014	0.170	0.014	25.9	19.3	16 5
10	10.125	0.008	+0.002 −0.006	0.338	0.018	0.250	0.016	0.212	0.015	40.3	30.1	25.6
12	12.125	0.008	+0.002 −0.006	0.405	0.020	0.280	0.018	0.254	0.016	57.8	40 4	36.7

[a] From Standard Specifications for Copper Water Tube of the *American Society for Testing Materials A.S.T.M.* Designation B88-41.

Note 1:—For copper gas and oil burner tubes, the tolerances shown above for various wall thicknesses (type K) apply irrespective of diameter.

Note 2:—For tubes other than round no standard tolerances are established. These tolerances do not apply to condenser and heat exchanger tubes.

extra-strong thicknesses in Schedules 40 and 80, respectively, and eight other schedules of varying wall thickness to provide for different service conditions. Dimensions and other useful data for Schedules 30 and 40 pipe are given in Table 1. Table 2 from *A.S.T.M.* Specifications A53 and A120 combines the schedule thicknesses of *ASA* B36.10 and the old series designations.

Standard-weight pipe is generally furnished with threaded ends in random lengths of 16 to 22 ft, although when ordered with plain ends, 5 percent may be in lengths of 12 to 16 ft. Five percent of the total number of lengths ordered may be *jointers* which are two pieces coupled together.

Extra-strong pipe is generally furnished with plain ends in random lengths of 12 to 22 ft, although 5 percent may be in lengths of 6 to 12 ft.

In addition to *IPS* copper pipe, several varieties of copper tubing are in use with either flared or compression couplings or soldered joints. Dimensions of copper water tubing intended for plumbing, underground water service, fuel-oil lines, gas lines, etc., have been standardized by the U. S. Government and the *American Society for Testing Materials*. There are

TABLE 4. THERMAL EXPANSION OF PIPE IN INCHES PER 100 FT[a]

(For superheated steam and other fluids refer to temperature column)

Saturated Steam			Elongation in Inches per 100 ft from −20 F up				Saturated Steam		Elongation in Inches per 100 ft from −20 F up			
Vacuum Inches of Hg.	Pressure Psig	Temperature Fahrenheit Degrees	Cast-Iron Pipe	Steel Pipe	Wrought-Iron Pipe	Copper Pipe	Pressure Psig	Temperature Fahrenheit Degrees	Cast-Iron Pipe	Steel Pipe	Wrought-Iron Pipe	Copper Pipe
		−20	0	0	0	0	2.5	220	1.634	1.852	1.936	2.720
		0	0.127	0.145	0.152	0.204	10.3	240	1.780	2.020	2.110	2.960
		20	0.255	0.293	0.306	0.442	20.7	260	1.931	2.183	2.279	3.189
		40	0.390	0.430	0.465	0.655	34.5	280	2.085	2.350	2.465	3.422
29.39	....	60	0.518	0.593	0.620	0.888	52.3	300	2.233	2.519	2.630	3.665
28.89	....	80	0.649	0.725	0.780	1.100	74.9	320	2.395	2.690	2.800	3.900
27.99	....	100	0.787	0.898	0.939	1.338	103.3	340	2.543	2.862	2.988	4.145
26.48	....	120	0.926	1.055	1.110	1.570	138.3	360	2.700	3.029	3.175	4.380
24.04	....	140	1.051	1.209	1.265	1.794	180.9	380	2.859	3.211	3.350	4.625
20.27	....	160	1.200	1.368	1.427	2.008	232.4	400	3.008	3.375	3.521	4.870
14.63	....	180	1.345	1.528	1.597	2.255	293.7	420	3.182	3.566	3.720	5.118
6.45	....	200	1.495	1.691	1.778	2.500	366.1	440	3.345	3.740	3.900	5.358

[a] From *Piping Handbook*, by Walker and Crocker. This table gives the expansion from −20 F to the temperature in question. To obtain the amount of expansion between any two temperatures take the difference between the figures in the table for those temperatures. For example, if a steel pipe is installed at a temperature of 60 F and is to operate at 300 F, the expansion would be 2.519 − 0.593 = 1.926 in.

three standard wall-thickness schedules of copper water tubing classified in accordance with their principal uses as follows:

Type *K*—Designed for underground services and general plumbing service.
Type *L*—Designed for general plumbing purposes.
Type *M*—Designed for use with soldered fittings only.

In general, Type *K* is used where corrosion conditions are severe, and Types *L* and *M* where such conditions may be considered normal as, for instance, in heating work. Types *K* and *L* are available in both hard and soft tempers; Type *M* is available only in hard temper. Where flexibility is essential as in hidden replacement work, or where as few joints as possible are desired as in fuel-oil lines, the soft temper is commonly used. In new or exposed work copper pipe of a hard temper is generally used. All three classes are extensively used with soldered fittings.

Standard dimensions, weights, and diameter and wall-thickness tolerances for these classes of copper tubing are given in Table 3. Copper pipe is also available with dimensions of steel pipe.

Refrigeration lines used in connection with air conditioning equipment also employ copper tubing extensively. For refrigeration use where tubing absolutely free from scale and dirt is required, bright annealed copper tubing that has been deoxidized is used. This tubing is available in a variety of sizes and wall thicknesses.

EXPANSION AND FLEXIBILITY

The increase in temperature of a pipe from room temperature to an operating steam or water temperature 100 deg or more above room temperature, results in an increase in length of the pipe for which provision must be made. The amount of linear expansion (or contraction in the case of refrigeration lines) per unit length of material per degree change in temperature is termed the *coefficient of linear expansion*, or commonly, the *coefficient of expansion*. This coefficient varies with the material.

The linear expansion of cast-iron, steel, wrought-iron, and copper pipe, the materials most frequently used in heating and ventilating work, can be determined from Table 4.

The three methods by which the elongation due to thermal expansion may be taken care of are: (1) expansion joints; (2) swivel joints; (3) inherent flexibility of the pipe itself utilized through pipe bends, right-angle turns, or offsets in the line.

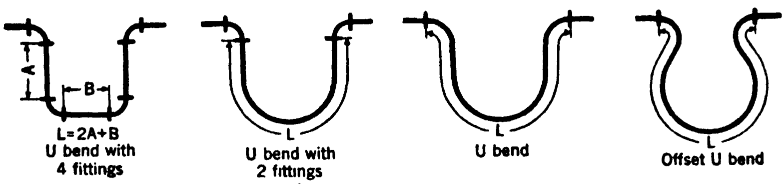


FIG. 1. MEASUREMENT OF L ON VARIOUS PIPE BENDS

Expansion joints of the slip-sleeve, diaphragm, or corrugated types made of copper, rubber, or other gasket material are all used for taking up expansion, but generally only for low pressures or where the inherent flexibility of the pipe cannot readily be used as in underground steam or hot water distribution lines.

Swivel joints are used to some extent in low-pressure steam and hot-water heating systems, and in hot-water supply lines. Since swivel joints permit the expansive movement of the pipe by turning of threaded joints, which may ultimately result in a leak, it is preferable to provide sufficient flexibility without resorting to swiveling in the threads.

Probably the most economical method of providing for expansion of piping in a long run is to take advantage of the directional changes which must necessarily occur in the piping, and proportion the offsets so that sufficient flexibility is secured. Ninety-degree bends with long, straight tangents in either a horizontal or a vertical plane are an excellent means for securing adequate flexibility with larger sizes of pipe. When flexibility cannot be obtained in this manner, it is necessary to make use of some type of expansion bend. The exact calculation of the size of expansion bends required to take up a given amount of thermal expansion is relatively complicated.[1] The following approximate method, however, has been found to give reasonably good results and is deemed to be sufficiently accurate for most heating installations.

Fig. 1 shows several types of expansion bends commonly used for taking up thermal expansion. The amount of pipe, L, required in each of these bends may be computed from Equation 1.

$$L = 6.16\sqrt{D\Delta} \quad (1)$$

where

L = length of pipe, feet.
D = outside diameter of the pipe used, inches.
Δ = the amount of expansion to be taken up, inches.

This formula, based on the use of mild-steel pipe with wall thicknesses not heavier than extra-strong, assumes a maximum safe value of fiber stress of 16,000 psi. When square type bends are used, the width of the bend should not exceed about twice the height, since for a given total length of pipe in the bend, the height of the bend becomes progressively less with increase in width until the height approaches zero and no flexibility exists. Actually, wide bends utilize to best advantage the inherent flexibility of the line, but such bends cannot be proportioned on the basis of Equation 1. For such applications, more accurate methods[1] should be employed. It is further assumed that the corners are made with screwed or flanged elbows or with arcs of circles having radii five to six times the pipe diameter. Use of welding elbows with radii of 1½ times the pipe diameter will decrease the end thrusts somewhat, but will raise the fiber stress correspondingly.

All risers must be anchored and safeguarded so that the difference in length when hot, from the length when cold, shall not disarrange the normal and orderly provisions for drainage of the branches.

Proper anchoring of piping is especially necessary with light-weight radiators, to allow for freedom of expansion in order that no pipe strain will distort the radiators. When expansion strains from the pipes are permitted to reach these light metal heaters, they usually emit disturbing sounds.

HANGERS AND SUPPORTS

Heating system piping requires careful and substantial support. Where changes in temperature of the line are not large, such simple methods of support may be utilized as hanging the line by means of rods or perforated strip from the building structure, or supporting it by brackets or on piers.

When fluids are conveyed at temperatures of 150 F or above, however, hangers or supporting equipment must be fabricated and assembled to permit free expansion or contraction of the piping. This can be accomplished by the use of long rod hangers, spring hangers, chains, hangers or supports fitted with rollers, machined blocks, elliptical or circular rings of larger diameter than the pipe giving contact only at the bottom, or trolley hangers. In all cases, allowance should be made for rod clearance to permit swinging without setting up severe bending action in the rods.

For pipes of small size, perforated metal strip is often used. For horizontal mains, the rod or strip usually is attached to the joists or steel work of the floor above. For long runs of vertical pipe subject to considerable thermal expansion, either the hangers should be designed to prevent excessive load on the bottom support due to expansion, or the bottom support should be designed to withstand the entire load.

THREADING PRACTICE

In all threaded pipe for heating and ventilating installations the American Standard taper pipe thread, *ASA* B2.1-1942 is used. This thread is cut with a taper of 1 in 16 measured on the diameter of the pipe so as to secure a tight joint. The number of threads per inch varies with the pipe size. Threads for fittings are the same, except that it is regular practice to furnish straight tapped couplings for Schedule 40 pipe 2 in. and smaller. For steam pressures in excess of 25 psi, it is recommended that taper-tapped couplings be used to obtain a tight joint. These may be secured by ordering *line* pipe[2] which is used for oil piping, the couplings of which are provided with taper-tapped threads and may be used with regular mill-threaded standard weight pipe. Thread lengths should be in accordance with *ASA* B2.1. Right-hand threads are used unless otherwise ordered. To facilitate drainage, some elbows have the thread tapped at an angle to provide a pitch of the connecting pipe of $\frac{1}{4}$ in. to the foot. These elbows are

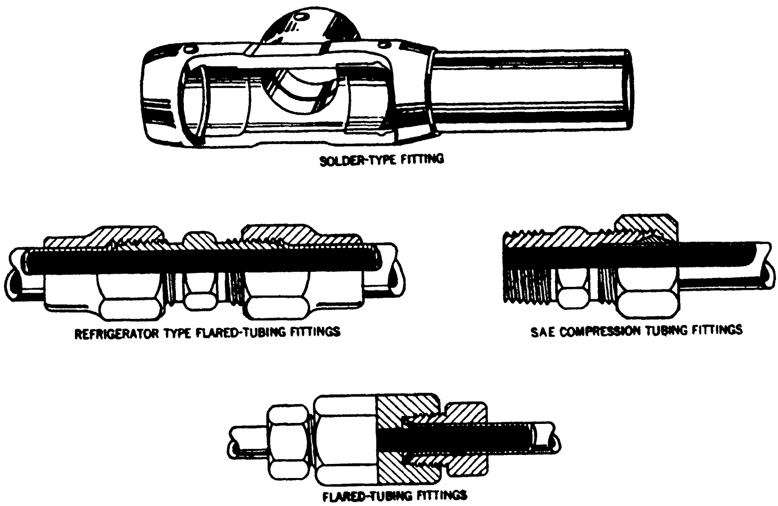


Fig. 2. Copper or Brass Tubing Fittings

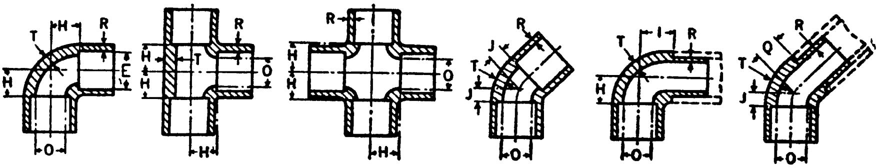
known to the trade as pitch elbows and are commercially available. All threaded pipe joints should be made up with a thread paste suitable for the service for which the pipe is to be used.

TYPES OF FITTINGS

Fittings for joining the separate lengths of pipe together are made in a variety of forms, and are either screwed or flanged, the former being generally used for the smaller sizes of pipe up to and including $3\frac{1}{2}$ in., and the latter for the larger sizes, 4 in. and above. Screwed fittings of large size, as well as flanged fittings of small size, are also made and are used for certain classes of work at the proper pressure.

The material used for fittings is generally cast-iron, but in addition to this, malleable-iron, steel and steel alloys are also used, as well as various grades of brass or bronze. The material to be used depends on the character of the service and the pressure. Malleable iron fittings, like brass fittings, are cast with a round instead of a flat band or bead, or with no bead at all. Fittings are designated as male or female, depending on whether the threads are on the outside or inside, respectively. Screwed galvanized fittings are made according to the 150 lb American Standard.

TABLE 5. AMERICAN STANDARD DIMENSIONS OF ELBOWS, TEES, CROSSES, AND 45 DEG ELBOWS, SOLDERED-JOINT FITTINGS, ASA A40.3-1941



Nominal Size[a]	Cast Brass[b]							Wrought Metal	Bore of Fittings
	Laying Length, Tee, Ell, and Cross[b]	Laying Length, Ell With External Shoulder	Laying Length, 45 Deg Ell	Laying Length, 45 Deg Ell External Shoulder	Inside Diameter of Fittings,[e] Min.	Metal Thickness[d]		Metal Thickness Min.[f]	
	H	I	J	Q	O	T	R	T and R	E, Min.
1/4	1/4	3/8	3/16	1/4	0.31	0.08	0.048	0.030	0.378
3/8	5/16	7/16	3/16	5/16	0.43	0.08	0.048	0.035	0.503
1/2	7/16	9/16	3/16	5/16	0.54	0.09	0.054	0.040	0.628
3/4	9/16	11/16	1/4	3/8	0.78	0.10	0.060	0.045	0.878
1	3/4	7/8	5/16	7/16	1.02	0.11	0.066	0.050	1.1285
1 1/4	7/8	1	7/16	9/16	1.26	0.12	0.072	0.055	1.3785
1 1/2	1	1 1/8	1/2	5/8	1.50	0.13	0.078	0.060	1.629
2	1 1/4	1 3/8	9/16	3/4	1.98	0.15	0.090	0.070	2.129
2 1/2	1 1/2	1 5/8	5/8	7/8	2.46	0.17	0.102	0.080	2.629
3	1 3/4	1 7/8	3/4	1	2.94	0.19	0.114	0.090	3.129
3 1/2	2	2 1/8	7/8	1 1/8	3.42	0.20	0.120	0.100	3.629
4	2 1/4	2 3/8	15/16	1 1/4	3.90	0.22	0.132	0.110	4.129
5	3 1/8	......	1 7/16	......	4.87	0.28	0.168	0.125	5.129
6	3 5/8	......	1 5/8	......	5.84	0.34	0.204	0.140	6.129

All dimensions given in inches.

[a] This size is the nominal bore of the tube.

[b] These dimensions may be used for wrought-metal fittings as well as for cast-brass fittings at manufacturer's option.

[c] This dimension is the same as the inside diameter Class L tubing (American Standard Specifications for Copper Water Tube, *ASA* H23.1-1939 (*A.S.T.M.* B88).

[d] Patterns shall be designed to produce body thicknesses given in the table. Metal thickness at no point shall be less than 90 percent of the thicknesses given in the table.

[e] This dimension has the same thicknesses as Type L tubing.

[f] These dimensions are minimum, but in every case the thickness of wrought fittings should be at least as heavy as the tubing with which it is to be used.

NOTE 1:—Wrought fittings, as well as cast fittings, must be provided with a shoulder or stop at the bottom end of socket.

NOTE 2:—Street fittings with male ends are for use in connection with other fittings illustrated.

As in the case of pipe, several weights of fittings are manufactured. Recognized American Standards for the various weights are as follows:

Cast-iron pipe flanges and flanged fittings for 25 lb (sizes 4 in. and larger), 125 lb, and 250 lb maximum saturated steam pressure, *ASA* B16b2, B16a, and B16b, respectively.

Malleable iron screwed fittings for 150 lb maximum saturated steam pressure, *ASA* B16c.

Cast-iron screwed fittings for 125 and 250 lb maximum saturated steam pressure *ASA* B16d.

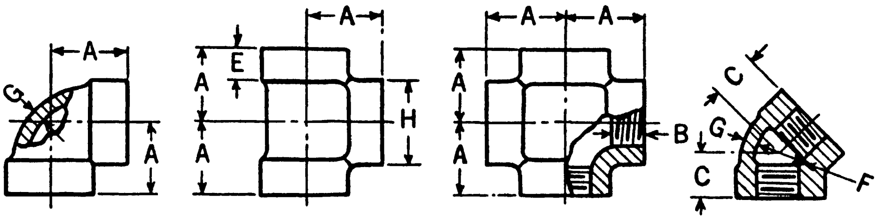
Steel flanged fittings for 150 and 300 lb maximum steam service pressure, *ASA* B16e.

The allowable cold water working pressures for these standards vary from 43 lb for the 25 lb standard, to 500 lb for the 300 lb steel standard.

War standard ratings in effect for the duration of the emergency permitted higher ratings for certain sizes of the 125 lb cast-iron flanged fitting standard, and for 300 lb steel flanges and flanged fittings, than those shown in the regular American Standards mentioned previously.

Screwed fittings include: nipples or short pieces of pipe of varying lengths; couplings of steel or wrought-iron; elbows for turning angles of either 45 deg or 90 deg; return bends, which may be of either the close or open pattern, and may be cast with either a back or side outlet; tees;

Table 6. American Standard Dimensions of Elbows, 45-Deg Elbows, Tees and Crosses (Straight Sizes) for Class 125 Cast-Iron Screwed Fittings, *ASA* B16a-1939



Elbow Tee Cross 45 deg elbow

Nominal Pipe Size	*A*	*C*	*B*	*E*	*F*		*G*	*H*
	Center to End, Elbows, Tees and Crosses	Center to End, 45 Deg Elbows	Length of Thread, Min.	Width of Band, Min.	Inside Diameter of Fitting		Metal Thickness,[a] Min.	Outside Diameter of Band, Min.
					Min.	Max.		
¼	0.81	0.73	0.32	0.38	0.540	0.584	0.110	0.93
⅜	0.95	0.80	0.36	0.44	0.675	0.719	0.120	1.12
½	1.12	0.88	0.43	0.50	0.840	0.897	0.130	1.34
¾	1.31	0.98	0.50	0.56	1.050	1.107	0.155	1.63
1	1.50	1.12	0.58	0.62	1.315	1.385	0.170	1.95
1¼	1.75	1.29	0.67	0.69	1.660	1.730	0.185	2.39
1½	1.94	1.43	0.70	0.75	1.900	1.970	0.200	2.68
2	2.25	1.68	0.75	0.84	2.375	2.445	0.220	3.28
2½	2.70	1.95	0.92	0.94	2.875	2.975	0.240	3.86
3	3.08	2.17	0.98	1.00	3.500	3.600	0.260	4.62
3½	3.42	2.39	1.03	1.06	4.000	4.100	0.280	5.20
4	3.79	2.61	1.08	1.12	4.500	4.600	0.310	5.79
5	4.50	3.05	1.18	1.18	5.563	5.663	0.380	7.05
6	5.13	3.46	1.28	1.28	6.625	6.725	0.430	8.28
8	6.56	4.28	1.47	1.47	8.625	8.725	0.550	10.63
10	8.08[b]	5.16	1.68	1.68	10.750	10.850	0.690	13.12
12	9.50[b]	5.97	1.88	1.88	12.750	12.850	0.800	15.47

All dimensions given in inches.

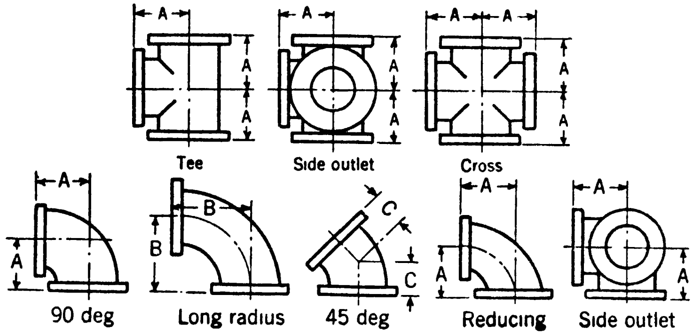
[a] Patterns shall be designed to produce castings of metal thickness given in the table. Metal thickness at no point shall be less than 90 percent of the thickness given in the table.

[b] Applies to elbows and tees only.

crosses; laterals or *Y* branches; and a variety of plugs, bushings, caps, lock-nuts, flanges and reducing fittings. Reducing fittings as well as bushings, both of which are used in changing from one pipe size to another, may have the smaller connection tapped eccentrically to permit free drainage of the water of condensation in steam lines or free escape of air in water lines.

Fittings for copper tubing are available in the soldered, flared, or compression types. Illustrations of each of these types are shown in Fig. 2.

Table 7. American Standard Dimensions of Tees, Crosses[a] (Straight Sizes), and Elbows for Class 125 Cast-Iron Flanged Fittings, *ASA* B16a-1939



Nominal Pipe Size[b-e]	*A* Center to Face Tees, Crosses[c-d] and Elbows	*AA* Face to Face Tees and Crosses[c-d]	*B* Center to Face Long Radius Elbows[e,f,g]	*C* Center to Face 45 Deg Elbows[f]	Diameter of Flange	Thickness of Flange, Min.	Metal[h] Thickness of Body
1	3½	7	5	1¾	4¼	7/16	5/16
1¼	3¾	7½	5½	2	4⅝	½	5/16
1½	4	8	6	2¼	5	9/16	5/16
2	4½	9	6½	2½	6	⅝	5/16
2½	5	10	7	3	7	11/16	5/16
3	5½	11	7¾	3	7½	¾	⅜
3½	6	12	8½	3½	8½	13/16	7/16
4	6½	13	9	4	9	15/16	½
5	7½	15	10¼	4½	10	15/16	½
6	8	16	11½	5	11	1	9/16
8	9	18	14	5½	13½	1⅛	⅝
10	11	22	16½	6½	16	1 3/16	¾
12	12	24	19	7½	19	1¼	13/16
14 O.D.	14	28	21½	7½	21	1⅜	⅞
16 O.D.	15	30	24	8	23½	1 7/16	1
18 O.D.	16½	33	26½	8½	25	1 9/16	1 1/16
20 O.D.	18	36	29	9½	27½	1 11/16	1⅛
24 O.D.	22	44	34	11	32	1⅞	1¼
30 O.D.	25	50	41½	15	38¾	2⅛	1 7/16
36 O.D.	28	56	49	18	46	2⅜	1⅝
42 O.D.	31	62	56½	21	53	2⅝	1 13/16
48 O.D.	34	68	64	24	59½	2¾	2

All dimensions given in inches.

[a] Crosses both straight and reducing sizes 18 in. and larger shall be reinforced to compensate for the inherent weakness in the casting design.

[b] Size of all fittings listed indicates nominal inside diameter of port.

[c] Tees, side outlet tees, and crosses, 16 in. and smaller, reducing on the outlet, have the same dimensions center to face, and face to face as straight size fittings corresponding to the size of the larger opening. Sizes 18 in. and larger, reducing on the outlet, are made in two lengths, depending on the size of the outlet.

[d] Tees and crosses, reducing on run only, carry same dimensions center to face and face to face as a straight size fitting of the larger opening.

[e] Reducing elbows and side outlet elbows carry same dimensions center to face as straight size elbows corresponding to the size of the larger opening.

[f] Special degree elbows, ranging from 1 to 45 deg, inclusive, shall have the same center to face dimensions as given for 45-deg elbows, and those over 45 deg and up to 90 deg, inclusive, shall have the same center to face dimensions as given for 90-deg elbows. The angle designation of an elbow is its deflection from straight line flow and is the angle between the flange faces.

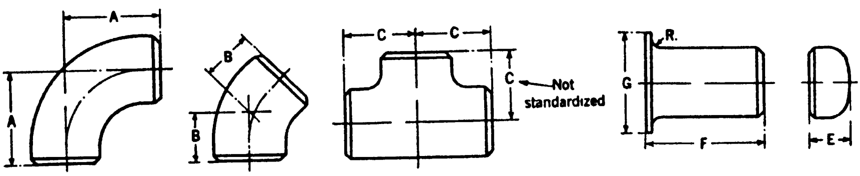
[g] Side outlet elbows shall have all openings on intersecting center lines.

[h] Body thickness at no point shall be less than 87½ percent of the dimensions given in the table.

Fittings for copper pipe of *IPS* dimensions are available in screwed or soldered types of connection. Table 5 from *ASA* Standard A40.3-1941 contains dimensions for soldered joint elbows, tees, crosses, and 45 deg elbows.

The compression type fitting is generally limited to smaller size tubing, while the flared and soldered types are used in both large and small sizes. An American Standard, *ASA* A40.2-1936 has been prepared to standardize dimensions for brass fittings for flared copper water tubes. Flared tube fittings are widely used in refrigerating work where *S.A.E.* dimensions

TABLE 8. AMERICAN STANDARD DIMENSIONS FOR BUTT-WELDING ELBOWS, TEES, CAPS, AND LAPPED-JOINT STUB ENDS, *ASA* B16.9-1940



Nominal Pipe Size	Outside Diameter	Center-to-End			Caps E[b,c]	Lapped-Joint Stub Ends		
		90-Deg Elbows A	45-Deg Elbows B	Of Run Tee C[a]		Length F[b]	Radius of Fillet R	Diam. of Lap G[d]
1	1.315	1½	⅞	1½	1½	4	⅛	2
1¼	1.660	1⅞	1	1⅞	1½	4	3/16	2½
1½	1.900	2¼	1⅛	2¼	1½	4	¼	2⅞
2	2.375	3	1⅜	2½	1½	6	5/16	3⅝
2½	2.875	3¾	1¾	3	1½	6	5/16	4⅛
3	3.500	4½	2	3⅜	2	6	⅜	5
3½	4.000	5¼	2¼	3¾	2½	6	⅜	5½
4	4.500	6	2½	4⅛	2½	6	7/16	6 3/16
5	5.563	7½	3⅛	4⅞	3	8	7/16	7 5/16
6	6.625	9	3¾	5⅝	3½	8	½	8½
8	8 625	12	5	7	4	8	½	10⅝
10	10.750	15	6¼	8½	5	10	½	12¾
12	12.750	18	7½	10	6	10	½	15

All dimensions given in inches.

[a] The dimensions of welding tees cover those which have side outlets from one size less than half the size of the run-way opening of the tees to full size.

[b] Dimensions E and F are applicable only to these fittings in schedules up to and including Schedule 80, *ASA* Standard B36.10-1939.

[c] The shape of these caps shall be ellipsoidal and shall conform to the requirements of the *A.S.M.E.* Boiler Construction Code.

[d] This dimension is for standard machined facings in accordance with American Standard for Steel Pipe Flanges and Flanged Fittings (*ASA* B16e-1939). The back face of the lap shall be machined to conform to the surface of the flange on which it seats. Where ring joint facings are to be applied, use dimension K as given in *ASA* B16e-1939.

and a 45-deg flare render most fittings interchangeable, although for refrigeration use, thread fits and tolerances on thread gages must be maintained within close limits. Brass fittings with *S.A.E.* dimensions are not interchangeable with the American Standard fittings for water tubes.

Ammonia pipe fittings made of cast-iron were formerly used extensively in handling refrigerants in large installations. Replacement of ammonia by other refrigerants operating at lower pressures has seriously curtailed the market for these fittings. For this reason formulation of an American Standard for these fittings was abandoned by the *ASA* in 1936.

FLANGE FACINGS AND GASKETS

A number of different flange facings in common use are plain face, raised face, tongue and groove, and male and female. Cast-iron fittings for 125 psi and below are normally furnished with a plain face, while the 250 lb cast-iron fittings are supplied with a $\frac{1}{16}$-in. raised face. The standard facing for steel flanged fittings for 150 and 300 psi is a $\frac{1}{16}$-in. raised face, although these fittings are obtainable with a variety of facings. The gasket surface of the raised face may be finished smooth or may be machined with concentric or spiral grooves often referred to as serrated face or phonograph finish, respectively.

The dimensions of elbows, tees, and crosses for 125 lb cast-iron screwed fittings are given in Table 6, whereas the dimensions for 125 lb cast-iron flanged fittings are given in Table 7.

For low temperature service not to exceed about 220 F, a number of paper or vegetable fiber gasket materials will prove satisfactory; for plain raised face flanges, rubber or rubber inserted gaskets are commonly employed. Asbestos composition gaskets are probably the most widely used, particularly where the temperature exceeds 250 F. Jacketed asbestos and metallic gaskets may be used for any pressure and temperature conditions, but preferably only with a narrow recessed facing.

WELDING

Erection of piping in heating and ventilating installations by means of fusion welding has been commonly accepted in the past few years as an alternate method to the screwed and flanged joint. Since the question of economy of welding as against the use of screwed and flanged fittings is dependent on the individual job, the use of welding is generally recommended on the basis of a greatly reduced cost of maintenance and repair, of less weight resulting from the use of a lighter-weight pipe, and of increased economy in pipe insulation, hangers, and supports rather than on the basis of any economy that might be effected in actual erection by welding on low to medium pressure heating jobs

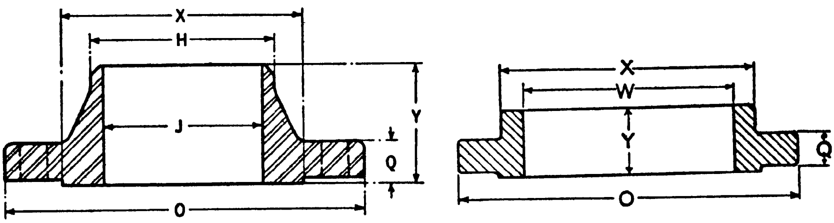
Fusion welding, commonly used in erection of piping, is defined as the process of joining metal parts in the molten, or molten and vapor states, without the application of mechanical pressure or blows. Fusion welding embraces gas welding and electric arc welding, both of which are commonly used to produce acceptable welds. Welding processes and procedure are described in various publications.

Welding application requires the same basic knowledge of design as do the other types of assembly, but, in addition, requires a generous knowledge of the sciences involved, particularly as to welding qualities of metal, their reaction to extremely high temperatures, and the ability to determine and use only the best quality welding rods. This requirement applies equally to employer and employee, with the employer accepting all of the responsibility. Thus the employer should select his welding mechanics with good judgment, provide them with first-class equipment and tools, arrange for their training and use of acceptable workmanship standards, and at regular intervals subject their work to prescribed tests.

Rules for fusion welding of pipe joints, the qualification of welding operators, welding procedures and the testing thereof, are contained in the *Standard Manual on Pipe Welding* of the *Heating, Piping & Air Conditioning Contractors National Association*, and in publications of other groups.[3,4,5] In general, the wall thickness and chemical analysis of the pipe are the governing factors, not the working pressure. There are a number of safety codes which govern the installation of welded piping in many cities and states. Some of the more prominent are listed at the end of this chapter.[4,5,6]

A complete line of manufactured steel welding fittings is now available,

Table 9. American Standard Dimensions of Steel Welding Neck and Slip-on Welding Flanges for Steam Service Pressure Rating of 150 PSI (Gage) at a Temperature of 500 F, and 100 PSI (Gage) at 750 F, *ASA* B16e-1939



WELDING NECK

SLIP-ON NECK

Nominal Pipe Size	Diameter of Flange	Thickness of Flg.[a] Min.	Diameter of Hub	Hub Diam. Beginning of Chamfer[b-c]	Length Thru Hub[a]	Inside Diam. of Pipe Schedule 40[c]	Bore of Slip-on Flanges Min.	Diam. of Bolt Circle	No. of Bolts	Size of Bolts
	O	*Q*	*X*	*H*	*Y*	*J*	*W*			
1/2	3 1/2	7/16	1 3/16	0.84	1 7/8	0.62*	0.88	2 3/8	4	1/2
3/4	3 7/8	1/2	1 1/2	1.05	2 1/16	0.82*	1.09	2 3/4	4	1/2
1	4 1/4	9/16	1 15/16	1.32	2 3/16	1.05*	1.38	3 1/8	4	1/2
1 1/4	4 5/8	5/8	2 5/16	1.66	2 1/4	1.38*	1.72	3 1/2	4	1/2
1 1/2	5	11/16	2 9/16	1.90	2 7/16	1.61*	1.97	3 7/8	4	1/2
2	6	3/4	3 1/16	2.38	2 1/2	2.07*	2.44	4 3/4	4	5/8
2 1/2	7	7/8	3 9/16	2.88	2 3/4	2.47*	2.94	5 1/2	4	5/8
3	7 1/2	15/16	4 1/4	3.50	2 3/4	3.07*	3.56	6	4	5/8
3 1/2	8 1/2	15/16	4 13/16	4.00	2 13/16	3.55*	4.06	7	8	5/8
4	9	15/16	5 5/16	4.50	3	4.03*	4.56	7 1/2	8	5/8
5	10	15/16	6 7/16	5.56	3 1/2	5.05*	5.66	8 1/2	8	3/4
6	11	1	7 9/16	6.63	3 1/2	6.07*	6.72	9 1/2	8	3/4
8	13 1/2	1 1/8	9 11/16	8.63	4	7.98*	8.72	11 3/4	8	3/4
10	16	1 3/16	12	10.75	4	10.02*	10.88	14 1/4	12	7/8
12	19	1 1/4	14 3/8	12.75	4 1/2	....	12.88	17	12	7/8
14 O.D.	21	1 3/8	15 3/4	14.00	5	To Be	14.19	18 3/4	12	1
16 O.D.	23 1/2	1 7/16	18	16.00	5	Specified	16.19	21 1/4	16	1
18 O.D.	25	1 9/16	19 7/8	18.00	5 1/2	by	18.19	22 3/4	16	1 1/8
20 O.D.	27 1/2	1 11/16	22	20.00	5 11/16	Purchaser	20.19	25	20	1 1/8
24 O.D.	32	1 7/8	26 1/8	24.00	6		24.19	29 1/2	20	1 1/4

All dimensions given in inches.

[a] A raised face of 1/16 in. is included in *thickness of flange minimum* and in *length through hub*.

[b] The outside surface of the welding end of the hub shall be straight or tapered at not more than 6 deg.

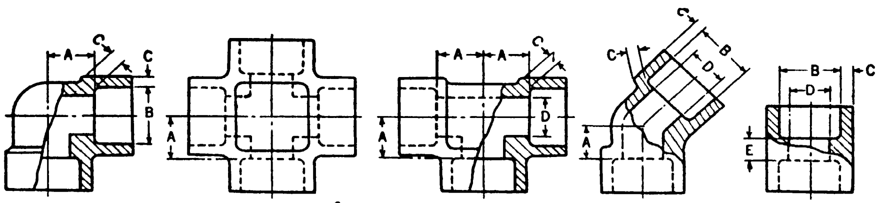
[c] Dimensions *H* and *J* correspond to the outside and inside diameters of pipe as given in *ASA* B36.10-1939, Schedule 40.

* These diameters are identical with the diameters of what was formerly designated as *Standard Weight Pipe* of the corresponding sizes.

and a dimensional standard[7] has been prepared under the procedure of the *American Standards Association* to unify heretofore divergent dimensions for the same type welding fittings as produced by different manufacturers. Standard dimensions for steel butt-welding elbows, tees, caps, and lapped-joint stub ends are given in Table 8. Dimensions for eccentric and concentric reducers, and 180-deg return bends are not shown in Table 8, but are included in the American Standard. Larger sizes also are available in some types of fittings. The welding bevel which is a straight 37½-deg V for wall thickness ¾ in. and below, and a U-bevel for thicknesses heavier than ¾ in., conforms to the recommended practice of *ASA* Standard B16e-1939, American Standard for Steel Pipe Flanges and Flanged Fittings. The latter also contains dimensions for steel welding neck flanges for pressures up to 2500 psi, and slip-on welding flanges for 150 and 300 psi. Table 9 gives these dimensions for welding-neck and slip-on welding flanges suitable for 150 psi gage pressure.

Socket-welding fittings also are commercially available. These fittings have a machined recess for inserting the pipe which is attached by a

TABLE 10. AMERICAN STANDARD DIMENSIONS OF SOCKET-WELDING ELBOWS, TEES, CROSSES, 45-DEG ELBOWS, AND COUPLINGS



Nominal Pipe Size	Minimum Depth of Socket	Center to Bottom of Socket: 90-Deg Ells, Tees, Crosses, Sched. 40 & 80	Center to Bottom of Socket: 90-Deg Ells, Tees, Crosses, Sched. 160	Center to Bottom of Socket: 45-Deg Ells, Sched. 40 & 80	Center to Bottom of Socket: 45-Deg Ells, Sched. 160	Couplings Distance Between Bottom Sockets	Bore Diameter of Socket, Minimum	Minimum Socket Wall Thickness, Sched. 40	Minimum Socket Wall Thickness, Sched. 80	Minimum Socket Wall Thickness, Sched. 160	Bore Diameter of Fittings, Sched. 40	Bore Diameter of Fittings, Sched. 80	Bore Diameter of Fittings, Sched. 160
		A		A		E	B	C[a]			D		
⅛	⅜	7/16	.....	5/16	.....	⅛	0.420	0.125	0.125	......	0.269	0.215	....
¼	⅜	7/16	.....	5/16	.....	¼	0.555	0.125	0.149	....	0.364	0.302	....
⅜	⅜	17/32	.....	5/16	....	¼	0.690	0.125	0.158	......	0.493	0.423	......
½	⅜	⅝	¾	7/16	½	⅜	0.855	0.136	0.184	0.234	0.622	0.546	0.466
¾	½	¾	⅞	½	9/16	⅜	1.065	1.141	0.193	0.273	0.824	0.742	0.614
1	½	⅞	1 1/16	9/16	11/16	½	1.330	0.166	0.224	0.313	1.049	0.957	0.815
1¼	½	1 1/16	1¼	11/16	13/16	½	1.675	0.175	0.239	0.313	1.380	1.278	1.160
1½	½	1¼	1½	13/16	1	½	1.915	0.181	0.250	0.351	1.610	1.500	1.338
2	⅝	1½	1⅝	1	1⅛	¾	2.406	0.193	0.273	0.429	2.067	1.939	1.689
2½	⅝	1⅝	2¼	1⅛	1¼	¾	2.906	0.254	0.345	0.469	2.469	2.323	2.125
3	⅝	2¼	2½	1¼	1⅜	¾	3.535	0.270	0.375	0.546	3.068	2.900	2.626

All dimensions are given in inches.

[a] Dimension C is 1¼ times the nominal pipe thickness, minimum, but not less than ⅛ in.

Reducing sizes have same center to bottom of socket dimension as the largest size of reducing fitting.

fillet weld between the pipe wall and socket end. Use of socket-welding fittings generally is restricted to nominal pipe sizes 3 in. and smaller in which range commercial fittings are available. This type of fitting has gained rapid acceptance owing to its ease of installation, low cost, and ability to make a pressure tight joint without weakening the pipe, as is the case with threading. Dimensions for socket-welding fittings, in accordance with *ASA* Standard B16.11-1946, are given in Table 10.[8]

VALVES

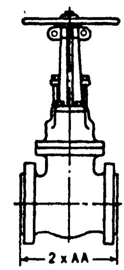
Valves are made with both threaded and flanged ends for screwed and bolted connections just as are pipe fittings.

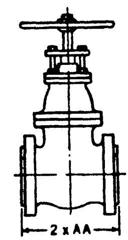
The material used for valves of small size is generally brass or bronze for low pressures and forged steel for high pressures, while in the larger sizes either cast-iron, cast-steel or some of the steel alloys are employed. Practically all iron or steel valves intended for steam or water work are bronze-mounted or trimmed.

Brass, bronze, and iron valves are generally designed for standard or extra heavy service, the former being used up to 125 lb and the latter up to 250 lb saturated steam working pressure, although most manufacturers also make valves for medium pressure up to 175 lb steam working pressure.

Table 11. American Standard Contact Surface to Contact Surface Dimensions of Cast-iron and Steel Flanged Wedge Gate Valves, *ASA* B16.10-1939

Nominal Pipe Size	Contact Surface to Contact Surface Dimensions, (2 × AA)				
	Cast-Iron[a]			Steel	
	125	175[bc]	250[b]	150[b]	300[b]
1	....	....	....	....	....
1¼	....	....	....	....	....
1½	....	....	....	....	7½
2	7	7¼	8½	7	8½
2½	7½	8	9½	7½	9½
3	8	9¼	11⅛	8	11⅛
3½	8½	10	11⅞	8½	11⅞
4	9	10½	12	9	12
5	10	11½	15	10	15
6	10½	13	15⅞	10½	15⅞
8	11½	14¼	16½	11½	16½
10	13	16¾	18	13	18
12	14	17½	19¾	14	19¾
14 O.D.	15	....	22½	15	30
16 O.D.	16	....	24	16	33
18 O.D.	17	....	26	17	36
20 O.D.	18	....	28	18	39
24 O.D.	20	....	31	20	45





All dimensions given in inches.

[a] These dimensions are the same for Cast-Iron Double Disc Flanged Gate Valves.

[b] These are pressure designations which refer to the primary service ratings in pounds per square inch of the connecting end flanges.

[c] The connecting end flanges of 175 lb valves are the same as those on 250 lb valves.

Note 1:—Where dimensions are not given, the sizes either are not made or there is insufficient demand to warrant the expense of unification.

Note 2:—Female and groove joint facings have bottom of groove in same plane as *flange edge*, and center to contact surface dimensions for these facings are reduced by the amount of the raised face.

The more common types are gate valves or straightway valves, globe valves, angle valves, check valves and automatic valves, such as reducing and back-pressure valves.

Gate valves are the most frequently used of all valves since in their open position the resistance to flow is a minimum, but they should not be used where it is desired to throttle the flow; globe valves should be used for this purpose. Gate valves may be secured with either a rising or a non-rising stem, although in the smaller size the rising stem is more commonly used. The rising stem valve is desirable because the positions of the handle and stem indicate whether the valve is open or closed, although space limitations may prevent its use. The globe valve is less expensive to manufacture than the gate valve, but its peculiar construction offers a high resistance to flow and may prevent complete drainage of the pipe line. These objections are of particular importance in heating work.

An American Standard, *ASA* B16.10-1939, has been prepared giving the face-to-face dimensions of ferrous flanged and welding-end valves. The following types are covered: wedge gate, double disc gate, globe and angle, and swing check. One purpose of establishing these dimensions is to insure that gate valves of a given rating and flange dimension of either the wedge or double disc design will be interchangeable in a pipe line. Contact surface to contact surface dimensions of cast-iron and steel flanged wedge-gate valves are given in Table 11. End-to-end dimensions for steel butt-welding valves in sizes up to 8 in., inclusive, are the same as those given in Table 11 for steel valves.

Check valves are automatic in operation and permit flow in only one direction, depending for operation on the difference in pressure between the two sides of the valve. The two principal kinds of check valves are the swing check in which a flapper is hinged to swing back and forth, and the lift check in which a dead weight disc moves vertically from its seat

Valves commonly used for controlling steam or water supply to radiators constitute a special class since they are manufactured to meet heating system requirements. These valves are generally of the angle type and are usually made of brass. Graduations on the heads or lever handles are often supplied to indicate the relative opening of the valve.

Automatic control of steam supply to individual radiators can be effected by use of direct-acting radiator valves having a thermostatic element at the valve, or near to it. The direct-acting valve is usually an angle-type valve containing a thermostatic element which permits the flow of steam in accordance with room temperature requirements. These valves usually are capable of adjustment to permit variation in room temperature to suit ndividual taste.

Ordinary steam valves may be used for hot water service by drilling a $\frac{1}{16}$-in. hole through the web forming the seat to insure sufficient circulation to prevent freezing when the valve is closed. Valves made for use in hot water heating systems are of simpler design, one type consisting of a simple butterfly valve, and another of a quick opening type in which a part in the valve mechanism matches up with an opening in the valve body.

In one-pipe steam-heating systems, automatic air valves are required at the radiators. Two common types of air valves available are the vacuum type and the straight-pressure type. Vacuum valves permit the expulsion of air from the radiators when the steam pressure rises and, in addition, act as checks to prevent the return of air into the radiator when a vacuum is formed by the condensation of steam after the supply pressure has dropped. Ordinary air valves permit the expulsion of air from the radiator when steam is supplied under pressure, but when a vacuum tends to be formed the air is drawn back into the radiator.

REFERENCES

[1] See (1) *Piping Handbook*, by Walker and Crocker (McGraw-Hill Co.); (2) *A Manual for The Design of Piping for Flexibility by the Use of Graphs*, by E. A. Wert, S. Smith, E. T. Cope (The Detroit Edison Company).

[2] See *API Specification* 5L *for Line Pipe, American Petroleum Institute.*

[3] *Standard Manual on Pipe Welding (Heating, Piping and Air Conditioning Contractors National Association*, Second Edition, 1951). *Welding Handbook* (American Welding Society, 1942).

[4] ASME Power Boiler Code, *American Society of Mechanical Engineers.*

[5] American Standard Code for Pressure Piping, *ASA* B-31.1—1942, *American Standards Association.*

[6] Marine Engineering Regulations of the Coast Guard, *American Bureau of Shipping.*

General Specifications for Inspection of Material, Appendix VII, Welding, *U. S. Navy.* Specifications for Welding, Appendix 5, Part 1—General—for vessels of the *U. S. Navy, Bureau of Ships,* April, 1940.

[7] American Standard, Steel Butt-Welding Fittings, *ASA* B16.9-1940, *American Standards Association.*

[8] American Standard, Steel Socket-Welding Fittings, *ASA* B16.11-1946, *American Standards Association.*

CHAPTER 27

PIPE INSULATION

Heat Losses from Bare and Insulated Pipes, Low Temperature Pipe Insulation, Insulation of Pipes to Prevent Freezing, Economical Thickness of Pipe Insulation, Underground Pipe Insulation

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THE heat loss from uninsulated pipes may be of considerable magnitude if the temperature of the surrounding medium differs appreciably from that of the fluid conveyed. Losses are increased by rapid motion of the surrounding air or by contact of the pipe with bodies of high conductivity. Careful consideration must, therefore, be given to this factor in a properly designed system, and adequate insulation provided, if necessary.

HEAT LOSSES FROM BARE PIPES

Heat losses from horizontal bare steel pipes, based on tests at *Mellon Institute* and calculated from the fundamental radiation and convection equations (Chapter 5), are given in Table 1. Heat losses from horizontal copper tubes and pipes with tarnished surfaces, are given in Table 2.[1]

Heat losses from bare pipe of materials having lower emissivities may be calculated from data appearing in Chapter 5.

The area in square feet per linear foot of pipe is given in Table 3 for various standard pipe sizes, and Table 4 for copper tubing, while Table 5 gives the area in square feet of flanges and fittings for various standard pipe sizes. These tables can be used to advantage in estimating the amount of insulation required.

Very often, when pipes are insulated, flanges and fittings are left bare so as to allow for easy access to the fittings in case of repairs. The fact that a pair of 8-in. standard flanges having an area of 2.41 sq ft would lose, at 100 lb steam pressure, an amount of heat equivalent to more than a ton of coal per year, shows the necessity for insulating such surfaces.

Examples 1 and *2* show how the annual heat loss from uncovered pipe and its dollar value may be computed from the data in Table 1.

Example 1: Compute the total annual heat loss from 165 ft of 2 in. bare pipe in service 4000 hr per year. The pipe is carrying steam at 10 lb pressure and is exposed to an average air temperature of 70 F.

Solution: The pipe temperature is taken as the steam temperature, which is 239.4 F, obtained by interpolation from Steam Tables. The temperature difference between the pipe and air = 239.4 − 70 = 169.4 F. By interpolation of Table 1 between temperature differences of 157.1 and 227.7 F, the heat loss from a 2-in. pipe at a temperature difference of 169.4 F is found to be 1.624 Btu per (hr) (linear ft) (F deg). The total annual heat loss from the entire line = 1.624 × 169.4 × 165 (linear ft) × 4000 (hr) = 181,600 Mb. (Mb = 1000 Btu.)

Example 2: Coal costing $11.50 per ton and having a calorific value of 13,000 Btu per pound is being burned in the furnace supplying steam to the pipe line given in the previous example. If the system is operating at an overall efficiency of 55 percent, determine the monetary value of the annual heat loss from the line.

Solution: The cost of heat per 1000 Mb supplied to the system = 1,000,000 × 11.5 (dollars) ÷ [13,000 (Btu) × 2000 (lb) × 0.55 (efficiency)] = $0.804. The total cost of heat lost per year = 0.804 × 181.6 (thousand Mb) = $146.00.

PIPE INSULATIONS

Pipe insulations are of several general forms and are made of various types of material The most common form is the rigid sectional covering either split longitudinally into halves or cut through on one side and scored on the other, to facilitate assembling on pipes. Preformed ma-

TABLE 1. HEAT LOSSES FROM HORIZONTAL BARE STEEL PIPES

Expressed in Btu per (hour) (linear foot) (Fahrenheit degree difference between the pipe and surrounding still air at 70 F)

Nominal Pipe Size (Inches)	Hot Water				Steam		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	299.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.455	0.495	0.546	0.584	0.612	0.706	0.760
¾	0.555	0.605	0.666	0.715	0.748	0.866	0.933
1	0.684	0.743	0.819	0.877	0.919	1.065	1.147
1¼	0.847	0.919	1.014	1.086	1.138	1.324	1.425
1½	0.958	1.041	1.148	1.230	1.288	1.492	1.633
2	1.180	1.281	1.412	1.512	1.578	1.840	1.987
2½	1.400	1.532	1.683	1.796	1.883	2.190	2.363
3	1.680	1.825	2.010	2.153	2.260	2.630	2.840
3½	1.900	2.064	2.221	2.433	2.552	2.974	3.215
4	2.118	2.302	2.534	2.717	2.850	3.320	3.590
5	2.580	2.804	3.084	3.303	3.470	4.050	4.385
6	3.036	3.294	3.626	3.886	4.074	4.765	5.160
8	3.880	4.215	4.638	4.960	5.210	6.100	6.610
10	4.760	5.180	5.680	6.090	6.410	7.490	8.115
12	5.590	6.070	6.670	7.145	7.500	8.800	9.530

TABLE 2. HEAT LOSS FROM HORIZONTAL TARNISHED COPPER PIPE

Expressed in Btu per (hour) (linear foot) (Fahrenheit degree difference between the pipe and surrounding still air at 70 F)

Nominal Pipe Size (Inches)	Hot Water (Type K Copper Tube)				Steam (Standard Pipe Size Pipe)		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.250	0.287	0.300	0.321	0.433	0.500	0.530
¾	0.340	0.381	0 409	0.429	0.533	0.543	0.654
1	0.440	0.475	0.509	0.536	0.636	0.746	0.803
1¼	0.500	0.559	0.618	0.622	0.764	0.878	0.934
1½	0.580	0.656	0.710	0.750	0.904	1.053	1.120
2	0.730	0.825	0.890	0.957	1.101	1.273	1.364
2½	0.880	1.000	1.091	1.143	1.305	1.490	1.605
3	1.040	1.175	1.272	1.343	1.560	1.800	1.940
3½	1.180	1.350	1.454	1.535	1.750	2.020	2.170
4	1.460	1.500	1.635	1.715	1.941	2.240	2.430
4½	....	....	....	....	2.131	2.465	2.650
5	1.600	1.812	1.980	2.071	2.387	2.770	2.990
6	1.840	2.125	2.270	2.430	2.740	3.210	3.440
8	2.400	2.685	2.910	3.110	3.310	4.050	4.370

terials are supplied in segments for assembly on large pipes. The sectional coverings are generally supplied with a pasted-on canvas jacket. Blanket insulations are sometimes used for wrapping large pipes, particularly where removal for frequent servicing of the pipe is necessary. Fittings and bends are commonly covered with portions of standard preformed

TABLE 3. EXTERNAL SURFACE PER LINEAR FOOT OF PIPE

Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)
½	0.22	2	0.622	5	1.456
¾	0.275	2½	0.753	6	1.734
1	0.344	3	0.917	8	2.257
1¼	0.435	3½	1.047	10	2.817
1½	0.498	4	1.178	12	3.338

insulation or, when irregular in contour, with plastic materials known as insulating cements. Insulation is secured to pipes with staples which are used to bridge the joint between half sections, and with metal pipe covering bands or rings of wire which secure individual sections and effect a junction between abutting sections. Surfacc finishes used over pipe insulation depend upon the service encountered and appearance desired. Canvas jackets are most common, although asbestos paper or asbestos finishing cements are sometimes employed. Insulation outdoors should be waterproof, and is generally protected with an asphalt felt for piping and asphaltic cements for fittings. Insulation on lines carrying cold water, brine, or other cold fluids is carefully finished to obtain adequate sealing against the penetration of water vapor.

The selection of pipe insulation for a particular service condition must be made with full consideration of a number of properties in addition to thermal conductivity. Factors which may be of more importance than the thermal conductivity are: ease of application, fire resistance, heat stability, weathering stability, resistance to damage by physical abuse, and others which may apply to a particular installation. A complete evaluation of pipe insulation cannot be included here. Insulation manufacturers should be consulted in regard to the selection of insulation which is to meet specific requirements.

HEAT LOSSES FROM INSULATED PIPES

The conductivities of various materials used for insulating steam and hot water systems are given in Table 6. They are given as functions of the mean temperatures or the arithmetic mean of the inner and outer surface temperatures of the insulations. It should be emphasized that they are the average values obtained from a number of tests made on each type of material; also, that in the use of *conductivity* all variables due to differences in thickness, pipe sizes, and air conditions, are eliminated. Individual manufacturer's materials will, of course, vary in conductivity to some extent from these values.

The heat losses through 1, 1½, and 2-in. thick, 85 percent magnesia type of insulation for temperature differences between the pipe and the surrounding atmosphere up to 280 F, are shown in Figs. 1, 2, and 3.

TABLE 4. EXTERNAL SURFACE PER LINEAR FOOT OF COPPER TUBING
Outside diameter ⅛ in. greater than nominal size

Tube Size (Inches)	Surface Area (Sq Ft)	Tube Size (Inches)	Surface Area (Sq Ft)	Tube Size (Inches)	Surface Area (Sq Ft)
½	0.164	2	0.556	5	1.342
¾	0.229	2½	0.687	6	1.604
1	0.295	3	0.818	8	2.128
1¼	0.360	3½	0.949	....	....
1½	0.426	4	1.080	....	....

Table 5. Area of Flanged Fittings, Square Feet[a]

Nominal Pipe Size (Inches)	Flanged Coupling		90 Deg Ell		Long Radius Ell		Tee		Cross	
	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy
1	0.320	0.438	0.795	1.015	0.892	1.083	1.235	1.575	1.622	2.07
1¼	0.383	0.510	0.957	1.098	1.084	1.340	1.481	1.925	1.943	2.53
1½	0.477	0.727	1.174	1.332	1.337	1.874	1.815	2.68	2.38	3.54
2	0.672	0.848	1.65	2.01	1.84	2.16	2.54	3.09	3.32	4.06
2½	0.841	1.107	2.09	2.57	2.32	2.76	3.21	4.05	4.19	5.17
3	0.945	1.484	2.38	3.49	2.68	3.74	3.66	5.33	4.77	6.95
3½	1.122	1.644	2.98	3.96	3.28	4.28	4.48	6.04	5.83	7.89
4	1.344	1.914	3.53	4.64	3.96	4.99	5.41	7.07	7.03	9.24
4½	1.474	2.04	3.95	5.02	4.43	5.46	6.07	7.72	7.87	10.07
5	1.622	2.18	4.44	5.47	5.00	6.02	6.81	8.52	8.82	10.97
6	1.82	2.78	5.13	6.99	5.99	7.76	7.84	10.64	10.08	13.75
8	2.41	3.77	6.98	9.76	8.56	11.09	10.55	14.74	13.44	18.97
10	3.43	5.20	10.18	13.58	12.35	15.60	15.41	20.41	19.58	26.26
12	4.41	6.71	13.08	17.73	16.35	18.76	19.67	26.65	24.87	34.11

[a] Including areas of accompanying flanges bolted to the fitting.

Standard thicknesses of 85 percent magnesia pipe covering are not exactly 1 in. However, the loss through any given thickness of insulation can be obtained by interpolation. Also, the losses through any of the insulations given in Table 6 can be obtained by multiplying the losses obtained from Figs. 1, 2, or 3 by the factors given in Table 7.

Pipes operating at high temperatures are frequently insulated to the

Table 6. Thermal Conductivity (k) of Various Type Pipe Insulations for Medium and High Temperature Pipe[a]

Expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference per inch)

Types of Insulating Materials	Density Lb/Cu Ft	Temp. Range of Accepted Use	Mean Temperature, F Deg				
			100	200	300	400	500
85% Magnesia—Type	11-14	Up to 600 F	0.39	....	0.45	....	0.51
Corrugated Asbestos—Type							
4 Ply per 1 in.	11-13	Up to 300 F	0.57	0.68	0.80	...	....
6 Ply per 1 in.	15-17	Up to 300 F	0.51	0.59	0.69	....	....
8 Ply per 1 in.	18-20	Up to 300 F	0.49	0.57	0.65		...
Laminated Asbestos—Type (35-40 laminations per 1 in.)	30-35	Up to 700 F	0.39	0.44	0.49	0.54	...
Mineral Wool—Type	10-15	Up to 800 F	0.40	0.45	0.50	0.55	.
Diatomaceous Silica—Type	25-30	Up to 1900 F	0.63	0.66	0.69	0.72	0.75
Brown Asbestos Fiber—Type	13-15	Up to 1200 F	0.34	0.39	0.44	0.49	0.54

[a] Average values from laboratories for insulating materials of various manufacturers.

Table 7. Pipe Covering Factors

Types of Insulating Materials	Temperature Difference, Pipe to Air, F Deg				
	100	200	300	400	500
Corrugated Asbestos—Type					
4 Ply per 1 in.	1.30	1.36	1.42	....	....
6 Ply per 1 in.	1.19	1.23	1.27	....	....
8 Ply per 1 in.	1.15	1.19	1.23	....	....
Laminated Asbestos—Type	0.96	0.98	1.00	1.02	1.04
Mineral Wool—Type	0.98	1.00	1.02	1.05	1.07
Diatomaceous Silica—Type	1.37	1.36	1.35	1.35	1.34
Brown Asbestos Fiber—Type	0.86	0.88	0.91	0.93	0.96

best advantage by combining a high temperature insulation near the pipe with a moderate or low temperature insulation around it as an outer layer. By this method an efficient material may be used for each of the two temperature ranges encountered. In calculating the heat loss through such a

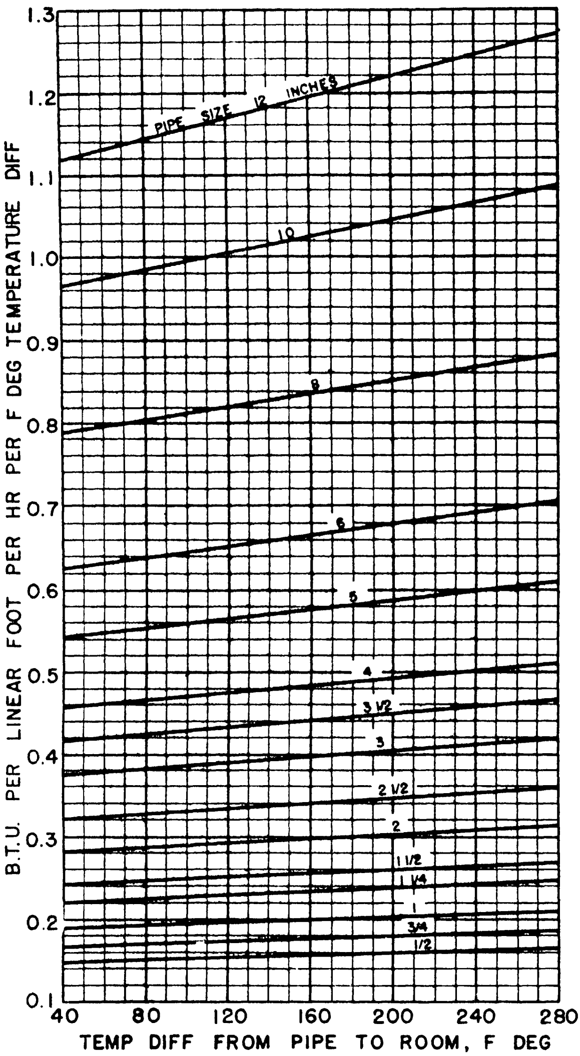


FIG. 1. HEAT LOSS THROUGH 1 IN. THICK 85 PERCENT MAGNESIA TYPE COVERING

combination the mean temperature of each layer must be determined along with the thickness of each. This is readily done in three or four calculations performed as a series of approximations, in which assumptions of thickness and mean temperature are adjusted as indicated in the discussion which follows.

In the case of a single thickness of pipe covering, the quantity of heat transferred per square foot of outer surface of the insulation is given by the equation:

$$q_o = \frac{k(t_1 - t_2)}{r_2 \log_e \frac{r_2}{r_1}} \tag{1}$$

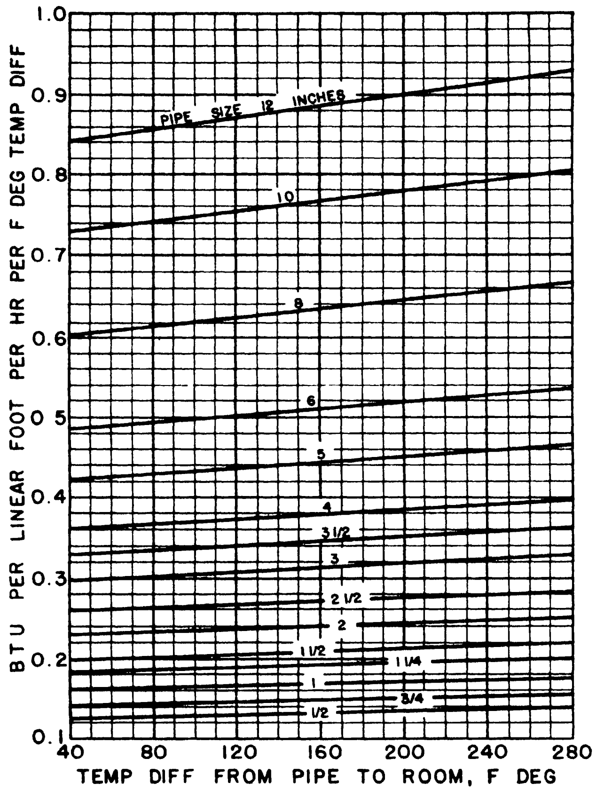


FIG. 2. HEAT LOSS THROUGH 1½ IN. THICK 85 PERCENT MAGNESIA TYPE COVERING

where

q_o = Btu per (hour) (square foot of outer surface of insulation).

r_1 = outer radius of pipe or inner radius of insulation, inches.

r_2 = outer radius of insulation, inches.

k = thermal conductivity of insulation, Btu per (hour) (square foot) (Fahrenheit degree per inch).

t_1 = temperature of inner surface of insulation, Fahrenheit degrees.

t_2 = temperature of outer surface of insulation, Fahrenheit degrees.

It is convenient to work from the outer surface of the insulation, since the loss through the covering must be determined from the outer surface loss by means of surface loss curves such as given in Fig. 4.

After the true heat loss is obtained, the loss per square foot of pipe surface can be calculated from the relationship:

$$q_i = q_o(r_2/r_1)$$

where

q_i = Btu per (hour) (square foot outer surface of pipe).

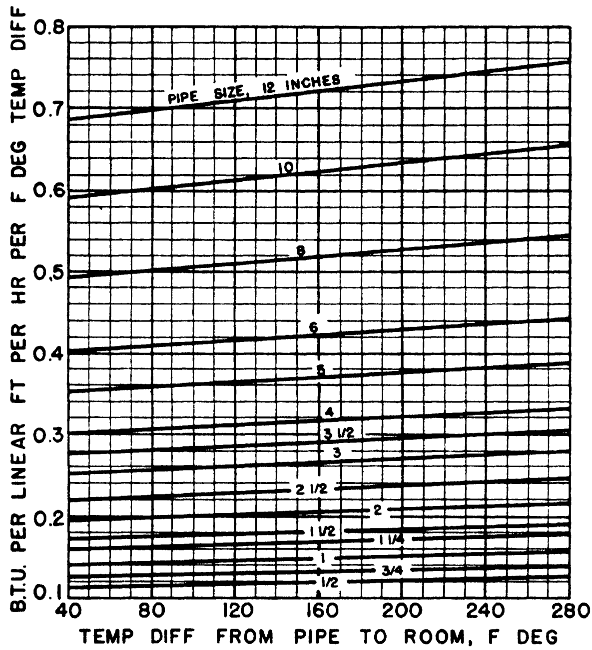


FIG. 3. HEAT LOSS THROUGH 2 IN. THICK 85 PERCENT MAGNESIA TYPE COVERING

The heat loss through two or more thicknesses of insulation applied to a pipe can be calculated by means of the equation:

$$q_o = \frac{t_1 - t_2}{\dfrac{r_s \log_e \dfrac{r_2}{r_1}}{k_1} + \dfrac{r_s \log_e \dfrac{r_3}{r_2}}{k_2} + \cdots} \qquad (2)$$

where

r_3 = outer radius of second layer of insulation, inches.

r_s = outer radius of last layer of insulation, inches.

The method of solving Equation 2, which is the most difficult of the two, is given in Example 3.

Example 3: Compute the heat loss per linear foot of pipe surface per hour from a 6-in. pipe, insulated with a 3-in. thickness of diatomaceous silica, and a 2-in. thickness of 85 percent magnesia. The pipe is operating at a temperature of 1200 F and is exposed to a room temperature of 80 F.

Solution: In figuring the heat loss from Equation 2, it is necessary to first make an assumption for the outer surface temperature t_2 and the temperature between the diatomaceous silica and 85 percent magnesia insulation, so that the mean temperature of each material can be obtained and the thermal conductivity corresponding to the mean temperature of each material substituted in the formula. First assume an outer surface temperature of 140 F and a temperature of 570 F between the two materials corresponding to a mean temperature of (1200 + 570) ÷ 2 or 885 F for the diatomaceous silica and (570 + 140) ÷ 2 or 355 F for the 85 percent magnesia insulation. The conductivities of these two materials at mean temperatures of 885 and 355 F, interpolated from Table 6, are 0.865 and 0.5 Btu, respectively.

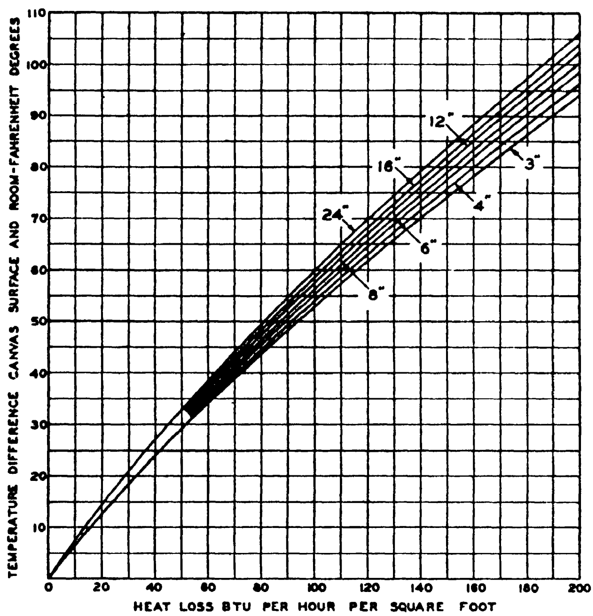


FIG. 4. HEAT LOSS FROM CANVAS-COVERED CYLINDRICAL SURFACES OF VARIOUS DIAMETERS

These values are substituted in Equation 2 and a trial calculation made. For a nominal 6-in. steel pipe $r_1 = 3.312$, $r_2 = 6.312$ and $r_3 = 8.312$ then,

$$q_0 = \frac{1200 - 140}{\dfrac{8.312 \log_e \dfrac{6.312}{3.312}}{0.865} + \dfrac{8.312 \log_e \dfrac{8.312}{6.312}}{0.5}} = \frac{1060}{6.2 + 4.85} = 98.3 \text{ Btu.}$$

The temperature drop from the outer surface of the insulation to the surrounding air for a heat loss of 98.3 Btu is found from Fig. 4 to be 57 F for a 16-in. O.D. cylindrical surface, or 57 + 80 F room temperature = 137 F surface temperature. Since a surface temperature of 140 F was assumed, it is evident that a temperature closer to 137 F, or, for instance, 138 F should be used for recalculation:

$$q_0 = \frac{1200 - 138}{6.2 + 4.58} = 98.4 \text{ Btu.}$$

Since the temperature drop through each material is equal to the heat flow times the actual resistance of each material, the temperature drop through the diatomaceous silica is 98.4 × 6.2 = 610 F, or the temperature between the two insulating materials is (1200 − 610) = 590 F. Since a temperature of 570 F between the two materials was assumed, it is obvious that a temperature closer to 590, or for instance 586 F may be selected. The mean temperatures of the two insulations corresponding to the new assumptions are (1200 + 586) ÷ 2 = 893 and (586 + 138) ÷ 2 = 362, and the interpolated conductivities corresponding to the new mean temperatures are 0 87 and 0.505 for the diatomaceous silica and 85 percent magnesia, respectively. By substituting in Equation 2

$$q_o = \frac{1200 - 138}{\frac{5.36}{0.87} + \frac{2.29}{0.505}} = \frac{1062}{6.16 + 4.58} = 99.3 \text{ Btu.}$$

Again referring to Fig. 4, it is seen that the temperature drop from the outer surface of the insulation to the surrounding air for a heat loss of 99.3 Btu = 58 F, which corresponds to the surface temperature of 138 F last assumed. The temperature drop through the diatomaceous silica is 99.3 × 6.16 = 612 F, corresponding to a temperature of 588 F between the two materials, which checks very closely with the temperature of 585 F last assumed. The heat loss is therefore 99.3 × 8.312 ÷ 3.312 or 249 Btu per sq ft of pipe surface. Since the surface area per linear foot of 6-in. pipe is 1.734 sq ft (Table 3), the heat loss per linear foot of pipe will be 249 × 1.734 = 432 Btu per hr.

The rate of heat loss from a surface maintained at constant temperature is greatly increased by air circulation over the surface. In the case of well-insulated surfaces, the increases in losses due to air velocity are very small as compared with increases from bare surfaces, because of the fact that air flowing over the surface of the insulation can increase only the conductance of heat from surface to air, and cannot change the internal conductance of the insulation itself. The maximum increase in heat loss due to air velocity ranges from about 15 percent in the case of 1-in. thick insulation, to about 5 percent in the case of 3-in. thick insulation, provided that the insulation is thoroughly sealed so that air can flow only over the surface. If the conditions are such that the air may circulate through cracks and crevices in the insulation, the increases may be far greater than those given. Therefore, it is essential that insulation be applied in such a manner that air circulation within it, or between it and the pipe, is avoided.

Fig. 4 shows the loss of heat from canvas-covered, cylindrical surfaces of various outside diameters when the surface to air temperature difference is low. The data are from tests made at Mellon Institute.

The frequent practice of omitting insulation on that portion of a pipe which passes through a masonry wall, or which may be in contact with other metals, should be avoided. Physical contact between the pipe surface and other structural materials of high thermal conducitivity will result in heat transfer much greater than that shown in Tables 1 and 2 for transfer from bare pipe to air.

The saving due to use of insulation on piping is illustrated in *Example 4*.

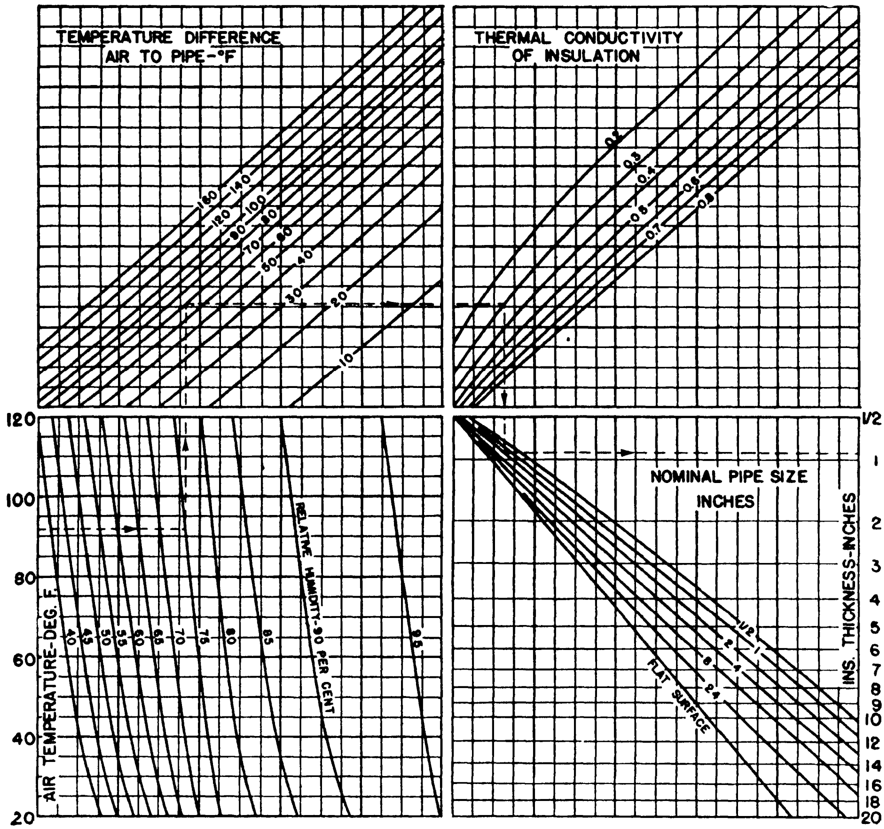
Example 4: If the steam line given in *Examples 1* and *2* is covered with 1 in. thick 85 percent magnesia, determine the resulting total annual loss through the insulation. Also compute the monetary value of the annual saving and the percentage of saving over the heat loss from the bare pipe.

Solution: By referring to Fig. 1, the coefficient for 1 in. magnesia on a 2-in. pipe is found to be 0.300 Btu per (hr) (linear ft of pipe) (deg temperature difference) at a temperature difference of 169.4 F. The total hourly loss per linear foot of pipe will then be 0.300 × 169.4 = 50.8 Btu. The total annual loss through the insulation = 50.8 × 165 (linear ft) × 4000 (hr) = 33,500 Mb. The annual bare pipe loss as determined in the solution of *Example 1* was found to be 181,600 Mb. The saving due to insulation is then 181,600 − 33,500 = 148,100 Mb per year.

From the solution of *Example 2*, it was found that the heat supplied to the system cost \$0.804 per thousand Mb. Therefore, the monetary value of the saving = 0.804 (dollars) × 148.1 (thousand Mb) = \$119.07, or 81.5 percent of the cost when using uninsulated pipe.

LOW TEMPERATURE PIPE INSULATION

Surfaces maintained at temperatures lower than the surrounding air are insulated to reduce the flow of heat and to prevent condensation.



ª Solve problems as indicated by dotted line, entering chart at lower left-hand scale

FIG. 5. THICKNESS OF PIPE INSULATION TO PREVENT CONDENSATION ON OUTER SURFACEª

The insulating material should absorb a minimum amount of moisture, because the absorption of moisture substantially increases the conductivity of the material. This property is particularly important in the insulation of surfaces that are below the dew-point of the surrounding air. In such cases, due to vapor pressure difference, it is necessary to seal the surface of the insulating material against the penetration of water vapor which would condense within the material, causing a serious increase in heat flow, possible breakdown of the material, and corrosion of metal surfaces. An insulating material with a high degree of moisture absorption might pick up moisture before application and then, when the seal is in place and the temperature of the insulated surface reduced, release that moisture to the cold surface. There are a number of methods of producing vapor seals, some of which have been worked out by insulation manufacturers to suit their products, and others by applicators and users. Unless time-proven methods are known, specifications of insulation manufacturers should be obtained and followed carefully.

The thickness of insulation required to prevent condensation on the outer surface is that thickness which will raise the temperature of the outer surface of the insulation to a point slightly higher than the dew-point of the surrounding vapor. The dew-point for various humidities can be readily ascertained from a psychrometric chart.

The approximate required thickness of insulation to prevent condensation on pipes and flat metallic surfaces may be obtained from Fig. 5 in

TABLE 8. HEAT GAINS FOR INSULATED COLD PIPES

Rates of heat transmission given in Btu per (hour) (Fahrenheit degree temperature difference between fluid in pipe and surrounding still air)

Based on materials having conductivity, k = 0.30

Nominal Pipe Size (Inches)	Ice Water Thickness			Brine Thickness			Heavy Brine Thickness		
	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface
½	1.5	0.110	0.502	2.0	0.098	0.446	2.8	0.087	0.394
¾	1.6	0.119	0.431	2.0	0.111	0.405	2.9	0.094	0.340
1	1.6	0.139	0.403	2.0	0.124	0.352	3.0	0.104	0.294
1¼	1.6	0.155	0.357	2.4	0.131	0.300	3.1	0.113	0.260
1½	1.5	0.174	0.351	2.5	0.134	0.270	3.2	0.118	0.238
2	1.5	0.200	0.322	2.5	0.151	0.244	3.3	0.134	0.214
2½	1.5	0.228	0.303	2.6	0.170	0.226	3.3	0.147	0.197
3	1.5	0.269	0.293	2.7	0.186	0.202	3.4	0.162	0.176
3½	1.5	0.295	0.282	2.9	0 191	0.183	3.5	0.176	0.167
4	1.7	0.294	0.248	2.9	0.209	0.176	3.7	0.182	0.154
5	1.7	0.349	0.239	3.0	0.241	0.165	3.9	0.202	0.138
6	1.7	0.404	0.233	3.0	0.259	0.150	4.0	0.228	0.130
8	1.9	0.455	0.201	3.0	0.318	0.140	4.0	0.263	0.116
10	1.9	0.559	0.198	3.0	0 383	0.135	4.0	0.309	0.110
12	1.9	0.648	0.194	3.0	0.438	0.131	4.0	0.364	0.108

which a surface resistance of 0.606, corresponding to a film conductance of 1.65, was used in calculating the curves. This value provides a slight factor of safety and its use is known to give satisfactory field results. In using the chart it is advisable to specify the next thicker, rather than the next thinner, commercial insulation in cases where an intermediate thickness is indicated.

Heat gains for pipes insulated with a material having an installed conducitivity of 0.30 Btu per (sq ft) (hr) (F deg per in.) are given in Table 8. This table may be used for any of the commercial insulations offered for this purpose since they have conductivities very near the 0.3 value used.

INSULATION OF PIPES TO PREVENT FREEZING

If the surrounding air temperature remains sufficiently low for an ample period of time, insulation cannot prevent the freezing of still water, or of water flowing at such a velocity that the quantity of heat carried in the water is not sufficient to take care of the heat losses which will result and cause the temperature of the water to be lowered to the freezing point.

Insulation can materially prolong the time required for the water to give up its heat, and if the velocity of the water flowing in the pipe is maintained at a sufficiently high rate, freezing may be prevented.

Table 9 may be used for making estimates of the thickness of insulation necessary to take care of still water in pipes at various water and surrounding air temperature conditions. Because of the damage and service interruptions which may result from frozen water in pipes, it is essential that an efficient insulation be utilized. This table is based on the use of a material having a conductivity of 0.30. The initial water temperature is assumed to be 10 deg above, and the surrounding air temperature 50 deg below the freezing point of water (temperature difference, 60 F).

The last column of Table 9 gives the minimum quantity of water at initial temperature of 42 F which should be supplied every hour for each linear foot of pipe, in order to prevent the temperature of the water from

TABLE 9. DATA FOR ESTIMATING REQUIREMENTS TO PREVENT FREEZING OF WATER IN PIPES WITH SURROUNDING AIR AT −18 F

Nominal Pipe Size (Inches)	Number of Hours to Cool 42 F Water to Freezing Point			Water Flow Required at 42 F to Prevent Freezing, Pounds per Linear Foot of Pipe per Hour		
	Thickness of Insulation in Inches (Conductivity, k = 0.30)					
	2	3	4	2	3	4
½	0.42	0.50	0.57	0.54	0.45	0.40
1	0.83	1.02	1.16	0.68	0.55	0.48
1½	1.40	1.74	2.02	0.81	0.68	0.58
2	1.94	2.48	2.90	0.95	0.75	0.64
3	3.25	4.27	5.08	1.24	0.94	0.79
4	4.55	6.02	7.20	1.47	1.11	0.93
5	5.92	7.96	9.69	1.73	1.29	1.06
6	7.35	9.88	12.20	1.98	1.46	1.19
8	10.05	13.90	17.25	2.46	1.78	1.43
10	13.00	18.10	22.70	2.96	2.12	1.70
12	15.80	22.20	28.10	3.43	2.45	1.93

being lowered to the freezing point. The weights given in this column should be multiplied by the total length of the exposed pipe line expressed in feet. As an additional factor of safety, and in order to provide against temporary reductions in flow occasioned by reduced pressure, it is advisable to double the rates of flow listed in the table. It must be emphasized that the flow rates and periods of time designated apply only for the conditions stated. To estimate for other service conditions, the following method of procedure may be used.

If water enters the pipe at 52 F instead of 42 F, the time required to cool it to the freezing point will be prolonged to twice that given in the table, or the rate of flow of water may be reduced so that the quantity required will be one-half that shown in the last column of Table 9. However, if the water enters the pipe at 34 F, it will be cooled to 32 F in one-fifth of the time given in the table. It will then be necessary to increase the rate of flow so that five times the specified quantity of water will have to be supplied in order to prevent freezing.

If the minimum air temperature is −38 F (temperature difference 80 F) instead of −18 F, the time required to cool the water to the freezing point will be 60/80 of the time given in the table, or the necessary quantity of water to be supplied will be 80/60 of that given.

In making calculations to arrive at the values given in Table 9, the loss of heat stored in the insulation, the effect of a varying temperature difference due to the cooling of pipe and water, and the resistance of the outer surface of the insulation to the transfer of heat to the air, have all been neglected. When these factors enter into the computations it is necessary to enlarge the factor of safety. Also as stated, the time shown in the table is that required to lower the water to the freezing point. A longer period would be required to freeze the water, but the danger point is reached when freezing starts. The flow of water will stop and the entire line will be in danger as soon as the water freezes across the section of the pipe at any point.

When water must remain stationary longer than the times designated in Table 9, the only safe way to insure against freezing is to install a steam or hot water line, or to place an electric resistance heater along the

TABLE 10. THICKNESS OF PIPE INSULATION ORDINARILY USED INDOORS[a]

Steam Pressure Psig or Condition	Steam Temperature Fahrenheit Degrees	Thickness of Insulation		
		Pipes Larger Than 4 In.	Pipes 2 In. to 4 In.	Pipes ½ In. to 1½ In.
0 to 25	212 to 267	1 in.	1 in.	1 in.
25 to 100	267 to 338	1½ in.	1 in.	1 in.
100 to 200	338 to 388	2 in.	1½ in.	1 in.
Low Superheat	388 to 500	2½ in.	2 in.	1½ in.
Medium Superheat	500 to 600	3 in.	2½ in.	2 in.
High Superheat	600 to 700	3½ in.	3 in.	2 in.

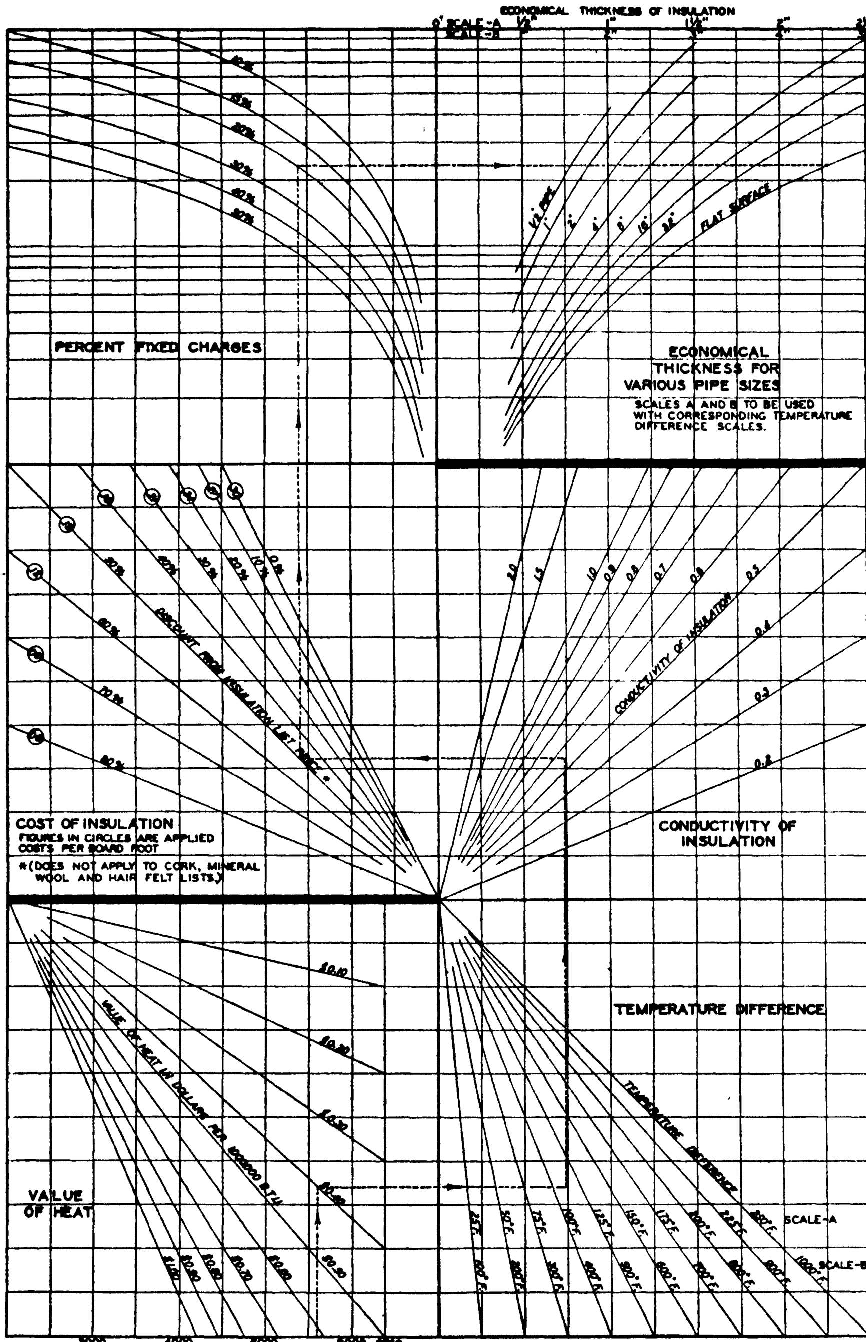
[a] All piping located outdoors or exposed to weather is ordinarily insulated to a thickness ½ in. greater than shown in this table, and covered with a waterproof jacket.

side of the exposed water line. The heating system and the water line are then insulated so that the heat losses from the heating system are not excessive, and the heating effect is concentrated against the water pipe where it is needed. For this form of protection 2 in. of an efficient insulation may be applied.

ECONOMICAL THICKNESS OF PIPE INSULATION

The thicknesses of insulation which ordinarily are used for various temperature conditions are given in Table 10. Where a thorough analysis of economic thickness is desired, this may be accomplished through the use of the chart, Fig. 6.

The dotted line on the chart illustrates its use in solving a typical example. In using the chart, start with the scale at the left bottom margin representing the given number of hours of operation per year; then proceed vertically to the line representing the given value of heat; thence horizontally to the right, to the line representing the given temperature difference; thence vertically to the line representing the conductivity of the given material; thence horizontally to the left, to the line representing the given discount on that material; thence vertically to the curve representing the required percent return on the investment; thence horizontally to the right, to the curve representing the given pipe



(L. B. McMillan, *Proc. National District Heating Association*, Vol. 18, p. 138).

FIG. 6. CHART FOR DETERMINING ECONOMICAL THICKNESS OF PIPE INSULATION

size; thence vertically to the scale at the top right margin where the economical thickness may be read off directly.

A rapid method for determining the economical thickness of insulation by use of tables has been published.[2]

UNDERGROUND PIPE INSULATION

Underground steam distribution lines are carried in protective structures of various types, sizes and shapes (see Chapter 28). Detailed data on commonly used forms of tunnels and conduit systems, have been published by the *National District Heating Association.*[3]

Pipes in tunnels are covered with sectional insulation to provide maximum thermal efficiency, and are also finished with good mechanical protection in the form of metal or waterproofing membrane outer jackets. In some instances, where actual submersion of hot lines may occur, it has been found good practice to firmly secure the covering with corrosion resistant wire, then sew on a wire-inserted asbestos fabric jacket with wire. This jacket is porous. The principle of withstanding submersion is that water may enter as water, then actually boil at the pipe surfaces and escape as steam without rupturing the insulation or jacket. Conduit systems are in more general use than tunnels. Pipes carried in conduits may be insulated with sectional insulation; however, the more usual practice is to fill the entire section of the conduit around the pipes with high quality, loose insulating material. The insulation must be kept dry at all times, and for this purpose effective waterproofing membranes enclose the insulation. A drainage system is also provided to divert water which may tend to enter the conduit.

Table 11. Thickness of Loose Insulation for Use as Fill in Underground Conduit Systems

Steam Pressure Psig or Condition	Steam Temperature Fahrenheit Degrees	Minimum Thickness of Insulation in Inches					Minimum Distance Between Steam and Return
		Steam Lines			Return Lines		
		Pipes Less than 4 In.	Pipes 4 In. to 10 In.	Pipes Larger than 12 In.	Pipes Less than 4 In.	Pipes 4 In. and Larger	
Hot Water, or 0 to 25	212 to 267	1½	2	2½	1¼	1½	1
25 to 125	267 to 352	2	2½	3	1¼	1½	1¼
Above 125, or superheat	352 to 500	2½	3	3½	1¼	1½	1½

The economical thickness of insulation for underground work is difficult to determine accurately due to the many variables which have to be considered. As a result of theories[4] previously developed, together with other experimental data which have been presented, the usual endeavor is to secure not less than 90 percent efficiency for underground piping. Table 11 can be used as a guide in arriving at the minimum thickness of loose insulation fills to use for laying out conduit systems. Other factors such as the number of pipes and their combination of sizes, as well as the standard conduit sizes, are primary controlling factors in the amount and thickness of insulation for use.

When sectional insulation is applied to lines in tunnels or conduits, usual practice is to apply the most efficient materials ½ in. less in thickness than that determined by the use of Fig. 6. The data in Fig. 6 are based on conditions of insulation exposed to the air, whereas normal ground temperature is substituted for air temperature in determining the temperature difference for use with the chart when applying it for underground pipe line estimates.

REFERENCES

[1] Heat Loss from Copper Piping, by R. H. Heilman (*Heating, Piping and Air Conditioning*, September, 1933, p. 458).

[2] Rapid Method of Determining the Economical Thickness of Pipe Insulation, by Utley W. Smith (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October 1947, p. 118).

[3] Handbook of the *National District Heating Association*, Second Edition, 1932.

[4] Theory of Heat Losses from Pipes Buried in the Ground, by J. R. Allen A.S.H.V.E. TRANSACTIONS, Vol. 26, 1920, p. 335).

CHAPTER 28

DISTRICT HEATING

Steam Distribution Piping, Pipe Sizes, Conduits for Piping, Pipe Tunnels, Overhead Distribution, Inside Piping, Metering, Steam Requirements, Rates

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THE term *district heating* refers to the heating of several buildings from a central plant as in the heating of portions of cities, or to the heating of groups of buildings as in institutions and factories. It is usually preferable, in a group of industrial or institutional buildings, that they be heated from a central plant rather than by individual plants. Fuel can generally be burned more efficiently, less labor is required, and often a central plant is cheaper to install. Those phases of district heating which frequently fall within the province of the heating engineer are outlined here, with data and information for solving incidental problems in connection with institutions and factories. Some data are included to cover the piping peculiar to heating systems which are to be supplied with purchased steam. A complete district heating installation should not be attempted without a thorough study of the entire problem by men competent and experienced in that industry.

Air Conditioning. In many cases steam from district heating mains can be used for air conditioning. There are three types of refrigeration machines which use steam as a source of energy. They are: (1) steam-driven compression machines, mostly turbine-driven; (2) steam jet machines; and (3) absorption machines. The method of using the steam for the cooling unit and for closed absorption systems is described in Chapter 36.

STEAM DISTRIBUTION PIPING

The methods used in district heating work for the distribution of steam are applicable to any problem involving the supply of steam to a group of buildings. The first step is to establish the route of the pipes, and in this matter, since the local conditions control the layout, little can be said regarding it.

Having established the route of the pipes, the next step is to calculate the pipe sizes. In district heating work it is common practice to design the piping system on the basis of pressure drop. The initial pressure and the minimum permissible terminal pressure are specified, and the pipe sizes are so chosen that the required amount of steam, with suitable allowances for future increases, will be transmitted without exceeding this pressure drop. The steam velocity is therefore almost disregarded and may reach a very high figure. Velocities of 35,000 fpm are not considered high. By the use of this method the pipe sizes are kept to a minimum, with consequent savings in investment.

The steam flowing through any section of the piping can be computed from a study of the requirements of the several buildings served. In general, a condensation rate of 0.25 lb per (hr) (sq ft of equivalent direct radiation) is a safe figure. This allows for line condensation which, however, is a small part of the total at times of maximum load. Miscellaneous steam requirements such as laundry, cooking, or process, should be individually calculated. The steam requirements for water heating should be taken into account, but in most types of buildings this load will be relatively small compared with the heating load, and will seldom occur at the time of the heating peak. Unusual features, such as large heaters for swimming pools, should not be overlooked.

The pressure at which the steam is to be distributed will depend upon (1) boiler pressure, (2) whether exhaust or live steam, and (3) pressure requirements of apparatus to be served. If steam has been passed through electrical generating units, the pressure will be considerably lower than if live steam, direct from the boilers, is used.

The advantages of low pressure distribution (2 to 30 psig) are: (1) smaller heat loss per square foot of pipe surface; (2) less trouble with traps and valves; (3) simpler problems in pressure reduction at the buildings; and (4) general reduction in maintenance costs. With distribution pressures not exceeding 40 psig, there is little danger even if the full distribution pressure should build up in the radiators through the faulty operation of a reducing valve; but with pressures higher than 50 psig, a second reducing valve or some form of emergency relief is usually desirable to prevent excessive pressures in the radiators.

The advantages of high pressure distribution are: (1) smaller pipe sizes; (2) greater adaptability of the steam to various operations other than building heating; and (3) wider flexibility in allowance for maximum pressure drop and ability to serve equipment requiring higher pressures.

Frequently the different kinds of apparatus which must be served require various minimum pressures. Kitchen equipment requires from 5 to 15 psig, the higher pressures being necessary for apparatus in which water is boiled, such as stock kettles and coffee urns. An increased amount of heating surface, which is easily obtained in some kinds of apparatus, results in quicker and more satisfactory operation at low pressures. For laundry equipment, particularly the mangle, a pressure of 75 psig is usually demanded, although 30 psi is sufficient if the flat work ironer is equipped with a large number of rolls, and if a slower rate of operation is permissible. Pressing machines and hospital sterilizers require about 50 psig. Where pressures are not as high as desired, higher pressures can be obtained by a steam compressor.

Important points in laying out underground conduits are:

1. The depth of the buried conduit should be kept at a minimum. Excavation costs are a large factor in the total cost.

2. An expansion joint, offset, or bend should be placed between each two anchors. Advantage should be taken of the flexibility of piping to absorb expansion wherever possible. Information on provisions for expansion will be found in Chapter 26.

3. A proper hydrostatic test should be made on the assembled line before the insulation and the top of the conduit are applied. The hydrostatic test pressure should be one and one-half times the maximum service pressure, and it should be held for a period of at least two hours without evidence of leakage.

Since it is difficult to make a concrete or masonry conduit absolutely water-tight, provision should be made for some seepage. The pipe should be protected by a waterproof jacket over the insulation, and the seepage drained from the inside of the conduit. Underdrainage of the conduit is generally provided for by a tile drain laid in crushed stone or gravel underneath the conduit. The tile underdrain should be carried to the sewer or some other drainage point. Manholes are required at intervals for access to valves, traps, and some types of expansion joints.

Where steam and return piping are installed in the same conduit, the return piping usually follows the same grade as the steam piping. In general, the condensation is pumped back under pressure.

Where it is possible to use basement or sub-sidewalk space for the distribution piping, the cost of installation and maintenance is greatly reduced.

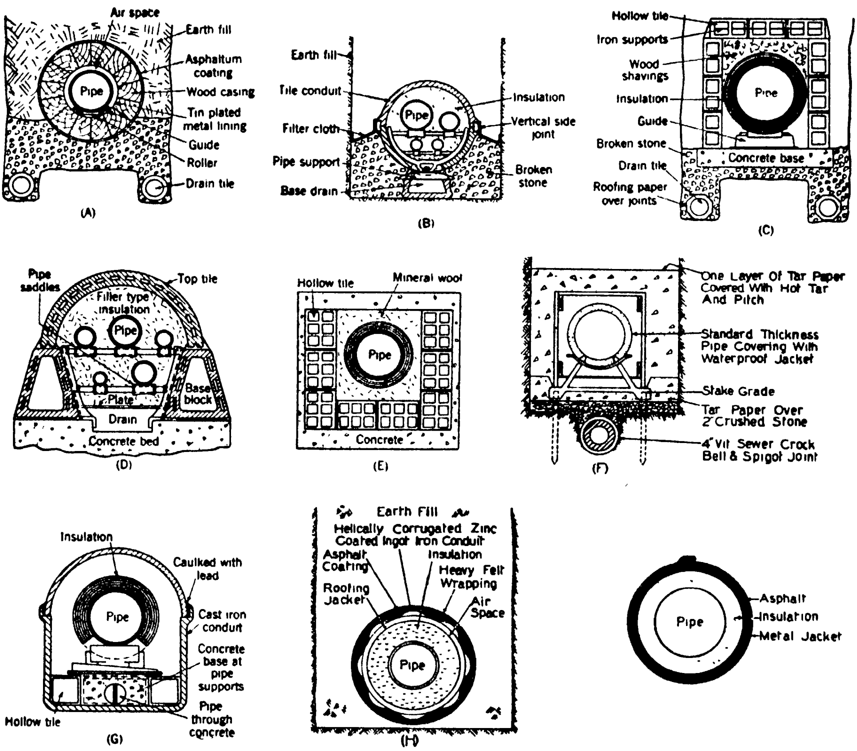


Fig. 1. Construction Details of Conduits Commonly Used

Pipe Sizes. The lengths of pipe, steam quantities, and initial and terminal pressures having been chosen, the pipe sizes can readily be calculated by means of Babcock's pressure drop formula given in Table 2 of Chapter 20.

CONDUITS FOR PIPING

Conduits for steam pipes buried underground should be reasonably waterproof, able to withstand earth loads and to take care of the expansion and contraction of the piping without strain or stress on the couplings, or without affecting the insulation or conduit. Expansion of the piping must be carefully controlled by means of anchors and expansion joints or bends so that the pipes can never come in contact with the conduit. Anchors can be anchor fittings or U-shaped steel straps which partially encircle the pipes, and are firmly bolted to a short length of structural or cast steel set in concrete. In general, cast steel is preferable to structural steel.

There are many types of conduits, some of which are manufactured products and some of which are built in the field. Some of the more common forms are illustrated in Fig. 1.

The conduit (A) is of a wood casing construction which has been widely used in the past. The wood casing is segmented, lined with tin, and bound with wire. The outside of the conduit is coated with asphaltum. It is not suitable for high temperatures or poorly drained soils.

In Fig. 1 (B), (C), (D), (H) and (I) are patented forms of conduits. The insulation is sometimes a loose filler packed into the conduit. Conduits (H) and (I) are prefabricated. Both of these conduits are enclosed in metal jackets.

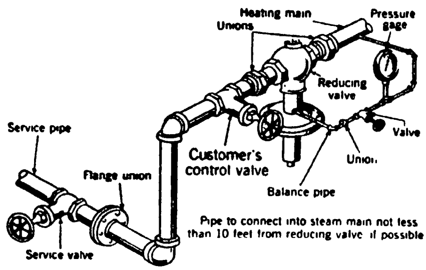


FIG. 2. CONNECTIONS FOR REDUCING VALVE WITHOUT BY-PASS

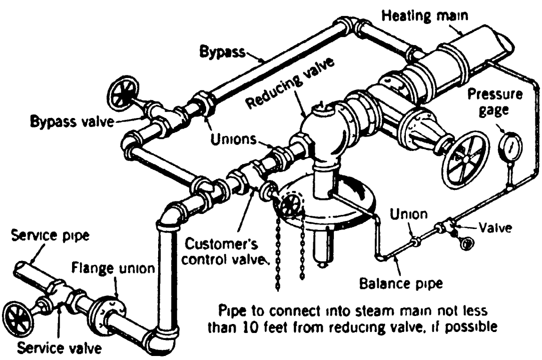


FIG. 3. CONNECTIONS FOR REDUCING VALVE WITH BY-PASS

At (C) and (E) are shown two tile conduits using sectional insulation. In these particular designs the space surrounding the pipe is filled partially or wholly with a loose insulating material. The addition of this loose insulating material to the sectional insulation is, of course, optional and is justified only where high pressure steam is used.

(E) and (F) are conduits used by two district heating companies, and have the advantage of being constructed of common materials.

Conduit (G) is of cast-iron construction, assembled with lead joints and is water-tight, if properly laid. It is obviously expensive and is justified only in exceptional cases.

There are, in addition to those mentioned, several conduits which use an insulating concrete as a pipe insulation. The insulating effect of the concrete is obtained by admixture of an insulating material with cement.

PIPE TUNNELS

Where steam heating lines are installed in tunnels large enough to provide walking space, the pipes are supported by means of hangers or roller frames on brackets or frame racks at the side or sides of the tunnel. The pipes are insulated with sectional pipe insulation over which is placed a sewed-on, painted canvas jacket or a jacket of asphalt-saturated asbestos water-proofing felt. The tunnel itself is usually built of concrete or brick, and water-proofed on the outside with membrane water-proofing.

Because of their relatively high first cost as compared with smaller conduits, walking tunnels are sometimes omitted along heating lines, unless they are required to accommodate miscellaneous other services, or provide underground passage between buildings.

OVERHEAD DISTRIBUTION

In some industrial and institutional applications, the distribution piping may be installed, entirely or in part, above ground. This method of con-

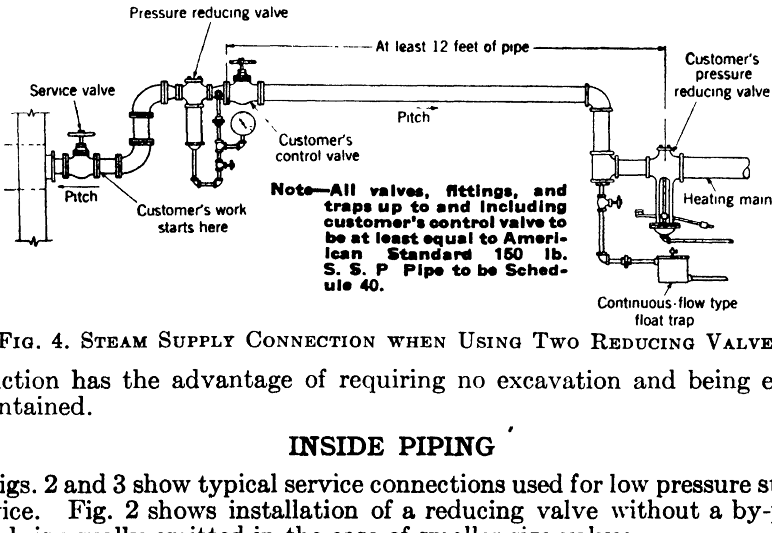


FIG. 4. STEAM SUPPLY CONNECTION WHEN USING TWO REDUCING VALVES

struction has the advantage of requiring no excavation and being easily maintained.

INSIDE PIPING

Figs. 2 and 3 show typical service connections used for low pressure steam service. Fig. 2 shows installation of a reducing valve without a by-pass, which is usually omitted in the case of smaller size valves.

Fig. 3 illustrates the use of a reducing valve, with a by-pass which is generally provided for larger installations. This latter construction permits the operation of the line in case of failure in the reducing valve. In the smaller sizes, the reducing valve can be removed, a filler installed, and the house valve used to throttle the flow of steam until repairs are made.

Fig. 4 shows a typical installation used for high pressure steam service.[1] The first reducing valve effects the initial pressure reduction. The second reducing valve reduces the steam pressure to that required.

In a heating system the pipes carrying condensate are more subject to corrosion than other parts of the system. Care must be taken to give proper pitch to the pipes and provide proper venting of non-condensable gases. (See Chapter 42, Corrosion).

Most district heating companies enforce certain regulations regarding the consumer's installation, partly to safeguard their own interests, but

[1] Code for Pressure Piping, B31-1, 1942, *American Standards A ociation*, Paragraph 408, p. 115.

principally to insure satisfactory and economical service to the consumer. There are certain fundamental principles that should be followed in the design of a building heating system which is to be supplied from street mains. Although some of these apply to any building, they have been demonstrated to be especially important when steam is purchased.

1. Provision should be made for conveniently shutting off the steam supply at night and at other times when heat is not needed.

It has been thoroughly demonstrated that a considerable amount of heat can be saved by shutting off steam at night. Although there is, in some cases, an increased consumption of heat when steam is again turned on in the morning, there is a large net saving which may be explained by the fact that the lower inside temperature maintained during the night obviously results in lower heat loss from the building, and less heat need therefore be supplied.

Steam can be entirely shut off at night in most buildings, even in very cold weather, without endangering plumbing. It is necessary, however, to have an ample amount

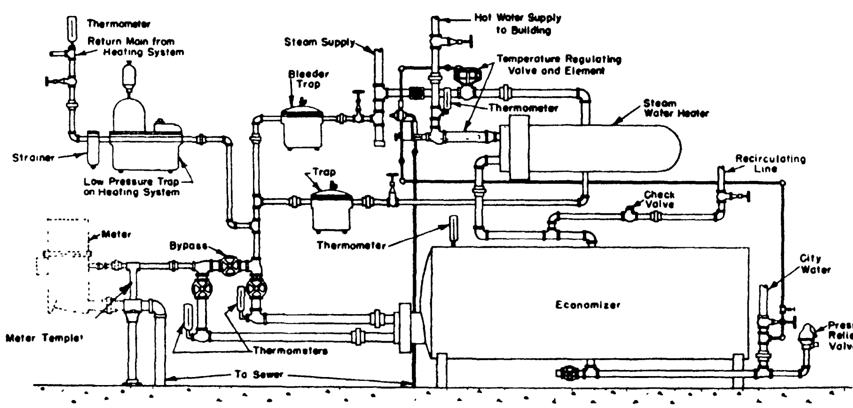


FIG. 5. METHOD OF INSTALLING A WATER HEATER AND ECONOMIZER IN A GRAVITY HEATING SYSTEM

of heating surface so that the building can be quickly warmed in the morning. Where the hours of occupancy differ in various parts of the building, it is good practice to install separate supply pipes to the different parts. For example, in an office building with stores or restaurants on the first floor which are open in the evening, a separate main supplying the first floor will permit the steam to be shut off from the remainder of the building in the late afternoon. The division of the building into zones, each with a separately controlled heat supply, is sometimes desirable, as it permits the heat to be adjusted according to variations in sunshine and wind.

2. Residual heat in the condensate should be salvaged.

This heat may be salvaged by means of a cooling coil, or as is more frequently done, by a water heating economizer (see Fig. 5) which preheats the hot water supply to the building.

The condensate from the heating system, after leaving the trap, passes through the economizer. The supply to the hot water heater passes through the economizer, absorbing heat from the condensate. If the hot water system in the building is of the recirculating type, the recirculating connection should be tied in *between* the economizer and the water heater proper, not at the economizer inlet, because the recirculated hot water is itself at a high temperature.

Because of the lack of coincidence between the heating system load and the hot water demand, a greater amount of heat can be extracted from the condensate if storage capacity is provided for the preheated water. Frequently, a type of economizer is used in which the coils are submerged in a storage tank.

3. Heat supply should be graduated according to variations in the outside temperature. The maximum in economical operation and satisfactory heating can only be obtained by the use of automatic temperature control (See Chapter 38).

METERING

The perfection of fluid meters has contributed as much to the advancement of district heating as any other one thing. Meters are classified into two groups: *Condensate Meters* and *Rate of Flow Meters.*

Condensate Meters

The one type of quantity meter used is the condensate meter, which may be of the *tilting bucket* or *revolving drum* type.

The condensate meter is a popular type for use on small and medium

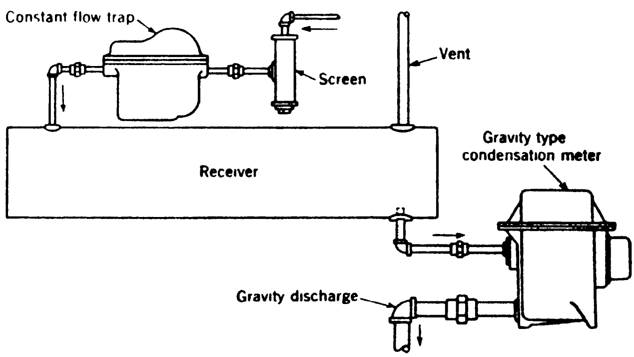


FIG. 6. GRAVITY INSTALLATION FOR CONDENSATION METER USING VENTED RECEIVERS

sized installations, where all the condensate can be brought to a common point for metering purposes. Its simplicity of design, ease in testing, accuracy at all loads, low cost, and adaptability to low pressure distribution have made it standard equipment with many heating companies.

Condensate meters should not be operated under pressure; they are made for either gravity or vacuum installations. Where bucket traps are used, a vented receiver is essential ahead of the meter. Where continuous flow traps are used, a vented receiver is not necessary, but is desirable. Fig. 6 illustrates a gravity condensate meter installation using a vented receiver.

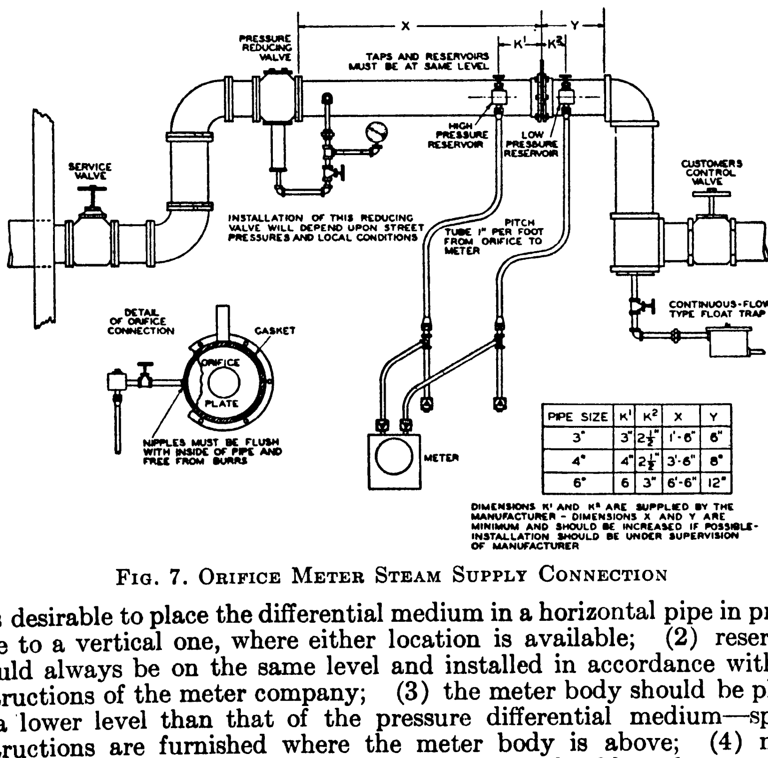
Rate of Flow or Flow Meters

Flow meters used for district heating work are of three types: *Area Meters*, *Head Meters* and *Velocity Meters*. (See Chapter 4, Fluid Flow.)

Area meters are those in the operation of which a variation in the cross-section of stream, under constant head, is used as an indication of the rate of flow. A tapered plug is suspended in an orifice and moves axially with the flow, which is vertically upward. The weight of the plug provides a definite pressure differential, and the plug floats at such a height as will provide enough orifice area to pass the flow at the pressure difference. The movement of the plug is transmitted by means of a lever to a pencil or marker which records the flow on a graduated strip chart.

Head meters are those in which the stream of fluid creates a difference of pressure, or differential head. This head is created by an orifice, Venturi tube, flow nozzle, or Pitot tube, and will depend upon the velocity and density of the fluid. The secondary element must contain a differential pressure gage, which will translate the pressure difference into rate of flow or total flow. This mechanism may be either mechanical or electrical. The electric flow meter has the advantage of being able to locate the instruments at some distance from the primary element.

Fig. 7 is a typical example of an orifice-type meter installation. A few general points to be considered in installing a meter of this type are: (1)



PIPE SIZE	K^1	K^2	X	Y
3"	3"	$2\frac{1}{2}$"	1'-6"	6"
4"	4"	$2\frac{1}{2}$"	3'-6"	8"
6"	6	3"	6'-6"	12"

Fig. 7. Orifice Meter Steam Supply Connection

it is desirable to place the differential medium in a horizontal pipe in preference to a vertical one, where either location is available; (2) reservoirs should always be on the same level and installed in accordance with the instructions of the meter company; (3) the meter body should be placed at a lower level than that of the pressure differential medium—special instructions are furnished where the meter body is above; (4) meter piping should be kept free from leaks; (5) sludge should not be permitted to collect in the meter body; (6) the meter body and meter piping should be kept above freezing temperatures; (7) it is best not to connect a meter body to more than one service; (8) special instructions are furnished for metering a turbulent or pulsating flow.

Velocity meters are those in which the primary element is some device that is kept in continual rotation by the linear motion of the stream. The secondary element is, essentially, a revolution counter. The primary and secondary elements are combined into one unit.

For steam metering, the shunt meter is an example of the velocity type. This unit is connected directly in 2, 3 and 4 in. pipe lines. Larger size mains are metered by installing a 2 in. meter in a by-pass with a restricting orifice in the main line.

Selection of Meter

In selecting a meter for a particular installation, the number of different makes and types of meters suitable for the job is usually limited by one or more of the following considerations: (1) its use in a new or an old installation; (2) method to be used in charging for the service; (3) location of the meter; (4) large or small quantity to be measured; (5) temporary or permanent installation; (6) cleanliness of the fluid to be measured; (7) temperature of the fluid to be measured; (8) accuracy expected; (9) nature of flow, *i.e.*, turbulent, pulsating, or steady; (10) cost: (*a*) purchase price, (*b*) installation cost, (*c*) calibration cost, (*d*) maintenance cost; (11) servicing facilities of the manufacturer; (12) pressure at which fluid is to be metered; (13) type of record desired as to indicating, recording or totalizing; (14) stocking of repair parts; (15) use of open jets where steam is to be metered; (16) metering to be done by one meter or by a combination of meters; (17) use as a check meter; (18) its facilities for determining or recording information other than flow; (19) whether or not the condensate can be returned to a central point.

STEAM REQUIREMENTS

Methods of estimating steam requirements for heating various types of buildings are given in Chapter 17.

Table 7 in Chapter 17 represents information obtained from all sections of the United States, and the group of buildings from which the information was taken represents a cross-section of all types of heating systems.

Steam requirements for water heating can be satisfactorily estimated by using a consumption of 0.0025 lb per (day) (cu ft of heated space) for office buildings, without restaurants, and 0.0065 lb per (day) (cu ft of heated space) for apartment buildings.

Complete information on water heating requirements is given in Chapter 48.

Additional data on steam requirements of various types of buildings in a number of cities may be found in the *Handbook of the National District Heating Association.*

RATES

Fundamentally, district heating rates are based upon the same principles as those recognized in the electric light and power industry, the main object being a reasonable return on the investment. However, there are other requirements to be met: the rate for each class of service should be based upon the cost to the utility company of the service supplied, and upon the value of the service to the consumer, and it must be between these two limits. District heating rates should be designed to produce a sufficient return on the investment regardless of weather conditions, although existing rate schedules do not always conform to this principle. Lastly, the rate schedule must be reasonably simple and understandable.

Glossary of Rate Terms

Load Factor. The ratio, in percent, of the average hourly load to the maximum hourly load. This is usually based on a one-year period, but may be applied to any specified period.

Demand Factor. The relation between the connected radiator surface, or required radiator surface, and the demand of the particular installation. It varies from 0.25 to 0.3 lb per (hr) (sq ft of surface).

Diversity Factor. The ratio of the sum of the individual demands of a number of buildings to the actual composite demand of the group.

Types of Rates

The various types of rates to be found in use in district heating systems are:

1. *Straight-Line Meter Rate*. The price charged per unit is constant, and the consumer pays in direct proportion to his consumption without considering the difference in costs of supplying the individual customers.

2. *Block Meter Rate*. The pounds of steam consumed by a customer are divided into blocks of thousands of pounds each, and lower rates are charged for each successive block consumed. This type of charge predominates in steam heating rate schedules, having the advantage of proportioning the bill according to the consumption and the cost of service. It has the disadvantage of not discriminating between customers having a high load factor (relatively low demand), and those having a low oad factor (relatively high demand). The utility company must maintain sufficient capacity to serve the high demand customers, and the cost of the increased plant investment is divided equally among the users, so the high demand customers are benefited at the expense of the others.

3. *Demand Rates*. These refer to any method of charge based on a measured maximum load during a specified period of time.

The *flat demand rate* is usually expressed in dollars per thousand pounds of demand per month or per annum. It is based on the size of a customer's installation, and is seldom used except where a meter is not practicable.

The *Wright demand rate* is similar in calculation to the block rate, except that it is expressed in terms of hours' use of the maximum demand. It is seldom used, but forms the basis for other forms of rates.

The *Hopkinson demand rate* is divided into two elements:

(*a*) A charge based upon the demand, either estimated or measured.

(*b*) A charge based upon the amount of steam consumed.

This rate may be modified by dividing the quantities of steam demanded and consumed into blocks charged for at different rates.

The *Doherty rate* is divided into three elements:

(*a*) A charge based upon demand.

(*b*) A charge based upon steam consumed.

(*c*) A customer charge.

In the *Hopkinson rate*, the last two elements are combined into one element.

Demand rates are comparatively new and are not yet widely used. While they are equitable and competitive, they are difficult for the average layman to understand. They are of benefit to utility companies and to consumers because the investment and operating costs can be divided, to suit the particular circumstances, into demand, customer, and consumption groups through the use of some modification of the Hopkinson rate. Demand rates are an advantage to the customer in that the use of such a rate reduces the rate per thousand pounds to the long-hour user.

Fuel Price Surcharge. It is usually desirable to establish a rate upon a specified basic cost of fuel to the utility company. Where there are wide variations in the price of fuel, it is also desirable to add a definite charge per thousand pounds of steam sold for each increment of increase in the price of fuel. This surcharge automatically compensates for the variations without necessitating frequent changing of the whole rate structure.

Some utility companies include a labor surcharge as well as a coal surcharge.

CHAPTER 29

CENTRAL SYSTEMS FOR AIR CONDITIONING

Features of Systems, Zoning, Apparatus Dew-Point, Cooling Load, Heating Load, Air Quantity and Effectual Temperature Difference, Low and High Pressure Induction Convectors, Evaporative Cooling, Precooling, Sensible Cooling with Unwetted Coils, Run-Around System, Selection of Type of System, Location of Apparatus, Design Procedure

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THE term, central, applied to an air conditioning system implies that the equipment such as fans, coils, filters and their encasement are designed for assembly in the field rather than in a factory as a unit. As a central system usually serves several different rooms, individual controls are required for each room.

FEATURES OF CENTRAL SYSTEMS

One advantage of a central air supply system is that one apparatus serving many rooms may involve a lower investment cost than that for a number of self-contained plants, each serving a single room. A central system may occupy basement or attic space that is relatively unimportant, whereas individual factory-assembled apparatus placed in each room may occupy otherwise valuable space. Another advantage of a central system is accessibility for servicing, since it is possible to provide doors in the encasement for cleaning and inspecting all of the component parts in a manner usually superior to that practicable with compact factory-assembled equipment.

Central air conditioning systems usually are connected by ducts with the various rooms served, and preferably have exhaust fans that may effect complete removal and disposal of any desired proportion of the air. The exhaust fan may return air to the supply system for recirculation, as a measure of economy of fuel or refrigeration.

Central air conditioning systems are served by heating and refrigerating equipment which may be located at some distance from the air supply apparatus, and which may serve one or more central air supply systems.

Year-Round Air Supply System

Fig. 1 is a plan of a year-round air supply system. Outside air may enter from the left at A, desirably from an intake on the side of the building least exposed to solar heat, and not close to the ground or to a sun-heated or dust-gathering roof. The damper B for proportioning the volume of outside air, is interlocked with the return air damper C in such manner that as the outside air volume increases the return air volume decreases. The return air duct D, shown diagrammatically, comes from the exhaust fan. All the air, it will be observed, must pass through the filters E, and there is ample room on both the inlet and outlet sides of the filters for servicing them. The filters may be of mechanically cleaned type, of replaceable cell type, or may be electronic, as described in more detail in Chapter 33.

The cleaned air passes to the equipment that changes its temperature and humidity. Except in very warm climates, a heating or tempering coil F is required to warm the air to a temperature above freezing. Usually, the heat is supplied by means of hot water or steam. During many hours of most days it is practicable to recirculate enough of the air so that the air drawn from outside, after mixing with the relatively warm return air, will not be cold enough to freeze the water in the humidifier.

Upon leaving the tempering coil, the air enters the humidifier G. This may be a spray of warmed water, circulated by a small pump from a water tank under the spray chamber, or may be other means of supplying water vapor. The supply of moisture must be under automatic and very reliable control. Following the humidifier a heater I is required, for controlling the temperature of the air entering the supply fan.

The second group of heat transfer devices in a year-round system includes an air cooling component H, for use in warm weather. Its surface may be chilled by direct expansion of an approved refrigerant within

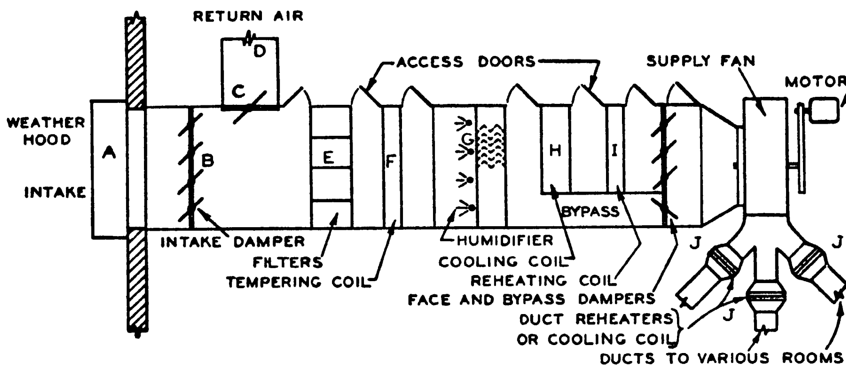

Fig. 1. Arrangement of Equipment for Year-Round Air Supply System

its tubes, or the surface may be cooled by a pump-circulated liquid such as water or brine. The coil must be sufficiently cold to cool the summer air to a temperature below the existing dew-point, and may be expected to be wetted constantly by the moisture condensed from the air. A water-tight drainage tank must be installed under the cooling coil and should extend for a distance toward the fan. Water should be drained by means of a trapped waste through a vented air-break. The second group of heat transfer devices also includes an air-heating component (reheater) similar to the tempering coil and capable of warming the chilled, saturated air leaving the cooling surface, to a temperature sufficiently high to prevent complaint of drafts when the air is delivered into the rooms.

ZONING AND ZONE CONTROL

It is apparent that while an apparatus like that of Fig. 1 would be very desirable for any single room, since in that case the air could be delivered at optimum conditions, the cost of a complete individual system for each room and the space required for the equipment generally would be prohibitive. Economy is favored if the varying requirements of numerous rooms or zones can be simultaneously satisfied by air from a single central supply system.

Various methods are practicable for controlling the temperature, humidity and air movement in various rooms or zones. A measure of control is attainable merely by proportioning the flow of air to each room, though usually such control by throttling dampers is difficult to maintain and should be avoided when possible.

Another scheme is to install a properly proportioned coil in the branch air supply duct serving each room to warm the air to suit the occupants. The air, for example, leaving the fan that serves several rooms, may be cooled before entering the fan, to the condition favorable for one room, and the air for each other room may be reheated by the branch duct coil to the required temperature. It is also possible to circulate a heat absorbing medium in the branch duct coils to reduce the temperature of the air passing to rooms that would be overheated if they received air at the condition leaving the central air supply system. In Fig. 1 such coils J are indicated in the three branch ducts leaving the supply fan. When

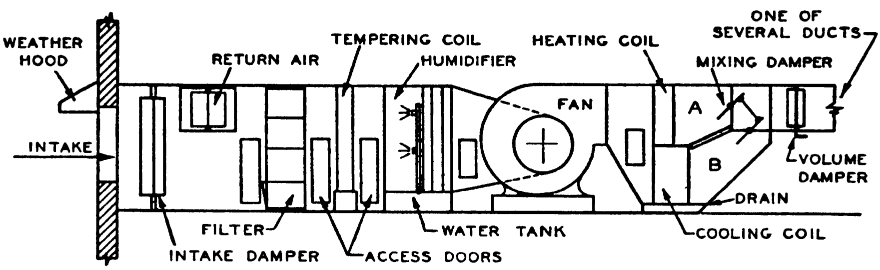

FIG. 2. ALTERNATE ARRANGEMENT OF EQUIPMENT FOR CONTROLLING AIR CONDITION IN CENTRAL AIR SUPPLY SYSTEM

heat transfer devices are placed in branch ducts for improved temperature control, mechanically circulated water gives excellent results as a heat carrier. The water usually is warmer than the air but it is possible to use water colder than the air.

It is practicable also to use single central air conditioning equipment similar to that shown in Fig. 1, in conjunction with several fans, one for each room or zone. In such cases there may be a separate reheater on the suction side of each relatively small supply fan.

The designer must remember that the various supply fans will compete with each other for air, against the resistance interposed by the filters, coils, etc., that are used in common under such circumstances, and consequently, unless the fans are of backward-curved blade, non-overloading type, they may alternate in carrying more than their share of the air, and thereby cause the air distribution to be chaotic and unsatisfactory.

Another method of controlling temperature in various rooms served by a central air supply system is shown in the sectional elevation, Fig. 2. The supply fan is placed immediately after the humidifier. When cooling the air in hot weather, the humidifier is not operated. The fan will deliver the air through the heating coil and through the cooling coil to the two air pressure chambers A and B at the right of these coils. From these chambers many separate ducts, one of which is shown, each with a double-blade mixing damper, may convey the air to the various rooms. The mixing dampers, one of which is shown, are interlocked so that as the upper one closes, the lower one opens; selecting between them, air in the required quantity from either the warmer chamber A or the cooler one B. In cold weather no refrigerant is required in the cooling coil, and in hot weather no heating medium is circulated in the heating coil. With this scheme, the control of relative humidity in warm weather is not always sufficiently precise to meet requirements, since the untreated air delivered through the upper coil may be so high in relative humidity that it cannot sufficiently compensate for the nearly saturated air leaving the lower coil. A reheater could be placed if desired, to the right of the lower coil to bring the air in the lower chamber to the desired relative humidity. The simple arrangement of Fig. 2 is admirable in winter and, except where close control of relative humidity is important, may be acceptable in summer.

Another method of attaining temperature control in individual rooms with a year-round central air supply system, is to install a booster fan

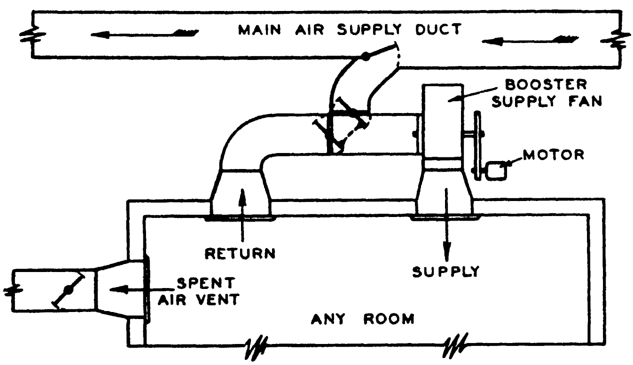

FIG. 3. ARRANGEMENT FOR INDIVIDUAL ROOM TEMPERATURE CONTROL WITH CENTRAL AIR SUPPLY SYSTEM

between the main air supply duct and the air delivery opening to each zone or room, as shown in Fig. 3. Air can then be delivered from the central supply fan through the main duct at some desired condition, for instance, 60 F, 45 percent relative humidity. A double mixing damper near the intake opening of the booster fan, controlled by a thermostat in the room or zone that is served by the fan, is interlocked with an outlet exhaust damper in the spent air opening, so that as more of the room air is recirculated, and as less new air from the main air supply duct is delivered into the room, the spent air outlet is throttled in proportion. In many large installations this principle is applied successfully for zoning different stories in multi-story office buildings, the main supply fan being on the roof, and each booster fan used for supplying the rooms of one orientation of each story. In other cases the booster fans serve only single offices, and therefore are small enough to be concealed above ceilings alongside the main supply duct.

There may be installations in which the use of recirculated air for mixing with new refrigerated and nearly saturated air to control temperature and relative humidity is objectionable. In such cases the general recirculation arrangements of Fig. 1 may be omitted, and heat transfer coils located in the ducts may be used. In some cases where general recirculation is not acceptable, as for all the rooms in an entire building, use of the local circulation of Fig. 3 may solve the problem.

APPARATUS DEW-POINT

In ordinary practice, with commercial apparatus, complete saturation of the air is seldom obtained. Four-row finned cooling coils contact approximately 80 percent of the air, whereas six-row finned coils contact approximately 95 percent of the air. In spray type dehumidifiers of good design the air leaves the dehumidifier at 1 to 2 deg higher wet-bulb temperature than the spray water leaving the dehumidifier, and the difference between the dry-bulb and wet-bulb temperatures leaving the dehumidifier may be as low as 1 deg. A spray type dehumidifier having sufficient length of spray chamber and density of spray, together with proper arrangement of nozzles, may approach saturation very closely.

As explained in Chapter 3, the slope of the line on the psychrometric chart connecting the room condition with the apparatus dew-point on the saturation line, determines the ratio of sensible heat absorbing capacity to the moisture absorbing capacity of the supply air. Therefore the room condition can be maintained as long as the supply air temperature lies on this line, but a greater volume of supply air must be used to satisfy the room load if the cooling coil does not contact 100 percent of the air. For a given room load, the same apparatus dew-point will be required whether the cooling appliance contacts all the air or only part of the air.

From the point of view of satisfying the given cooling load requirements, the air passing through the apparatus without being cooled below the dew-point temperature produces two effects:

1. The air quantity which must be passed through the dehumidifier must be increased. Thus, if 20 percent of the air passing is contacted, then $(20 \div 80) \times 100 = 25$ percent more air must be used than would be necessary if all of it were contacted.

2. Passing untreated air may change the room cooling load, which in turn may change the sensible heat factor. If return air only is passed through the dehumidifier or if room air only is by-passed, the room load will not change, but if some outside air is passed through, the room sensible heat gain and room latent heat gain will be changed due to the addition of untreated outside air, which changes the sensible heat factor. When a load calculation is made, it is necessary to know the percentage of air affected in the dehumidifier, and calculation must be made accordingly.

If the ventilation air is drawn through the dehumidifier before it goes into the room, only that portion of the air not saturated must be included in the room load for the purpose of determining the apparatus dew-point and supply air quantity. It should be noted when evaluating the load added by untreated outside air that the temperature difference between room air and outside air, and the moisture content difference between room air and outside air, should be used, rather than the difference between outside air and apparatus dew-point, since the rise from the apparatus dew-point to room condition is charged against the dehumidifier as the cooling and dehumidifying load.

In winter, room relative humidities in excess of 30 percent are seldom required in a system designed for comfort conditioning only, and a low saturating efficiency may be desirable, or even necessary, especially if the same volume of air is handled as in summer. With a spray type dehumidifier the main sprays may be shut off and only the eliminators need be flooded; which may give sufficient moisture. In other cases, such as those in which cooling coils are sprayed, the spray water supply may be throttled. If the saturation efficiency of the sprays is too low, the spray water may be heated. The amount of heat put into the spray water by open or closed water heaters will be equal to that required to bring the dew-point temperature of the air entering the sprays up to that required before entering the preheater. It is possible, where clean steam is available, to introduce steam directly into the air stream to produce the desired dew-point temperature of supply air. However, the steam must be exceptionally clean, or objectionable odors will result.

It should be noted that the quantity of outdoor air to be introduced is affected by infiltration and leakage. Infiltration will reduce the quantity to be introduced by the system, while leakage may have to be offset by an increase in the quantity of outdoor air.

COOLING LOAD

The method of determining the cooling load for a conditioned space or spaces is outlined in Chapter 12. As pointed out therein, many of the items of heat gain are variable and do not reach their maximum values simultaneously. Proper consideration of these peaks and the avoidance of pyramiding these peaks in the cooling load calculations are stressed. Maximum solar heat gain on an east exposure is seldom coincident with the maximum outdoor wet-bulb.

A large difference in the time-incidence of the peaks between various spaces or parts of the same space indicates the necessity for zoning. In a building having an east and west exposure, where solar heat gain forms a fair share of the cooling load, the times of individual zone peaks are apt to be some hours apart, and the peak load of one plus the off-peak load of the other will be substantially less than their combined peak loads. Proper zoning will permit operation to take full advantage of this condition or of similar conditions of non-simultaneous peaks, and will result in a lower total load and in savings in equipment.

A factor, similar in effect and closely related to the non-simultaneous occurrence of peak loads, is diversity. Typical of this is the case of a large department store where the air handling equipment serving a certain space must be sufficient to handle the load created by the throngs of people attending sales in that space. Under such a condition the number of people in other spaces is usually normal or below. While this means that the air handling equipment for certain departments must be large enough to cope with the situation, the refrigeration equipment need be only large enough to handle the average maximum. If a system employing zone recirculating fans and a single central fan and dehumidifier were used, the saving would be reflected in the capacity of the central fan and dehumidifier. Another example of this diversity is found in an office building having restaurants and stores of certain types in the first story and basement. At noon, when the restaurants and stores are crowded, the offices are below normal occupancy.

Heat lag should be carefully considered in the cooling load calculations. In certain types of buildings the effect of solar radiation is still apparent several hours after the sun has shifted from that exposure. In other types having a much lighter construction, the heat gain due to solar radiation decreases markedly with the passing of the sun. Some walls, having been warmed by the sun, may radiate heat long after the passing of the sun, thus requiring lower inside temperatures to offset the radiant energy.

Buildings have considerable heat storage capacity which can often be utilized to great advantage, and which has more than once provided an unexpected safety factor. If a space is kept below the design inside temperature for some time, the interior walls, floors, furniture and fixtures begin to assume the temperature of the space. Where the time is sufficient the entire mass, rather than merely its surface, may reach the room temperature. Thus, when a space has been precooled below the design maximum temperature for a period of time prior to the advent of the peak load, and the heat gain begins to increase the peak conditions, some of the increase is used in raising the temperature of the furniture, fixtures, etc., to the design conditions and the cooling load can be reduced accordingly. However, unless very accurate data with regard to the mass, surface, specific heat, etc., of the items within the space are available, due caution must be used in discounting the cooling load for this storage effect. In the absence of reliable data this allowance is often a matter of experience rather than calculation.

Where air conditioning supply and return ducts pass through unconditioned spaces, there will be a transfer of heat from these spaces to the air in the ducts, even though these ducts are well insulated. An allowance should be made for this heat gain and included in the heat estimate so that air can be supplied at a temperature low enough to offset the rise caused by this heat gain (see Chapter 31). There will also be some heat gain to the air in ducts passing through conditioned spaces, but since a cooling effect is produced in the space through which the duct passes, this is not a loss and usually can be compensated for by adjustment of air quantities between the various spaces.

HEATING LOAD

Methods of calculating the heating load are shown in Chapter 11. Many of the factors outlined previously under Cooling Load, such as zoning, non-simultaneous peaks, and diversity, apply in the reverse manner due to the heating requirements instead of the cooling requirements. However, these factors affect the heating load from the standpoint of control of inside conditions, overall performance, and economy of operation more than from a capacity of equipment standpoint. It is not only necessary to heat a building or space to its design conditions when there is but the merest fraction of normal occupancy, and when there are practically no lights, internal heat, or solar radiation, but it is also necessary to provide capacity to heat the building quickly when sudden cold follows relatively warm weather, as may occur after a week-end or holiday shutdown. However, in normal operation during week-ends and holidays, buildings are usually kept at a holding temperature to prevent the freezing of services. In many cases, less fuel is required to continue operation of the heating plant at a near-normal rate and maintain the building or space at a temperature of 50 to 65 F at such times, than to shut the system down and then bring the temperature back to normal through forced operation of the heat generating equipment with a consequent loss in efficiency.

AIR QUANTITY AND EFFECTUAL TEMPERATURE DIFFERENCE

The difference between the room air temperature and the supply air temperature at the outlet to the room is known as the effectual temperature difference. In the theoretical case of a dehumidifier having 100 percent saturating efficiency, and where this air is delivered directly to the room without temperature increases due to heat gain, then the effectual temperature difference is the difference between room temperature and apparatus dew-point temperature. If duct heat gains are considered a part of the room load, this still holds true. The apparatus dew-point, as outlined previously, is fixed by the latent and sensible loads of the space, but in many cases, it is desirable to deliver more air to the spaces than is indicated by the difference between the room temperature and the apparatus dew-point.

It has been indicated that where a percentage of air is passed through the dehumidifier without being treated, the relationship is modified in direct proportion, and that if room air is passed through untreated, no effect on the heat balance results. Similarly, if room air is passed around the dehumidifier and mixed with the treated air, the heat balance is not adversely affected. Therefore, if the quantity of air passed through the dehumidifier is determined by the usual methods, room air can be passed around the dehumidifier and mixed with the dehumidified air, increasing the supply air quantity and temperature and decreasing the effectual temperature difference. Thus if the difference between the room temperature and the apparatus dew-point indicates that 10,000 cfm at 30 deg below room temperature will be required to hold conditions, that quantity can be passed through the dehumidifier and cooled to 30 deg below the room temperature, then mixed with 10,000 cfm of room air, resulting in a supply air quantity of 20,000 cfm and an effectual temperature difference of 15 deg instead of 30 deg. Air supply outlets and grilles having a high induction ratio are available, and through their induction effect cause a large amount of room air to be mixed with the supply air within a short distance of the grille. A proper selection of outlets may make it possible to introduce air at low temperatures and high velocities without causing objectionable drafts or cold spots, but care must be used to see that too little air motion is not a result. Low effectual temperature differences may be required for this reason. While the use of a high effectual temperature difference results in a saving in initial cost of fans and ducts, and in the operating cost of fans, this difference should be carefully considered. If the sensible heat load of a space is subjected to substantial variations, low effectual temperature differences should be considered, since systems employing low effectual temperature differences require less precision in controls.

Reduction of air quantity by slowing down the fans for the winter season, and increasing the temperature difference, often is feasible. A saving in fan power can thus be effected, provided the air distribution remains adequate.

Extremes should be avoided in all cases. For summer air conditioning, low supply air temperatures result in larger heat gains to the air passing through the ducts, as well as in poor control. Too high a supply air temperature may result in excessive initial and operating costs. Suggested limits for the effectual temperature difference are from 12 to 20 deg, the actual selection being based on the requirements of the particular case. For winter air conditioning, too high supply air temperatures result in excessive heat losses from the ducts and stratification within the room unless thorough mixing is assured, while too low supply air temperatures may cause drafts, high operating costs, etc. Suggested limits are from

15 to 35 deg. There can be no set rule, and each case should be judged according to its particular requirements of the installation.

Reference may be made to Chapter 30 for further discussion of the most satisfactory design difference between the entering air temperature and volume in relation to the desired room condition.

INDUCTION CONVECTORS—LOW PRESSURE TYPE

Induction convectors located in the room that is to be served, utilize a jet of primary conditioned air to mix with a stream of secondary room air as shown in Fig. 4. The mixture is discharged into the room through a grille at the top of the convector. Heating coils are located in the secondary air stream. The output is controlled either by manually or automatically throttling the air jet. Heat may be supplied to the coil in summer as well as in winter. These induction convectors present several

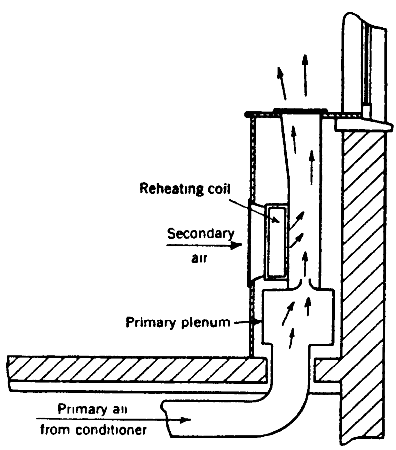

FIG. 4. INDUCTION UNIT (LOW PRESSURE TYPE)

advantages. Since the secondary air stream is thoroughly mixed with the high velocity low temperature air stream before leaving the discharge outlet of the device, the resultant temperature of the mixture is satisfactory even though the primary air is introduced at a temperature too low for ordinary methods of distribution. One of these devices usually is provided under each window in place of the customary direct radiator, and combines the air distribution system with the heating system. An air conditioning system without induction convectors may require installation of direct radiation for maintenance of minimum temperatures during air conditioning shut-down periods, but when induction convectors are used they may be selected with heating coils of sufficient capacity to maintain, by thermal circulation, a reasonable temperature when the primary air supply system is shut off. The use of low temperature, dehumidified air which has not been reheated or mixed with room air before delivery to the room, may permit a reduction in fan capacity and the use of smaller ducts. In some cases a by-pass may be desirable in order to maintain the primary air volume and to provide additional control. This system can provide a degree of zoning that is usually difficult with conventional design since the air delivered by each unit can be controlled individually.

Selection of induction convectors should be made with due regard to noise level. The inductive capacity of the device increases with the jet velocity, but high jet velocities may result in objectionable noise.

INDUCTION CONVECTORS—HIGH PRESSURE TYPE

Another type of induction convector, Fig. 5, employs nozzles which produce a high velocity air jet without objectionable noise. The term, high pressure, is to some extent inaccurate, since the air pressure at the nozzles, while several times that used with a low pressure induction convector, is still less than the total resistance pressure of a conventional central system. The high velocity jet of primary air induces a flow of air rom the room through coils located in the secondary air stream and

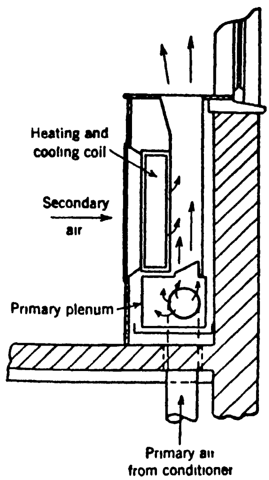

Fig. 5. Induction Unit (High Pressure Type)

supplied with chilled water in summer and with hot water in winter. The chilled water removes a large portion of the sensible heat in summer and the hot water supplies the sensible heat loss in winter. The primary air is delivered at a sufficiently low dew-point to compensate for the latent heat gain in summer. In winter the primary air is supplied at a sufficiently high dew-point to take care of latent heat losses. Control of temperature is obtained by throttling the water quantity supplied to the secondary coil. The required flow of primary air is greatly reduced due to the fact that a portion of the sensible heat load is carried by the secondary air stream. Since the primary quantity is small, very high velocities can be maintained in the supply ducts without requiring fan power in excess of that for a conventional system. Therefore, the supply ducts or pipes can be very small and can be run in chases, or furred in at columns along with the water pipes. The primary air is treated in the usual manner to reach the required dew-point and a surface or spray dehumidifier or a dehydrator may be used. The primary air quantity is sufficient for ventilation purposes and frequently consists entirely of outdoor air. The water piping for the coils can be so valved that hot water will be supplied to one zone that may require heating, while cold water may be supplied at the same time to a zone that requires cooling.

This system usually is limited in application to hotels, apartments, office buildings and other multi-room installations having a large perimeter with relation to the floor area. The units usually are installed beneath the windows, replacing direct radiation or thermally-circulating enclosed convectors. Where the spaces to be conditioned extend a considerable distance from the outer wall into the interior of the building, a separate system or zone for the conditioning of the interior portions may be required.

EVAPORATIVE COOLING

In climates where, on the hottest days, the outdoor wet-bulb depression is relatively great, it may be possible to replace mechanical refrigeration, or other cooling sources, and use the evaporative cooling effect. A well designed air washer using recirculating sprays will reduce the entering dry-bulb temperature to within a degree or two of the entering wet-bulb condition. Thus, it may be possible that, with air entering at 100 F dry-bulb, 60 F wet-bulb, a leaving condition of 62 F dry-bulb, nearly saturated, can be obtained. Under some conditions of latent and sensible heat load, this evaporative cooling may be adequate.

At times when the outdoor wet-bulb temperature is not low enough to permit the use of straight evaporative cooling, it is still possible to use pre-cooling convectors with refrigeration, well water, or a cooling tower, as the basic source of sensible heat removal to reduce the wet-bulb temperature of the air before it enters the spray chamber. Where internal heat loads are high, this scheme may be more economical than one using return air. Where the required supply air dew-point is too low to permit straight evaporative cooling, and where the sensible heat load is not too great, intentional partial saturation may be employed. That is, the low dew-point of the outdoor air is utilized by permitting some of this air to pass through the humidifying sprays untreated, or to by-pass the humidifier. All of these remarks with regard to evaporative cooling are based on the assumption that all of the supply air will be taken from outside. Provision should be made in most cases for the return of some air from the conditioned spaces for control purposes, as well as for economy of fuel in winter.

PRECOOLING

Where sufficiently cold water from wells or streams is available, a saving in refrigeration may be obtained by the use, in location ahead of the dehumidifier, of precooling coils through which the cold water is circulated. The resultant cooling of the air decreases the load to be carried by the dehumidifier and refrigeration plant. In normal practice the water, after passing through the precooling coils, may be further utilized in the refrigeration plant condenser. The economic advantages of this scheme are apparent, and it is frequently used.

SENSIBLE COOLING WITH UNWETTED COILS

Under favorable atmospheric conditions where a large wet-bulb depression exists and the dew-point of the outdoor air is sufficiently low at all times, acceptable cooling may be obtained by removing only the sensible heat from the outdoor air delivered to the rooms. Under this condition of a great wet-bulb depression, a temperature-reducing coil may be located in the air stream and supplied with water from a cooling tower. When humidity control is desired, sprays to saturate or partially saturate the air may be used down-stream from the unwetted coil. Saturation or partial saturation after the coil will reduce further the dry-bulb temperature and the air quantity required. This system has very definite application in hot dry climates.

RUN-AROUND SYSTEM

An interesting method of control is found in the use of combined reheating and precooling, usually termed the run-around system. Typically, three coils are placed in series in the air stream. The primary one receives liquid that has been cooled in the third coil. The center coil is maintained at a temperature colder than the dew-point of the air. The primary coil thus precools the air, and the third coil reheats the saturated air from the center coil. The third coil is heated by the relatively warm water coming to it from the primary coil. The run-around scheme has the advantage of permitting a higher supply air dew-point temperature than would be possible otherwise. This is due to the fact that continuous reheating is available, which is not a large penalty on the refrigeration plant since it provides precooling at the same time. This reheating at peak load creates an artificial sensible heat gain which increases the ratio of sensible heat to total heat and, for a given room temperature, results in a higher apparatus dew-point. Thus, while the volume of supply air is increased, the low-side temperature level of the refrigeration plant is raised and this may effect savings in initial and operating costs. The run-around system has the disadvantage of providing a decreasing amount of heat for reheating as the demand for reheating increases.

SELECTION OF TYPE OF SYSTEM

If the perimeter of the building is large with regard to the area, and if there are many rooms, induction convectors of either the low or high pressure type may be employed. Occasionally a dual system, one duct carrying air at a warmer temperature than the other, may be considered.

Low buildings with large floor areas may be divided into sections or zones with separate central air supply systems to facilitate temperature control. In the case of large department stores it may be possible to provide a single conditioner, with a fan delivering the conditioned air to local mixing fans which supply the various departments or spaces. This application is limited by the practicability of running the large conditioned air ducts to the various recirculating fans. Each vertical section of the building also may be supplied by a separate fan delivering conditioned air to local mixing fans. In many cases the most economical and satisfactory scheme may be to employ a hot water or steam reheater in each branch duct. Where vertical sectionalizing is not indicated, the building may be divided into horizontal groups, each handled by a central system and adequately zoned. In some large buildings, apparatus rooms for the systems may be located in the basement and in the attic and on intermediate floors.

In high buildings the necessity for horizontal sectionalizing may be suggested by the size of air supply and return risers, and by the extent to which they encroach upon usable space. Each story should be cut off by doors from other stories, as otherwise the cool air tends to collect in the lower story and the warm air is forced to the upper story.

Balconies and large lobbies in theatres and similar high rooms frequently justify the use of separate zoning fans, to counteract the tendency of the heavier, cooler air to collect at the lower levels of these spaces.

Fans operate at full capacity continuously in many systems, and therefore should be selected for good efficiencies. In winter when higher temperature differentials are used, it is sometimes practicable to deliver smaller air quantities than when cooling.

In climates where winter temperatures fall below freezing, the tempering coils should be of the steam-distributing type; or if they are heated by a liquid, this liquid should contain some anti-freezing substance such as ethylene glycol. If hot water is employed in cold climates, the temperature control of the air should be obtained through use of face and by-pass dampers rather than by throttling the valves, to prevent damage due to freezing.

If zone reheaters placed in supply duct branches are employed, they should be of such type as to be heated over the entire surface so that no temperature-stratification can occur in the delivered air. Steam-distributing-tube coils or mechanically circulated water coils are satisfactory in such cases, and throttling valves may be used.

Refrigeration equipment must be carefully selected to satisfy the particular requirements of each installation. For some small plants the evaporator may be placed in the air stream, when type of refrigerant and nature of occupancy permit. In many cases, chilled water coils are required by considerations of safety. Where low temperature and relative humidity are necessary, brine, often of calcium chloride, may be indicated.

Condensing requirements must have economic analysis. Wells, public water service, cooling towers and evaporative condensers present possibilities for consideration. Condenser water may have a secondary use for roof sprays in hot weather, and is usually suitable for lawn sprinkling Most health department rules in cities prohibit any connection from refrigerant condensers that might permit the water to be used for drinking or lavatory purposes.

Practically without exception, air cleaners should be provided for both outside and recirculated air.

Control of temperature and of relative humidity by automatic means is vital, if comfort and economical operation of air conditioning equipment are to be attained.

The insulation of ductwork is not merely a matter of economics, but sometimes is a necessity from the standpoint of limiting the temperature change of the air between the conditioning apparatus and the point of final delivery. Such temperature change of the air should be taken into account when apportioning the air and sizing the ducts. When computing heating or cooling loads, due allowance must be made for the effect of any hot or cold ducts or pipes contained in the space under consideration, and insulation must be incorporated where necessary or justified. Consideration must also be given to the possibilities of condensation of moisture on either the inside or outside surfaces of pipes, ducts, housings, fan encasements, etc., and insulation should be applied to prevent corrosion and water damage and to conserve refrigeration.

The location of the apparatus room often is determined by building construction or available space. The closer the apparatus room is to the conditioned space, the less expensive are the ducts. If the equipment is noisy, it should be located at some distance from the occupied spaces or be provided with adequate sound and vibration treatment. The scattering of wet apparatus throughout a building is to be avoided unless suitable precautions are taken. It must be remembered that encroachment on spaces that are otherwise usable should be charged against the system as an operating cost.

In general, the apparatus should be arranged to have straight-line air flow. Each change in direction increases air resistance and, in addition, elbows and offsets may cause eddy currents resulting in stratification. The usual order of equipment location, beginning at the outside air intake, is: weather hood of louvers, outside air dampers, return air connection, filters, tempering coils, cooling coils or sprays, by-pass connection with or without reheaters, reheaters, fan and distributing ducts.

Screens at the intake prevent the entry of large foreign matter, birds, etc. A hood or louver at the outside air intake prevents the entry of rain and snow. Since in most climates there are many days during which use of 100 percent outside air unheated or uncooled may be economical, the areas of all air-passing and treating apparatus should be large enough for such a volume, and the exhaust or spent air equipment should be capable of discharging out of doors, all of the air admitted.

The by-pass connection normally connects the return air duct system with the apparatus casing between the conditioner and the supply fan. Usually the by-pass opening is sized to handle about 50 percent of the fan capacity where a variable by-pass is used, though extreme load variations may require a larger size. It is at times good design to locate a reheating coil in the by-pass connection to permit using some by-pass air when heating is required. Since the relatively high resistance of the cooling coil or spray is to be balanced by the heating coil and by-pass connection, enough heating surface can be provided to raise the temperature of the by-pass air to the point where the mixture of by-passed air and conditioned air will have the required temperature. When a variable by-pass is used, a damper working in opposition to the by-pass damper should be placed across the face of the dehumidifier, for unless the resistances of the two are carefully balanced at all operating points, the proper mixtures of air will not be obtained. Outside air that has not been dehumidified should not be by-passed around a cooling coil or spray dehumidifier if accurate control of the delivered relative humidity is desired. Where the by-pass is made a part of the dehumidifier or conditioner and is located on the top or side of it, the return air connection should be arranged so that stratification of return air is insured, baffles being provided to accomplish this purpose if necessary. Where return air and by-pass air connections are taken off a return duct system, it may be necessary to install a back-draft damper between the return air connection and the by-pass connection. When the by-pass damper is at maximum opening it may be much easier for outside air to pass through the return damper, into the return duct connection and through the by-pass, than for return air to pass through the by-pass connection into the fan. Air tends to take the path of least resistance and, if the dehumidifier resistance is high, and if the return duct resistances are low, this situation is apt to occur. A recirculating air fan, instead of a back-draft damper, may be required for this case, if the failure of return air to reach the dehumidifier or conditioner is a serious matter under reduced load conditions.

LOCATION OF APPARATUS

In general, the outside air intake, preheaters, and return air connections precede the conditioner, while the by-pass, reheaters and fan follow it. In the case of a blow-through system, where the fan is located ahead of the conditioner, the leakage of air at the conditioner is outward, instead of inward, and may be accompanied by water leakage.

The location of the complete apparatus assembly, including the dehumidifier, will be dependent on the type of building, spaces available, structural characteristics, etc. The type of conditioner used may limit the location under certain conditions. Where cooling coils employing chilled water or brine as the cooling agent are used, there are few limitations with regard to location other than those of pumping power, working pressures, piping costs, etc. Where spray dehumidifiers are used, very definite limitations present themselves, and these may require certain extraneous equipment to make the system workable. If several spray type dehumidifiers are located on different levels, thus involving different water pressures, a surge or storage tank, to which the return water from each dehumidifier can be taken, is required. Should the water level in the pan of the dehumidifiers be low in relation to that of the surge tank, return water pumps will be required, and these pumps will have to be operated until the water supply lines are drained in order to prevent flooding of the lower dehumidifiers. Where spray dehumidifiers are on the same level, equalizing lines between the pans may be required if a storage tank is not provided. It is exceedingly important that water-tight drained floors be provided under all overhead cooling systems, since water condensed out of the air generally will be present and may damage the interior finish of the rooms below the apparatus.

All of the various pieces of equipment from the outdoor air intake through the fan usually are connected together by sheet metal casings. Frequently, the building structure or specially constructed walls or partitions may be used to form a portion of the casing. In any case the casing or connection must be sufficiently sturdy for the required duty. Sheet metal work must be well braced not only to prevent vibration under pulsations in air flow, but also to withstand the abuse of normal usage. Casings should be braced wherever access doors are installed, and all large panels should be adequately reinforced by structural steel.

Accessibility for Service

Each apparatus layout should be made with accessibility in mind. Where cooling convectors are used, space for removing and repairing or replacing them should be provided. Adequate space should be provided for the servicing and replacement of eliminators. Whether these accompany sprays or wetted coils, filters must be so located that the proper cleaning, replacement or routine servicing can be accomplished without difficulty. Free access to the bearings of all moving machinery is a necessity. Provision should be made for the complete removal and replacement of any parts of the apparatus that are subject to wear, deterioration or damage, whether they be filter, fanwheel, motor, pump impeller, or heat transfer surface.

DESIGN PROCEDURE

The customary design procedure is outlined herewith. For simplification the procedure is set up on the basis of a year-round system. For systems designed only for winter or for summer, the unrelated parts may be omitted.

1. Selection of design conditions (inside and outside).
   a. Summer.
   b. Winter.

2. Determination of outside air requirements.
3. Determination of cooling load.
   a. Room sensible heat gain.
   b. Room latent heat gain.
   c. Room total heat gain.
   d. Grand total heat gain.
4. Determination of heating load.
   a. Room sensible heat loss.
   b. Room moisture loss.
   c. Humidification requirement.
   d. Total heating requirement.
5. Determination of apparatus dewpoint and dehumidified or humidified air quantity.
   a. Summer (full load and part load).
   b. Winter.
6. Supply air temperature difference and quantity.
   a. Summer.
   b. Winter.
7. Equipment selection.
8. Equipment layout.

The foregoing steps are merely typical. Many applications will require at least a preliminary investigation of some of the latter steps before proceeding with the earlier steps. A permanent record of all design assumptions and computations should be made and preserved for comparison with the performance of the installation.

CHAPTER 30

AIR DISTRIBUTION

Standards for Satisfactory Conditions, Definitions, Mechanics of Air Distribution, Outlet Performance, Types of Air Outlets, Outlet Location and Selection, Directional and Volume Control, Return and Exhaust Intakes, Specific Applications

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CORRECT air distribution contributes as much or more to the success of a forced air heating, ventilating, cooling or air conditioning system as does any other single factor. An air conditioning system may deliver the required quantity of conditioned air and still fail to give satisfactory room conditions because of poor air distribution. The scope of the chapter is limited to the air distribution within the conditioned space. Reference is made to the distributing duct system only insofar as it affects the performance of the air distribution outlet. (See Chapter 31 for information on air duct design).

STANDARDS FOR SATISFACTORY CONDITIONS

The object of air distribution is to create within the space the proper combination of room temperature, air motion and humidity, whether by cooling, heating or ventilating. The purpose to be accomplished determines the factors to be controlled. For instance, in many industrial applications it is necessary to maintain proper standards throughout a large portion of the space; sometimes almost throughout the entire enclosure. In these cases design room temperature, room air motion and humidity will depend entirely upon the requirements of the product and its manufacturing processes.

If, however, comfort of the occupants is the principal objective, consideration of the occupied zone (floor to 6 ft above floor level) is primarily required. In order to obtain comfort conditions within this zone, standard limits have been set up as acceptable *effective temperatures*. This term comprises air temperature, motion, humidity and their physiological effect on the surface of the human body. Any variation from accepted standards of one of these elements may result in discomfort to the occupants. The same effect may be caused by lack of uniformity of conditions within the space or by excessive fluctuation of conditions in the same part of the space. Such discomfort may arise due to excessive room air temperature variations (horizontally, vertically, or both), excessive air motion (draft), failure to deliver or distribute the air according to the load requirements at the different locations, or too rapid fluctuation of room temperature or air motion (gusts).

In addition the noise level created by the introduction of supply air should be kept within acceptable limits, and streaking or smudging of walls or ceilings should be prevented.

With reference to permissible room air motion it is not possible to establish a specific standard covering the entire complex problem of air distribution. Velocities less than 15 fpm generally cause a feeling of air stagnation, whereas velocities higher than 65 fpm will disturb loose paper sheets on desks and may result in a sensation of draft. Air velocities of 25 to 35 fpm in the occupied zone are most satisfactory, but air motion of 20 to 50 fpm will usually be acceptable, particularly when the lower part of this range of velocity is used in cooling applications, and the higher values on heating jobs. In any case, it is certain that the effect of room air motion on comfort or discomfort depends on air temperature and direction as well as on velocity.

Reference should be made to Chapter 6, Physiological Principles, for information on effective temperature and comfort zones. Material in Chapter 40, Sound Control, covers acceptable room noise levels and noise generated by air outlets.

DEFINITIONS

The following definitions referring to air distribution equipment have gained general acceptance.

1. *Supply Opening or Outlet:* Any opening through which air is delivered into a space which is being heated, or cooled, or humidified, or dehumidified, or ventilated.

2. *Exhaust Opening or Return Intake:* Any opening through which air is removed from a space which is being heated, or cooled, or humidified, or dehumidified, or ventilated.

3. *Outside Air Opening:* Any opening used as an entry for air from outdoors.

4. *Damper:* A device used to vary the volume of air passing through a confined cross-section by varying the cross-sectional area.

5. *Grille:* A covering for any opening and through which air passes. A supply grille discharges air axially with a limited spread.

6. *Register:* A grille equipped with a damper.

7. *Free Area:* The total minimum area of the openings in the air outlet or inlet through which air can pass.

8. *Core Area:* The total plane area of the portion of a grille, bounded by a line tangent to the outer edges of the outer openings through which air can pass.

9. *Mean Area:* The total of the core and free areas divided by two.

10. *Percentage Free Area:* The ratio of the free area to the core area expressed in percentage.

11. *Aspect Ratio:* The ratio of length of the core of a grille to the width.

12. *Vane Ratio:* The ratio of depth of vane to shortest opening width between two adjacent vanes.

13. *Plaque:* A ceiling outlet in which the supply air impinges against a plate or series of parallel plates, and is deflected horizontally in all directions.

14. *Diffuser:* An outlet discharging supply air in various directions and planes thereby effecting its mixture with the room air.

15. *Primary Air:* The air delivered to the outlet by the supply duct.

16. *Induction:* The entrainment of room air by an air stream.

17. *Internal Induction:* The induction of room air drawn into an outlet by the primary air stream. (Commonly called *aspiration.*)

18. *External Induction:* The induction of room air by the air stream discharged from the outlet (commonly called *secondary air motion*).

19. *Induced Air:* The room air entrained by the primary air through internal induction, or by the discharged air through external induction or both.

20. *Total Air:* The mixture of primary air and induced air.

21. *Induction Ratio:* The total air divided by the primary air.

22. *Throw (Blow):* The horizontal or vertical axial distance an air stream travels on leaving the outlet (grille) to a position at which air motion reduces to a maximum velocity of 50 fpm.

23. *Drop:* The vertical distance, the lower edge of a horizontally projected air stream drops between the outlet and the end of its throw.

24. *Rise:* The converse of drop.

25. *Envelope:* The outer boundary of an air stream moving at a perceptible velocity.

26. *Spread:* The divergence of the air stream in a horizontal or vertical plane after it leaves the outlet.

27. *Diffusion:* Distribution and mixing of air within a space, accomplished by an outlet discharging supply air in various directions and planes in order to effect the desired air conditions in the occupied zone of that space.

28. *Radius of Diffusion:* The horizontal distance from the diffuser outlet to the perimeter of the space, within which effective diffusion is accomplished and air motion in the occupied zone is reduced to 50 fpm maximum.

29. *Outlet Velocity:* The average velocity of air emerging from the outlet measured n the plane of the opening.

30. *Terminal Velocity:* The average air stream velocity at the end of the throw.

31. *Temperature Differential:* Temperature difference between primary and room air.

32. *Temperature Variation:* Temperature difference between points of the same space.

MECHANICS OF AIR DISTRIBUTION

In the mechanics of air distribution, two major problems are involved: (1) complete mixing of the primary air and air outside of the zone of occupancy in order to reduce the temperature difference and air motion to acceptable limits before the air enters the occupied zone; and (2) counteraction of the natural convection and radiation effects within the room.

The theory concerning the distribution of conditioned air within an enclosure is still incomplete, and no general law governing outlet performance has been formulated. The characteristics and performances of the various existing types of outlets must therefore be evaluated largely by experimental work. Some progress has been made concerning the theoretical analysis of the characteristics of a primary air stream discharged in an unconfined space, *i.e.*, a space large enough so that the primary air stream is not disturbed by contact with surfaces, or by adjacent streams. The approach to this problem is usually made by means of the momentum theory. Development of this theory has so far been confined to side wall distribution of air, because this is its most elementary application. Fundamentally, the same laws apply also to ceiling distribution, but a great amount of additional research is still required to adapt them to the more complicated conditions of deflection of air up to 90 deg, spread up to 360 deg and the resulting rapid induction.

Momentum Theory

When air is discharged from an outlet into a free open space, the primary air stream entrains room air as it traverses the space. This entraining effect increases the cross-sectional area and reduces the velocity of the resulting air stream. When the air stream is projected horizontally, induction takes place with the conservation of linear momentum. This has been confirmed by tests which indicate that the momentum remains almost constant throughout the entire measurable length of the air stream. This relationship may be expressed by Equation 1:

$$M_1V_1 + M_2V_2 = (M_1 + M_2)V_3 \tag{1}$$

where

M_1 = mass of primary air.

M_2 = mass of induced air.

V_1 = velocity of primary air.

V_2 = velocity of induced air (for practical use, $V_2 = 0$).

V_3 = velocity of the mixture.

If the velocity of induced air is zero, Equation 1 changes to

$$M_1V_1 = (M_1 + M_2)V_3$$

or,

$$\frac{V_1}{V_3} = \frac{M_1 + M_2}{M_1} = r \tag{2}$$

Since in many applications the densities of primary and room air are about equal, air volumes may be substituted for mass and Equation 2 becomes

$$\frac{V_1}{V_3} = \frac{Q_1 + Q_2}{Q_1} = \frac{Q_3}{Q_1} = r \tag{3}$$

where

Q_1 = volume of primary air, cubic feet per minute.

Q_2 = volume of secondary air, cubic feet per minute.

Q_3 = volume of mixture of primary air and induced air, cubic feet per minute.

r = induction ratio.

Jet Pattern from Round or Rectangular Openings in a Large Room

The relation between the shape of the discharge of a jet and the shape of the conventional outlet has long been the subject of research. It has been proved to be incorrect to assume that the jet retains the outlet shape when it discharges into a free open space.[1] Air streams from rectangular outlets having low aspect ratios develop a symmetrical or cone shape within a few diameters from the outlet face. From there on, the jet continues to expand at a fairly constant rate. Beyond 20 diameters there is very little difference between round and rectangular jets. The assumption can be made that the apex of the cone is in the same position for any jet having a small aspect ratio. For the more usual problems of the conventional room with outlets near the ceiling, there are insufficient experimental data to justify a definite statement on the effect of aspect ratio.

If the round or rectangular opening is divided into a number of orifices having straight sides, the performance of the air stream will be similar to that of a plain opening.

Velocity Across Jets

Results of many tests[1] indicate that the ratio of centerline velocity to average velocity is about 3, irrespective of outlet size, shape or initial velocity. This statement is true for stream cross-sections located beyond 10 diameters from the outlet, and is fairly accurate for distances up to 50 diameters. Experimental data are lacking for distances beyond 50 diameters.

Effect of Aspect Ratio on Entrainment

In slotted outlets, the air entrainment of the primary jet is a function of aspect ratio.[1] This effect is most pronounced when large changes in the ratio are made. A comparison between a slot of aspect ratio 24 and a square opening of the same area is given in curves A and B of Fig. 1. At a distance of 8 ft from the outlet, the entrainment of the slot is 8.1 as compared with 6.9 for the square, or an increase of about 17 percent.

Curve *C* shows the further increase in entrainment obtained by using an aspect ratio of 48. An increase of 40 percent is obtained over the 24 in. x

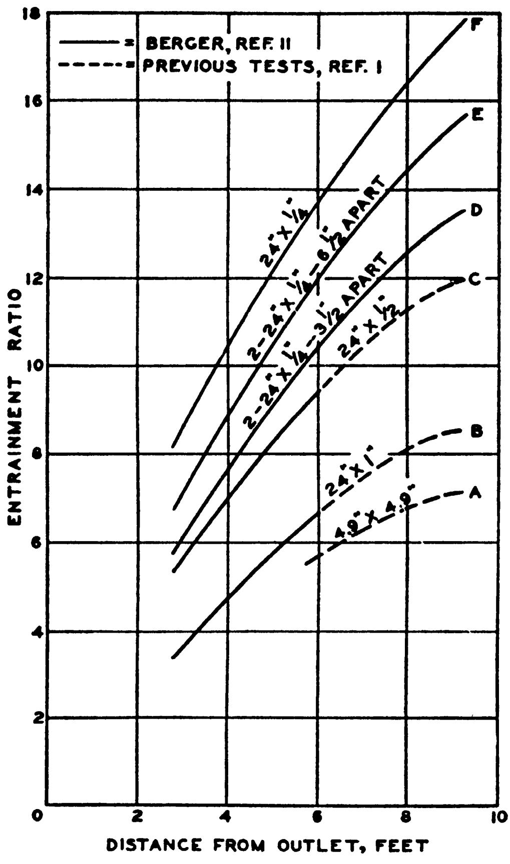


Fig. 1. Typical Relation of Entrainment Ratio to Distance from Outlet for Slotted Outlets. (Based on 800 fpm Outlet Velocity.)

1 in. slot. This indicates that long narrow slots produce air streams that give high induction of secondary air.

Parallel Slots

The use of several slots in parallel to vary the rate of air entrainment depends mainly on the distance between the slots. If close together, the air pattern is about the same as for a single opening of equal area. Spacing the openings farther apart gives an increase in entrainment as shown on curves D and E of Fig. 1. It will be noted that 2 openings 24 in. x $\frac{1}{4}$ in. located very close together will obtain an entrainment which is about the same as obtained with one 24 in. x $\frac{1}{2}$ in. opening. However, if the slots are spaced $6\frac{1}{2}$ in. apart there is a marked increase in entrainment.

Throw

Equations for the throw of straight flow side wall outlets have been developed on the basis of the momentum theory. Equation 4 states the throw in terms of the area of the outlet and the primary air volume.[2]

$$L = 0.82 \frac{Q_1}{\sqrt{A_1}} \tag{4}$$

where

L = throw, feet.

A_1 = effective outlet area, in square inches = (gross measured area) × (percentage of free area/100) × (discharge coefficient).

The discharge coefficient is approximately 0.8.

Equation 4 has been developed under the assumption that the temperature of the supply air is the same as the temperature of the room air. It applies only to straight flow outlets with aspect ratios less than 16.

Equation 5 for the performance of straight flow outlets evolved from research[1] allows the calculation of the maximum residual velocity at any distance perpendicular to the outlet face. It applies for aspect ratios up to 50.

$$V_r = K \frac{V_1 \sqrt{A_1}}{X} = K \frac{Q_1}{X \sqrt{A_1}} \tag{5}$$

where

V_r = maximum residual velocity in air stream, *i.e.*, the highest maintained velocity at the given cross section in the room, feet per minute.

V_1 = average initial velocity across outlet, feet per minute.

K = constant of proportionality.

A_1 = effective outlet area in square feet = (gross measured area) × (percentage of free area/100) × (discharge coefficient).

X = normal distance from outlet face, feet.

Equation 5 together with Equation 6 (which reduces to Equation 7 if the jet angle is 20 deg) for the entrainment ratio,

$$\text{Entrainment Ratio} = \frac{0.785\,K}{RX\sqrt{A_1}} \left(\sqrt{\frac{A_1}{0.785}} + 2\,X \tan \frac{\theta}{2} \right)^2 - 1 \tag{6}$$

where

R = ratio of maximum residual velocity to average residual velocity.

θ = jet angle or spread angle in degrees.

$$\begin{matrix}\text{Entrainment Ratio} \\ \text{20 deg jet angle)}\end{matrix} = \frac{0.785\,K}{RX\sqrt{A_1}} \left(\sqrt{\frac{A_1}{0.785}} + 0.35\,X \right)^2 - 1 \tag{7}$$

has been used to develop charts[3] which provide the graphical solution of problems involving the determination of the throw of air from slots and jets, the residual velocity, and the size of openings. (See Figs. 2 and 3). The charts apply only to *air discharging into room air of same temperature as the stream.* They can be used to determine the throw of air and entrainment ratios up to 40:1 with initial velocities of 1000 to 6000 fpm, and with residual velocities of 100 to 1000 fpm. The charts furthermore are for use with sharp-edged orifices or slots, and *include the coefficient of discharge.* If air is discharged from an orifice with a well-rounded entrance or from a length of straight duct, the coefficient of discharge is unity and the actual area of the opening is the effective area. For such rectangular openings the effective diameter is the diameter of a circle with an area equal to the actual area of the rectangle. The following examples will illustrate the use of the charts:

Example 1: Air is delivered to a cooler through independent slots each 24 in. x 2 in.

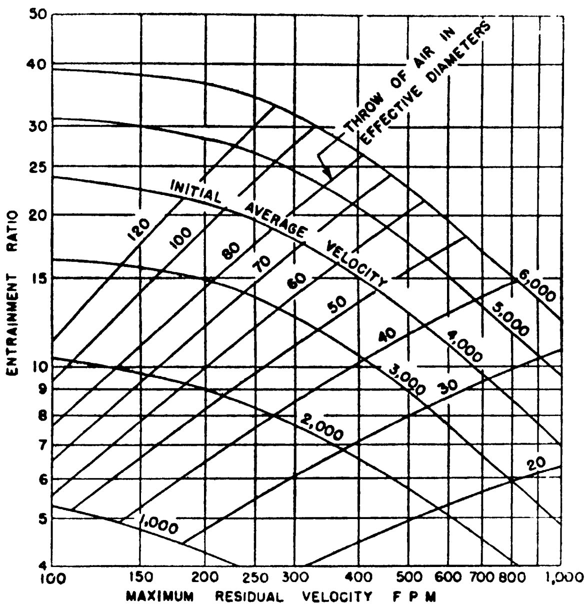


FIG. 2. RELATION BETWEEN INITIAL VELOCITY, RESIDUAL VELOCITY, ENTRAINMENT RATIO AND THROW OF AIR FROM JETS AND SLOTS

with an initial velocity of 2000 fpm. Determine the maximum residual velocity and the entrainment ratio at a distance of 15 ft from the slot.

From Fig. 3 the effective diameter = 6.2 in. = 0.52 ft. The number of effective diameters in 15 ft = 15/0.52 = 28.8.

From Fig. 2 at 2000 ft initial velocity read entrainment ratio = 6.6 and maximum residual velocity = 390 fpm. From tests it has been shown that the *average* residual velocity may be taken as ⅓ of the *maximum* or 130 fpm in this case.

Example 2: Using the data from *Example 1* determine the distance at which the maximum residual velocity will be 150 fpm.

From Fig. 2 at V_1 = 2000 and V_r = 150, the number of effective diameters is read directly as 73 and the throw of the air is therefore 73 x 0.52 = 38 ft.

Example 3: Air issues from a round orifice plate with an initial average velocity of 4000 fpm. It is to have a maximum residual velocity of 400 fpm at a distance of 30 ft from the opening. Calculate the size of the opening required and the entrainment ratio.

On Fig. 2 at the intersection of the curve of 4000 fpm, the entrainment ratio is read directly as 15 and the effective diameters of throw = 55.

Since 55 effective diameters are equal to 30 ft as required, 1 effective diameter = 30/55 = 0.545 ft or 6.56 in.

On Fig. 3 vertically below intersection of 6.56 in. effective diameter line, and equivalent round opening line, read 8.5 in. in lower margin.

Example 4: A jet of air issues from a pipe or from air orifice having a well-rounded

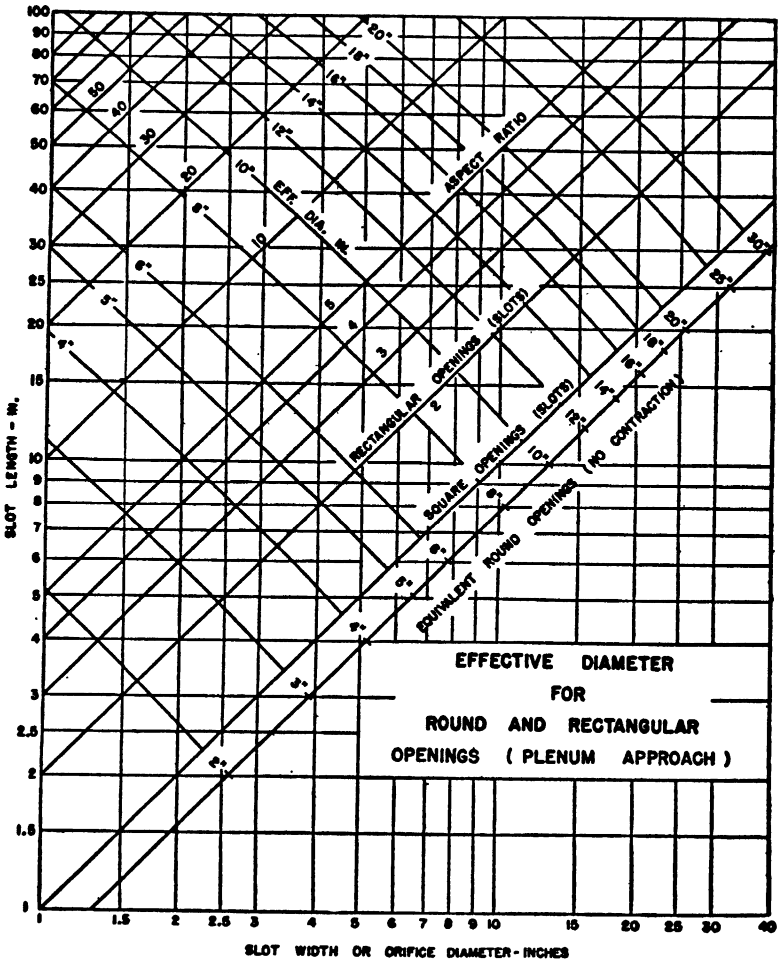


FIG. 3. EFFECTIVE DIAMETERS FOR ROUND AND RECTANGULAR OPENINGS (PLENUM APPROACH)

entrance (coefficient of discharge = 1.0) and delivers air with the same velocities and with the same throw as in *Example 3*. What is the required diameter?

In this case since the coefficient of discharge is unity, the effective diameter of the jet is the actual diameter of the pipe or orifice, or 6.56 in., as obtained in *Example 3*

Spread

The induction effect results in the spreading of the air stream. The total angle included by the air stream from straight flow outlets has been measured and found to be between 14 and 25 deg. The angle will depend on the type of approach, type of outlet and velocity.

The effect of vertical bars placed in the face of the outlet to increase the spread, may also be deduced from the momentum theory. Assuming that there are no horizontal deflecting bars, and that the air spreads vertically through a total angle of 14 deg; that a uniform velocity exists at any section of the air stream; and that the conservation of momentum principle applies down to a velocity of 60 fpm; the following approximate equations for throw are to be substituted for Equation 4:[2]

For a spread of 15 deg on each horizontal side $$L = 0.55 \frac{Q_1}{\sqrt{A_1}} \tag{8}$$

For a spread of 30 deg on each horizontal side $$L = 0.37 \frac{Q_1}{\sqrt{A_1}} \tag{9}$$

For a spread of 45 deg on each horizontal side $$L = 0.28 \frac{Q_1}{\sqrt{A_1}} \tag{10}$$

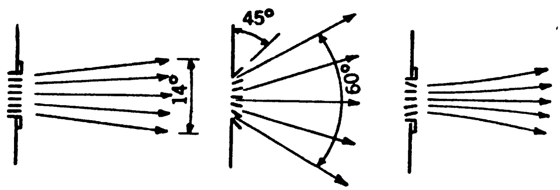


FIG. 4. SPREAD OF AIR STREAM WITH VARIOUS VANES

Guide Vanes

Vanes should have a depth of one to two times the spacing between the vanes. If the ratio of vane depth to spacing is less than one, effective control by means of the vanes cannot be obtained. Little improvement is obtained by increasing the ratio beyond two. The effect of various types of vanes is given in following paragraphs.

Straight Vanes. As mentioned previously, the included angle between both planes will be in the neighborhood of 14 deg, for a straight setting of the vanes as shown in Fig. 4.

Diverging Vanes. Such vanes set for an angular spread will have a marked effect on the direction and distance of travel of an air stream. An outlet having vertical vanes set straight forward in the center, with uniformly increasing angular deflection to a maximum at each end of 45 deg, will produce an air stream with a horizontal included angle of approximately 60 deg as shown in Fig. 4. The throw will be reduced one-half for such a vane setting. Increasing the divergence of the vanes reduces the air quantity handled by an outlet, for a given duct static pressure. The primary function of the vanes is to spread the air horizontally. Spreading the air vertically entails the risk of hitting beams or other obstructions, or of blowing primary air at excessive velocities into the occupied zone.

Converging Vanes. The blow of an outlet may be somewhat increased by converging the vanes of an outlet as illustrated in Fig. 4. Even with converging vanes, the resultant angle of spread of an air stream will not be less than 14 deg. The air converges for a few feet in front of the outlet, and then diverges more than if the vanes had been set straight.

Both the horizontal and vertical vanes of an outlet are important. After an installation has been made, many conditions of draftiness or stuffiness can be alleviated by some vane adjustment, provided an independent means for regulation of static pressure behind the vanes is included.

Vertical Drop and Rise

The distance that the lower edge of the air stream drops below the bottom of the outlet is important, since the air stream should not reach the occupied zone until the velocity has fallen to about 50 fpm. The drop (H, feet) is influenced by two forces; the natural vertical spread of the stream and the gravitational force due to the difference in density between supply air and room air. For air emerging at room temperature, the drop will be a function of the spread only and will be equal to:

$$H_1 = L \times \tan\left(\frac{\text{Spread Angle}}{2}\right) \tag{11}$$

where

H_1 = drop due to spread (when emerging air and room temperature are the same), feet.
L = throw, feet.

When there is a temperature difference between the air stream and the room, there is an *additional* drop which is approximately:[4]

$$H_2 = \frac{n_1(t_r - t_{as})L^{n_2}}{V_1} \tag{12}$$

where

H_2 = additional drop due to temperature difference, feet.
n_1 and n_2 = constants (tentative suggested values $n_1 = 5$, $n_2 = 1.2$).
t_r = room temperature, degrees Fahrenheit.
t_{as} = supply air temperature, degrees Fahrenheit.
V_1 = jet velocity, feet per minute.

It should be remembered that the total drop $H = H_1 + H_2$. H_1 is positive for either heating or cooling; H_2 is positive for cooling, negative for heating. In consequence, there will always be vertical drop in cooling, and a vertical rise in heating only if $H_2 > H_1$.

Another empirical equation for the *total drop* is:[2]

$$H = \frac{m(t_r - t_{as})L}{U_1} \tag{13}$$

where

m = constant (tentatively suggested value of $m = 16$).

In other words, for a given throw L, the drop or rise increases as the temperature difference increases and the outlet velocity decreases. This equation is only valid if a temperature difference exists between room air and supply air.

Room Air Motion (Wall Outlet)

One lf the most important problems in air distribution is to achieve air motion in the occupied zone within acceptable velocity limits. Therefore, outlet performance and characteristics of the space have to be related to this air motion.

The air moving in the occupied zone is (for a side wall outlet) equal in quantity to the total air contained in the outlet stream at the end of the throw, and it is generally moving in a direction opposite to the stream. Assuming that the maximum volume of air is in circulation when the air stream velocity V_3 drops to 200 fpm, that the free area for return flow is 0.6 of the area of the wall in which the outlets are located, then, according to the momentum theory:[2,4]

$$V = \frac{Q_3}{0.6 \times A_w} \tag{14}$$

where

V = average room velocity, fpm.
Q_3 = volume of room air in motion, cfm.
A_w = area of wall in which outlet is located, square feet.

Since $Q_3 = Q_1 \times r$, (by definition); and $r = \frac{V_1}{V_3}$ according to Equation 3; the average room velocity is:

$$V = \frac{Q_1 r}{0.6 A_w} = \frac{Q_1}{0.6 A_w}\left(\frac{V_1}{V_3}\right)$$

or, with $V_3 = 200$ fpm

$$V = \frac{Q_1 V_1}{120 A_w} \tag{15}$$

When the volume Q_1, of primary air, the velocity V_1, of primary air and the wall area A_w, are known, the average room velocity may be calculated from Equation 15 in order to determine the acceptability of the air distribution system.

OUTLET PERFORMANCE

The factors of outlet performance, (1) throw, (2) drop, (3) room air motion, (4) capacity, (5) temperature differential, (6) dirt and (7) noise, place considerable limitations on the design of a satisfactory distribution system.

1. *Throw.* The throw of a wall outlet must be sufficient to produce satisfactory conditions over the area to be conditioned. Underblowing may cause heated air to rise too rapidly above the occupied zone and thus create excessive vertical temperature variation (stratification); in cooling operation it may cause cold air to drop into the occupied zone before a satisfactory mixing of supply and room air has been accomplished by induction and thereby create a condition of acute discomfort (draft). On the other hand, overblowing will result in objectionable downdrafts from any surface the primary air stream may strike.

On the average, it is considered most practicable to select a throw which is three-fourths of the distance toward an exposed wall or window, as shown in A of Fig. 5. However, structural characteristics, mounting height, temperature differential and resultant drop or rise, or location of greatest heating or cooling loads strongly affect the selection of the optimum throw. In spaces with beamed ceilings, the outlets should be located below the bottom of the lowest beam level, and preferably low enough so that an upward or arched blow may be employed. The blow should be arched sufficiently to miss the beams and, at the same time, in such a manner as to prevent the primary or induced air stream from striking furniture and obstacles, and producing objectionable drafts.

In the case of ceiling diffusers, air is distributed with a horizontal spread of 360 deg. In addition there is a downward component of air motion. Therefore, both throw (radius of diffusion) and mounting height are important and interdependent factors. Due to the 360 deg spread of air diffusion, the rate of induction will be higher, and the throw shorter, than that of a wall grille opening handling the same air quantity at the same outlet velocity. Therefore, ceiling diffusers will frequently permit the use of higher air velocities than wall outlets, and consequently may be sized smaller to handle the same air volumes. If such ceiling outlets are installed flush with the ceiling, impingement of the air stream along the ceiling surface restricts induction of secondary air, and the throw is increased approximately 20 percent above that of an unrestricted air stream.

In the use of perforated ceiling plates as air distributing devices, the term throw could hardly be applied in its proper meaning. Although this type of outlet can handle the greatest amount of air in proportion to room size, jet velocities must be kept low.

In all types of ceiling air distribution the following should be noted:

If cold air is used, it must be brought to the proper temperature by mixing with room air before entering the zones of occupancy.

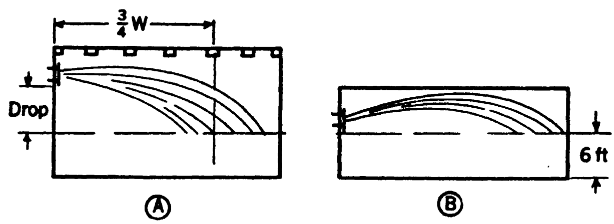


FIG. 5. THROW OF WALL OUTLETS

Air slightly above room temperature will usually be properly distributed by outlets selected for cooling.

When delivering warm air the same may be projected downward, and the amount of dispersal of the jet varied to obtain proper mixing and control.

2. *Drop*. The outlets should be located so that the air stream at the termination of the blow is not less than 5 or 6 ft above the floor level. As illustrated in *B* of Fig. 5 the maximum permissible blow for a given ceiling height may be obtained by locating the outlet low on the wall, arching the blow, and sweeping the air across the flat ceiling. The air, as it traverses the room, will adhere to the ceiling. The objection to this method is the possible streaking of the ceiling with dirt.

3. *Room Air Motion*. Various features may cause room air motion to exceed acceptable standards. Some of these are: excessive air discharge velocities; high air volume per cubic foot of space (often referred to as number of air changes per hour); premature drop of cold air into the occupied zone; overblow causing spilling of high velocity air into the occupied zone; heating in severe climates by means of downward projection of hot air. It should be realized that these factors will not equally affect all types or designs of outlets at different temperature differentials, mounting heights, etc. For instance, certain outlets may safely handle more air per cubic foot of space at higher discharge velocities than others, and downward projection of supply air will sometimes not be considered excessive if the supply air temperature is substantially higher than the room temperature.

4. *Capacity*. The quantity of air to be handled is determined by the heating, cooling, or ventilating requirements. Manufacturers' rating sheets are usually consulted for selection of the proper number, size and type of outlets for a given air quantity. The basis of rating used should be carefully noted to make certain that resulting velocities are suitable for the application.

5. *Temperature Differential.* This is one of the most important factors affecting outlet performance. The quality of the temperature control, or the extent of the control problem, is directly a function of temperature difference. Obviously, a system which carries under design conditions only a 5 deg difference between supply air stream and room temperature, would require no control at all, for even a 50 percent change in load could only effect a 2½ deg change in room temperature under the worst conditions. Because of the self-equalizing nature of most load factors, even this extreme is never realized. It is obvious that the greater the temperature differential between supply air and room temperature, the greater will be the change in room temperature for a given change in load. The use of outlets that give rapid mixing, permits greater temperature differentials. These principles apply in both heating and cooling practice.

6. *Dirt.* Although the primary air may be carefully filtered, small particles of dirt and dust will not be captured by mechanical filters, and may finally be deposited on the walls or ceiling. With ceiling outlets, dirt streaking may be minimized by carefully controlling the discharge of the outlets. With wall outlets, dirt streaking may be minimized by preventing direct impingement of the air on any ceiling or room surface. Floor outlets may offer objection as dirt collectors.

7. *Noise.* The increase of noise level caused by an outlet is primarily a function of its air discharge velocity and its size. The maximum acceptable noise level in a space may dictate completely the selection of the permissible outlet velocity. In addition, however, noise may be caused by excessive restriction of free outlet area due to outlet design; by unnecessary turbulence due to one sided air flow through the outlet; or by the impingement of high velocity air on sharp edges. Such high frequency noises due to excessive turbulence are especially annoying (see Chapter 40 for discussion of permissible room noise levels and noise generation by outlets).

TYPES OF AIR OUTLETS

Two types of air supply outlets are commonly used; side wall and ceiling. A variety of designs has been developed for both types, and the final selection depends to a large degree upon the specific problems arising in the air distribution system to be used.

In addition to the comments on use and application of outlets which follow, reference should also be made to sections of this chapter on Outlet Location and Selection, as well as on Specific Applications.

Wall Outlets

Wall type openings in general use are: (1) perforated grilles, (2) vaned outlets, (3) registers, (4) slotted outlets, (5) ejector nozzles, and (6) wall diffusers.

1. *Perforated Grilles.* Due to the non-adjustibility and small vane ratio, these outlets, although inexpensive, have not met with favor as wall type supply openings. They are useful primarily where directional air control is unnecessary, and for return air intakes.

2. *Vaned Outlets.* Outlets equipped with either vertical or horizontal adjustable vanes or both are particularly suited to sidewall distribution. For proper control over the air flow, the vane ratio should be from 1 to 2. Outlets with non-adjustable vanes may be employed, but they should only be used where the performance is not critical or can be adequately predicted. Vanes should be properly designed to prevent an increase of noise above permissible level.

3. *Registers.* Perforated grilles or vaned outlets equipped with a vane damper are termed registers. They are used primarily for residential heating systems, where the outlet distribution is not critical and low cost is of importance.

4. *Slotted Outlets.* Slotted outlets consist essentially of either flat steel plates containing a number of long narrow slots, or a single long narrow slot. In order to give a good conversion from static pressure to velocity pressure, the sides of the slots are rounded to give a venturi effect. Due to their high aspect ratio, the slotted outlets have a greater induction effect than the comparable vaned outlets of equal area and consequently, the throw is reduced. They are primarily useful where an un-

obtrusive means of distribution is desired, and where it is desirable to submerge the outlets into the room decoration and to minimize the effect of obstructions in the line of discharge. They are adaptable to narrow rooms having low ceilings. In this case the slots should extend the full length of the room. In all applications air quantity and distribution must be carefully planned, as correction after installation is difficult.

5. *Ejector Nozzles*. These are outlets operating at high static pressure. They give a high conversion from static in the duct to velocity pressure in the outlet, and have a high induction effect due to their high outlet velocity. They are chiefly used for long throw and industrial process installations, such as drying, freezing, cooking, etc. Another type of ejector is sometimes referred to as a *louver nozzle* and has a 45 to 90 deg elbow, which can be rotated similarly to a universal joint about an axis perpendicular to the surface to which it is fastened. These outlets give a considerable degree of adjustability and are, therefore, desirable for use in confined spaces where spot cooling is employed. The use of very high velocities is gradually disappearing due to noise difficulties.

6. *Wall Diffusers*. These outlets incorporate design features originally developed for ceiling outlets, and use, therefore, semi-conical or semi-pyramidal guide vanes instead of the straight vanes of the conventional side wall outlet.

Ceiling Outlets

Generally used ceiling outlets are: (1) plaques, (2) ceiling diffusers, and (3) perforated ceilings and panels. A discussion of each follows.

1. *Plaques*. Plaques are of simple design. The air from the supply opening impinges on a plate, which permits the air to be deflected horizontally in all directions. Plaques, although inexpensive, are difficult to control and are not generally satisfactory. In certain applications, a properly designed plaque yields satisfactory results.

2. *Ceiling Diffusers*. Ceiling diffusers are round or rectangular outlets installed on, or parallel to, the ceiling, and discharge supply air in a variety of directions and planes. Performance of the different designs varies according to principle employed. Some have no internal induction, but hasten external induction by supplying air in multiple layers. Others have internal induction and distribute air over an entire hemisphere. The induction effect is greatest in the direction of the axis of the outlet, and least in the plane perpendicular to the axis and located at the ceiling level. Thus the induction is greatest in the vertical direction where the least throw can be tolerated, and least in the horizontal plane at the ceiling where the greatest blow is both desired and permissible.

3. *Perforated Ceilings and Perforated Panels*. These devices obtain air diffusion by discharging air through perforations in the ceiling or part of the ceiling or walls. The advantages are unobtrusive appearance, adaptability for application of sound absorbing material to the design, and large outlet areas (when desired) to limit outlet air velocities.

Some perforated panels feature a control plate frame which is inserted in the conventional ceiling duct. Supply air enters the plenum above the distribution plates through an adjustable air valve which can be set for varying air quantities and velocities. Means are provided in all cases for distributing the primary air over the entire panel in order to obtain even air distribution and proper air velocities.

The large outlet areas are advantageous where there is a high room load, a high ventilating requirement, or where a low ceiling, combined with high ventilating air requirement, necessitates a low outlet velocity. This last condition may be illustrated by a drafting room with a 9 ft ceiling having a high cooling load. In this case, the air velocity at the head level of standing occupants must be barely perceptible, and consequently, the velocity from the outlets must be low in order to be dissipated in a distance of approximately 4 ft.

OUTLET LOCATION AND SELECTION

In selecting the location of outlets, consideration must be given to the factors of (1) physical construction, (2) physical appearance, (3) location of heating or cooling loads, and (4) outlet performance.

1. The *physical construction* of a building, particularly of old buildings, immedi-

ately places limitations on the type of distribution system which can be employed. The first factor in the selection of outlet locations, therefore, is a consideration of the possible location of the supply duct, that is, whether it is above the ceiling, within the walls, through furred spaces above corridors, or in the conditioned space, etc. A particular method of distribution may be highly desirable, but its execution, due to the location of beams and masonry walls, may be impossible.

2. The *physical appearance* of the outlets should conform to the esthetic appearance of the room. In factories, warehouses, etc., the esthetic demand may not be high; however, in department stores, clubs, theaters, etc., the location of the grilles may be dictated largely by such demands.

3. The *location of heating or cooling loads* in a room dictates to a great extent the general location of the outlets. The outlets should be located to neutralize any undesirable cold drafts or radiation effects set up by a concentration of the heating or cooling load. The problem can be divided into *natural loads* due to outside weather and *internal heat loads*.

In winter the natural or primary heating load is caused by exposed walls, windows and skylights. Heat is lost primarily through convection to these exposed surfaces. The convection currents or cold drafts drop down the exposed surfaces and seriously impair the comfort conditions in the room, particularly at the floor level near the exposed surfaces. The outlets should be located to counteract these down drafts. Methods which may be employed are:

a. Direct counteraction of convection currents from cold surfaces can be obtained by locating the outlets to blow upward from beneath windows or exposed walls, or to blow across the exposed wall. This method is desirable in small offices or bedrooms, or any location where people are seated or working near exposed surfaces. In northern climates, where the outside temperature may be constantly below 40 F, and the construction consists of uninsulated walls and single glass, this method of distribution is particularly useful for the maintenance of comfort requirements.

b. High induction by ceiling or wall outlets may be employed to nullify the convection currents from exposed surfaces. If outside temperatures are consistently below 40 F, and the exposed surfaces are not well insulated, the induction effort required for neutralization of the downdrafts is so great that the air motion in the room may exceed comfort limits, unless care is taken in selection and location of the outlet. Where comfort conditions are not critical as in factories for heavy manufacturing, warehouses, etc., satisfactory results can be obtained even in cold climates. For uninsulated walls and glass areas some supplementary heating is often valuable. Wall diffusers, direct radiation or warm panels will satisfy these requirements for supplementary heating.

c. The location of exhaust or recirculated air openings at the base of large areas of glass is sometimes effective in reducing cold downdraft into the occupied space.

If a *concentrated source of heat* creating an *internal heat load* is located at the occupancy level of the room, the heating effect may be counteracted by blowing the supply air toward the heat source, or by locating an exhaust or return grille adjacent to the heat source. The latter method will prove more economical, as heat will be withdrawn at its source rather than be dissipated into the conditioned space. Where a lighting load is particularly heavy (five watts per square foot) and located high in a conditioned space, it may be economically desirable to locate the outlets below the lighting load. Warm air from the lights will stratify near the ceiling and can be removed by an exhanst or return fan, the former being advisable if the wet-bulb temperature of the air is above the outside temperature, and the latter being preferable if the wet-bulb temperature is below that of the outside air. Either method reduces the requirements for supply air. If the lamps are exposed, less saving can be realized than if enclosed, as a considerable portion of the total energy is radiant.

4. *Outlet Performance.* The laws of air distribution, previously discussed, will be found to exercise an important influence upon the design of an acceptable distribution system. This applies particularly to such features as throw, drop, capacity and room air motion.

Procedure for Outlet Location and Selection

In determining outlet location and selecting the type of outlets, it is customary to proceed as follows:

1. Study the plan of the building and note the amount of air to be supplied to each enclosure.

2. Select number of outlets for each enclosure, considering air quantity required and distance available for throw or as radius of diffusion. The same factors, as well as distance from floor level available as mounting height, structural characteristics of the space and consideration of appearance, will determine the type of outlet used.

3. Arrange location of outlets in space. Usually the outlets will be evenly spaced to distribute air uniformly throughout the enclosure. Sometimes, however, more air should be supplied and directed towards zones having exceptional heating or cooling loads. An important point to consider is the combination of proper outlet location and efficient duct design (see Chapter 31). Consult manufacturers' tables for recommended location and spacing of outlets.

4. Select size of outlets according to air quantity handled, permissible throat or discharge velocities or effective throw, taking into consideration other factors such as noise level, static pressure resistance, etc. It will generally be found that most selection tables for grille type outlets are based on capacity and throw, whereas data for ceiling or wall diffusers are usually based upon capacity and permissible outlet velocity. Choice and arrangement of either type of outlet should, however, satisfy the requirements of all aspects of air distribution. Therefore, type, location and size of any outlet should be checked against manufacturers' ratings to determine whether the selection made would satisfy the requirements of the job. The most important questions to be considered are:

a. Can drafts occur because of divergence between rated throw (radius of diffusion) and distance between outlet and nearest obstacle of air stream (wall, beam, pillar, ledge, etc.)?
b. Can drafts occur because of excessive cooling temperature differential and too low mounting height of the outlet?
c. Can drafts occur because of too low velocity causing a drop in cooling installations?
d. Will the outlet operate at too high a velocity and thereby cause an excessive increase in noise level?
e. Will the outlet operate against an excessive static pressure resistance?

Balancing the Sysem

In designing an air conditioning system it should be the aim of the engineer to size ducts and outlets in such a manner that proper distribution of supply air takes place. In practice, however, this is almost impossible and therefore additional means for regulating air distribution are required to *balance* the system. Some of these means are:

1. Reducing the effective area of some supply openings by blank-offs.
2. Placing dampers in the supply and return (exhaust) openings.
3. Placing dampers in the supply and return (exhaust) ducts.
4. Using combinations of dampers in both supply and return (exhaust) ducts.

In selecting the desired type of damper or balancing method, the following points should be kept in mind:

1. Unfavorable effect on air stream and noise level should be avoided. This will often eliminate blank-offs and dampers installed in the supply and return (exhaust) openings, unless such dampers are of special design.

2. It should be possible to alter the volume control setting and measure the amount of air handled without difficulty. This will be particularly difficult to achieve in the case of blank-offs.

Generally speaking, it is most satisfactory to install dampers in the supply duct at some distance back of the outlets, so as to avoid disturbing the air flow. Dampers in both supply and return air ducts, form the most flexible means of controlling supply of air to the room and static pressure within the room. Means of volume and directional control are discussed in detail in a following section of this chapter. Many types of air distribution control devices are now commercially available.

DIRECTIONAL AND VOLUME CONTROL

Duct Approaches to Outlets

In order to obtain proper direction of flow and distribution of air from outlets, it is necessary that the air stream approaching the outlet be of uniform velocity over the entire connection to duct, and perpendicular to the face.

Grilles and directional outlets cannot compensate for improper approach. Any attempt to secure a low face velocity and a high duct velocity by con-

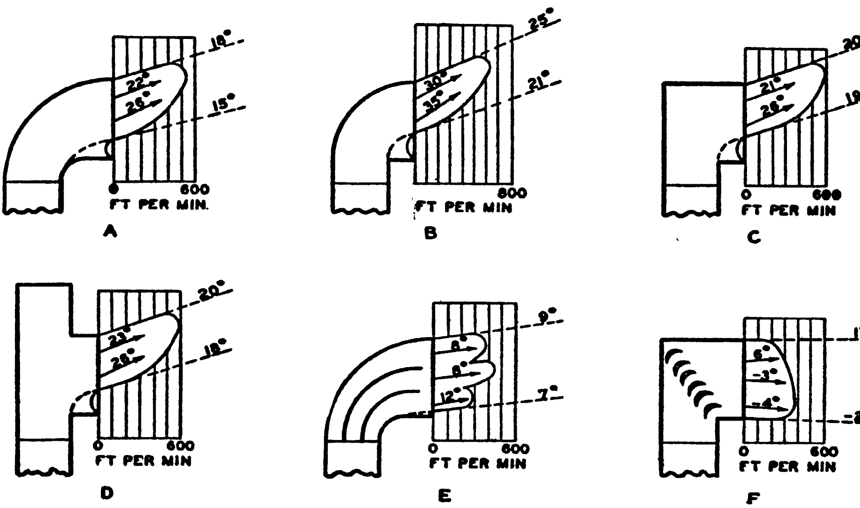


FIG. 6. OUTLET VELOCITY AND AIR DIRECTION DIAGRAMS FOR STACK HEADS WITH EXPANDING OUTLETS

Stack 14 in. x 6 in Outlets 14 in. x 9 in. Stack Velocity 500 fpm

A. Rounded Throat and Rounded Back.
B. Square Throat and Round Back.
C. Square Throat and Back.
D. Square Throat and Cushion Chamber.
E. Rounded Throat and Back and 2 Splitters.
F. Square Throat and Back and 6 Guide Vanes.

structing an expanding chamber directly behind the grille, is likely to be unsuccessful because the enlargement angle, even in a straight duct, cannot be greater than 7 deg at each side if the stream is to fill the outlet without turbulence.

In elbow outlets or stack heads at the top of vertical stacks, it is necessary to provide splitters or guide vanes in the elbows regardless of the shape of the elbows, whether of rounded, square or expanding types. Cushion chambers at the top of the stack heads have no beneficial effect. The direction of flow, distribution and velocity (measured 12 in. from outlet) of the air, based on tests,[5] are shown in Fig. 6 for various types of stack heads expanding from a 14 in. x 6 in. stack to 14 in. x 9 in. outlets, without grilles. The air velocity for each was 500 fpm in the stack below the elbow, but the direction of flow and the distribution patterns are generally indicative of performance obtainable with non-expanding elbows of similar shapes for a range of velocities 200 to 1400 fpm. Some of the conclusions drawn from the tests were:

1. Experiments with various elbow outlets on the 14 in. x 6 in. vertical stack[5] with stack air velocities of 200 to 1400 fpm, indicated that enlargement of the outlet area, whether used in connection with square or rounded elbows, would not reduce either the angle of discharge (which was 20 to 30 deg above the horizontal) or the outlet velocity. The effect of the enlargement of the outlet was mainly to increase the reverse flow area in the lower part of the outlet, but in each case enlargement of the outlet reduced the static pressure in the duct below the elbow.

2. Splitters in the elbows had the effect of dividing the air stream into a number of streams flowing through rounded elbows, and therefore lowered the angle of discharge, reduced or eliminated the reverse flow area, and made the outlet velocity quite uniform.

3. Turning vanes having 2 in. inner and 1 in. outer radii located in the center of the elbow were found most effective in improving performance in regard to angle of discharge, outlet velocity, and elimination of reverse flow area.

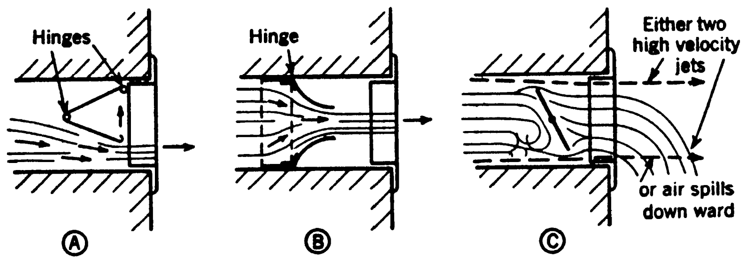


FIG. 7. EFFECT OF VARIOUS DAMPER ARRANGEMENTS DESIGNED FOR STRAIGHT BLOW

4. Pressure loss through stack heads may be reduced by use of splitters or turning vanes, or by increasing the inner radius of an elbow. Considering the sum of the velocity and static pressure as a measure of the energy required to change the direction of the air stream and to deliver the air into the atmosphere, and considering the energy required for a plain fitting as 100 percent, it was found that *turning vanes* dropped the energy requirement of square type stack heads to 45 percent. Splitters reduced the energy requirement to 90 percent in long radius elbows, and to 74 percent in short radius turns. In expanding heads, splitters reduced the energy requirement to 58 percent.

Side Outlets in Horizontal Air Ducts

When air is supplied to a room from side outlets in horizontal ducts, it is necessary to use directive devices within the duct at each outlet in order to obtain a uniform velocity of delivered air, and to obtain a direction of flow perpendicular to the face of the outlet. In tests[6] conducted with 3 in. x 10 in., 4 in. x 9 in., and 6 in. x 6 in. outlets in a 6 in. x 20 in. horizontal duct at duct velocities of 200 to 1400 fpm (in the 6 in. x 20 in. section) it was found that multiple curved deflectors produced the best flow characteristics. Vertical guide strips in the outlet were not so effective as curved deflectors. A single scoop type deflector at the outlet did not improve the flow pattern obtained from a plain outlet, and was therefore not found to be desirable.

Ceiling Outlets on Horizontal Ducts

Ceiling outlets are usually installed below horizontal supply ducts so that the supply air has to make a 90 deg turn before entering the outlet itself. The shorter the connection between bottom of duct and outlet, the greater is the need for directive devices to obtain uniformity of flow. Generally speaking, conditions and remedy in such cases strongly resemble those for side outlets in horizontal air ducts. Ceiling ducts often have a rectangular cross section, while the connections to the ceiling outlets are circular. It will then be quite difficult to install turning vanes successfully, particularly if the ducts are shallow and the connection areas are comparatively large. This will be the case when more than one outlet is installed on one duct run, and restrictions of duct area must be avoided. In such cases good results have been obtained by using a series of vertical guide strips, installed at right angles to the direction of air approach in the outlet connection where it leaves the horizontal air duct.

Volume Control

Various methods are used to regulate volume of supply and return (exhaust) air. Some of these accomplish only minor changes in volume; most

TABLE 1. RECOMMENDED RETURN INTAKE FACE VELOCITIES

Intake Location	Velocity Over Gross Area Fpm
Above occupied zone	800 up
Within occupied zone, not near seats	600–800
Within occupied zone, near seats	400–600
Door or wall louvers	500–700
Undercutting of doors (through undercut area)	600

of them however permit a range of adjustment from maximum air supply to complete shut-off.

When selecting type and location of such dampers, the following points must be considered, especially when the volume control feature is to be located near the air outlet itself: (1) deflection of air stream by the damper; (2) need and feasibility of directional control; (3) increase of noise level due to irregular and localized high air velocities caused by damper operation.

The following types of volume control are most frequently encountered:

1. *Slide Damper.* A single plate which can be pushed across the duct. Since its operation changes the free area of air passage in a one-sided manner, it should not be located near any air outlet, and its use is practicable only where no intermediate setting between full open and closed is required.

2. *Hit-and-Miss Damper.* Two slotted plates or discs, closely adjacent; by moving one of the two plates the respective slots may be opened or closed. This type of volume control may be installed close to an air outlet and it is easy to operate, but its main disadvantage is that even in the open position the air passage area is blocked by at least 50 percent. This requires oversizing of the air outlet in order to avoid excessive increase of noise level.

3. *Splitter Damper.* A single blade sheet metal plate hinged at one edge, usually located at the branch connection of a duct or outlet. It is easy to operate, but often causes irregular air flow in the duct. When used in connection with, and near an outlet, additional directional control is required.

4. *Butterfly Damper.* A single blade sheet metal plate hinged in the middle, usually located in a straight duct run. It is easier to handle than a splitter damper, since only half the motion is necessary to change its setting. However, if located too close to an air outlet, it is objectionable because its operation frequently results in a

condition whereby two high velocity jets are created along the sides of the duct, or the air spills immediately downward into the occupied zone (See C Fig. 7).

5. *Louver Dampers*. Numerous designs have been developed incorporating a series of splitter or butterfly dampers across the duct or air outlet. Their main advantage consists in retaining greater uniformity of air flow, and in requiring less depth for installation. Some designs provide for louver blades moving in opposite directions, and while decreasing free air passage area, retain a constant air flow direction along the axis of the duct air outlet connection (see A and B in Fig. 7).

RETURN AND EXHAUST INTAKES

The *selection of return and exhaust intakes* depends on: (1) velocity in occupied zone near intake; (2) permissible pressure drop through intake; and (3) noise.

1. *Velocity*. The effect of air flow through return intakes upon air movement in the room is slight. Air handled by the intake is drawn from all directions, the velocity dropping off rapidly as distance from intake increases. The only locality where drafts may prove objectionable is adjacent to the intake. To prevent excessive air motion in this area due to the return intake, it is advisable to compute the total air motion toward the exhaust opening as outlined in Equation 14 where A is the exhaust wall area in square feet. Recommended return intake face velocities are given in Table 1.

TABLE 2. APPROXIMATE PRESSURE DROPS FOR LATTICE RETURN INTAKES
Inches Water Gage—Standard Air

Per Cent Free Area	Face Velocity, Fpm						
	400	500	600	700	800	900	1000
50	0.06	0.09	0.13	0.17	0.22	0.28	0.35
60	0.04	0.06	0.09	0.12	0.16	0.20	0.24
70	0.03	0.05	0.07	0.09	0.12	0.15	0.18
80	0.02	0.03	0.05	0.07	0.09	0.11	0.14

2. *Permissible Pressure Drop*. The permissible pressure drop will depend on the choice of the designer. Table 2 gives pressure drop through plain lattice intakes as a function of free area and face velocity.

Proper pressure drop allowance should be made for control or directive devices.

3. *Noise*. **The problem of *noise* generated by return intakes is the same as that for supply outlets. In computing resultant room noise levels from the operation of an air conditioning system, the return intake must be included as a part of the total grille area. The major difference between the supply outlets and return intakes is the frequent installation of the latter at ear level. When so located, it is recommended that the return intake velocity be not in excess of 75 percent of the maximum permissible outlet velocity.**

Outlet Location

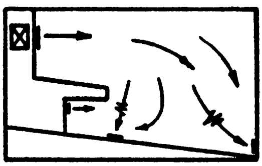
The control of the room air motion for the maintenance of comfort conditions depends on the proper selection of the *supply* outlets. The location of the return or exhaust intakes does not critically affect air motion, unless room air velocities in the occupied zone adjacent to the intake exceed comfort limits. The locations of return or exhaust intakes are, however, important for obtaining the desired room temperature equalization.

Ceiling locations for exhaust outlets are recommended for bars, kitchens, lavatories, dining rooms, club rooms, etc., where warm air will rise to the ceiling level. In heating installations, location of the return grilles in the ceiling or high on the wall will result in stratification of the conditioned air, and a high percentage of the heated air will be drawn into the return duct before it has served its purpose. (Refer also to considerations outlined previously in section *Outlet Location and Selection* in this chapter.)

Some circular ceiling outlets combine the supply and return openings in a single unit. The return duct is in the center with the supply pattern on the outside. This method gives best results for cooling applications. The application for heating is more critical and requires consideration of ceiling height, amount of outside wall area, and number of air changes required. In some cases, stratification of warm air may cause short circuiting. Where the wall losses are a small part of the total, little difficulty is encountered with stratification.

Floor locations of returns are used in heating installations for ceiling or side wall supply. When located so that air is drawn across exposed walls, the performance of the system may be somewhat improved. In general, floor locations tend to collect dirt and refuse.

Wall and door locations of exhaust outlets depending on their elevation, have the characteristics of either floor or ceiling returns. In large buildings


Rear wall distribution

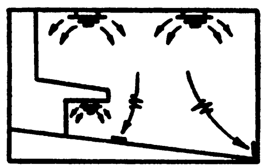

Ceiling distribution

FIG. 8. AIR DISTRIBUTION METHODS FOR THEATERS, CHURCHES, AND AUDITORIUMS

with many small rooms, the return air may be brought through door grilles or door undercuts into the corridors, and then to a common return or exhaust. The pressure drop through door returns should not be excessive; otherwise the air distribution to the room may be seriously unbalanced with the opening or closing of the doors. Outward leakage through doors or windows cannot be counted upon for dependable results.

SPECIFIC APPLICATIONS

For *theaters and auditoriums* the air distribution methods used are the downward distribution system with ceiling diffusers, and the horizontal distribution system with ejector nozzles or wall diffusers. Fig. 8 shows both methods. Ceiling distribution is accomplished by ceiling outlets under main ceiling and balcony. It is indicated when main ceiling or balcony is cut up by architectural treatment or beams. The only critical points are under the balcony, and (occasionally) above the very rear of the balcony, where ceiling heights are low and where direct impingement of air is sometimes a hazard.

Wall or ejector distribution is particularly applicable for relatively long and narrow theaters. It is essential with this type of distribution that there be no interference with the movement of air throughout its entire path from the high velocity nozzles to the front of the theater. The ceiling should be smooth, without projecting beams or obstructing ornamentation. For large

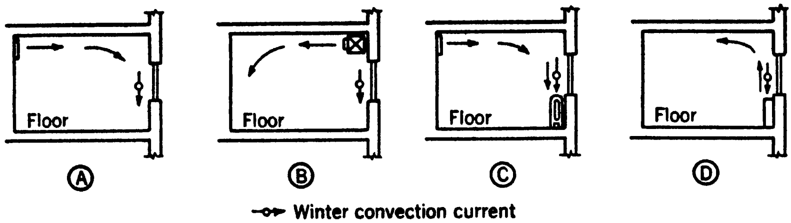


FIG. 9. DISTRIBUTION METHODS FOR SMALL ROOMS

A. Satisfactory for cooling. Unsatisfactory for heating in severe climates where the outside temperature is consistently below 40 F, and single glass and uninsulated walls are prevalent.

B. Performance approximately that of A when small diffusers are used in bottom of the duct.

C. Satisfactory for cooling. Satisfactory for heating if direct radiation is properly controlled.

D. Satisfactory for both cooling and heating. The air should be discharged slightly away from the wall, and for low velocities, should be fanned out parallel to the wall.

theaters, relatively high velocities can be used. These will work satisfactorily if adjustable outlets are used to avoid areas of local turbulence.

In small or medium size theaters, it is sometimes practicable to use side wall or front wall distribution. For the satisfactory operation of such a system during the winter heating period, the returns should preferably be located at the floor level and near the front of the theater to prevent cold spots which may result from exposed wall convection or infiltration from exits.

For *multi-room buildings* diagrams shown in Fig. 9 illustrate distribution methods for small rooms with exposed wall, such as offices, hotel (guest) rooms, hospital (patients) rooms, apartments, etc.

For *a small store* the cooling performance of various distribution methods

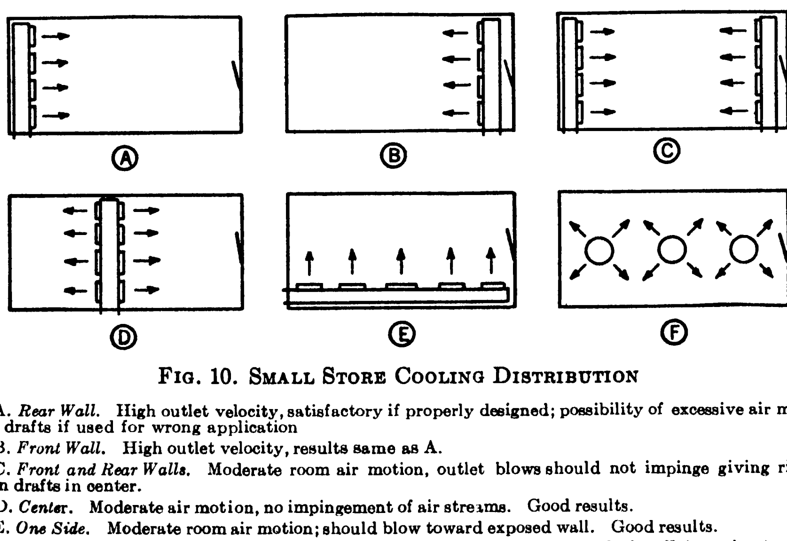


FIG. 10. SMALL STORE COOLING DISTRIBUTION

A. *Rear Wall.* High outlet velocity, satisfactory if properly designed; possibility of excessive air motion and drafts if used for wrong application

B. *Front Wall.* High outlet velocity, results same as A.

C. *Front and Rear Walls.* Moderate room air motion, outlet blows should not impinge giving rise to down drafts in center.

D. *Center.* Moderate air motion, no impingement of air streams. Good results.

E. *One Side.* Moderate room air motion; should blow toward exposed wall. Good results.

F. *Ceiling.* Low room air motion. Good results. Outlets should be selected of sufficient size to allow for blocking when not located in perfect squares.

is illustrated in Fig. 10. *Marine applications* of air distribution systems are given in Chapter 47.

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CHAPTER 31

AIR DUCT DESIGN

Pressure Losses, Friction Losses, Circular Equivalents of Rectangular Ducts, Dynamic Losses, Pressure Loss in Elbows, Losses Due to Area Changes, Pressure Changes, Duct Design Methods and Examples, Duct Construction Details, Heat Losses from Ducts, Maintenance

AIR ducts for the transmission of the air in forced air heating, ventilating, cooling, or air conditioning systems must be carefully designed for functional as well as economical reasons. The design should be based upon the fundamental laws of fluid flow in pipes, and should take into account recent analytical and experimental studies which complement and substantiate the fundamental laws. The basic equations of the flow of fluids will be found in Chapter 4, Fluid Flow.

PRESSURE LOSSES

Air ducts impose resistances to air flow which must be overcome by pressure differences resulting from the expenditure of energy in maintaining the flow. Since the flow of air, in ventilating and air conditioning work, takes place under very small pressure differences, the assumption that the gas density remains constant throughout the flow will cause only a negligible error. It is therefore possible to use the equation for incompressible fluids (liquids) for the flow of air in a duct, instead of the complicated thermodynamic formulas for air discharge under conditions of adiabatic flow, which would be necessary if pressure differences were large.

A reasonably precise estimate of the flow resistances offered by the system is essential for satisfactory duct design. The theoretical resistance of an air handling system can be computed from the methods and data given in this chapter. The actual resistance for any given installation, however, may vary considerably from the calculated resistance because of variation in the smoothness of materials, the type of joints used and the ability of the workmen to manufacture the system in accordance with the design. It is best to select fans and motors of sufficient size to provide a factor of safety. Dampers should be installed in each branch outlet to balance the system.

The drop in pressure in air transmission systems is due to *friction* losses and *dynamic* losses. Pressure increases and decreases may also be caused by changes in duct areas, with resulting conversion of velocity pressure to static pressure, and vice versa. The friction losses for turbulent flow (which occur in all practical air flow problems) are due to the friction of air against the sides of the duct, and to internal friction between the air molecules. The dynamic losses are caused by changes in the direction or in the velocity of air flow, and may be caused by changes in size and shape of the cross-section of the duct, by bends (elbows), and by obstructions to flow offered by dampers.

FRICTION LOSSES

Pressure drop in a *straight* duct is caused by surface friction, and this friction loss is most readily calculated by means of the Air Friction Charts,

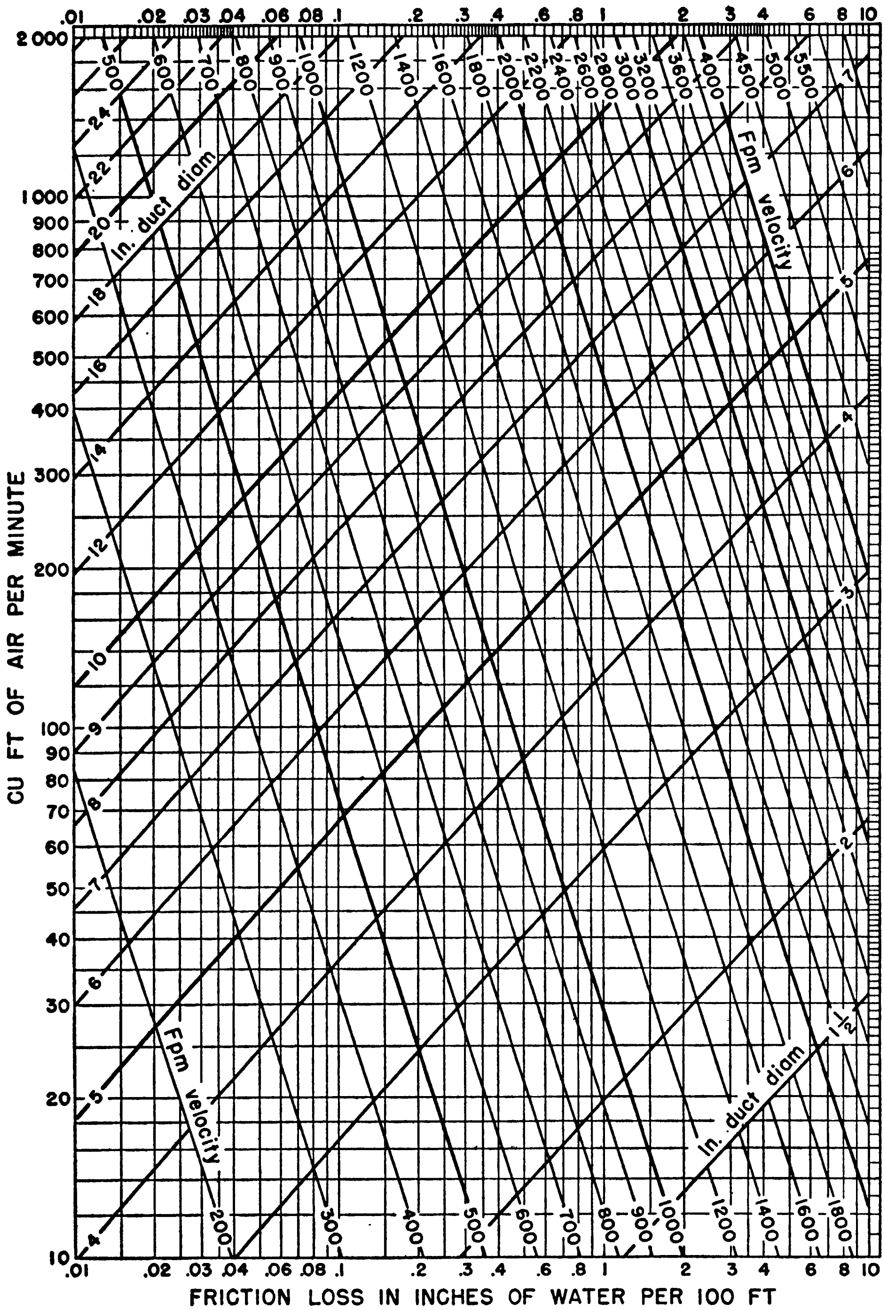


FIG. 1. FRICTION OF AIR IN STRAIGHT DUCTS

For Volumes of 10 to 2000 cfm

(Based on Standard Air of 0.075 lb per cu ft density flowing through average, clean, round, galvanized metal ducts having approximately 40 joints per 100 ft.) **No safety factor included.** Caution: Do not extrapolate below chart.

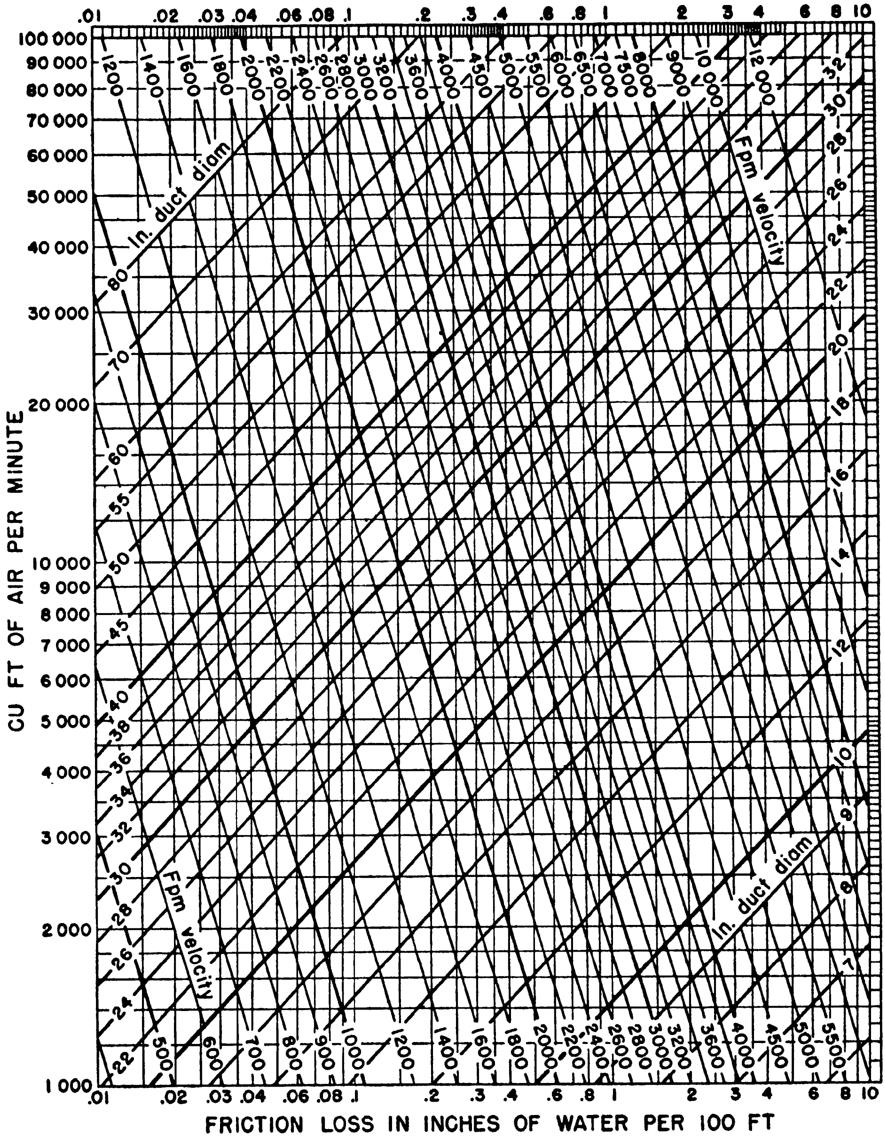


FIG. 2. FRICTION OF AIR IN STRAIGHT DUCTS

Vor Volumes of 1000 to 100,000 cfm

(Based on Standard Air of 0.075 lb per cu ft density flowing through average, clean, round, galvanized metal ducts having approximately 40 joints per 100 ft.) **No safety factor included.**

Figs. 1 and 2, covering volume ranges of 10 to 2000 cfm, and 1000 to 100,000 cfm, respectively. These charts were developed by the A.S.H.V.E. Research Laboratory.[1] *They do not include any safety factor.*

The charts, Figs. 1 and 2, were constructed from the basic flow equation for the pressure loss in circular ducts (see Chapter 4):

$$h_f = f \frac{l}{D} \frac{v^2}{2g} \tag{1}$$

where

h_f = head loss due to friction, in feet of fluid flowing.

l = length of conduit, feet.

D = inside diameter of conduit, feet.

v = mean fluid velocity, feet per second.

g = acceleration due to gravity, 32.17 feet per (second) (second).

f = a non-dimensional friction coefficient, which for ventilation work depends upon Reynolds Number and the relative roughness of the conduit. Appropriate values of f were taken from the work of Moody[2] where ϵ = 0.0005 ft. See Chapter 4, Fig. 4, Relation Between Friction Factor and Reynolds Number.

The air friction chart is based on standard air[3] with a density of 0.075 lb per cu ft, flowing through average, clean, round, galvanized metal ducts having approximately 40 joints per 100 ft. Fig. 1 should not be used to obtain values below the charts by extrapolation, because critical flow would occur in this region and values so obtained would be unreliable. For the average application, values from the charts should have sufficient precision, without corrections, for any air temperature from 50 F to 90 F, for any relative humidity, and for any normal variation in barometric pressure. For widely varying air pressures or temperatures, or for unusual duct conditions, the friction values obtained from the chart should be corrected.[4]

For ordinary ventilating work, friction may be assumed to vary directly as the density without serious error, and therefore

$$h_o = h_s \left(\frac{\rho_o}{\rho_s}\right) \tag{2}$$

where

h_o = friction loss under actual operating conditions, any consistent units.

h_s = friction loss under standard conditions, any consistent units.

ρ_o = density of air under actual operating conditions, any consistent units.

ρ_s = density of air under standard conditions, any consistent units.

For ducts of other than standard sheet metal construction, correction factors may be obtained from Fig. 3.[4] The correction factors shown in Fig. 3 were computed for the values of ϵ, the roughness in feet, shown in Table 1.[4] The correct friction loss for such ducts may then be determined by multiplying the losses obtained from Figs. 1 and 2 by these factors.

Examples 1 and 2 illustrate the use of Fig. 2 to determine friction loss, and the use of Fig. 3 to apply a correction for roughness.

Example 1: Determine the friction loss when circulating 10,000 cfm of air through 75 ft of 24 in. diameter galvanized duct.

Solution: Find 10,000 cfm on the left scale of Fig. 2 and move horizontally right to the diagonal line marked 24 in. The other intersecting diagonal shows that the velocity in the pipe is 3200 fpm. Directly below the intersection it is found that the friction per 100 ft is 0.50 in.; then for 75 ft the friction will be 0.75 × 0.50 = 0.38 in. In a like manner, any two variables may be determined by the intersection of the lines representing the other two variables.

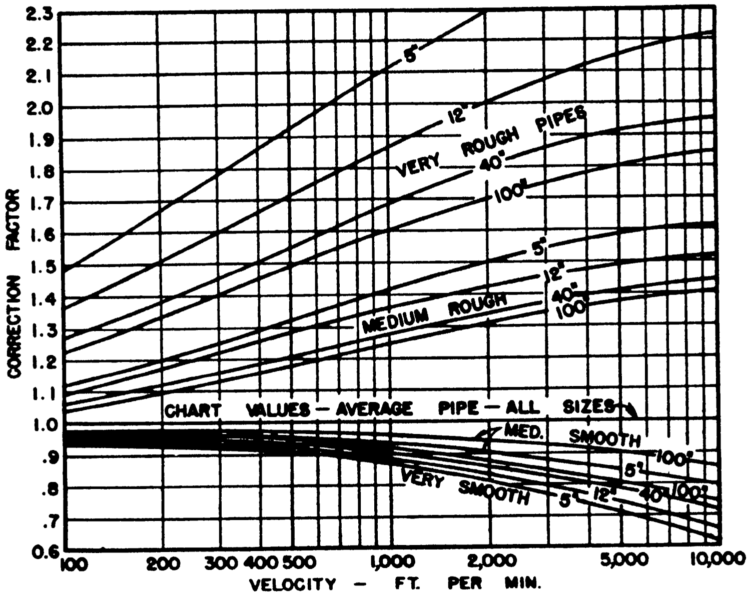


FIG. 3. CORRECTION FACTORS FOR PIPE ROUGHNESS

To correct for pipe roughness multiply friction loss obtained from Figs. 1 and 2 by correction factor obtained from Fig. 3.

Example 2: If the duct in Example 1 is very rough, instead of galvanized, with 40 joints per 100 ft, find the total friction.

Solution: On Fig. 3 find (by interpolation between 12 in. and 40 in. pipe) the intersection of the 24 in. *very rough pipe* line and the 3200 fpm velocity ordinate, and at the left margin read a correction factor of 2. The friction loss in the rough duct is therefore $2 \times 0.38 = 0.76$ in.

CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS

An air handling system is usually sized first for round ducts. Then, if rectangular ducts are desired, their sizes are selected to provide air carrying capacities equivalent to those of the round ducts originally selected.

A recent comprehensive study at the A.S.H.V.E. Research Laboratory proved that for most practical purposes rectangular ducts of aspect ratios not exceeding 8:1 will have the same static friction pressure loss for equal

TABLE 1. VALUES OF ROUGHNESS ϵ FOR DIFFERENT PIPES[a]

PIPE	DEGREE OF ROUGHNESS	ϵ ROUGHNESS IN FEET
Drawn Tubing	Very smooth	0.0000015
New Steel or Wrought-Iron Pipe	Medium smooth	0.00015
Galvanized Iron	Average	0.0005
Average Concrete	Medium rough	0.003
Average Riveted Steel	Very rough	0.01

[a] Used in computing values for Fig. 3.

lengths and mean velocities of flow as a circular duct of the same hydraulic diameter. When duct sizes are expressed in terms of hydraulic diameter, and when equations for friction loss in round and rectangular ducts are equated for equal capacity and equal length, an equation giving the circular equivalent of a rectangular duct is obtained[5] (Equation 3).

$$d_0 = 1.30 \frac{(ab)^{0.625}}{(a+b)^{0.250}} = 1.30 \sqrt[8]{\frac{(ab)^5}{(a+b)^2}} \qquad (3)$$

where

a = length of one side of rectangular duct, inches. (Other side is b.)

b = length of one side of rectangular duct, inches. (Other side is a.)

d_0 = circular equivalent of a rectangular duct for equal friction and capacity, inches.

Table 2 gives the circular equivalents of rectangular ducts for equal friction and capacity for aspect ratios not greater than 11.7:1 based on Equation 3.[5]

Multiplying or dividing the length of each side of a duct by a constant is the same as multiplying or dividing the equivalent round size by the same constant. Thus, if the circular equivalent of an 80 x 24 in. duct is required, it will be twice that of 40 x 12 in. duct, or 2 x 23.0 = 46.0 in.

DYNAMIC LOSSES

Wherever eddying flow is present, brought about by sudden changes in the direction or magnitude of the velocity of the air flowing, a greater loss in pressure takes place than would occur in a steady flow through a similar length of straight duct having a uniform cross-section. The amount of this loss, in excess of straight duct friction, is termed *dynamic loss*. Dynamic losses generally are greater with decelerating flow in duct enlargements than with accelerating flow in reducing fittings. Although dynamic

TABLE 2. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION AND CAPACITY

Dimensions in Inches

Side Rectangular Duct	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0	9.5	10.0
3.0	3.8	4.0	4.2	4.4	4.6	4.8	4.9	5.1	5.2	5.4	5.5	5.6	5.7
3.5	4.1	4.3	4.6	4.8	5.0	5.2	5.3	5.5	5.7	5.8	6.0	6.1	6.3
4.0	4.4	4.6	4.9	5.1	5.3	5.5	5.7	5.9	6.1	6.3	6.4	6.6	6.8
4.5	4.6	4.9	5.2	5.4	5.6	5.9	6.1	6.3	6.5	6.7	6.9	7.0	7.2
5.0	4.9	5.2	5.5	5.7	6.0	6.2	6.4	6.7	6.9	7.1	7.3	7.4	7.6
5.5	5.1	5.4	5.7	6.0	6.3	6.5	6.8	7.0	7.2	7.4	7.6	7.8	8.0

Side Rectangular Duct	10.0	10.5	11.0	11.5	12.0	12.5	13.0	13.5	14.0	14.5	15.0	15.5	16.0
3.0	5.7	5.9	6.0	6.1	6.2	6.3	6.4	6.5	6.6	6.7	6.8	6.9	7.0
3.5	6.3	6.4	6.5	6.7	6.8	6.9	7.0	7.1	7.2	7.3	7.4	7.5	7.6
4.0	6.8	6.9	7.1	7.2	7.3	7.5	7.6	7.7	7.8	7.9	8.1	8.2	8.3
4.5	7.2	7.4	7.5	7.7	7.8	8.0	8.1	8.2	8.4	8.5	8.6	8.7	8.9
5.0	7.6	7.8	8.0	8.1	8.3	8.4	8.6	8.7	8.9	9.0	9.1	9.3	9.4
5.5	8.0	8.2	8.4	8.6	8.7	8.8	9.0	9.2	9.4	9.5	9.6	9.8	9.8

TABLE 2. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION AND CAPACITY (CONTINUED)

Dimensions in Inches

Side Rectangular Duct	6	7	8	9	10	11	12	13	14	15	16	17	18	19
6	6.6													
7	7.1	7.7												
8	7.5	8.2	8.8											
9	8.0	8.6	9.3	9.9										
10	8.4	9.1	9.8	10.4	10.9									
11	8.8	9.5	10.2	10.8	11.4	12.0								
12	9.1	9.9	10.7	11.3	11.9	12.5	13.1							
13	9.5	10.3	11.1	11.8	12.4	13.0	13.6	14.2						
14	9.8	10.7	11.5	12.2	12.9	13.5	14.2	14.7	15.3					
15	10.1	11.0	11.8	12.6	13.3	14.0	14.6	15.3	15.8	16.4				
16	10.4	11.4	12.2	13.0	13.7	14.4	15.1	15.7	16.3	16.9	17.5			
17	10.7	11.7	12.5	13.4	14.1	14.9	15.5	16.1	16.8	17.4	18.0	18.6		
18	11.0	11.9	12.9	13.7	14.5	15.3	16.0	16.6	17.3	17.9	18.5	19.1	19.7	
19	11.2	12.2	13.2	14.1	14.9	15.6	16.4	17.1	17.8	18.4	19.0	19.6	20.2	20.8
20	11.5	12.5	13.5	14.4	15.2	15.9	16.8	17.5	18.2	18.8	19.5	20.1	20.7	21.3
22	12.0	13.1	14.1	15.0	15.9	16.7	17.6	18.3	19.1	19.7	20.4	21.0	21.7	22.3
24	12.4	13.6	14.6	15.6	16.6	17.5	18.3	19.1	19.8	20.6	21.3	21.9	22.6	23.2
26	12.8	14.1	15.2	16.2	17.2	18.1	19.0	19.8	20.6	21.4	22.1	22.8	23.5	24.1
28	13.2	14.5	15.6	16.7	17.7	18.7	19.6	20.5	21.3	22.1	22.9	23.6	24.4	25.0
30	13.6	14.9	16.1	17.2	18.3	19.3	20.2	21.1	22.0	22.9	23.7	24.4	25.2	25.9
32	14.0	15.3	16.5	17.7	18.8	19.8	20.8	21.8	22.7	23.6	24.4	25.2	26.0	26.7
34	14.4	15.7	17.0	18.2	19.3	20.4	21.4	22.4	23.3	24.2	25.1	25.9	26.7	27.5
36	14.7	16.1	17.4	18.6	19.8	20.9	21.9	23.0	23.9	24.8	25.8	26.6	27.4	28.3
38	15.0	16.4	17.8	19.0	20.3	21.4	22.5	23.5	24.5	25.4	26.4	27.3	28.1	29.0
40	15.3	16.8	18.2	19.4	20.7	21.9	23.0	24.0	25.1	26.0	27.0	27.9	28.8	29.7
42	15.6	17.1	18.5	19.8	21.1	22.3	23.4	24.5	25.6	26.6	27.6	28.5	29.4	30.4
44	15.9	17.5	18.9	20.2	21.5	22.7	23.9	25.0	26.1	27.2	28.2	29.1	30.0	31.0
46	16.2	17.8	19.2	20.6	21.9	23.2	24.3	25.5	26.7	27.7	28.7	29.7	30.6	31.6
48	16.5	18.1	19.6	20.9	22.3	23.6	24.8	26.0	27.2	28.2	29.2	30.2	31.2	32.2
50	16.8	18.4	19.9	21.3	22.7	24.0	25.2	26.4	27.6	28.7	29.8	30.8	31.8	32.8
52	17.0	18.7	20.2	21.6	23.1	24.4	25.6	26.8	28.1	29.2	30.3	31.4	32.4	33.4
54	17.3	19.0	20.5	22.0	23.4	24.8	26.1	27.3	28.5	29.7	30.8	31.9	32.9	33.9
56	17.6	19.3	20.9	22.4	23.8	25.2	26.5	27.7	28.9	30.1	31.2	32.4	33.4	34.5
58	17.8	19.5	21.1	22.7	24.2	25.5	26.9	28.2	29.3	30.5	31.7	32.9	33.9	35.0
60	18.1	19.8	21.4	23.0	24.5	25.8	27.3	28.7	29.8	31.0	32.2	33.4	34.5	35.5
62	18.3	20.1	21.7	23.3	24.8	26.2	27.6	29.0	30.2	31.4	32.6	33.8	35.0	36.0
64	18.6	20.3	22.0	23.6	25.2	26.5	27.9	29.3	30.6	31.8	33.1	34.2	35.5	36.5
66	18.8	20.6	22.3	23.9	25.5	26.9	28.3	29.7	31.0	32.2	33.5	34.7	35.9	37.0
68	19.0	20.8	22.5	24.2	25.8	27.3	28.7	30.1	31.4	32.6	33.9	35.1	36.3	37.5
70	19.2	21.	22.8	24.5	26.1	27.6	29.1	30.4	31.8	33.1	34.3	35.6	36.8	37.9

losses may be assumed to be caused by changes in area actually occupied by the air flow, for convenience they are divided into two general classes: (1) those caused by *changes in direction* of the duct at bends and branches, and (2) those caused by *changes in cross-sectional area* of the duct at transitions.

Dynamic losses vary substantially as the square of the mean velocity of

TABLE 2. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION AND CAPACITY (CONCLUDED)

Dimensions in Inches

Side Rectangular Duct	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48	50	52	56	60	64	68	72	76	80	84	88	Side Rectangular Duct
20	21.9																										20
22	22.9	24.1																									22
24	23.9	25.1	26.2																								24
26	24.8	26.1	27.2	28.4																							26
28	25.7	27.1	28.2	29.5	30.6																						28
30	26.7	28.0	29.3	30.5	31.6	32.8																					30
32	27.5	28.9	30.1	31.4	32.6	33.8	35.0																				32
34	28.3	29.7	31.0	32.3	33.6	34.8	36.0	37.2																			34
36	29.0	30.5	32.0	33.0	34.6	35.8	37.0	38.2	39.4																		36
38	29.8	31.4	32.8	34.2	35.5	36.7	38.0	39.2	40.4	41.6																	38
40	30.5	32.1	33.6	35.1	36.4	37.6	39.0	40.2	41.4	42.6	43.8																40
42	31.2	32.8	34.4	35.9	37.3	38.6	39.9	41.1	42.4	43.6	44.8	45.9															42
44	31.9	33.5	35.2	36.7	38.1	39.5	40.8	42.0	43.4	44.6	45.8	46.9	48.1														44
46	32.5	34.2	35.9	37.4	38.9	40.3	41.7	43.0	44.3	45.6	46.8	47.9	49.1	50.3													46
48	33.1	34.9	36.6	38.2	39.7	41.2	42.6	43.9	45.2	46.5	47.8	48.9	50.2	51.3	52.6												48
50	33.7	35.5	37.3	38.9	40.4	42.0	43.5	44.8	46.1	47.4	48.8	49.8	51.2	52.3	53.6	54.7											50
52	34.3	36.2	38.0	39.6	41.2	42.8	44.3	45.7	47.1	48.3	49.7	50.8	52.2	53.3	54.6	55.8	56.9										52
54	34.9	36.8	38.7	40.3	42.0	43.6	45.0	46.5	48.0	49.2	50.6	51.8	53.2	54.3	55.6	56.8	57.9										54
56	35.5	37.4	39.3	41.0	42.7	44.3	45.8	47.3	48.8	50.1	51.5	52.7	54.1	55.3	56.5	57.8	58.9	61.3									56
58	36.0	38.0	39.8	41.7	43.4	45.0	46.6	48.1	49.6	51.0	52.4	53.7	55.0	56.2	57.5	58.8	60.0	62.3									58
60	36.5	38.6	40.4	42.3	44.0	45.8	47.3	48.9	50.4	51.8	53.3	54.6	55.9	57.1	58.5	59.8	61.0	63.3	65.7								60
62	37.1	39.2	41.0	42.9	44.7	46.5	48.0	49.7	51.2	52.6	54.2	55.5	56.8	58.0	59.4	60.7	62.0	64.3	66.7								62
64	37.6	39.7	41.6	43.5	45.4	47.2	48.7	50.4	52.0	53.4	55.0	56.4	57.7	59.0	60.3	61.6	62.9	65.3	67.7	70.0							64
66	38.1	40.2	42.2	44.1	46.0	47.8	49.5	51.1	52.8	54.2	55.8	57.2	58.6	59.9	61.2	62.5	63.9	66.3	68.7	71.1							66
68	38.6	40.7	42.8	44.7	46.6	48.4	50.2	51.8	53.5	55.0	56.6	58.0	59.5	60.8	62.1	63.4	64.8	67.3	69.7	72.1	74.4						68
70	39.1	41.3	43.3	45.3	47.2	49.0	50.9	52.5	54.2	55.8	57.3	58.8	60.3	61.7	63.0	64.3	65.7	68.3	70.7	73.1	75.4						70
72	39.6	41.8	43.8	45.9	47.8	49.7	51.5	53.2	54.9	56.5	58.0	59.6	61.1	62.6	63.9	65.2	66.6	69.2	71.7	74.1	76.4	78.8					72
74	40.0	42.3	44.4	46.4	48.4	50.3	52.1	53.9	55.6	57.2	58.8	60.4	61.9	63.3	64.8	66.1	67.5	70.1	72.7	75.1	77.4	79.9					74
76	40.5	42.8	44.9	47.0	49.0	50.8	52.7	54.6	56.3	57.9	59.5	61.2	62.7	64.1	65.6	67.0	68.4	71.0	73.6	76.1	78.4	80.9	83.2				76
78	40.9	43.3	45.5	47.5	49.5	51.5	53.3	55.2	57.0	58.6	60.3	62.0	63.4	64.9	66.4	67.9	69.3	71.8	74.5	77.1	79.4	81.8	84.2				78
80	41.3	43.8	46.0	48.0	50.1	52.0	53.9	55.8	57.6	59.3	61.0	62.7	64.1	65.7	67.2	68.7	70.1	72.7	75.4	78.1	80.4	82.8	85.2	87.5			80
82	41.8	44.2	46.4	48.6	50.6	52.6	54.5	56.4	58.2	60.0	61.7	63.4	64.9	66.5	68.0	69.5	71.0	73.6	76.3	79.0	81.4	83.8	86.2	88.6			82
84	42.2	44.6	46.9	49.2	51.1	53.2	55.1	57.0	58.9	60.7	62.4	64.1	65.7	67.3	68.8	70.3	71.8	74.5	77.2	79.9	82.4	84.8	87.2	89.6	91.9		84
86	42.6	45.0	47.4	49.6	51.6	53.7	55.7	57.6	59.5	61.3	63.0	64.8	66.4	68.0	69.5	71.1	72.6	75.4	78.1	80.8	83.3	85.8	88.2	90.6	92.9		86
88	43.0	45.4	47.9	50.1	52.2	54.3	56.3	58.2	60.1	62.0	63.7	65.4	67.0	68.7	70.3	71.8	73.4	76.3	79.0	81.6	84.2	86.8	89.2	91.6	93.9	96.3	88
90	43.4	45.9	48.3	50.6	52.8	54.8	56.9	58.8	60.7	62.6	64.4	66.0	67.8	69.4	71.1	72.6	74.2	77.1	79.9	82.5	85.1	87.8	90.2	92.6	94.9	97.3	90
92	43.8	46.3	48.7	51.1	53.4	55.4	57.4	59.4	61.3	63.2	65.0	66.8	68.5	70.1	71.8	73.3	74.9	77.9	80.8	83.4	86.0	88.7	91.2	93.6	95.9	98.3	92
94	44.2	46.7	49.1	51.6	53.9	55.9	57.9	60.0	61.9	63.8	65.6	67.5	69.2	70.8	72.5	74.1	75.6	78.7	81.7	84.3	86.9	89.6	92.1	94.6	96.9	99.3	94
96	44.6	47.2	49.5	52.0	54.4	56.3	58.4	60.5	62.4	64.4	66.2	68.2	69.8	71.5	73.2	74.3	76.3	79.4	82.6	85.2	87.8	90.5	93.0	95.6	97.9	100.3	96

the air, and are therefore conveniently expressed as a fraction of the velocity head:

$$h_v = C\frac{v^2}{2g} \tag{4}$$

and for standard air

$$H_v = C\left(\frac{V}{4005}\right)^2 \tag{5}$$

where

h_v = dynamic pressure loss, feet of fluid flowing.
H_v = dynamic pressure loss, inches of water.
v = velocity of fluid, feet per second.

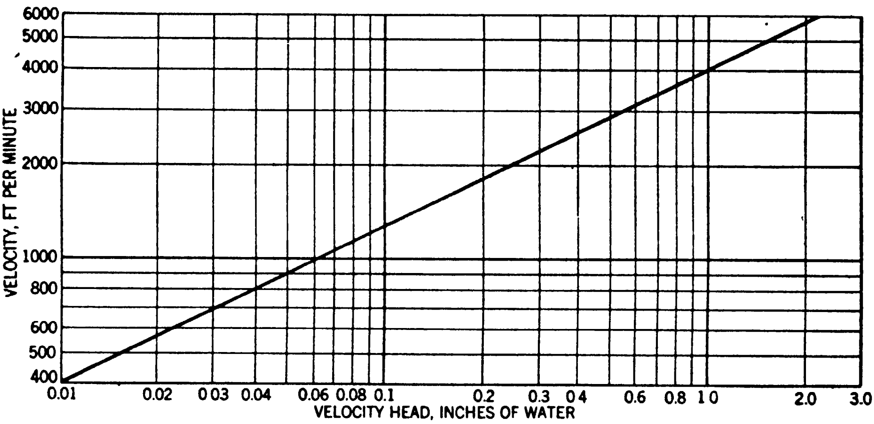


FIG. 4. RELATION BETWEEN VELOCITY AND VELOCITY HEAD FOR STANDARD AIR

$\frac{v^2}{2g}$ = the velocity pressure corresponding to the mean velocity of flow, feet of fluid flowing.
V = mean velocity of standard air, feet per minute.
C = an experimentally determined constant (dynamic loss coefficient).

It can be seen from Equation 4 that the dynamic loss coefficient is independent of both density and the units used, and that it represents the number of velocity heads lost at the conduit transition or bend. Values of the dynamic loss coefficient for various duct elements are sometimes tabulated,[6, 7, 8] though it should be kept in mind that absolutely reliable dynamic loss coefficients have not yet been fully established for all duct elements.

Fig. 4, which shows the relation of velocity pressure to velocity for standard air ($V = 4005\sqrt{H_v}$), can be conveniently used to find the total dynamic pressure loss for any duct element with known *dynamic loss coefficient C.* This coefficient is nearly independent of the air velocity and the roughness of the duct walls; therefore dynamic losses cannot theoretically be computed as friction losses. For duct components where intense eddying flow is not appreciable, as in elbows of good design, it is customary to include the dynamic loss with the friction loss, thereby facilitating design calculations.

PRESSURE LOSSES IN ELBOWS

It is convenient to express the combined dynamic and friction losses due to an elbow as equivalent to the loss in a length L of similar straight duct. A recent A.S.H.V.E. survey[9] of available data has indicated that this method of expressing the loss is justified for design purposes, owing to the relation of the loss to the corresponding friction factor, f.

Fig. 5 gives the *additional equivalent length* of duct in terms of widths W for elbows in rectangular ducts; Fig. 6 gives the equivalent length in terms

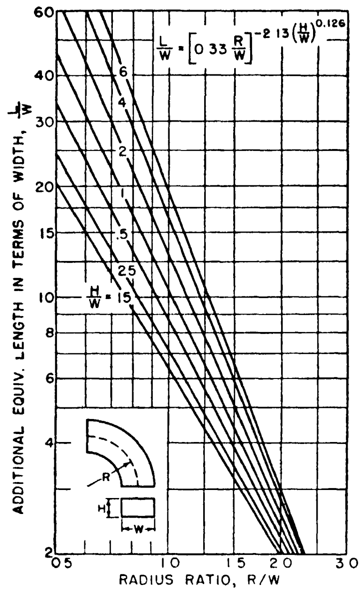


FIG. 5. LOSS IN 90-DEG ELBOWS OF RECTANGULAR CROSS-SECTION

of diameters D for round ducts. When these curves for *additional* equivalent length are used, the straight lengths of duct between elbows should be measured to the intersection of their center lines. The data of Figs. 5 and 6 may be readily converted[9] to the loss as a percentage of the velocity head.

Example 3: (Use of Fig. 5 for the calculation of elbow losses.)

Given the portion of a duct system shown in Fig. 7, it is required to determine the pressure loss between points A and D. Air at standard conditions is being supplied at the rate of 2000 cfm in a 6 by 24 in. galvanized duct of average construction. Elbows No. 1 and 2 have centerline radii of 18 and 9 in., respectively.

Solution: For elbow No. 1 the radius ratio is $\frac{R_1}{W_1} = \frac{18}{24} = 0.75$ and the aspect ratio is $\frac{H_1}{W_1} = \frac{6}{24} = 0.25$. The additional equivalent length for elbow No. 1 in terms of W is obtained from Fig. 5: $(L/W)_1 = 11.5$. Thus $L_1 = 11.5 \times 24/12 = 23$ additional equivalent feet. Similarly for elbow No. 2, the ratio radius is $\frac{R_2}{W_2} = \frac{9}{6} = 1.5$ and the aspect ratio is $\frac{H_2}{W_2} = \frac{24}{6} = 4.0$; Fig. 5 gives $(L/W)_2 = 6$, so $L_2 = 6 \times 6/12 = 3$ additional equivalent feet.

The total length of the straight runs from A to D is $l = l_{A-B} + l_{B-C} + l_{C-D} = 7 + 20 + 5 = 32$ ft and the additional equivalent length due to the elbows is $L = L_1 + L_2 = 23 + 3 = 26$ ft. Thus the equivalent length of the system from A to D is $l + L = 32 + 26 = 58$ ft of 6 by 24 in. duct.

The diameter of a circular duct, equivalent in friction and capacity to this rectangular duct, is 12.4 in. as given by the table of circular equivalents, Table 2. At a delivery rate of 2000 cfm, the A.S.H.V.E. Friction Chart, Fig. 2, gives a loss of 0.6 in. of water per 100 ft of 12.4 in. diameter duct. Thus the loss from A to D is $0.6 \times 58/100 = 0.348$ in. of water.

The use of elbows of radius ratio, $R/W = 1.5$, is considered good practice, with respect to both installation and operation. In a given rectangular duct 6 by 24 in., for example, the additional equivalent length L necessary to represent the elbow loss will generally be greater for a flat bend where the

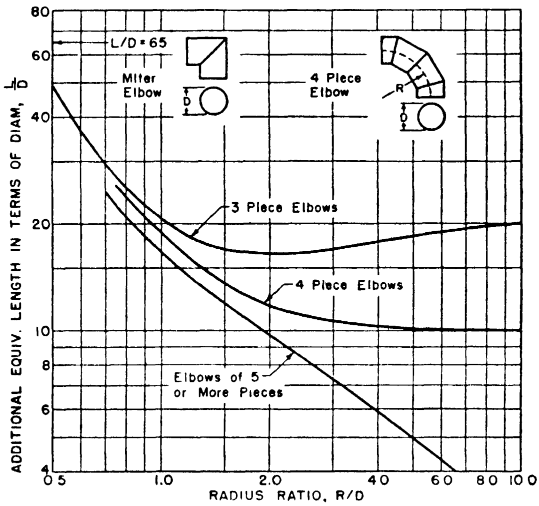


FIG. 6. LOSS IN 90-DEG ELBOWS OF ROUND CROSS-SECTION

aspect ratio, $H/W = 1/4$, than if the bend of the same radius ratio had been made in the plane of the narrow dimension giving an aspect ratio, $H/W = 4$.

Data presently available for losses in compound bends,[10, 11] where two or more elbows are close together, do not warrant refinement of design calculations beyond use of the sum of the additional equivalent lengths L for the individual elbows. Where angles of other than 90-deg bend are encountered, the loss may be considered as directly proportional to the angle of bend. Losses[11] for elbows discharging air directly into a large space are higher than those indicated in Figs. 5 and 6 for elbows within duct systems. Data[12] for losses at branch take-offs are at present quite meager. An A.S.H.V.E. cooperative investigation is underway for the purpose of obtaining more data on losses in typical take-off fittings.

Turning vanes may be advantageously employed in elbows, both to reduce the pressure loss and to provide a more uniform velocity distribution downstream from the bend. Vanes and concentric splitters are particularly recommended where miter elbows are used, because even the simplest vane

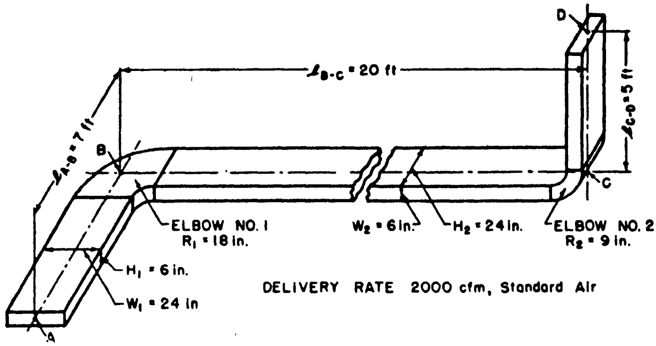


FIG. 7. PORTION OF DUCT SYSTEM FOR EXAMPLE 3

forms will produce a substantial saving in pressure loss. The addition of vanes or splitters divides an elbow into parallel channels, each having more favorable radius and aspect ratios than the original elbow. Values of additional equivalent length L, for elbows of square cross-section having various vane forms and combinations, may be found from Table 3 in which values of L/W are shown.

LOSSES DUE TO AREA CHANGES

Area changes in ducts, generally unavoidable, are necessitated frequently by the building construction or changes in the volume of air carried. Experimental investigations[14,15] of pressure changes, and pressure losses at changes of the area of duct cross-sections, indicate that the excess pressure loss over the normal friction loss is a dynamic loss due to a faster stream expanding into a slower stream, as determined by the actual areas occupied by the flow rather than the areas of the duct. No perceptible dynamic loss is due to the converging of the air stream itself where the flow is contracted, but the air stream continues to converge beyond the edge of the contraction and reaches a minimum at the *vena contracta.* This contraction of the air stream is shown in Fig. 8. For contraction, therefore, the dynamic loss is caused by expansion from the vena contracta to the full area following the contraction. Enlargement in area may be considered as a special condition of general expansion following contraction. Fig. 8 illustrates (a) abrupt enlargement and (b) abrupt contraction.

For a sudden symmetrical enlargement, a theoretical expression for the loss is

$$h_e = \left(1 - \frac{A_1}{A_2}\right)^2 \frac{v_1^2}{2g} = \frac{(v_1 - v_2)^2}{2g} \qquad (6)$$

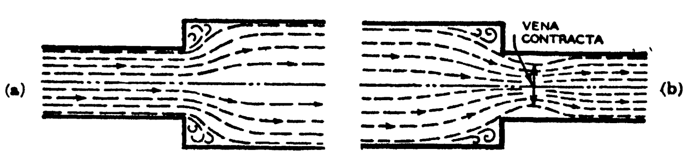


FIG. 8. AIR FLOW AT ABRUPT ENLARGEMENT OR CONTRACTION OF AIR STREAM

or for standard air;

$$H_e = \left(1 - \frac{A_1}{A_2}\right)^2 \left(\frac{V_1}{4005}\right)^2 = \left(\frac{V_1 - V_2}{4005}\right)^2 \tag{7}$$

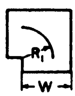
where

h_e = pressure loss due to sudden enlargement, feet of fluid flowing.
H_e = pressure loss due to sudden enlargement, based on standard air, inches of water.

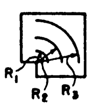
TABLE 3. PRESSURE LOSS IN VANED ELBOWS OF SQUARE CROSS-SECTION EXPRESSED IN ADDITIONAL EQUIVALENT DUCT LENGTH[a, b]

Additional Equivalent Length L = Duct Width W, in Feet, Multiplied by L/W Values Shown

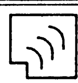
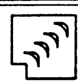
MITER ELBOW



R1/W	0	.2	.4	.6	.8	1.0
L/W	70	34	28	33	54	60

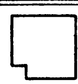


R1/W	0	.2	.3	.2	.3
R2/W	0	.4	.5	.4	.5
R3/W	0	0	0	.6	.7
L/W	70	22	22	18	20

	A	B	C
L/W	20	14	15

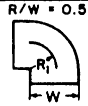




	D	E	F
L/W	15	28	70

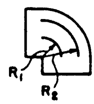
ELBOWS WITH VARIOUS RADIUS RATIOS

R/W = 0.5



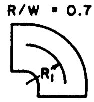
R1/W	0	.2	.4	.6	.8	1.0
L/W	60	20	19	24	30	60

R/W = 0.5



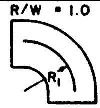
R1/W	0	.2	.3	.4	.5	.6
R2/W	0	.4	.5	.6	.7	.8
L/W	60	16	19	20	21	24

R/W = 0.7



R1/W	0	.4	.6	.8	1.0	1.2
L/W	24	13	12	14	21	24

R/W = 1.0



R1/W	0	.7	.8	.9	1.0	1.2
L/W	10	8.0	8.0	7.4	7.2	7.4

[a] These values are based upon vane test data of Reference 13 which have been modified by the finding of Reference 9.

[b] Vanes: A = a large number of small arc vanes, B = a small number of large arc vanes; C = hollow vanes having different outside and inside curvature; D = four vanes with radius of 0.4 W; E = single splitter with radius of 0.5 W, F = no vanes or splitters.

v_1 = velocity in the inlet duct, feet per second.
v_2 = velocity in the outlet duct, feet per second.
V_1 = velocity of standard air in the inlet duct, feet per minute.
V_2 = velocity of standard air in the outlet duct, feet per minute.
A_1 = area of the inlet duct, square feet.
A_2 = area of the outlet duct, square feet.

The loss for a sudden symmetrical contraction, h_c can similarly be expressed as

$$h_c = \left(1 - \frac{A_2'}{A_1}\right)^2 \frac{(v_2')^2}{2g} \tag{8}$$

where

h_c = pressure loss due to sudden contraction, feet of fluid flowing.
v'_2 = the velocity at the vena contracta, feet per second.
A'_2 = the area of the vena contracta, square feet.

Introduction of the *contraction coefficient* $\alpha = \frac{A_2'}{A^1}$ and the *loss coefficient* $C = \left(\frac{1}{\alpha} - 1\right)^2$ in Equation 8 gives (with $A_2' \times v_2' = A_1 \times v_1$):

$$h_c = C \frac{v_1^2}{2g} \quad (9)$$

or for standard air:

$$H_c = C\left(\frac{V_1}{4005}\right)^2 \quad (10)$$

where

H_c = pressure loss due to sudden contraction, inches of water.
V_1 = velocity of standard air in the inlet duct, in feet per minute.

Values of α and C for sharp corners for increasing ratios of A_2/A_1 are given in the following table:

A_2/A_1	0.01	0.1	0.2	0.4	0.6	0.8	1.00
α	0.6	0.61	0.62	0.65	0.7	0.77	1.00
C	0.44	0.41	0.37	0.29	0.19	0.09	0.00

For discharge to atmosphere from a pipe, $C = 1.0$ in Equation 10.

For a gradual enlargement, Equation 7 changes to

$$H_{ge} = C_1\left(\frac{V_1 - V_2}{4005}\right)^2 \quad (11)$$

where

H_{ge} = pressure loss due to gradual enlargement, based on standard air, inches of water.
C_1 = coefficient of loss, dependent upon the total angle included between the sides of the duct.

Values for C_1 are given in the following table:

Total Included Angle, Degrees	5	7	10	20	30	40	50	60
C_1	0.20	0.15	0.16	0.35	0.65	0.80	0.92	1.0

Pressure losses for various duct transitions and area changes have been determined experimentally, although the available information is generally restricted to symmetrical area changes.[6, 7, 14, 15, 16, 17]

PRESSURE CHANGES

The fundamental energy equation for standard air flow in a horizontal duct can be written[18]

$$H_1 + \left(\frac{V_1}{4005}\right)^2 = H_2 + \left(\frac{V_2}{4005}\right)^2 + \text{Loss of pressure (head), inches of water} \quad (12)$$

where

H_1 and H_2 = the static pressure (head) at two given points (1) and (2), inches of water.

$\left(\frac{V_1}{4005}\right)^2$ and $\left(\frac{V_2}{4005}\right)^2$ = the velocity pressure (head) at the same points, inches of water.

Equation 12 states that the mechanical energy at a given point (1) must be equal to the mechanical energy at another point (2), plus any dissipation of mechanical energy to internal energy (loss of pressure). Equation 12 is valid only if no work is done by or upon the air between the sections (1) and (2), and if there is no heat transfer to or from the air.

In Equation 12, H is a measure of the potential energy and $\left(\frac{4005}{v}\right)^2$ a measure of the kinetic energy or energy of motion. The sum of static pressure and velocity pressure is called total pressure, and is a measure of the total energy.

Static pressure and velocity pressure are mutually convertible, that is to say, static pressure may be converted into velocity pressure, and vice versa. Every change in the cross-sectional area of a duct results in such a conversion of energy and is always accompanied by some loss in efficiency, or loss in total pressure.

In the final analysis of pressure losses in ducts, dynamic losses are due to accelerations and decelerations of the air stream as a whole. In a *converging* duct, the air velocity will be accelerated; some pressure head will be converted into velocity pressure. This conversion is generally a stable and efficient process, the energy losses are small, and there is no eddy formation.

In an *expanding* duct section, on the other hand, the air will be decelerated and an opposing pressure gradient be required to reduce the velocity. If the angle of divergence is appreciable, the flow becomes unstable, there is danger of separation of the flow from the duct wall, and large energy losses and eddy formation are possible.[19]

In order to keep losses in an expanding duct section to a minimum and to convert the velocity pressure efficiently into static pressure, the angle of divergence should be kept small.[20] Theoretically, it might seem possible to increase the duct area so gradually that the reduction in velocity and accompanying loss of velocity pressure would occur reversibly, and thus permit 100 percent conversion to static pressure. Such an ideal application of the principle of *static regain* in duct design is, of course, impossible for various reasons[21] such as: the necessity of using sections of uniform diameter because of cost, the need for using ducts of dimensions varying in full inches, the changing of duct sizes mainly at branch connections, and the inevitable loss due to turbulence. The principle of static pressure regain is, however, of importance in the economical design of duct systems.

Fig. 9 shows the application of static pressure regain to a simple fan and discharge duct.[22] The fan in the upper part of the figure has a free inlet and discharges air through a straight duct, the diameter of which is equal to the fan outlet. The total pressure which must be provided by the fan is therefore the sum of the pressure that is necessary to overcome the friction in the duct (no dynamic pressure loss), plus the velocity pressure which, in this case, is the same at any location along the length of the duct.

In arrangement B in the lower part of Fig. 9, a diverging section, with after section, has been added to the straight duct. The velocity in the diverging section is therefore decreased, and velocity pressure converted

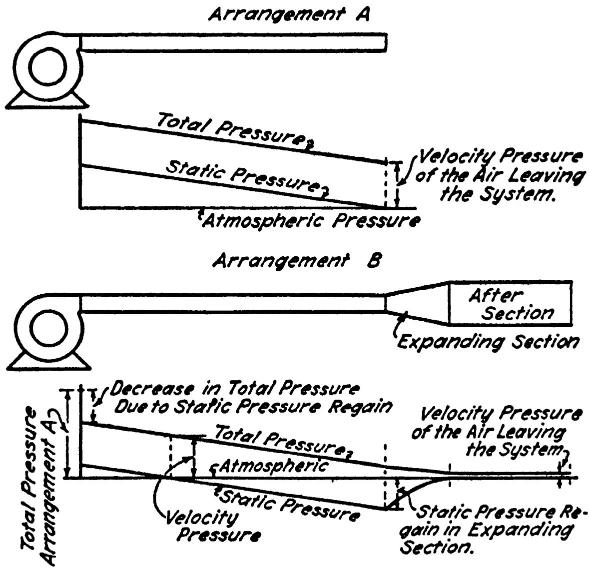


Fig. 9. Application of Static Pressure Regain to a Simple Fan and Discharge Duct

into static pressure before the air is released to the atmosphere. It can be seen that in case B, the total pressure at the fan outlet is less than in case A, and thus a saving in horsepower can be effected.

The regain in static pressure h_r in an abruptly expanded section is the difference in the velocity pressures of the small and the large duct, minus the dynamic pressure loss (Equation 6):

$$h_r = \left[\frac{v_1^2}{2g} - \frac{v_2^2}{2g}\right] - \left[\frac{(v_1 - v_2)^2}{2g}\right] \tag{13}$$

or simplified

$$h_r = \frac{v_1 v_2 - v_2^2}{g} = \frac{v_2(v_1 - v_2)}{g} \tag{14}$$

where

h_r = regain in static pressure, feet of fluid flowing.

v_1 and v_2 = mean velocities in inlet and outlet duct sections, respectively, feet per second.

The static pressure regain in a gradually expanding transition, followed by an after section, may be expressed as

$$h_r = \left[\frac{v_1^2}{2g} - \frac{v_2^2}{2g}\right] - \left[\frac{c_1(v_1 - v_2)^2}{2g}\right] \tag{15}$$

or

$$h_r = \frac{v_1^2 - v_2^2 - c_1(v_1 - v_2)^2}{2g} \tag{16}$$

where

c_1 = an experimentally determined regain constant depending on nature of construction.

Curves have been developed showing the static pressure regain and the *theoretical* efficiency of conversion in abrupt expansion, and in diverging sections in smooth circular ducts.[15,16]

DUCT DESIGN

The discussion of duct design in this chapter refers to ducts in fan systems for central heating, ventilating and air conditioning. Additional data for heating ducts used in residences are to be found in Chapter 18 (Gravity Warm Air Systems) and Chapter 19 (Forced Warm Air Systems). The design of ducts in industrial exhaust systems is discussed in Chapter 45.

The following general rules should be followed in design:

1. The air should be conveyed as directly as possible at the permissible velocities to obtain the desired results with greatest economy of power, material, and space.
2. Sharp elbows and bends should be avoided. Splitters and turning vanes should be used to reduce the elbow or outlet pressure loss.
3. Diverging transformation pieces should be made as long as practicable. As shown in the section on area changes, losses in sudden enlargements are high, and abrupt enlargements should be avoided. The included angle of divergence for enlargements should not exceed 20 deg. Losses in contractions are low, but the included angle of convergence should not be larger than 60 deg.
4. Special care should be taken to avoid restriction of flow in elbows or transformation pieces.
5. Where the greatest air carrying capacity per square foot of sheet metal is desired, rectangular ducts should be made as nearly square as possible. Aspect ratios greater than 10 to 1 should be avoided.
6. Ducts should be constructed of smooth material, such as steel or aluminum sheet metal. For ducts made from other materials, for example masonry, proper allowance for the surface friction coefficient should be made.

Procedure for Duct Design

The general procedure for design is outlined as follows:

1. Study the plan of the building and draw in roughly the most convenient system of ducts, taking cognizance of the building construction, avoiding all obstructions in steel work and equipment, and at the same time maintaining a simple design.
2. Arrange the positions of duct outlets to insure the proper distribution of air.
3. Divide the building into zones and proportion the volume of air necessary for each zone.
4. Determine the size of each outlet, based on the volume as obtained in the preceding paragraph, for the proper outlet velocity and throw. In case of some ceiling diffusers, determine size of outlet for proper throat velocity and radius of diffusion.

5. Calculate the sizes of all main and branch ducts by one of the three methods of sizing air supply systems in common use, the velocity reduction method, the equal friction method or the static regain method.

6. Calculate the losses for the duct offering the greatest resistance to the flow of air, using the A.S.H.V.E. Friction Charts, Figs. 1 and 2, and the other data given in this chapter.

Recommended Design Velocities

The air velocities given in Table 4 have been found to give satisfactory results in engineering practice. Where the higher velocities are used, the ducts should be cross-broken to prevent breathing, buckling or vibration, and should be constructed of heavier gage metal. At the higher velocities, it is particularly important to design the ducts for minimum resistance. Since high velocities at one point offset the effect of proper design in all other parts of the system, emphasis should be placed on the importance of air velocities, elbow design, location of dampers, fan connections, grille and register approach connections, and similar details. For industrial buildings, noise is seldom given much consideration, and main duct velocities as high as 2800 or 3000 fpm are sometimes used, but when these velocities are used due consideration should be given to duct design, resistance pressure, fan efficiencies and motor horsepower. For department stores and similar buildings, 2000 to 2200 fpm are sometimes used in main ducts where noise is not objectionable.

TABLE 4. RECOMMENDED AND MAXIMUM DUCT VELOCITIES

Designation	Recommended Velocities, fpm			Maximum Velocities, fpm		
	Residences	Schools, Theaters, Public Buildings	Industrial Buildings	Residences	Schools, Theaters, Public Buildings	Industrial Buildings
Outside Air Intakes[a]	500	500	500	800	900	1200
Filters[a]	250	300	350	300	350	350
Heating Coils[a]	450	500	600	500	600	700
Air Washers	500	500	500	500	500	500
Suction Connections	700	800	1000	900	1000	1400
Fan Outlets	1000–1600	1300–2000	1600–2400	1700	1500–2200	1700–2800
Main Ducts	700–900	1000–1300	1200–1800	800–1200	1100–1600	1300–2200
Branch Ducts	600	600–900	800–1000	700–1000	800–1300	1000–1800
Branch Risers	500	600–700	800	650–800	800–1200	1000–1600

[a] These velocities are for total face area, not the net free area; other velocities in table are for net free area.

Where high velocity diffusing outlets are used, the duct velocity should be, if possible, equal to, or somewhat lower than the throat (neck) velocity of the diffuser, in order to utilize the effect of higher static pressure in the duct for equalization of air discharge.

The velocities in main ducts, and particularly in branch ducts and branch risers, should be correlated to the throat (neck) velocity of the air outlets, and manufacturers' data should be consulted for permissible throat velocity for the particular type of application.

If it is necessary to use a duct velocity that is twice the velocity for an outlet mounted on the side of the duct, a collar with directing vanes should be used to straighten the flow of air from the outlet. Sometimes it is desirable to mount the outlet flush with the side of the duct, in which case the duct velocity should be kept below twice that of the outlet velocity, and even then an outlet larger than normally required should be used, as the entire outlet area will not be effective. Manufacturers' selection tables base sizing of outlets on required volume of air, temperature differential, and distance of throw or radius of diffusion. In following their recommendations, maxima should be avoided. See Chapter 30 for a discussion of air outlets.

DESIGN METHODS

The design of the air transmission system is generally the last step in the design of the heating, ventilating or air conditioning system, but it should always be kept in mind that the type of air transmission used will, to some extent, depend on the type of equipment used, as well as on the purpose of the system. Various factors such as zoning and zone control, and their influence on the transmission and air distribution system, are briefly discussed in Chapter 29 (Central Systems for Air Conditioning).

The methods used for the design of duct systems reflect, to some degree, certain developments in the arts of heating, ventilating and air conditioning, and it took a long time before empirical methods gave way to more refined and scientific calculations. Some engineers prefer speed and simplicity to scientific exactness, but experience is then needed and proper judgment must be exercised to prevent mistakes. Both the *Velocity Reduction Method* and the *Equal Friction Method* take no account of the static regain resulting from the difference between the velocity of fan discharge and velocities of pipe discharge, and are therefore, to some degree, approximate methods. However, they are more easily applied than the static regain method which is based on proper theory, *but is subject to an assumption (based on tests) regarding the efficiency of conversion from kinetic energy to static regain.*

1. Velocity Reduction Method

When this method is used, arbitrary velocities for the various sections of the ducts are selected, with the highest velocity at the fan outlet, and lower velocities down the run as various branch ducts are taken off the main duct. Since the quantities of air that are to be delivered through each section of the duct are known, the area of each duct section can be easily determined by using the formula:

$$A = \frac{Q_a}{V_m} \tag{17}$$

where

A = duct area in square feet.

Q_a = air quantity in cubic feet per minute.

V_m = air velocity in feet per minute.

To find the total static pressure against which the fan must operate, the static pressure loss of each section is calculated separately, and the total loss found by adding the individual losses of the sections of the duct which has the highest resistance. This may be the duct with the longest run, but not necessarily so.

The velocity method has the advantage that the duct area can be determined very easily. It should be used only for simple layouts. The air velocities given earlier in this chapter are helpful in choosing proper velocities. Balancing is obtained by use of dampers. The method is illustrated in *Example 4*.

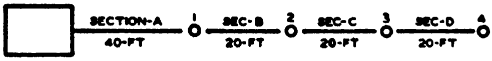


FIG. 10. DUCT LAYOUT FOR EXAMPLE 4

Example 4: (Velocity Reduction Method). A duct layout is shown in Fig. 10. The fan delivers 8000 cfm. Four outlets deliver 2000 cfm each. Find duct dimensions and total pressure loss.

Solution: Select velocity for Section A (2200 fpm) and reduce velocity arbitrarily along run. Find duct areas by using Equation 17. For selection of circular equivalents of rectangular ducts refer to Table 2, and for determination of friction loss in duct refer to Fig. 2 (See *Example 1*). Results are tabulated in Table 5.

TABLE 5. TABULATION OF RESULTS (EXAMPLE 4)

SECTION	AIR VOLUME cfm	VELOCITY fpm	AREA sq ft	AREA sq in.	DUCT SIZE in.	DIAM in.	FRICT PER 100 FT in. H_2O	FRICT LOSS in. H_2O
A	8000	2200	3.64	524	26 x 20	24.8	0.25	0.10
B	6000	2000	3.00	432	22 x 20	22.9	0.23	0.05
C	4000	1800	2.22	320	20 x 16	19.5	0.24	0.05
D	2000	1600	1.25	180	12 x 16	15.1	0.24	0.05
							Total resistance,	0.25

2. Equal Friction Method

When the equal friction method of design is used, the duct system is designed for equal friction per foot of length. This prevents one section of the duct from having an excessive resistance compared with another. The usual procedure in this method is to select the main duct velocity to be consistent with good practice from a standpoint of noise for a particular type of building. This velocity should be less than the fan outlet velocity. All ducts are then sized for equal friction per unit length by the use of Figs. 1 or 2 and Table 2. The equal friction method has the advantage of automatically reducing the velocities in the various sections of the system, and also of allowing a quick check of the total duct resistance.

In cases where the fan or factory assembled air conditioning unit can operate against only a limited external resistance, it is necessary to divide the permissible total resistance by the total *equivalent* length of the longest or most complicated run of duct to determine the design resistance per 100 ft, and then to size all ducts at this resistance value. This will automatically determine the duct velocities and give the desired total duct resistance. A further refinement, which is sometimes used in large systems, is to size each branch duct so that it has a resistance equal to the resistance of the main system at the point of juncture. Even when this refinement is added, regulating dampers are recommended in each branch.

Example 5: (Equal Friction Method). A duct layout is shown in Fig. 11. The fan delivers 2500 cfm. Outlets No. 1 and 2 deliver 750 cfm each and outlet No. 3 delivers 1000 cfm. Trunk velocity is assumed as 1500 fpm; the area will be 1.67 sq ft (240 sq in.); and the size will be 20 x 12 in. Determine sizes of ducts for sections B, C, D and E and find the total pressure loss.

Solution: The equivalent round diameter of a 20 x 12 in. rectangular duct is 16.8 in. (from Table 2). Referring to Friction Chart, Fig. 2, a volume of 2500 cfm through

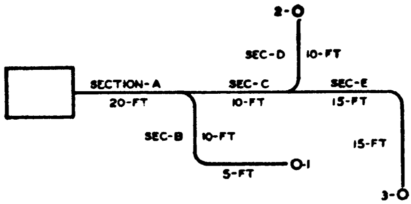


FIG. 11. DUCT LAYOUT FOR EXAMPLE 5

a 16.8 in. duct gives a resistance of 0.2 in. per 100 ft. The amount of air to be handled by each section is known, and the corresponding round duct sizes with equal pressure drop for these values can be located on the 0.2 in. friction line. The equivalent rectangular duct sizes are then selected from Table 2.

Results are tabulated in Table 6.

TABLE 6. TABULATION OF RESULTS (EXAMPLE 5)

SECTION	AIR VOLUME cfm	FRICTION PER 100 FT. in	DIAM in	VELOCITY fpm	RECTANGULAR DUCT in.	FRICTION PER 100 FT. in	DIAM. in	VELOCITY fpm	RECTANGULAR DUCT in.
A	2500	0.2	16.8	1620	20 x 12	0.2	17	1600	20 x 12
B	750	0.2	10.7	1190	10 x 9	0.286	10	1350	10 x 8
C	1750	0.2	14.8	1400	15 x 12	0.2	14.5	1450	15 x 12
D	750	0.2	10.7	1190	10 x 9	0.4	9.8	1350	10 x 8
E	1000	0.2	12	1300	10 x 12	0.2	12	1300	10 x 12

The total pressure loss in the longest run is the friction loss in Sections (A + C + E), plus the loss in one elbow and the loss through the outlet (3). The additional pressure loss in the elbow will be assumed as $\frac{L}{W} = 12$ (Fig. 5); thus, the additional equivalent length of duct is 10 ft, and the design loss will be 0.02 in.

Friction loss (A + C + E)	0.12	(Duct length = 20 ft + 10 ft + 15 ft + 15 ft)
Elbow loss	0.02	
Loss through outlet	0.12	
Total pressure loss in duct	= 0.26 in.	

The pressure required at the beginning of the main run is therefore 0.26 in. The fan selected for the duct system must not only deliver the required volume of air against this loss, but also against the losses in all air conditioning apparatus such as washers or spray chambers, heating or cooling coils and filters. The static head required of the fan for the usual air conditioning installation is between 1 and 1.5 in. of water. About one-third of this represents losses in the duct system. The losses in the air conditioning apparatus can be obtained from manufacturers' catalogs.

Resizing of Ducts

In order to equalize the pressure drop in the system, the following additional procedure is recommended:

Assume ΔH_1, ΔH_2, ΔH_3 to be the total pressure loss through ducts (1), (2) and (3); r_a, r_b, r_c, r_d, r_e the friction losses in the straight sections of the system; r_{be}, r_{de}, r_{ee} the elbow losses, and r_1, r_2, r_3 the loss through the outlets. Then,

$$\Delta H_1 = r_a + r_b + 2r_{be} + r_1$$
$$\Delta H_2 = r_a + r_c + r_d + r_{de} + r_2$$
$$\Delta H_3 = r_a + r_c + r_e + r_{ee} + r_3$$

If

$$\Delta H_1 = \Delta H_2 = \Delta H_3 = \Delta H$$
$$r_b + 2r_{be} = \Delta H - r_a - r_1$$
$$r_d + r_{de} = \Delta H - r_a - r_c - r_2$$

or, using the values from *Example 5*:

$$r_b + 2r_{be} = 0.26 - 0.04 - 0.12 = 0.10 \text{ in.}$$

$$r_d + r_{de} = 0.26 - 0.04 - 0.02 - 0.12 = 0.08 \text{ in.}$$

The loss in the elbows will be assumed to be $\frac{L}{W}$ = 12 or 10 additional equivalent feet, the friction loss of head per 100 equivalent ft is then

for duct (1) $$\frac{0.10}{0.15 + 2 \times 0.10} = 0.286 \text{ in.}$$

for duct (2) $$\frac{0.08}{0.10 + 0.10} = 0.4 \text{ in.}$$

Using Friction Chart Fig. 1, the duct diameter of Section B, to carry 750 cfm with a loss of 0.286 in. per 100 ft, is found as 10 in., and the duct diameter of Section D, to carry 750 cfm with a loss of 0.4 in. per 100 ft is 9.8 in. Equivalent rectangular ducts are 10 × 8 in., the velocity in both ducts is 1350 fpm.

The actual loss in ducts (1) and (2) is:

$$0.04 + 20 \times \frac{0.286}{100} + 15 \times \frac{0.286}{100} + 0.12 = 0.26$$

$$0.06 + 10 \times \frac{0.4}{100} + 10 \times \frac{0.4}{100} + 0.12 = 0.26$$

For final survey of ducts selected see Table 6 (Tabulation of Results).

3. Static Regain Method

When this method is used, the velocity is reduced at each branch or take-off so that the recovery in static pressure due to this reduction will offset the friction in the succeeding section. This method is based on the convertibility of static pressure and velocity pressure, as discussed in a preceding section on Pressure Changes. If no friction or dynamic losses occurred, the change in velocity head would be completely converted into a regain in static pressure, which for standard air would be:

$$H_r = \left(\frac{V_1}{4005}\right)^2 - \left(\frac{V_2}{4005}\right)^2 \tag{18}$$

where

H_r = theoretical head recovered (static regain), inches of water.
V_1 = initial velocity of standard air, feet per minute.
V_2 = velocity of standard air after reduction, feet per minute.

Under ideal conditions, 0.7 to 0.8 of the velocity head is actually recovered, but for practical design an average recovery of 0.5 is assumed. The actual velocity head recovered H_r, then becomes

$$H_r = 0.5\left[\left(\frac{V_1}{4005}\right)^2 - \left(\frac{V_2}{4005}\right)^2\right] \tag{19}$$

The advantage of the static regain method is that it provides a convenient means of designing a long run of duct (or an entire system) so that essentially the same static pressure will be obtained at each outlet. This simplifies *outlet selection* and *system balancing*. On large systems or very long runs, where it may not be feasible or economically desirable to design for zero static pressure loss between outlets, the method may be used to size

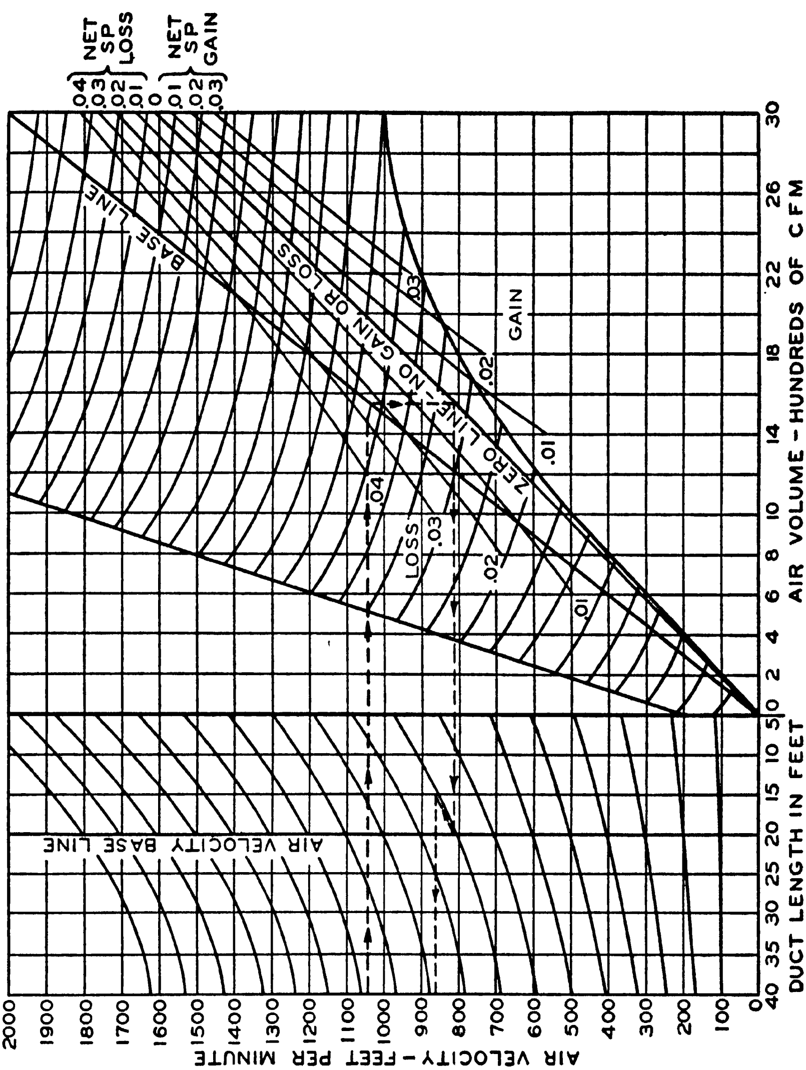


FIG. 12. STATIC REGAIN CHART*

For Determining Velocity in Ducts Carrying 0 to 3000 cfm

* Based on Equation 19.

ducts for a *uniform predetermined loss.* This loss or gain is *net*, that is, it is the friction loss compensated by any static pressure gain made available by a change in velocity. (The latter effect is commonly neglected in the Equal Friction Method.)

Charts for the practical application of the principles of static regain to duct design, are presented in Figs. 12 and 13. These charts are based on Equation 19, as applied to rectangular ducts of average construction with dimension ratios of 3 to 1 or less. Note that the *gain* or *loss* indicated on the charts is the net gain or loss in the duct section considered (normally the distance between two outlets); it should not be confused with static pressure loss per 100 ft, or total pressure loss in the duct. The total loss or gain in the *outlet run* is the summation of the losses or gains in the successive sections figured. (Losses in outlets, coils, or similar items are

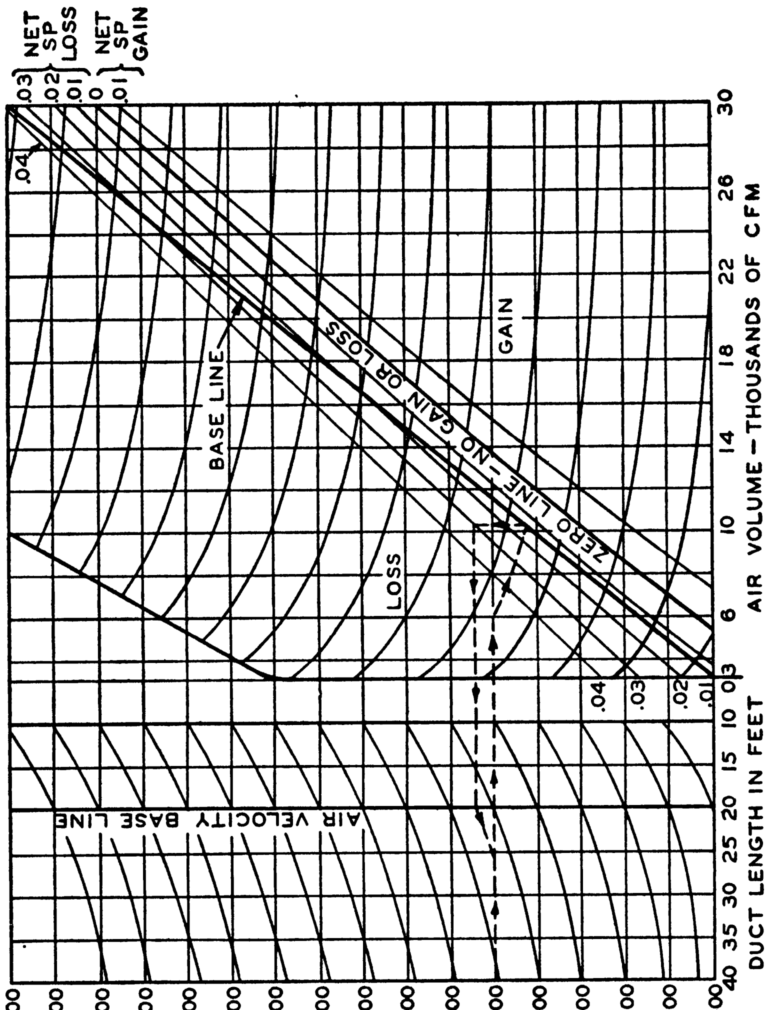


Fig. 13. Static Regain Chart[a]

For Determining Velocity in Ducts Carrying 3000 to 30,000 cfm

[a]Based on Equation 19.

figured separately.) The *duct length* of any section should include the equivalent length of any elbows occurring within the section.

The static regain charts are intended primarily for constructions where regain takes place unaccompanied by radical change in direction; thus, in Fig. 14 they are strictly applicable along the main run A to E, and at the junction of Sections F and G, but not at the junction of Sections A and F. Although some regain will usually occur at the branch take-off (where velocity is generally reduced), there are so many varieties of elbows and branch take-off connections, that estimation of an *average* value of regain would be quite impracticable.

The static regain method finds its widest application in the design of long duct runs containing numerous successive outlets (usually designated the *outlet run*). An outlet run is typified by Sections C–D–E in Fig. 14. On larger systems several methods of duct sizing may be combined to secure equal or approximately equal pressures at all outlets. Example 6 shows the method of approach as applied to a very small system.

As with any other method of duct design, balancing dampers should be installed in each branch, and each outlet should be equipped with means of regulating air volume.

Example 6: (Static Regain Method). A duct layout is shown in Fig. 14. The fan delivers 8000 cfm. Outlets 1, 2, 3 and 4 deliver 1500 cfm each, and outlets 5 and 6, 1000 cfm each. The operating pressure loss at all outlets is 0.12 in. water. Initial trunk velocity is assumed as 1500 fpm; the area of the trunk duct will then be 5.33 sq ft, and the size will be 48 x 16 in. It is assumed that for this example it is desirable to maintain a 16 in. depth on all duct sections. Determine the sizes of duct sections B, C, D, E, F and G so that substantially the same static pressure will be obtained at each of the outlets, and find the total pressure loss of the system.

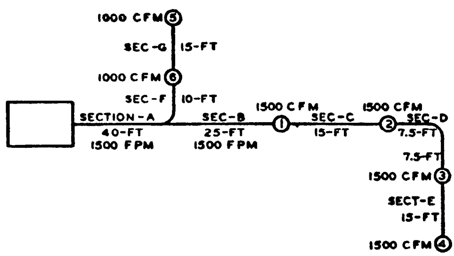


FIG. 14 DUCT LAYOUT FOR EXAMPLE 6

Solution:

1. Size Section F by the equal friction method so that it has the same rate of friction loss as Section A. Section A is equivalent to a 29.2 in. round duct (Table 2) and the pressure loss from Fig. 2 is 0.13 in. per 100 ft. For 2000 cfm flowing at this rate of pressure loss, the indicated round duct diameter for Section F is approximately 17 in. (Fig. 2). This is equivalent to a 15 x 16 in. duct, which will be used for Section F. The velocity in Section F will be 1200 fpm.

2. Determine the pressure loss in Section F. Actual length of duct is 10 ft; equivalent length of elbow take-off is assumed as 10 W, or 12.5 ft. Therefore, the total equivalent length is 22.5 ft. The pressure loss in F = 0.13 in. $\times \frac{22.5}{100}$ = 0.03 in. water.

3. Using Static Regain Chart, Fig. 13, size Section B for a net pressure loss equal to the loss in F, or 0.03 in. water as follows:

The operation is indicated by arrow heads on the dotted line on Fig. 13. On Fig. 13, start at the velocity in Section A (1500 fpm) at left margin. Proceed horizontally to 6000 cfm ordinate, and then run parallel to the curved lines to intersect the diagonal Base Line. From this point, rise vertically to the 0.03 net static pressure loss line, and from this intersection proceed horizontally to the Air Velocity Base Line. Proceed parallel to curved lines to intersect ordinate for 25 ft equivalent duct length, and then move horizontally to left margin and read the velocity (1500 fpm).

Since Section B carries 6000 cfm, the area required will be $\frac{6000}{1500}$ = 4 sq ft, and the size of duct will be 36 x 16 in.

4. Using Static Regain Charts (Figs. 12 and 13) determine size of Sections C, D, E and G, but instead of allowing 0.03 in. net loss, which was used for Section B, proceed from the diagonal Base Line vertically to the *no gain or loss* diagonal. The procedure for Section E is shown by the dotted line and arrows on Fig. 12: starting from 1040 fpm velocity, which is the velocity in Section D.

Duct sizes determined by the given procedure are listed in Table 7 on next page.

TABLE 7. TABULATION OF RESULTS (EXAMPLE 6)

SECTION	AIR VOLUME cfm	EQUIVALENT LENGTH ft	VELOCITY fpm	RECTANGULAR DUCT in.	DIAM. in.	FRICTION PER 100 FT in. H_2O	NET PRESSURE LOSS in. H_2O
A	8000	40	1500	48 x 16	29.2	0.13	.05
B	6000	25	1500	36 x 16	—	—	.03
C	4500	15	1300	31 x 16	—	—	0
D	3000	26[a]	1040	26 x 16	—	—	0
E	1500	15	860	16 x 16	—	—	0
F	2000	22.5	1200	15 x 16	17.	0.13	.03
G	1000	15	900	10 x 16	—	—	0

[a] Includes additional equivalent length of elbow between outlets 2 and 3, which is assumed as 5.5 W, or 11 ft equivalent length of duct. (Based on 3000 cfm at estimated velocity of 1100 fpm).

5. Total pressure loss of the system is the loss in Section A, plus the loss in Section F (or B), plus the loss in the outlet as follows:

$$\text{Loss in Section A} = 0.13 \text{ in.} \times \frac{40}{100} = 0.05$$

$$\text{Loss in Section F (or B)} = 0.03$$

$$\text{Outlet Loss} = \underline{0.12}$$

$$\text{Total Pressure Loss} = 0.20 \text{ in.}$$

DUCT CONSTRUCTION DETAILS

Straight sections of round duct are usually formed from sheets, rolled to the proper radius with a longitudinal grooved seam. Each section is swedged 1.5 in. from each end and assembled with the larger end of the adjoining section butting against the swedge. The sections are held in place by rivets, sheet metal screws, or by soldering.

Rectangular ducts are generally constructed by breaking the corners and grooving the longitudinal seam, although some fabricators still use the standing seam. Elbows and transformation sections are generally formed with Pittsburgh corner seams because this seam is easier to lock in place than the double seam, but complicated fittings such as double compounded elbows are usually constructed with double seam corners. The construction of these various seams, as well as the types of girth connections, are shown in Fig. 15. The application of the various slips and connections is outlined in Table 8. The end slip may be used wherever *S* slips are recommended. Where drive slips are used, the end slip may be applied on the narrow side of the duct, and the drive slips on only the maximum side. Ducts 25 to 30 in. in size should be reinforced between the joints, but not necessarily at the joint. Ducts 31 in. and up should be reinforced at the joint and between the joints; if drive slips are used the angles are usually rivited to the duct about 2 in. from the slips. It is good practice to cross-break or kink all flat surfaces to prevent vibration or buckling due to the air flow and accompanying variations in internal pressure

The construction of elbows and changes of shape cannot be definitely outlined, because of the varied conditions encountered in the field, but in general, long radius elbows and gradual changes in shape tend to maintain uniform velocities accompanied by decreased turbulence, lower resistance and a minimum of noise.

Heavy canvas connections (asbestos cloth if there is a fire hazard) are recommended on both the inlet and outlet to all fans. Self-vulcanizing adhesive tapes are available for this purpose and for sealing joints in duct work. The fan discharge connections shown in Fig. 15 are marked *good*, *fair*, and *poor* in the order of the amount of turbulence produced. An inspection of the heater connections shown in Fig. 15 will readily show that uniform velocity through the heater cannot be expected in the diagram

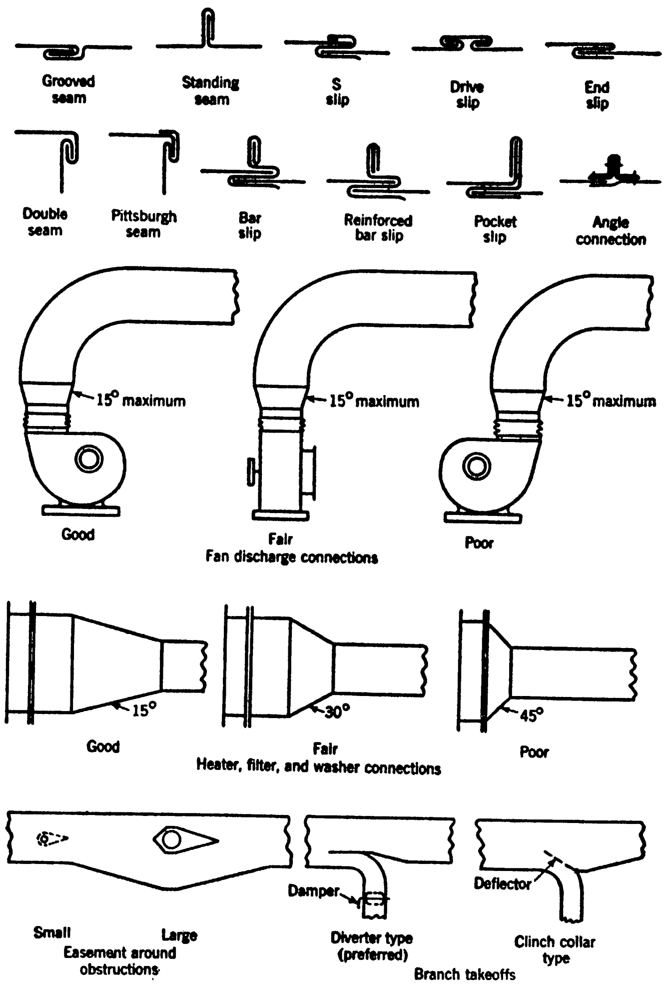


Fig. 15. Sheet Metal Duct and Arrangement Details

noted *poor*. When obstructions cannot be avoided, the duct area should never be decreased more than 10 percent, and then a *streamlined* collar should be used. Larger obstructions require an increase in the duct size in order to maintain as nearly uniform velocity as possible. Branch take-offs should always be arranged to cut or slice into the air stream in order to reduce as far as possible the losses in velocity head.

Wherever ducts pass through fire walls or connect two fire areas of a building, automatic fire dampers should be provided. For design of such dampers and other fire protective details, see Pamphlet No. 90 of the *National Board of Fire Underwriters*.[22]

The recommended gages for steel (or iron) and aluminum sheet metal rectangular ducts are given in Table 8. Steel or iron sheets are specified according to the Manufacturers or U. S. Standard Gage System. Aluminum sheets are specified according to the American or Brown & Sharpe Gage System. Weights of black and galvanized steel and iron sheets per square foot of surface for various gages are given in Table 9. Similar data for 2S aluminum sheets will be found in Table 10. Weights of standard

Table 8. Recommended Sheet Metal Gages for Rectangular Duct Construction[a]

Aluminum B.&S. Gage	Steel U. S. Std. Gage	Maximum Side, Inches	Type of Transverse Joint Connections[b]	Bracing
24	26	Up to 12	S, Drive, Pocket or Bar Slips, on 7 ft 10 in. centers	None
22	24	13 to 24	S, Drive, Pocket or Bar Slips, on 7 ft 10 in. centers	None
		25 to 30	S, Drive, 1 in. Pocket or 1 in. Bar Slips, on 7 ft 10 in. centers[c]	1 x 1 x ⅛ in. angles 4 ft from joint
20	22	31 to 40	Drive, 1 in. Pocket or 1 in. Bar Slips, on 7 ft 10 in. centers[c]	1 x 1 x ⅛ in. angles 4 ft from joint
		41 to 60	1½ in. Angle Connections, or 1½ in. Pocket or 1½ in. Bar Slips with 1⅜ in. x ⅛ in. bar reinforcing on 7 ft 10 in. centers[c]	1½ x 1½ x [illegible] in. angles 4 ft from joint
18	20	61 to 90	1½ in. Angle Connections, or 1½ in. Pocket or 1½ in. Bar Slips 3 ft 9 in. maximum centers with 1⅜ x ⅛ in. bar reinforcing	1½ x 1½ x ⅛ in. diagonal angles, or 1½ x 1½ x ⅛ in. angles 2 ft from joint
16	18	91 and up	2 in. Angle Connections or 1½ in. Pocket or 1½ in. Bar Slips 3 ft 9 in. maximum centers with 1⅜ x ⅛ in. bar reinforcing[d]	1½ x 1½ x ⅛ in. diagonal angles, or 1½ x 1½ x ⅛ in. angles 2 ft from joint

[a] For normal pressures and velocities (see Table 4) utilized in typical ventilating and air conditioning systems. Where special rigidity or stiffness is required, ducts should be constructed of metal two gages heavier. All uninsulated ducts 18 in. and larger should be cross-broken. Cross-breaking may be omitted on uninsulated ducts if two gages of heavier metal are used.

[b] Other joint connections of equivalent mechanical strength and air tightness may be used.

[c] Duct sections of 3 ft 9 in. may be used with bracing angles omitted, instead of 7 ft 10 in. lengths with joints indicated.

[d] Ducts 91 in. and larger require special field study for hanging and supporting methods.

copper sheets are given in Table 11. In calculating the total weight of a given length of duct work from these tables, it is customary to add 20 percent for the weight of joints and bracings.

Aluminum sheets of the 2S and 3S type alloy and ¾ hard temper are readily workable, and can be used for practically all duct work. The 2S type (commercially pure aluminum) is suitable for all, except very large ducts. For large ducts, where more strength is desired, the 3S alloy with ¾ or ½ hard temper is frequently used. The higher tempers, particularly full hard, do not have the formability of the lower tempers. For very large ducts, where considerable strength is required, aluminum sheets

TABLE 9. WEIGHTS OF BLACK AND GALVANIZED SHEETS

U. S. Std. Gage	Black Sheets				Galvanized Sheets*			
	Approximate Thickness, In.		Weight Per Square Foot		Approximate Thickness, In.		Weight Per Square Foot	
	Steel	Iron	Ounces	Pounds	Steel	Iron	Ounces	Pounds
30	0.0123	0.0125	8	0.500	0.0163	0.0165	10.5	0.656
28	0.0153	0.0156	10	0.625	0.0193	0.0196	12.5	0.781
26	0.0184	0.0188	12	0.750	0.0224	0.0228	14.5	0.906
24	0.0245	0.0250	16	1.000	0.0285	0.0290	18.5	1.156
22	0.0306	0.0313	20	1.250	0.0346	0.0353	22.5	1.406
20	0.0368	0.0375	24	1.500	0.0408	0.0415	26.5	1.656
18	0.0490	0.0500	32	2.000	0.0530	0.0540	34.5	2.156
16	0.0613	0.0625	40	2.500	0.0653	0.0665	42.5	2.656
14	0.0766	0.0781	50	3.125	0.0806	0.0821	52.5	3.281
12	0.1072	0.1094	70	4.375	0.1112	0.1134	72.5	4.531
11	0.1225	0.1250	80	5.000	0.1265	0.1290	82.5	5.156
10	0.1379	0.1406	90	5.625	0.1419	0.1446	92.5	5.781

*Galvanized sheets are gaged before galvanizing and are therefore approximately 0.004 in. thicker.

should be 2 gages heavier than indicated in Table 8, and should be amply stiffened. Joints can be of any of the standard designs, and can be fabricated in the same manner as iron. Repeated sharp bending and rebending should be avoided, as aluminum has a tendency to crack under such treatment. Aluminum of 16 B. & S. gage or heavier can readily be welded by the metallic arc or acetylene process. Soldering is difficult and is not generally recommended. Riveting is done in the same manner as in iron or steel sheet. Self-tapping screws tend to loosen because of the softness of aluminum.

HEAT LOSSES FROM DUCTS

In designing duct systems, the heat gains or losses of ducts should not be neglected. Heat gains in large duct systems can be quite considerable, not only if the duct passes through unconditioned space, but also on long duct runs within conditioned space. Proper insulation will remedy this situation considerably, but sometimes a redistribution of the supply air to the various supply outlets is necessary in order to compensate for the heating effect of the duct surface.

The thermal transmittance U for ducts can be found as follows:

For uninsulated metal duct, $$U = \frac{1}{\dfrac{1}{f_i} + \dfrac{1}{f_o}} \qquad (20)$$

TABLE 10. WEIGHTS AND THICKNESSES OF 2S ALUMINUM (DENSITY 0.098 LB/CU IN.)

B. & S. Gage	Thickness, Inches		Weight per Square Foot	
	Decimal	Nearest Fraction	Ounces	Pounds
28	0.012	1/64	2.7	0.169
26	0.016	1/64	3.6	0.226
24	0.020	1/64	4.5	0.282
22	0.025	1/32	5.4	0.353
20	0.032	1/32	7.2	0.452
18	0.040	3/64	9.0	0.563
16	0.051	3/64	11.5	0.720
14	0.064	1/16	14.4	0.903

For uninsulated non-metallic ducts,

$$U = \frac{1}{\dfrac{1}{f_i} + \dfrac{x}{k} + \dfrac{1}{f_o}} \tag{21}$$

where

U = overall coefficient of heat transfer, Btu per (hour) (square foot) (Fahrenheit degree).

f_i = surface conductance (inside) Btu per (hour) (square foot) (Fahrenheit degree).

f_o = surface conductance (outside) Btu per (hour) (square foot) (Fahrenheit degree).

x = thickness, inches.

k = unit conductivity of material, Btu per (hour) (square foot) (Fahrenheit degree per inch thickness).

TABLE 11. WEIGHTS AND THICKNESSES OF STANDARD COPPER SHEETS[a]
Rolled to Weight

WEIGHT PER SQUARE FOOT		THICKNESS, INCHES		NEAREST GAGE NO.		
Ounces	Pounds	Decimal Equivalent	Nearest Fraction	B. & S.	Stubs	U. S. STD.
10	0.625	0.0135	1/64	27	29	29
12	0.750	0.0162	1/64	26	27	28
14	0.875	0.0189	1/64	25	26	26
16	1.000	0.0216	1/32	23	24	25
18	1.125	0.0243	1/32	22	23	24
20	1.250	0.0270	1/32	21	22	23
24	1.500	0.0324	1/32	20	21	22
28	1.750	0.0378	1/32	19	20	20
32	2.000	0.0432	3/64	17	19	19
36	2.250	0.0486	3/64	16	18	18
40	2.500	0.0540	3/64	15	17	17
44	2.750	0.0594	1/16	15	17	17
48	3.000	0.0648	1/16	14	16	16
56	3.500	0.0756	5/64	13	15	14
64	4.000	0.0864	5/64	11	14	13

[a]Variations from these weights must be expected in practice.

Where x is small and k is large, however, the factor $\frac{x}{k}$ is of little importance and may be neglected.

Film conductance f_i for air flowing in ducts apparently depends only on the velocity of the air and the diameter of the duct. A fairly reliable inside coefficient can be calculated from Schultz's modified equation:

$$f_i = \frac{0.32v^{0.8}}{D^{0.25}} \tag{22}$$

where

v = velocity of air in duct, feet per second.

D = inside diameter of duct, feet.

Film conductance f_o depends on a number of variables including temperature, diameter, and emissivity of the outer surface, and can be calculated from data in Chapter 5. From this explanation, it is seen that it is unwise to recommend a given value of U for all uninsulated metal ducts.

The heat loss from a given length of duct can be expressed by:

$$Q_w = UPl\left[\left(\frac{t_1 + t_2}{2}\right) - t_3\right] \tag{23}$$

where

Q_w = heat loss through duct walls, Btu per hour.
P = perimeter of duct, feet.
l = length of duct, feet.
t_1 = temperature of air entering duct, Fahrenheit degrees.
t_2 = temperature of air leaving duct, Fahrenheit degrees.
t_3 = temperature of air surrounding duct, Fahrenheit degrees.

The heat given up by the air in the duct is:

$$Q_w = 0.24w(t_1 - t_2) = 14.4AV_m\rho_v(t_1 - t_2) \tag{24}$$

where

w = weight of air through duct, pounds per hour.
A = cross-sectional area of duct, square feet.
V_m = mean velocity of fluid, feet per minute.
ρ_v = density of air at specified temperature at which velocity V_m, is measured pounds per cubic foot.

Equating (23) and (24):

$$\frac{t_1 + t_2 - 2t_3}{t_1 + t_2} = \frac{28.8AV_m\rho_v}{UPl}$$

Let $y = \frac{28.8AV_m\rho_v}{UPl}$ for rectangular ducts, and $\frac{7.2DV_m\rho_v}{Ul}$ for round ducts and solve for t_1 and t_2:

$$t_1 = \frac{t_2(y + 1) - 2t_3}{(y - 1)} \tag{25}$$

$$t_2 = \frac{t_1(y - 1) + 2t_3}{(y + 1)} \tag{26}$$

For low velocities and long ducts of small cross-section, a somewhat more accurate formula may be used as follows:

$$t_2 = \frac{t_1 - t_3}{e^z} + t_3 \tag{27}$$

where

$z = \frac{UPL}{14.4A\rho_v V_m}$.
e = Naperian base of logarithms = 2.718.

In using Equations 25, 26, and 27, one of the duct air temperatures will be unknown and will be obtained by substitution of the other known or assumed values.

Heat loss coefficients for insulated ducts with various conductivities are given in Fig. 16. The conductivities of various materials, which are based on mean temperatures, ranging from about 70 to 90 F, will be found in Table 2 of Chapter 9. For cases where the mean temperature is other than that at which the test was conducted, a correction should be made. However, in most cases the effect of this factor will be small and may be neglected.

Example 7: Determine the entering air temperature and heat loss for a duct 24 × 36 in. cross-section and 70 ft in length, insulated with ½ in. of a material having a

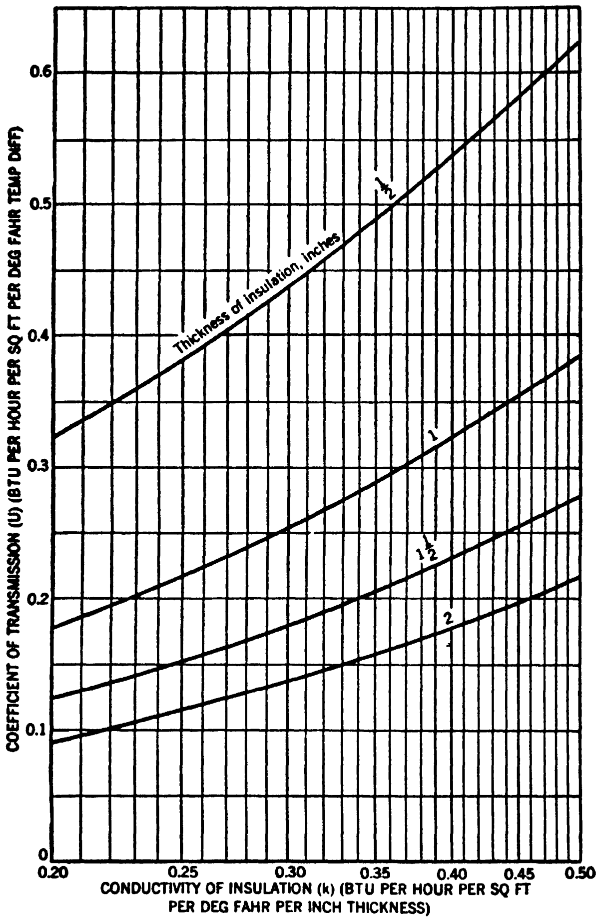


FIG. 16. HEAT LOSS COEFFICIENTS FOR INSULATED DUCTS[a]

[a] For round ducts less than 30 in. diameter, increase heat transmission values by the percentages shown below.

THICKNESS OF INSULATION (Inches)	½	1	1½	2
12 to 21 in. Duct Diameter	3%	5%	7%	9%
21 to 30 in. Duct Diameter	1%	2%	3%	4%

conductivity of 0.35 Btu at 86 F mean temperature, carrying air at a velocity of 1200 fpm, measured at 70 F, to deliver air at 120 F with air surrounding the duct at 40 F.

Solution: Referring to Fig. 16, the overall heat transmission coefficient is found to be 0.49 Btu. From Table 1, Chapter 3 the density of air at 70 F and 29.921 in. Hg

is found to be 1/13.348 = 0.0749 lb per cu ft. Substituting these and the other given values in Equation 25, y and t_1 will be as follows:

$$y = \frac{28.8 \times 6 \times 1200 \times 0.0749}{0.49 \times 10 \times 70} = 45.3$$

$$t_1 = \frac{120(45.3 + 1) - 80}{45.3 - 1} = 123.7F$$

Substituting in Equation 23:

$$Q_w = 0.49 \times 10 \times 70 \left[\left(\frac{123.7 + 120}{2} \right) - 40 \right] = 28{,}100 \text{ Btu per hr.}$$

For special considerations which apply to insulation of ducts in marine installations see Chapter 47.

MAINTENANCE

Ducts should be designed in such a manner as to enable easy maintenance.[23] They should have enough access doors, not only to enable inspection, but also to facilitate cleaning of the ducts.[24] The periodic cleaning of the ducts should be part of the regular maintenance schedule. It should be done efficiently and competently to avoid difficulties or hazards in the operation of the system.[25,26]

LETTER SYMBOLS USED IN CHAPTER 31

α = coefficient of contraction.
ϵ = absolute roughness, feet.
ρ_o = density of air under actual (operating) conditions, any consistent units.
ρ_s = density of air under standard conditions, any consistent units.
ρ_v = density at which V_m is measured, pounds per cubic foot.
A = cross-section area of duct, square feet.
A_1 = area of inlet duct, square feet.
A_2 = area of outlet duct, square feet.
A'_2 = area of vena contracta, square feet.
a = length of one side of rectangular duct, inches. (Other side is b.)
b = length of one side of rectangular duct, inches. (Other side is a.)
C and C_1 = dynamic loss coefficients, dimensionless.
c_1 = regain constant, dimensionless.
D = inside diameter of duct, feet.
d_e = circular equivalent of a rectangular duct for equal friction and capacity, inches.
e = Naperian base of logarithms = 2.718.
f = non-dimensional friction coefficient.
f_i = surface conductance (inside) Btu per (hour) (square foot) (Fahrenheit degree).
f_o = surface conductance (outside) Btu per (hour) (square foot) (Fahrenheit degree).
g = acceleration due to gravity, 32.17 feet per (second) (second).
H = duct dimension perpendicular to plane of bend, feet.
H/W = aspect ratio, dimensionless.
H_c = pressure loss due to sudden contraction, based on standard air, inches of water.
H_e = pressure loss due to sudden enlargement, based on standard air, inches of water.
H_{ge} = pressure loss due to gradual enlargement, based on standard air, inches of water.
H_v = dynamic loss for standard air, inches of water.
H_1 and H_2 = static pressure head at given points (1) and (2), inches of water.
H_r = regain static pressure for standard air, inches of water.
ΔH = total pressure loss, inches of water.

h_o = friction loss under actual (operating) conditions, any consistent units.
h_c = pressure loss due to sudden contraction, feet of fluid flowing.
h_e = pressure loss due to sudden enlargement, feet of fluid flowing.
h_f = friction loss, feet of fluid flowing.
h_r = regain in static pressure, feet of fluid flowing.
h_s = friction loss under standard conditions, any consistent units.
h_T = total dynamic pressure loss, feet of fluid flowing.
k = conductivity, Btu per (hour) (sq ft) (Fahrenheit degree per inch).
L = additional equivalent length of duct for loss in elbows, feet.
L/W and L/D = additional equivalent length in terms of width and diameter, dimensionless.
l = length of straight duct, feet.
P = perimeter of duct, feet.
Q_a = air quantity, cubic feet per minute.
Q_w = heat loss through duct walls, Btu per hour.
R = centerline radius of elbow, or vane radius as noted, feet.
R/W and R/D = radius ratio, dimensionless.
r = loss through specified duct sections, inches of water.
t_1 = temperature of air entering duct, Fahrenheit degrees.
t_2 = temperature of air leaving duct, Fahrenheit degrees.
t_3 = temperature of air surrounding duct, Fahrenheit degrees.
U = thermal transmittance coefficient, Btu per (hour) (square foot) (Fahrenheit degree).
V = mean velocity of standard air, feet per minute.
V_m = mean velocity of air or fluid, feet per minute.
V_1 = mean velocity of standard air in inlet duct section, feet per minute.
V_2 = mean velocity of standard air in outlet duct section, feet per minute.
v = mean fluid or air velocity, feet per second.
v_1 = mean velocity in inlet duct section, feet per second.
v_2 = mean velocity in outlet duct section, feet per second.
W = duct dimension in plane of bend, feet.
w = weight rate of air flow through duct, pounds per hour.
x = thickness, inches.

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CHAPTER 32

FANS

Types, Fan Performance, Fan Laws, Fan Performance Curves, System Characteristics, Fan Arrangements, Fan Control, Motive Power, Fan Selection, Fan Installation, Fan Applications

IN HEATING, ventilating and air conditioning practice, the devices used to produce air flow are variously known as fans, blowers, exhausters or propellers. The *A.S.M.E.* Test Code[1] limits fans to those in which the fluid density change does not exceed 7 percent (one psi at atmospheric pressure) and labels as compressors those devices operating beyond that pressure range. Since air conditioning rarely requires pressures of over $\frac{1}{3}$ psi, all such devices will be known as *fans* and the air will be considered non-compressible.

Types

Fans are divided into two general classifications: (1) *centrifugal or radial flow* in which the air flows radially through the impeller within a scroll type housing, and (2) *axial flow* in which the air flows axially through the impeller within a cylinder or ring.

Centrifugal fans are further subdivided into types denoted by the curvature or slope of the impeller blades, the angle of which largely determines the operating characteristics. For a given output, a forward inclination of blade indicates a relatively low speed of operation, and a backward inclination, a relatively high speed of operation. Many intermediate forms are also found.

Axial flow fans are subdivided into types differentiated mainly by their enclosures and refinements of impellers and appurtenances. All types vary in shape, number and angles of blades; ratios of hub diameter to impeller diameter; materials and methods of fabrication, depending upon design and preference of manufacturer. *Tubeaxial* and *vaneaxial* fans, usually used against appreciable resistance, commonly have relatively large hubs and helical blades (the angle varies radially along the blade). The blades may be of uniform thickness, either flat or cambered, and either cast or made of plates; or they may be of air foil sections, either cast or of double thickness sheet. Streamlining of both impeller and enclosure is common practice. *Vaneaxial* fans incorporate guide vanes to modify performance and increase efficiency. *Propeller* fans customarily used for free delivery, or against low resistance, also are found with a variety of blade conformations, but are simple in construction. They are merely mounted within a plate or ring.

The fan nomenclature in Fig. 1 has been standardized by the *National Association of Fan Manufacturers.*[2]

FAN PERFORMANCE

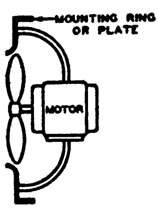
Fan performance is a statement of volume, total pressures, static pressures, speed, power input, mechanical and static efficiency, at a stated density. These terms are defined by the *National Association of Fan Manufacturers*[2] as follows:

1. *Volume* handled by a fan is the number of cubic feet of air per minute expressed at fan outlet conditions.

2. *Total pressure* of a fan is the rise of pressure from fan inlet to fan outlet.
3. *Velocity pressure* of a fan is the pressure corresponding to the average velocity determination from the volume of air flow at the fan outlet area.
4. *Static pressure* of a fan is the total pressure diminished by the fan velocity pressure.
5. *Power output* of a fan is expressed in horsepower and is based on fan volume and the fan total pressure.
6. *Power input* to a fan is expressed in horsepower and is measured horsepower delivered to the fan shaft.
7. *Mechanical efficiency* of a fan is the ratio of power output to power input.
8. *Static efficiency* of a fan is the mechanical efficiency multiplied by the ratio of static pressure to the total pressure.
9. *Fan outlet area* is the inside area of the fan outlet.
10. *Fan inlet area* is the inside area of the inlet collar.

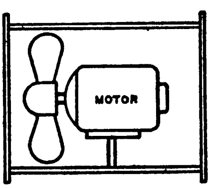
While the total pressure truly represents the actual pressure developed by the fan, the static pressure may best represent the useful pressure for overcoming resistance. In many installations, since the outlet velocity of the fan is greater than the duct velocity, some of the velocity pressure may be utilized by conversion to static pressure within the system. However, due to the uncertainty of the flow at the points of velocity change, the amount of conversion is seldom known and therefore, most fan tables list only the static pressure as available to overcome the system resistance.

Fig. 1. Names and Definitions of Types of Fans



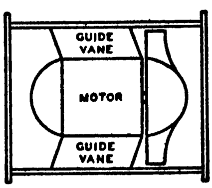
Propeller Fan

A propeller fan consists of a propeller or disc wheel within a mounting ring or plate.



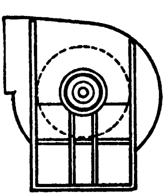
Tubeaxial Fan

A tubeaxial fan consists of an axial flow wheel within a cylinder.



Vaneaxial Fan

A vaneaxial fan consists of an axial flow wheel within a cylinder, combined with a set of air guide vanes located either before or after the wheel.



Centrifugal Fan

A centrifugal fan consists of a fan rotor or wheel within a scroll type of housing.

According to the Standard Test Code[3] the efficiencies may be determined by the formulas:

$$\text{Mechanical (total) Efficiency} = \frac{0.0001573 \times (\text{cfm}) \times \text{total pressure (inches water)}}{\text{horsepower input}}$$

$$\text{Static Efficiency} = \frac{0.0001573 \times (\text{cfm}) \times \text{static pressure (inches water)}}{\text{horsepower input}}$$

As the static pressure is often more useful than total pressure, static efficiency is likewise many times more useful than mechanical efficiency. However, where a high outlet velocity can be effectively utilized, the static efficiency fails to be a satisfactory measurement of performance; also when a fan operates against no resistance, the static efficiency becomes zero and is meaningless. Under such circumstances, many engineers prefer to use mechanical efficiency.

Sound developed by a fan is a characteristic which is becoming increasingly important. Unfortunately, no method has yet been devised for accurately measuring the sound actually discharged into a duct system. The A.S.H.V.E. Research Laboratory, in cooperation with the U. S. Navy, has a program underway seeking to find a method. Many manufacturers list the average sound (for various fan operating conditions) measured at seven stations near the fan. These stations, as specified in the *N.A.F.M.* Test Code,[4] are located in a horizontal plane passing through the fan shaft, and are at a distance of one wheel diameter (but not less than 5 ft) from the fan. Such values are useful in comparing the relative sound generated by various types and sizes of fans under comparable operating conditions.

FAN LAWS[5]

The performances of fans of all types follow certain laws which are useful in predicting the effect upon performance of changes in the conditions of operation, the duty required of the installation, or the size of the equipment due to the space, power, or speed limitations. In the following laws, groups 1 to 6, Q = air volume and P = static, velocity or total pressure. The laws pertaining to fan size apply only to fans geometrically similar, *i.e.*, those in which all dimensions are proportional to some linear dimension denoted as *size*. If the size number is also linearly proportional, it may be used; otherwise, wheel diameter is commonly used as a size criterion.

1. Variation in Fan Speed:

   Constant Air Density—Constant System

   (*a*) Q: Varies as fan speed.
   (*b*) P: Varies as square of fan speed.
   (*c*) Power: Varies as cube of fan speed.

2. Variation in Fan Size:

   Constant Tip Speed—Constant Air Density

   Constant Fan Proportions—Fixed Point of Rating

   (*a*) Q: Varies as square of wheel diameter.
   (*b*) P: Remains constant.
   (*c*) RPM: Varies inversely as wheel diameter.
   (*d*) Power: Varies as square of wheel diameter.

3. Variation in Fan Size:
   At Constant RPM—Constant Air Density
   Constant Fan Proportions—Fixed Point of Rating
   (*a*) *Q:* Varies as cube of wheel diameter.
   (*b*) *P:* Varies as square of wheel diameter.
   (*c*) Tip Speed: Varies as wheel diameter.
   (*d*) Power: Varies as fifth power of diameter.

4. Variation in Air Density:
   Constant Volume—Constant System
   Fixed Fan Size—Constant Fan Speed
   (*a*) *Q:* Constant.
   (*b*) *P:* Varies as density.
   (*c*) Power: Varies as density.

5. Variation in Air Density:
   Constant Pressure—Constant System
   Fixed Fan Size—Variable Fan Speed
   (*a*) *Q:* Varies inversely as square root of density.
   (*b*) *P:* Constant.
   (*c*) RPM: Varies inversely as square root of density:
   (*d*) Power: Varies inversely as square root of density.

6. Variation in Air Density:
   Constant Weight of Air—Constant System
   Fixed Fan Size—Variable Fan Speed
   (*a*) *Q:* Varies inversely as density.
   (*b*) *P:* Varies inversely as density.
   (*c*) RPM: Varies inversely as density.
   (*d*) Power: Varies inversely as square of density.

Examples 1 to 4 illustrate the application of the preceding fan laws.

Example 1: A certain fan delivers 12,000 cfm at a static pressure of 1 in. of water when operating at a speed of 400 rpm and requires an input of 4 hp. If in the same installation 15,000 cfm are desired, what will be the speed, static pressure, and power?

$$\text{Speed} = 400 \times \frac{15{,}000}{12{,}000} = 500 \text{ rpm}$$

$$\text{Static pressure} = 1 \times \left(\frac{500}{400}\right)^2 = 1.56 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{500}{400}\right)^3 = 7.81 \text{ hp}$$

Example 2: A certain fan delivers 12,000 cfm at 70 F and normal barometric pressure (density 0.075 lb per cubic foot) at a static pressure of 1 in. of water when operating at 400 rpm, and requires 4 hp. If the air temperature is increased to 200 F (density 0.0602 lb) and the speed of the fan remains the same, what will be the static pressure and power?

$$\text{Static pressure} = 1 \times \frac{0.0602}{0.075} = 0.80 \text{ in.}$$

$$\text{Power} = 4 \times \frac{0.0602}{0.075} = 3.20 \text{ hp}$$

Example 3: If the speed of the fan of *Example 2* is increased so as to produce a static pressure of 1 in. of water at the 200 F temperature, what will be the speed, capacity, and power?

$$\text{Speed} = 400 \times \sqrt{\frac{0.075}{0.0602}} = 446 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \sqrt{\frac{0.075}{0.0602}} = 13.392 \text{ cfm (measured at 200 F)}$$

$$\text{Power} = 4 \times \sqrt{\frac{0.075}{0.0602}} = 4.46 \text{ hp}$$

Example 4: If the speed of the fan of the previous examples is increased so as to deliver the same weight of air at 200 F as at 70 F, what will be the speed, capacity, static pressure, and power?

$$\text{Speed} = 400 \times \frac{0.075}{0.0602} = 498 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \frac{0.075}{0.0602} = 14{,}945 \text{ cfm (measured at 200 F)}$$

$$\text{Static pressure} = 1 \times \frac{0.075}{0.0602} = 1.25 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{0.075}{0.0602}\right)^2 = 6.20 \text{ hp}$$

The fan laws stated may be combined to give other overall values. One useful combination is the product of laws 1 and 3 which gives the following relations:

Capacity varies as the ratio of size cubed, times the ratio of the rpm.

Pressure varies as the ratio of size squared, times the ratio of the rpm squared.

Horsepower varies as the ratio of the size to fifth power, times the ratio of the rpm cubed.

Example 5: Assuming that a fan with a 36 in. diameter blast wheel will deliver 12,000 cfm at 70 F at 1 in. static pressure, requiring 4.0 brake hp when operating at 400 rpm, what is the capacity, pressure and horsepower of a homologous fan having a 45 in. wheel at the same speed?

$$\text{Capacity} = \left(\frac{45}{36}\right)^3 \times \left(\frac{400}{400}\right) \times 12{,}000 = 23{,}400 \text{ cfm}$$

$$\text{Static pressure} = \left(\frac{45}{36}\right)^2 \times \left(\frac{400}{400}\right)^2 \times 1 = 1.56 \text{ in.}$$

$$\text{Horsepower} = \left(\frac{45}{36}\right)^5 \times \left(\frac{400}{400}\right)^3 \times 4 = 1.22 \text{ hp}$$

FAN PERFORMANCE CURVES

Fan performance curves are the graphical presentation (for constant speed and air density) of the relation of total pressure, static pressure, power input, and mechanical and static efficiency, to actual volume, for the desired range of volumes. Figs. 2, 3 and 4 illustrate performance (sometimes called characteristic) curves of various types of fans.

Centrifugal fans[6] may be roughly divided into three classes: (1) those with the tip of the blades curved forward in the direction of rotation; (2) those with straight radial blades; and (3) those with the tip of the blades inclined backward away from direction of rotation. They are also characterized as slow speed, moderate speed and high speed types, respectively, although the actual speed range of each may be wide and overlapping. The highest speed type may operate as high as 200 percent of the speed of the lowest speed type, to deliver the same volume of air and the same pressure. The differentiating curvature is always the tip of the blade, since the inlet edge, if inclined, is always curved forward to minimize the shock loss at entrance. Straight radial blades are most frequently found in pressure fans and material handling fans.

Centrifugal fans produce pressure from two independent sources: (1) from the centrifugal force created by rotating the enclosed air column, and (2) from the kinetic energy imparted to the air by virtue of its velocity

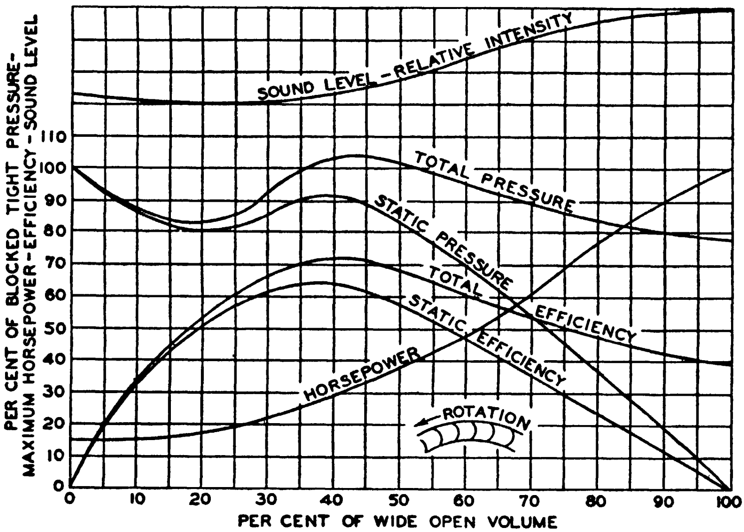


FIG. 2. PERCENTAGE PERFORMANCE CURVES OF A FORWARD-CURVED BLADE CENTRIFUGAL FAN

leaving the impeller. This velocity in turn is a combination of rotative velocity of the impeller and air speed relative to the impeller. When the blades tip forward, these two velocities are cumulative, and when backward, oppositional. Thus a fan with forward-curved blades depends less on centrifugal force for its pressure, and more on velocity pressure conversion in the scroll, with the result that it may run at relatively low speed. Conversely, a fan having backward-curved blades builds up more of its pressure by centrifugal force (a more efficient form of energy transfer) and less by velocity conversion and, therefore, must run at a higher speed. Likewise, a fan having forward-curved blades will produce the greatest capacity of any type of the same size when operating against no resistance.

Since the energy imparted to the air depends on the velocities,[7] and since the velocities are cumulative with a fan having forward-curved blades, the theoretical energy per pound of air rises rapidly with an increase of air delivery. With the velocities oppositional in the fan having backward-curved blades, the energy per pound of air may decrease, and in a fan having straight blades it is roughly constant. Thus the shape of the horsepower curve definitely identifies the blade angle.

Performance curves of a typical *forward-curved* blade centrifugal fan are shown in Fig. 2. The pressure rises from free delivery toward no delivery, with a characteristic drop at low capacities, because a large share of the pressure is being generated by conversion of velocity, which is small at low capacity. The maximum efficiency occurs at approximately maximum pressure. The horsepower curve reflects the increase in energy by rising rapidly from no delivery to free delivery. The sound is a minimum at maximum efficiency, and rises toward free delivery as the velocities increase.

Performance curves of a typical *backward-curved* blade centrifugal fan are shown in Fig. 3. The pressure is constantly rising from free delivery

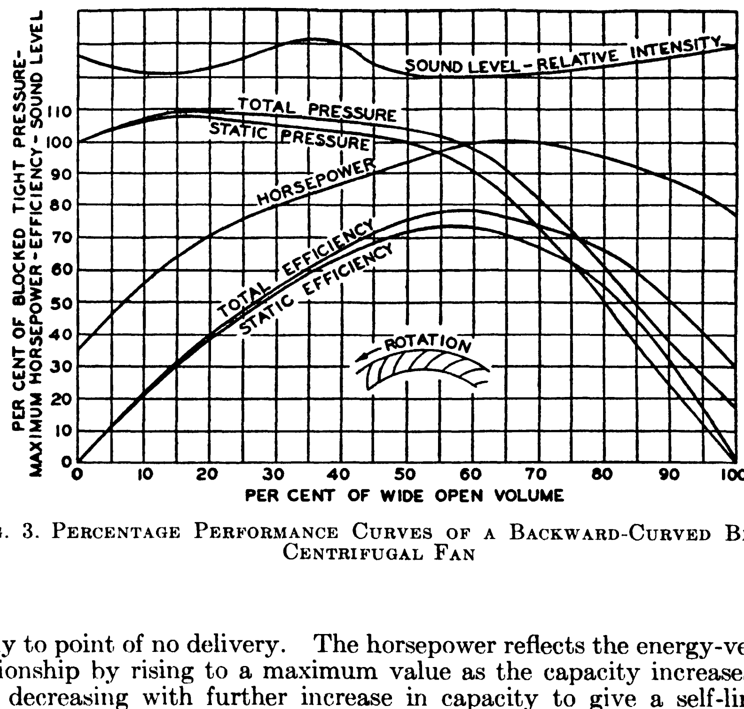


FIG. 3. PERCENTAGE PERFORMANCE CURVES OF A BACKWARD-CURVED BLADE CENTRIFUGAL FAN

nearly to point of no delivery. The horsepower reflects the energy-velocity relationship by rising to a maximum value as the capacity increases, and then decreasing with further increase in capacity to give a self-limiting horsepower characteristic. The maximum horsepower coincides approximately with the maximum efficiency. The sound is again a minimum near maximum efficiency, but is little or no higher at free delivery than at low capacities.

Between the extremes of forward and full-backward-curved blades, there exists a number of intermediate designs which show varying degrees of similarity to the curves in Figs. 2 and 3. A common variation is a fan having modified backward, single or double-curved blades and equipped with fixed inlet vanes. Such vanes applied to a partially backward-curved impeller give the steep, constantly rising pressure characteristic, and the self-limiting horsepower feature of the full-backward-curve. They also stabilize the flow entering the impeller when adverse flow conditions exist in the approach to the inlet.

Axial flow fans develop none of their static pressure by centrifugal force, but all from the change in velocity in passing through the impeller, and its conversion into static pressure. They are thus inherently high velocity fans, and are very dependent on blade conformation for good characteristics. For that reason, an air foil section, such as developed in wind tunnels for aircraft work[8] is frequently used. Since any shape of blade can only be correct for a narrow range of capacity at constant speed, the performance curves for any blade show definite characteristics. To absorb energy, the air must be given a tangential motion in passing the impeller,[7] and when operating against higher pressures, must have guide vanes (see vaneaxial fans) to obtain best efficiencies.

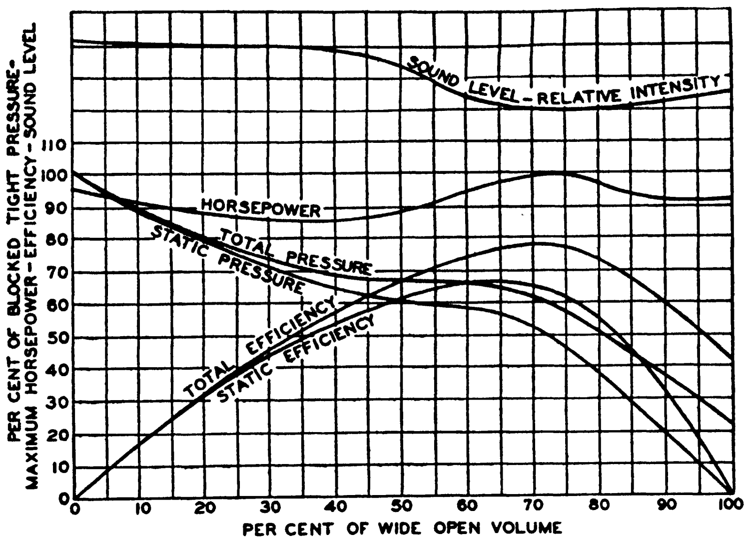


Fig. 4. Percentage Performance Curves of an Axial Flow Fan

While *axial flow* fans are inherently a higher capacity type than centrifugal fans, they, too, may be designed with widely varying characteristics. As with a centrifugal fan, the pressure rises generally from free delivery to no delivery, but tubeaxial and vaneaxial fans may have a drop in pressure when the capacity decreases below a certain volume, a condition also found in the case of the centrifugal fan having forward-curved blades. The pressure drop is caused by the same condition for both fans, *i.e.*, the static pressure is largely dependent on conversion of velocity pressure, and velocity pressure is small at low capacity. Tubeaxial and vaneaxial fans may also have performance curves resembling somewhat those of a centrifugal fan with backward-curved blades. Fig. 4 shows the performance curves for a typical design.

The horsepower curve may be flat with a self-limiting characteristic as in a backward-curved blade centrifugal fan, or it may have a generally downward trend from no delivery to free delivery with the maximum at no delivery, contrary to that of a centrifugal fan. The type of guide vanes in a vaneaxial fan has a distinct bearing on the shape of the horsepower curve. The maximum efficiency tends to occur at a percentage of free delivery capacity higher than for a centrifugal fan.

The sound curve, which may have a minimum value comparable to centrifugal fans, is again lowest near maximum efficiency, but has a characteristic rise when the fan is operating at low capacities and the *stall* point of the blade section is reached.

Since *propeller fans* are designed for operation near free delivery, less attention is given the regaining of velocity to static pressure, and the pressure curve rises constantly from free delivery to no delivery. The

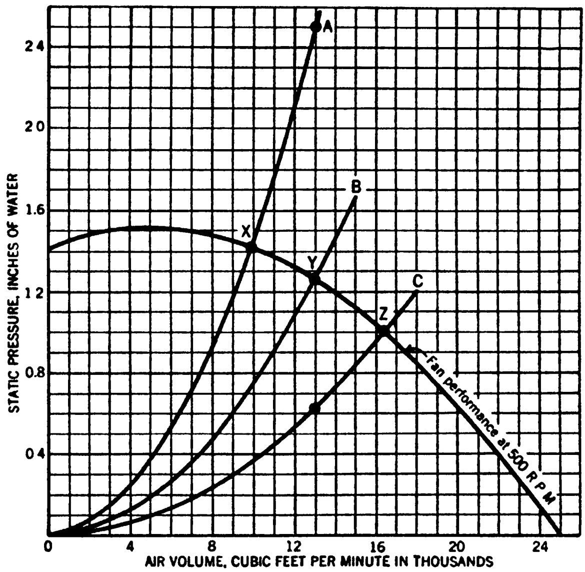


FIG. 5. PARABOLIC SYSTEM CHARACTERISTIC CURVES

horsepower is highest at no delivery, and decreases toward free delivery, in contrast to a centrifugal fan. Maximum total efficiency is obtained at a higher percentage of free delivery than for other types.

SYSTEM CHARACTERISTICS

Any ventilating system consisting of duct work, heaters, air washers, filters, etc., has a system characteristic which is individual to that system, and is independent of any fan which may be applied to the system. This characteristic may be expressed in curve form in exactly the same manner that fan characteristics may be shown. Typical system characteristic curves are shown as A, B and C in Fig. 5. These curves are drawn to follow the simple parabolic law in which the static pressure or resistance to flow of air varies as the square of the volume flowing through the system. Heating and ventilating systems follow this law very closely and no serious error is introduced by its use.

When a constant speed fan curve for a given size fan is super-imposed

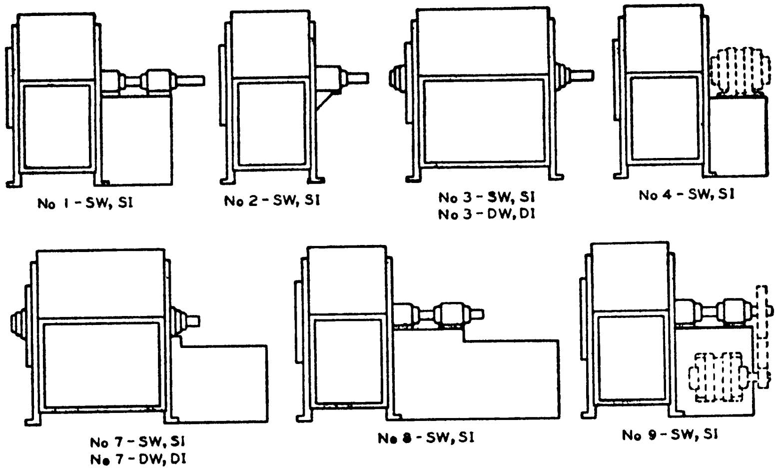


FIG. 6. ARRANGEMENT OF FAN DRIVES

Arr. 1, SW, SI. For belt drive or direct connection. Wheel overhung. Two bearings on base.

Arr. 2, SW, SI. For belt drive or direct connection. Wheel overhung. Bearings in bracket supported by fan housing.

Arr. 3, SW, SI. For belt drive or direct connection. One bearing on each side and supported by fan housing. Not recommended in sizes 27 in. diameter wheel and smaller.

Arr 3, DW, DI. For belt drive or direct connection. One bearing on each side and supported by fan housing.

Arr. 4, SW, SI. For direct drive. Wheel overhung on prime mover shaft. No bearings on fan Base or equivalent for prime mover.

Arr. 7, SW, SI. For belt drive or direct connection. Arrangement No. 3 plus base for prime mover. Not recommended in sizes 27 in. diameter and smaller.

Arr. 7, DW, DI. For belt drive or direct connection. Arrangement No. 3 plus base for prime mover

Arr. 8, SW, SI. For belt drive or direct connection. Arrangement No. 1 plus base for prime mover.

Arr. 9, SW, SI. For belt drive. Arrangement No. 1 designed for mounting prime mover on side of base.

upon a system characteristic curve, the relation between the two is at once apparent. The only point common to the two curves is the point at the intersection of the system characteristic curve and the fan characteristic curve, and it is at this point that the combination will operate. In Fig. 5, system characteristic curves A, B and C cross the fan characteristic curve at points X, Y and Z. The fan whose curve is shown, when applied to systems having characteristic curves A, B and C, will deliver 10,000, 13,000 or 16,400 cfm, respectively.

The curves in Fig. 5 also illustrate the effect of errors which may be made in calculating the resistance of a ventilating system. For instance, if a given system requires 13,000 cfm, and the resistance to flow of the system has been computed as 1.25 in. static pressure, such a system would be represented by system characteristic curve B in Fig. 5. If a 100 percent error had been made and the resistance were 2.5 in. instead of 1.25 in., then the system characteristic would be as shown in curve A, and would cross the fan curve at 10,000 cfm. Such an error would cause the flow of air to be decreased from a design volume of 13,000 cfm to 10,000 cfm. If the resistance to flow had been over-estimated and the resistance

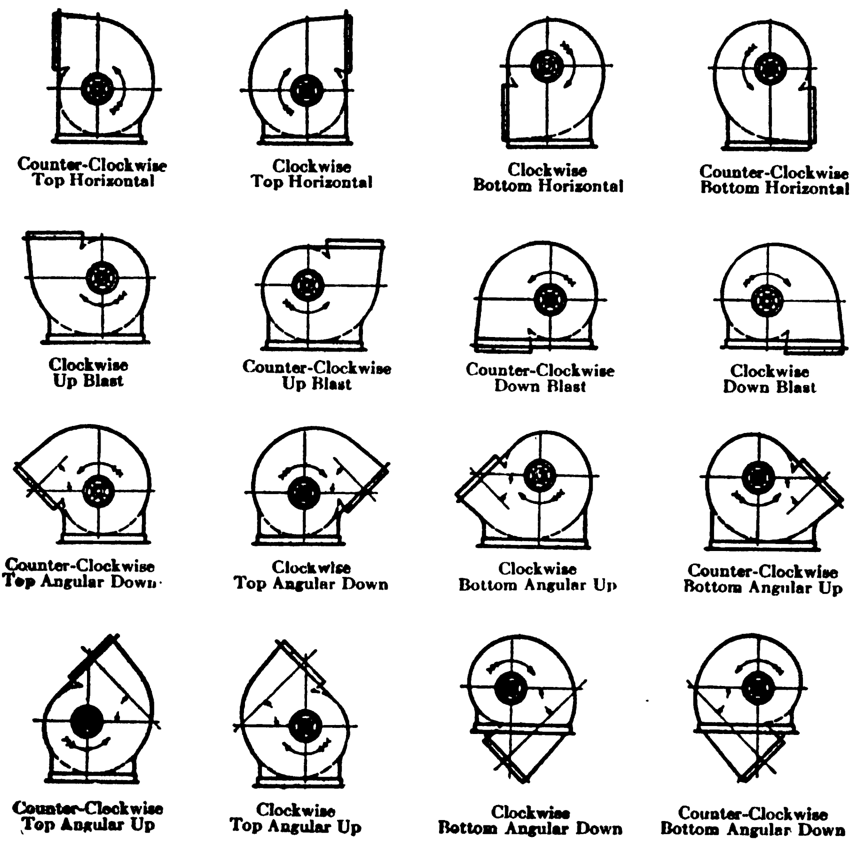


FIG. 7. DESIGNATION OF DIRECTION OF ROTATION AND DISCHARGE

Note: Direction of Rotation is determined from the drive side for either single or double width or single or double inlet fans. (The driving side of a single inlet fan is considered to be the side opposite the inlet regardless of the actual location of the drive.) For fan inverted for ceiling suspension, direction of rotation and discharge is determined when fan is resting on floor.

actually were 0.625 in., the system characteristic curve would be as shown in curve C, and the fan would deliver 16,400 cfm to the system instead of the design volume of 13,000 cfm.

In this example, extreme errors have been selected to emphasize the effect the square function of the system characteristic has in maintaining the fan performance within comparatively narrow limits. In the first example, a system estimated at half what it should have been resulted in a drop of 23 percent in volume; and in the second example, a system estimated at twice what it should have been resulted in an increase of 26 percent in volume.

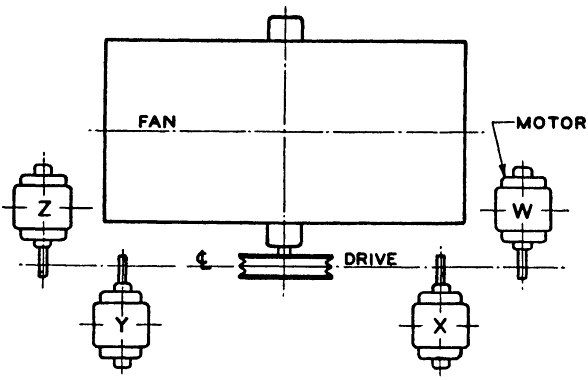
In some instances fans may be applied to variable flow systems. In such cases, the limiting systems may be plotted and the effect on fan performance examined. For instance, a system might have a characteristic curve between A, shown in Fig. 5, as one limit, and B as the other limit. The fan performance will then fall between points X and Y on the fan curve at a point determined by the system characteristics at that particular time. If A and B are the limiting characteristic curves of the systems, the fan performance will never be outside the points X or Y.

FAN ARRANGEMENTS

Centrifugal fan arrangements have been standardized by the *National Association of Fan Manufacturers.* Figs. 6, 7 and 8 show the accepted designation as to arrangement of drive, rotation, discharge and motor position, for belt drive. Axial flow fans are either belt driven or direct connected, in accordance with individual manufacturer's arrangements. Usually a choice of anti-friction or sleeve bearings is available.

FAN CONTROL

In some heating and ventilating systems it is desirable to vary the volume of air handled by the fan, and this may be accomplished by a



Location of motor is determined by facing the drive side of fan or blower and designating the motor position by letters W, X, Y or Z as the case may be.

FIG. 8. MOTOR POSITION, BELT OR CHAIN DRIVE

number of methods. Where the change is made infrequently, the pulley or sheave on the driving motor, or fan, may be changed to vary the speed of the fan and alter the air volume. Dampers may be placed in the duct system to vary the volume. Variable speed pulleys or transmissions, such as fan belt change boxes, or electric or hydraulic couplings, may be used to vary the fan speed. Variable speed motors and variable fan inlet vanes may also be used to adjust the fan volume. All of these methods will give control. From a power consumption consideration, a reduction of fan speed is most efficient. Inlet vanes save some power, while dampers save the least. From consideration of first cost, dampers usually are the lowest in cost. In some installations, adjustments of volume are desirable at various times during the day, or continuously. In others, an increased supply of air in summer, over that needed in winter, is demanded. The demands in each case will dictate which type of control is most desirable. Where noise is a factor, lowering the fan speed if possible is preferred as a control means, because of the resulting reduction in sound level.

In addition to the above types of control, *tubeaxial* and *vaneaxial* fans are sometimes made with adjustable blades to permit balancing the fan against the system, or making seasonal adjustment.

MOTIVE POWER

Heating, ventilating and air conditioning fans are usually driven by electric motors, although other prime movers may be used. The small sizes of fans, and especially those operating in the higher speed range, are equipped with direct-connected motors. For larger size fans, and those operating at lower speed, V-belt drives are generally used.

In selecting the size of motor for operating a fan, it is advisable to select at least the standard size next larger than the fan requirements. Direct-connected motors do not require so great a safety factor as belted units. Justification for liberal power provision exists only in systems where it is possible that larger volumes of air may be required at intervals, and made available by use of by-pass dampers, thus greatly reducing the system resistance. If such a system includes a fan with forward-curved blades, it would be necessary that the motor be sized for the maximum volume and duty. If such a system includes fans with backward-curved blades, the volume peak would not make it necessary to provide additional motor power. In selecting fans for such a system, sound ratings should be given careful consideration.

Where a system is constant, and has no provision for volume change that would materially reduce the resistance, and when the resistance calculations are reasonably precise, there is no necessity for too liberal a motor allowance (even where fans with forward-curved blades are used) if the fan has been properly selected. Fig. 5 shows that the system resistance varies as the square of the volume, and the fan static pressure varies approximately inversely as the volume, thus greatly offsetting the trend toward both increase in air delivery and motor load. Reference to Fig. 5 indicates that there is no justification for allowing large spare motor capacity. It is generally more economical to operate motors well loaded.

Since the power consumption of fans varies as the cube of the speed, very little starting torque is required of the motor. Refer to Chapter 39 for characteristics of various types of motors.

FAN SELECTION

The following information is required to select the proper type and size of fan:

1. Capacity in cubic feet per minute.
2. Static pressure or system resistance.
3. Air density if other than standard.
4. Type of application or service.
5. Arrangement of system.
6. Prevailing sound level or use of space served.
7. Nature of load.
8. Type of motive power available.

In order to facilitate the choice of apparatus, the various fan manufacturers supply fan tables or curves which usually show the following factors for each size of fan operating against a wide range of static pressures: (1) volume of air in cubic feet per minute (68 F, 50 percent relative humidity, 0.075 lb per cubic foot); (2) outlet velocity; (3) revolutions per minute; (4) brake horsepower; (5) tip or peripheral speed; and (6) static pressure. The most efficient operating point is usually shown by either bold-face or italicized figures in the capacity tables.

Often the service determines the type of fan. When operation occurs with little or no resistance, and particularly without a duct system, the propeller fan is indicated for convenience and low cost. When resistance is low the power required is low, and efficiency becomes a secondary importance. When a duct system is involved the choice is usually made between a centrifugal fan and a tubeaxial or vaneaxial. At times the capacity-pressure-speed relationship (specific speed)[9] dictates a choice. Usually, space, efficiency, sound, cost and serviceability must all be considered.[10] In general, centrifugal and axial fans are comparable in efficiency and sound, but the latter are lighter and require considerably less space, especially if arranged for straight-through operation. The comparison cannot be made on the cost of fans only, but the difference in cost of ductwork, mounting and servicing must be included. A vaneaxial is more efficient and quieter than a tubeaxial, but is more expensive, and frequently requires more space. While requiring less space than the centrifugal, the

TABLE 1. GOOD OPERATING VELOCITIES AND TIP SPEEDS FOR VENTILATING FANS

Static Pressure Inches of Water	Forward Curved Blade Fans		Backward Tipped and Double Curved Blade Fans		Tubeaxial and Vaneaxial Fans
	Outlet Velocity Feet per Minute	Tip Speed Feet per Minute	Outlet Velocity Feet per Minute	Top Speed Feet per Minute	Wheel Velocity[a] Feet per Minute
1/4	1000–1100	1520–1700	800–1100	2600–3100	1100–1500
3/8	1000–1100	1760–1900	800–1150	3000–3500	1250–1700
1/2	1000–1200	1970–2150	900–1300	3400–4000	1400–1900
5/8	1200–1400	2225–2450	1000–1500	3800–4500	1500–2100
3/4	1300–1500	2480–2700	1100–1650	4200–5000	1650–2350
7/8	1400–1700	2660–2910	1200–1750	4500–5300	1800–2500
1	1500–1800	2820–3120	1200–1900	4800–5750	1900–2700
1 1/4	1600–1900	3162–3450	1300–2100	5300–6350	2150–3000
1 1/2	1800–2100	3480–3810	1400–2300	5750–6950	2350–3300
1 3/4	1900–2200	3760–4205	1500–2500	6200–7550	2500–3550
2	2000–2400	4000–4500	1600–2700	6650–8050	2700–3800
2 1/4	2200–2600	4250–4740	1700–2800	7050–8550	
2 1/2	2300–2600	4475–4970	1800–2950	7450–9000	
3	2500–2800	4900–5365	2000–3200	8200–9850	

[a] Wheel velocity is the axial mean air velocity through the inside diameter of the housing cylinder at the point of wheel location.

axial flow fan is inherently less accessible for service. When high-temperature air or air containing corrosive elements is being conveyed, motors and bearings should be located outside of the air stream. This requirement may determine the type of fan to be used. Where the system resistance is indefinite or variable, the pressure, horsepower and noise characteristics of centrifugal fans usually indicate their selection. Under such conditions, a steep and constantly rising pressure curve permits less variation in air delivery when the resistance varies. Likewise, a flat sound curve minimizes the change of moving into a region of increased noise. A fan having a high efficiency over a wide range is more desirable than one which reaches an even higher maximum efficiency, but decreases more rapidly on either side of a narrow range. A self-limiting horsepower curve may permit more accurate selection of motor size.

The selection of size of fan usually involves balancing cost and space against sound and efficiency. Unless the pressure involved is so high that a smaller fan running at greater speed requires a higher class[2] of construction, the smallest fan is the cheapest in first cost of the fan only. However, as the cost of the driving equipment is also involved in the total installation cost, fan efficiency must be considered for that reason as well as its bearing on the cost of operation. In some cases where large fans operate long hours per year, selection at absolute maximum efficiency is indicated. Generally, however, the power saving by selecting for optimum efficiency does not justify the extra cost, and a slightly smaller fan gives the best balance of cost and efficiency. Reference to Figs. 2 to 4 shows that all types tend to have a minimum sound near maximum efficiency so, when noise is a consideration, selection approaching maximum efficiency is indicated. Too large a fan may not only mean an unnecessary investment and an increased power consumption and sound, but may also give faulty performance if of a type having an unstable pressure characteristic at low capacity.

Table 1 shows the outlet velocities and impeller tip speeds recognized as good practice for various static pressures for centrifugal, tubeaxial and vaneaxial fans applied to average heating, ventilating and air conditioning applications. Fans for churches, schools, residences and other buildings having a low prevailing noise level should be selected for lower than average outlet velocities.

FAN INSTALLATION

In designing heating, ventilating and air conditioning systems, the characteristics of the fans available for use therewith should not be ignored. If double inlet fans or multiple fans in parallel are used, care must be taken that both inlets have the same free area and general approach conditions.

The dimensions of the ductwork and the size of the various devices whose individual resistances determine the static pressure, dictate the fan selection. Often a minor modification of the system may permit use of a smaller motor, and even a lower class[2] of fan, with considerable saving of cost. Invariably, the sound generated is affected, as fans operating at high pressure produce more noise than at lower pressure (see Chapter 40). Minimizing the resistance may be the best insurance against noise. On the other hand, sometimes the lowest overall cost results from selecting the minimum size of system equipment, and then installing adequate acoustical and vibration treatment.

All ducts should be connected to fan outlets and inlets by means of unpainted canvas or other flexible material. Access should be provided in the connections for periodic removal of any accumulations tending to unbalance the rotor. When operating against high resistance, or when ambient noise levels are low, it is preferable to locate the fan in a room removed from occupied areas or acoustically treated to prevent sound transmission. The lighter building constructions which are common today, make it desirable to mount fans and driving motors on resilient bases designed to prevent transmission of vibrations through floors to the building structure. Conduits, pipes and other rigid members should not be attached to fans. Noises due to high velocities, abrupt turns, grilles and other items not connected with the fan, may be present. Treatment of such problems, as well as the design of sound and vibration absorbents, are covered in Chapter 40.

FAN APPLICATIONS

Many fan applications and the corresponding types of fan commonly used are listed in the following paragraphs. Reference is also made to the chapters where the applications are discussed.

Central System Supply Fans (Chapter 29) are usually of the centrifugal type, since this application requires a wide range of satisfactory and quiet operation against relatively high pressures. They can readily be connected to apparatus of large cross-section on the inlet side, and to relatively small ducts on the outlet side.

Comparative sizes have been standardized among manufacturers,[2] and most rating tables cover a range of 700 to 500,000 cfm, and static pressures from ¼ to 15 in. of water.

Central system exhaust fans are predominately centrifugal, but the space conservation of the axial is being increasingly utilized. Tubeaxial and vaneaxial fan sizes are not yet standardized, but several manufacturers list capacities from 2000 to 100,000 cfm, and static pressures up to 3 in. of water.

Exhaust fans are found in all types. *Wall fans* are predominantly of the propeller type, since they operate against little or no resistance. They are listed in capacities from 1000 to 75,000 cfm. They are sometimes incorporated in factory-built pent houses or roof caps, or are provided with matching automatic louvers. *Hood exhaust fans* (Chapter 45) involving ductwork, are predominantly centrifugal, especially if handling hot, corrosive or erosive fumes, where it is best to keep the bearings and drive remote from the air stream. Otherwise, axial fans are applicable, and where little or no ductwork is involved, propeller fans are suitable. *Spray booth exhaust fans* (Chapter 45) are frequently centrifugal, especially if built into self-contained booths. Tubeaxial fans lend themselves particularly well to this application where ease of cleaning and of suspension in a section of ductwork are advantageous. For such application built-in cleanout doors are desirable. *Material handling fans* (Chapter 45) are always straight radial (or modified) blade centrifugal type. They are of heavier construction, and have fewer blades and greater clearances than ventilating fans. Many characteristics are compromised to provide wear resistance and ease of maintenance. They are commonly listed in capacities from 600 to 60,000 cfm, and static pressures up to 15 in. water.

Mine fan applications[11] vary greatly and require fans ranging from small portable units for local ventilation, to immense slow speed centrifugal fans, often steam engine driven, for general or emergency ventilation. Vaneaxial fans are well suited to mine ventilation. For underground location, their compactness saves on cost of excavations, and above ground, their ready reversibility is valuable in emergencies, even if reversal causes a reduction in capacity.

Marine fans (Chapter 47) are of both centrifugal and vaneaxial types. The latter are particularly well adapted to both combatant and noncombatant ships, where compactness and light weight are invaluable.

Unitary systems, i.e., unit heaters, unit ventilators, unit humidifiers, unit air conditioners, unit air coolers and unit evaporative condensers are equipped with centrifugal or propeller fans, the latter usually being limited to the relatively small suspended type where no ductwork is involved. Fans for units having considerable internal or possible external resistance, are mostly of the forward-curved blade, or so-called mixed flow centrifugal type. The latter is really a centrifugal type with axial inlets, having a pressure curve resembling a backward-curved blade centrifugal fan. Both of these types have the high capacities (in relation to displacement) requisite for a compact unit. Ratings are frequently given for these units as separate fans, as well as in conjunction with the various internal resistances.

In multiple units on a common shaft they are listed up to 40,000 cfm capacities.

Cooling tower fans (Chapter 34) are predominantly of the propeller type, but axial types are also used for packed towers, and occasionally a centrifugal fan is used to supply forced draft.

Circulating fans are invariably of propeller or disk type, and are made in a vast variety of blade shapes and arrangements. They are designed for pleasing appearance, as well as utility.

General purpose fans are centrifugal fans of conventional design, built for service in the lower capacity ranges. They are built with the fan wheel mounted on the motor shaft, or connected to a self-contained belt driven arrangement. They are listed in capacities from 100 to 20,000 cfm, and static pressures up to $1\frac{1}{2}$ in. water.

Kitchen fans for domestic use are small propeller fans arranged for window or wall mounting, and with various useful fixtures. Their capacities range from 300 to 800 cfm.

Attic fans are used during the warm seasons to draw large volumes of outside air through a house or other building whenever the inside temperature exceeds the outside, and thereby utilize the cooling effect of the relatively cool evening or night air. Research by the A.S.H.V.E.[12] indicates that a two to three-minute air change is desirable in the North; in the South, a one-minute change is recommended to provide the additional cooling effect of air motion.

Fans may be centrally located in an attic or other unused space, or in a hallway, and arranged to draw proportionately from several rooms; or local window units may be installed in a single room. Central units draw from the living quarters and discharge into the attic, whence the air escapes through windows or grilles; or the air may be drawn through grilles into the attic with the fan discharging directly outdoors. Discharge openings on the lee side are preferred.

Attic fans are usually of propeller type, and should be selected to operate at low velocities to minimize noise. Noise is more of a problem on local units, but care should be taken to prevent transmission of noise or vibration on all installations, because they operate during sleeping hours. Central units are available in sizes from 3,000 to 30,000 cfm, and window units up to 8,000 cfm.

Fans For Special Applications

Fans are used to handle many gases other than air at normal temperature. These range from heated air to many types of fumes, vapors, and industrial gases which may be corrosive, explosive, toxic, radio active or merely noxious.

For handling corrosive gases fans should be constructed of a material suitable for the particular gas being handled, although at times it is more practicable to plan on replacing a unit of standard material at more frequent intervals. Explosive combinations usually dictate a non-sparking fan in which either all of the fan, or only those parts which might strike together if mal-adjusted, are built of non-sparking materials. Noxious, toxic, radio-active or pure or valuable gases call for special construction to prevent leakage. This usually consists of a welded gas-tight housing, using flanged inlet and outlet, and some form of shaft seal.

In most of these special applications, since it is desirable to keep all bearings, drives and motors outside the gas stream, centrifugal fans are usually used and in arrangements 1, 2, 4, or 8 (Fig. 6). The exception is in the case of fans handling explosive mixtures, where propeller or axial fans are permissible when built with explosion-proof motors and non-ferrous wheels.

Fans handling hot air or gas are generally of the centrifugal types and follow arrangements 1 or 8 so that bearings, drives and motors are remote from the heat. If of double inlet type, they have inlet boxes and long shafts to keep the bearings outside the stream. Propeller or axial types are rare when the temperature exceeds 150 F. With temperatures above 600 F special heat resistant materials may be required for the rotors. In addition to a remote location for the bearings, external cooling is frequently required when the temperature exceeds 200 to 250 F. This is often obtained by use of air cooled or water cooled jackets. With anti-friction bearings a heat radiating disk or a small circulating impeller between the fan and the nearest bearing is sometimes sufficient. Oil lubrication rather than grease lubrication is commonly recommended.

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CHAPTER 33

AIR CLEANING

Filters: Airborne Particulate Matter, Viscous Impingement Type Filters, Dry Air Filters, Electric Precipitators, Air Filter Performance, Testing, Selection and Maintenance, Filter Installation, Adsorption of Vapors; **Dust Collectors:** Degree of Dust Removal Required, Factors Affecting Selection, Types and Application of Dust Collectors, Electrostatic Precipitators, Fabric Collectors, Wet and Centrifugal Collectors, Settling Chambers, Sonic Agglomerators, Testing Methods

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AIR cleaning devices remove contaminants from an air or gas stream. They are available in a wide range of designs to meet various air cleaning requirements. Degree of removal required, quantity and characteristics of the contaminant to be removed, and conditions of the air or gas stream will all have a bearing on the device selected for a given application. Definitions and a discussion of contaminant characteristics are given in Chapter 8, together with some consideration of their origin.

Air cleaning devices are divided in this chapter into two basic groups: Air Filters, described in Part I, and Dust Collectors, described in Part II.

Air filters are designed to remove dust concentrations such as are found in outside air, and are employed in ventilation, air conditioning, and heating systems where dust content seldom exceeds four grains per thousand cubic feet of air.

Dust collectors are designed for the heavier industrial loads encountered in local exhaust ventilation where dust content (loading) ranges from 100 to 20,000 grains per thousand cubic feet of air. This heavy loading from industrial dust control systems is often overlooked. Because of it air cleaning devices in the air filter group can seldom be used for control of process dust in industrial applications.

PART I—AIR FILTERS

The conventional mechanical-type air filters are installed in ventilating systems to remove the airborne dusts which tend to settle out in the still air of the ventilated space and become a nuisance. Such dust comes under the classification of temporary atmospheric impurities listed in Fig. 1, Chapter 8, and includes most of the pollens, house dust, and similar allergens which motivate attacks upon persons of allergic sensitivity.[1]

Electrostatic precipitators, in addition to removing the temporary atmospheric impurities, exhibit high efficiency in the removal of fine dust or liquid particles which have no gravitational settling tendency.

The air filters classified as viscous impingement types are widely used in general ventilating systems, and will usually justify their cost through a reduction in housekeeping costs in the ventilated space, and in the protection of the component parts of the ventilating system itself, or both.

For application in locations where the larger part of the atmospheric impurities are of fine particle size, such as smoke and fumes, more effective dry filters and electric precipitators are proving economical. By use of combinations of filters, advantage may be taken of specific characteristics of the various types of filters. There are many applications in which the air is passed through a series of two or more different types of filters so that the special advantages of each type are utilized to obtain optimum air cleaning and least maintanance cost.

A description of various types of air filters available commercially will be found under *air filters* in the catalog section of THE GUIDE.

AIRBORNE PARTICULATE MATTER

Airborne solid matter, from the viewpoint of air filtering, is considered as lint and dust, the latter including smoke and fumes. Some lint originates outdoors, as animal hair, vegetable fibers, etc., but much is generated within buildings by the wear and brushing of fabrics in the form of clothes, draperies, carpets, etc. Lint is comparatively easy to capture in an air filter because of its comparatively great length. So far as air filter performance and testing are concerned, lint is chiefly important because of its tendency to impede or stop the flow of air through the filter. In general, lint, if not captured, will accumulate in corners and under furniture in a building in areas of slight air motion, and in some cases may seriously obstruct heating and cooling coils. Dust settles, or is precipitated by heat or air motion, upon furniture, fixtures and walls, and the only satisfactory treatment is washing or re-painting. Dust is more difficult to capture than lint, and, obviously, small particles are more difficult to capture than large ones. The air cleaning problem is complicated by the vast difference in size of dust particles, the range of which is shown in Fig. 1, Chapter 8. Further complications arise, due to variations in kind and quantity of airborne contaminants with the geographic location of the installation as well as the position of outside-air and return-air intakes. Due consideration should be given to the type and amount of ambient dust at the point of intake. The type of filter required is frequently dictated by the kind and amount of contaminant to be encountered in the installation under consideration.

Even if the discussion is limited to the range from 0.2 micron to 50 microns, that is, between the smallest particle observable in the conventional microscope and the smallest particle visible to the naked eye, this range is so far outside the usual experience that it is difficult to visualize. If particles could be examined through a super microscope having a magnification of 250,000 diameters, a tobacco smoke particle of 0.1 micron would appear to be 1 in. in diameter, or approximately the size of a golf ball; a soft coal smoke particle 0.3 micron in diameter would appear like a baseball; a ragweed pollen grain of 20 microns in diameter would appear 16.5 ft in diameter, while the 50 micron particle (just visible to the naked eye and able to pass through a 270 mesh screen) would appear to be 50 ft in diameter. Consideration of this range in particle size from a golf ball to a sphere 50 ft in diameter will emphasize the difficulty of devising any single test to measure adequately the performance of air cleaning devices under all conditions of service.

As a general rule, the removal of the coarser dust particles and lint from the ventilating air produces tangible results in so far as cleanliness in a house or building is concerned, because much of the finer dust remains suspended in the air and is removed from the building by the circulating air. However, since some of the fine dusts, especially smoke and fume particles, are undoubtedly deposited by means other than settling, such as electrical or thermal precipitation[2, 3] and by contact, the ability to remove small particles is desirable in an air cleaner if it can be obtained at not too great a cost.

VISCOUS IMPINGEMENT TYPE FILTERS

The viscous impingement type of filters consist of relatively coarse media constructed of a suitable fiber, screen, wire, mesh, metal stamping or plates, or a combination of medias. The filter may be either of the unit type manually-cleaned, the replaceable type, or the automatic self-cleaning design type.

The medium in a viscous impingement type filter is usually a fiber pack for non-automatic types, or a series of metal plates for automatic self-cleaning types. In either case, the medium is treated with a viscous substance, often an oil or grease, called the *adhesive* or the *saturant*, intended to retain dust particles which come in contact with it. Also, in either case, the arrangement is such that the air stream is broken up into many small air streams, and these are caused to change direction abruptly a number of times in order to throw the dust particles, by momentum, against the adhesive. Several desirable characteristics of an adhesive for air cleaners of this type are: (1) its surface tension should be such as to produce a homogeneous film or coating on the filter medium; (2) the viscosity should vary only slightly with normal changes of temperature; (3) it should prevent the development of mold spores and bacteria on the filter medium; (4) the liquid should have high capillarity, or ability to wet and retain the dust at all operating temperatures; (5) evaporation should be slight; (6) it should be fire resistant; (7) it should be odorless.

Various fibrous materials have been used as filtering media in unit filters of the viscous impingement type. These include glass fiber, steel wool, similar *wool* of non-ferrous metals, wire screen, animal hair, hemp fibers, and other materials. In such filters, the medium is often packed more densely on the discharge than on the approach side, in order to increase the dust holding capacity. This results in a selective arrestance of dust with the larger particles nearer the approach face. The arrangement also permits some penetration of lint into (but not through) the filter, so that the amount of lint which can be tolerated on the filter is also increased. Due to plane surface area the viscous impingement type filter, however, may be inferior to some dry types if the air carries a high percentage of lint.

The resistance of air filters obviously increases with the rate of air flow through them. Face velocities of about 300 fpm, and resistances in the range from 0.1 to 0.2 in. water, when the device is new and clean, are usual for ventilation system filters. Special filters with low resistances are available for use with gravity warm air furnaces, and for other applications where only low pressure is available.

Filters designated as high velocity units are also available. In most instances these filters employ a uniform media pack and give best performance at velocities about 500 fpm. Impingement type air filters should hold a substantial amount of adhesive so that the dirt collected by the filter will be retained. The design and construction should also provide a goodly proportion of free space through the media so that high velocities can be used without excessive resistances. The design should assure that the media throughout the thickness of the filter will be effectively utilized and therefore, the filter should be designed to minimize the formation of a clogging dirt mat on the filter face. High velocity filters are apt to be adversely affected if the amount of lint in the air to be cleaned is excessive.

The resistance of filters increases with dust or lint loading, and it is the resistance due to this cause which ordinarily necessitates servicing. The rate of loading obviously depends upon the amount as well as the kind of dust in the air, and for this reason, periods between servicing cannot be predicted. Manometers are often installed to indicate the pressure drop across filter banks, and they serve to indicate when the filter requires cleaning. The pressure drop tolerated differs between operators and system designs. The resistance of a filter bank can be kept desirably low by periodically servicing some, but not all, of the units in the bank at one time. It is to be noted that a decline in efficiency may be the limiting factor in filter life rather than increased resistance due to load.

The method of cleaning viscous impingement unit filters differs for different types of filters and kinds of dust. Much dry dust or lint can often be removed by rapping the filter.

Throw-away filters are constructed of inexpensive materials, and are designed to be discarded after one use. The frame is frequently a combination of cardboard and wire.

Cleanable filters usually have metal frames. Various cleaning methods have been recommended including: air jet, water jet, steam jet, washing in kerosene, and dipping in an oil. The latter may serve both to clean the filter and add the necessary adhesive.

It is not mandatory that unit filters be removed from their metal frames for cleaning outside of the system. Cleaning of unit filters in place is feasible by means of hot water sprayed from a hose or by means of fixed nozzles to accomplish washing and adhesive application. Where filters are cleaned in place, provision should be made to collect and drain the water so as to prevent leakage from the filter housing.

Automatic Viscous Filters

In an automatic air filter, means are provided to remove the dust from the medium mechanically. Automatic filters with moving cloth media have been constructed, but are not now in wide use.

The medium in a typical automatic filter at present consists of a series of specially formed metal plates mounted on a pair of chains. The chains are mounted on sprockets located at the top and bottom of the filter housing, so that the filter medium can be moved as a continuous curtain up one side and down the other side of the sprockets. The arrangement is such that, at the bottom, the medium passes through a bath of special oil which both serves to remove the dirt from the plates, and acts as an adhesive when the cleaned plates next pass through the air stream. The plates forming the filtering medium or curtain usually overlap each other, and due to their special shape, many small air passages are formed between them. These air passages turn abruptly one or more times in order to give the impingement effect.

An electrically driven rotating device is usually supplied with an automatic filter. The device may be set to move the curtain periodically, or a special switch, actuated by pressure drop, may be used to govern its motion. In operation, the resistance of an automatic filter will remain approximately constant as long as proper operation is obtained. A resistance of ½ in. water at a face velocity of 500 fpm is typical of this class.

DRY AIR FILTERS

The media in such filters are usually fabrics or fabric-like materials. Media of wool felt, cotton batting (both glazed and unglazed), cellulose fiber and other materials have been used commercially. The medium in a filter of this class is usually supported by a wire frame in the form of pockets or V-shaped pleats in order to increase the area exposed to the passage of air. A 2-ft square unit may contain from 15 to 30 sq ft of medium.

Dry air filters, by virtue of the large area of medium used, have a comparatively high lint holding capacity. The efficiency of dry type filters is usually higher than that of viscous impingement filters, while the life of the former based on dust holding capacity, may be lower. Dust tends to clog the fine pores or openings of dry filters more quickly, thereby causing a higher rate of resistance rise than for viscous impingement filters. Effi-

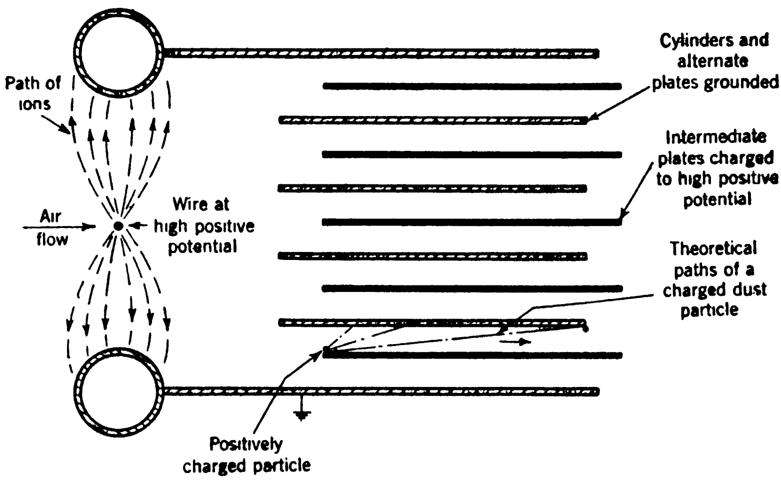

FIG. 1. DIAGRAMMATIC CROSS-SECTION OF IONIZING TYPE ELECTROSTATIC PRECIPITATOR

ciency with very fine particles equivalent to that of electrostatic precipitation is not uncommon with certain types of dry filters. A comparatively deep bed, ½ to 1 in. thickness, of filtering media having individual fibers in the order of 1 micron in diameter, is required for such high efficiency.

Dry filters may consist of a cleanable medium held in permanent frames, or a throw-away or replaceable filtering medium held in permanent frames, or the entire filter may be of the throw-away type. Usually the filtering medium alone is replaced after having collected its full dust load.

ELECTRIC PRECIPITATORS

Two principal types of electrostatic precipitators are available for air filter application: (1) ionizing type collectors where particles are given a definite charge by passing through an ionization zone, and (2) charged media type collectors where the energized, collecting medium induces a charge on the particles.

Ionizing Type Electrostatic Precipitator

The fact that a particle exposed to an electric field will assume a charge and migrate toward one of the electrodes, has been utilized for some years in boiler plants as a means of smoke abatement. The same principle has been used in equipment developed for air cleaning in air conditioning without generating ozone in intolerable quantities. The air stream in a precipitator passes first through a relatively high-tension electric field, known as the ionizing field, and then through a secondary field where the precipitation of the dust occurs. The arrangement is as shown in Fig. 1.

In a typical case, a potential of 12,000 volts may be used to create the ionizing field, and some 5000 volts between the plates upon which the precipitation of dust occurs. These voltages, which are capable of shock to personnel similar to that of a spark plug, necessitate some safety measures. A typical arrangement provides means for automatically making the unit inoperative when a door to the precipitator is opened. To resume operation the procedure necessitates closing the door and turning an electric switch, the latter of which should be located at a reasonable distance from the equipment. The voltages necessary for the operation of the precipitator are usually obtained from an alternating current building service line by means of a step-up transformer. Precipitation with alternating current is possible, but is not nearly so effective, so the current is usually rectified by means of vacuum tubes. The transformer and tubes are collectively termed the power pack.

The precipitators of this type offer negligible resistance to air flow and, therefore, care must be exercised in arranging the duct approaches on the entering and leaving sides of the precipitators in order that the air flow may be distributed uniformly over the cross-sectional area. The efficiency of the precipitator is very sensitive to air velocity, and the device itself has much less tendency to rectify the air stream than filters which have much higher resistances. In most systems, resistance is deliberately added in the form of a perforated plate attached to the air approach side of the precipitator in order to obtain a uniform distribution of air. Such plates, however, are only partially effective. The plate also serves to prevent lint agglomerates, paper, and similar foreign material from reaching the precipitator proper.

Electric precipitators of the ionizing type are available in both automatic and non-automatic types. The plates of non-automatic precipitators are commonly coated with a light oil as an adhesive. Cleaning is accomplished with a water hose and, for this reason, the bottom of the equipment is made water tight and provided with a drain. In one automatic type, the grounded groups of precipitation units are mounted on chains, and are alternately dipped in oil and exposed to the air stream with an action similar to that of an automatic impingement filter. The charged precipitator plates are mounted stationary in the air stream. These plates collect only a minor portion of the total dirt. They are cleaned and recoated with adhesive by means of a wiping device attached to the revolving chain.

Charged Media Electrostatic Precipitators

The charged media type precipitator consists of a dielectric filtering medium, usually arranged in pleats as in typical dry filters. No ionization means are employed. The dielectric filtering medium may consist of glass fiber mat, cellulose mat or other similar material, and is supported on or in contact with a gridwork consisting of alternately grounded and charged members, the latter usually being held at a potential of 12,000 volts d.c. An intense and non-uniform electrostatic field is thus created through the dielectric medium. Airborne particles approaching the field are polarized and drawn toward filaments or fibers of the media. The general arrangement illustrating this type of filter is shown in Fig. 2.

The precipitator of this type offers resistance to air flow, when clean, on the order of 0.10 in. of water at 250 fpm for velocity and, unlike the ionizing type, the resistance of the charged media type electrostatic precipitator rises as dust is accumulated on the media. Because of these characteristics the filter tends to equalize the air distribution over the face of the filter. Like typical replaceable media mechanical filters, the charged media precipitator is serviced by replacing the filtering medium.

AIR FILTER PERFORMANCE AND TESTING

Air filters are generally rated in terms of the total air flow for which they are designed, expressed in cubic feet per minute. Face velocity is defined as the average velocity of the air entering the filter, and it is determined

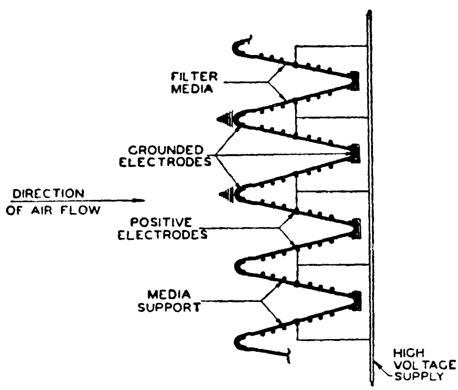

FIG. 2. CROSS-SECTION DIAGRAM OF CHARGED MEDIA TYPE ELECTROSTATIC PRECIPITATOR

by taking the air flow and dividing it by the area of the duct connection to the cleaner in square feet. Filters are often rated at a face velocity in the range of 250 to 500 fpm. Resistance to air flow is usually measured in inches of water column. The resistances of filters when new and clean, and when operated at rated capacity, are generally available from the manufacturer (see *Catalog Data Section*). A suitable allowance should be made in system design for increase of filter resistance due to the accumulation of dust.

The ability of air cleaners to clean air is called the efficiency or the arrestance, and may be denoted by the symbol E. The efficiency of an air cleaner differs with the size and nature of the dust on which the cleaner operates. The efficiency of an air cleaner, algebraically expressed, is

$$E = \frac{D_1 - D_2}{D_1} \tag{1}$$

where

D_1 = amount of dust per unit volume in uncleaned air.
D_2 = amount of dust per unit volume in cleaned air.

Several methods have been investigated for evaluating D_1 and D_2. The particle count method is not used for efficiency evaluation, except in investigation of filter performance on specific particles such as pollen, or on certain industrial dusts harmful to health. Dust particles can be captured on microscope slides by means of one of the various kinds of impingement devices. The process is useful if inspection and analysis of dust are desired, but particle counting is not sufficiently precise for evaluating the efficiency of a cleaner operating on a heterogeneous dust.

The weight method of evaluating efficiency has found wide utility, and was recognized by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and incorporated in a code.[4] For this test, a known weight of a prepared dust is injected into air supplied to the filter, and the quantity of dust in the cleaned air is determined by extracting and weighing the dust from a known volume of the cleaned air. Dust extraction from the air is accomplished by drawing the air through a porous crucible or thimble by means of a high vacuum.

Caution is to be exercised in interpreting published air filter arrestance data, since the test efficiency may be somewhat higher than that which will be obtained in an installation with respect to the ventilated space.

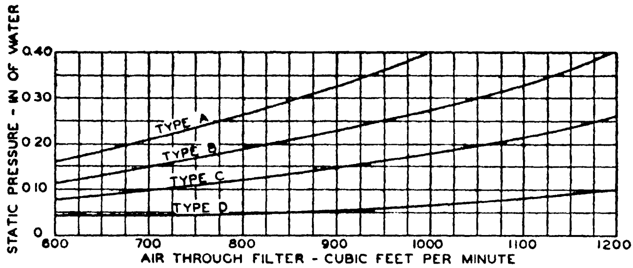

FIG. 3. RESISTANCE TO AIR FLOW OF TYPICAL UNIT AIR FILTERS

The *dust-spot* or *blackness* test for cleaner efficiency was developed at the *National Bureau of Standards*.[5] The test consists of drawing samples of cleaned air and of uncleaned air through filter papers simultaneously. The ratio of the areas of paper through which the air samples are drawn, and the ratio of the amount of air drawn through the papers, are adjusted during successive trials to yield spots of approximately equal blackness on the papers. The ratios of the areas and of the volumes of the air samples are then indicators of the filter effectiveness. A special photometer is provided for comparing the blackness or opacity of the papers by transmitted light. For tests of ordinary air filters by this method, a dust is injected into the air stream. The dust consists of precipitated smoke particles from a Cottrell precipitator used in a local power plant for smoke abatement. For tests of electrostatic air cleaners, no dust is added to the air. Tests are commonly made with the dust existing in the air at the location of the installation on a clear day.

Dust-holding capacity is defined as the amount of dust which a filter can retain and have a resistance less than some arbitrary value. The term applies only to non-automatic air cleaners. Determination of dust-holding capacity is an objective of each test under the A.S.H.V.E. Standard Code.[4] Curves are obtained during such tests to show the relation between dust load and resistance.

Fig. 3 indicates the general range of resistance to air flow through unit air filters. Type A is a dense pack used in bacterium control; Type B is a medium pack used for general ventilation work; Type C is a low resistance unit, for use where low resistance is the important factor and maximum cleaning efficiencies are not essential; and Type D is a high-velocity viscous-impingement filter.

At the *National Bureau of Standards* two injectors are provided on the air cleaner testing apparatus. One injector is used to contaminate the air stream with Cottrell precipitate, previously described. This dust is used to make both efficiency determination and dust-holding capacity tests. The other injector contaminates the air stream with cotton linters with which lint-holding capacity tests are made. The curves in Fig. 4 illustrate the difference in the characteristics of two filters, one a viscous-impingement type and the other a dry filter with a cellulose fiber medium. The two injectors can be operated either separately or simultaneously.

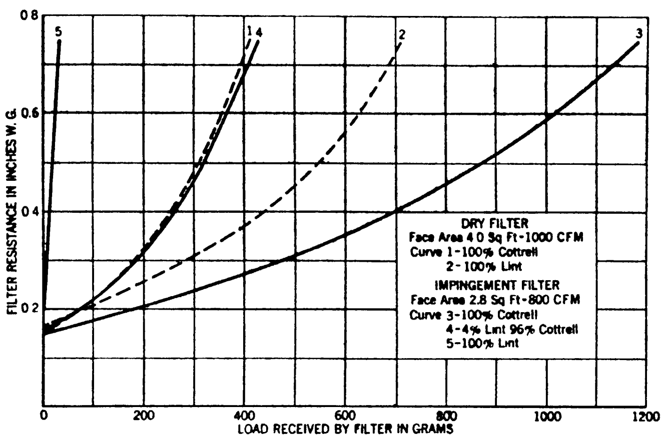

FIG. 4. DUST AND LINT HOLDING CAPACITY OF TWO FILTERS

SELECTION AND MAINTENANCE

For removing fine dust or liquid particles which show no gravitational settling tendency, electrical precipitators are highly effective in air cleaning applications. The electrostatic equipment is comparatively expensive to install and maintain. However, in many applications where the cleanest air possible is needed, this expense is justified by the results obtained with properly installed and operated electrostatic air filters. While the charged media type of electrostatic precipitator is less expensive to install and maintain than other types, it is still substantially more costly than the mechanical types of filters.

The advantage of the automatic impingement type filter consists in the small amount of attention which it requires. Such devices are therefore to be recommended where labor is scarce, or where reliable and frequent attention to filters cannot be assumed. The constant pressure drop of this type of filter is an advantage. The first cost is substantially greater than that of unit filters, and the dust arrestance may not be any higher.

Unit filters constitute the majority of air cleaners now in use, and some choice is possible between the types available. Where lint in an eminently dry state predominates, a dry filter obviously may be preferable to other types because of its lint-holding capacity. If the lint is greasy, or if oil vapor exists in the air, the dry filter, if it is of the cleanable type, may be troublesome, since grease tends to make it difficult to clean. Most dry types, however, employ a throw-away type of medium which is held in permanent metal frames so that the difficulty of cleaning the medium is avoided. Some dry filters are capable of high efficiencies, compared to other unit filters on fine particles, but their dust-holding capacity for such dust may be inferior to that of the viscous impingement type.

Viscous impingement unit filters represent the general type of air cleaner now in use. They have approached standards in size, and their overall dimensions are small when compared with their ratings.

Throw-away units are often installed in series so that the one in front, which usually becomes plugged with lint, can be discarded, after which the downstream unit is moved to the front and replaced by a new unit.

Viscous impingement unit filters do not have efficiencies as high as can be expected with some other types of unit filters, but their first cost and upkeep are generally lower, whether of the cleanable or the throw-away type. They require more careful attention than the automatic oil type if the resistance is to be maintained within reasonable limits.

FILTER INSTALLATION

Many air cleaners are available in units of convenient size for handling when installing, cleaning, or replacing. Such units are usually designated as filters or unit filters. A typical unit filter may be 20 in. square and from one to several inches thick, depending on the manufacture and proposed use. In large systems, a number of such units are installed adjacent to each other and collectively called a bank of filters.

Air cleaners are commonly installed in the outdoor air intake ducts of buildings, and generally in the recirculating and by-pass air ducts as well. Cleaners are logically placed ahead of heating or cooling coils and other air conditioning equipment in the system to protect them from dust. The character of the dust arrested by the filters in an air intake duct is likely to be mostly particulate matter of a greasy nature, while lint may predominate in dust from within the bulding.

The published performance data for all air filters are based on *straight through* unrestricted air flow. Filters should be installed so that the face area is at right angles to the air flow whenever possible. Eddy currents and dead air spaces should be avoided, and air should be distributed uniformly over the entire filter surface, using baffles or diffusers if necessary.

Failure of air filter installations to give satisfactory results can, in most cases, be traced to faulty installation or improper maintenance or both. The most important requirements of a satisfactory and efficiently operating air filter installation are:

1. The filter must be of ample size for the amount of air it is expected to handle. An overload of 10 to 15 percent is regarded as the maximum allowable. When air volume is subject to increase, a larger filter should be installed.

2. The filter must be suited to the operating conditions, such as degree of air cleanliness required, amount of dust in the entering air, type of duty, allowable pressure drop, operating temperatures, and maintenance facilities.

3. The filter type should be the most economical for the specific application. The first cost of the installation should be balanced against depreciation as well as expense and convenience of maintenance.

The following recommendations apply to filters (also washers) installed with central fan systems:

1. Duct connections to and from the filter should change size or shape gradually to insure even air distribution over the entire filter area.

2. Sufficient space should be provided in front as well as behind the filter to make it accessible for inspection and service. A distance of two feet may be regarded as the minimum.

3. Access doors of convenient size should be provided in the sheet metal connections leading to and from the filters.

4. All doors on the clean air side should be lined with felt to prevent infiltration of unclean air. All connections and seams of the sheet metal ducts on the clean air side should be as air-tight as possible.

5. Electric lights should be installed in the chamber in front of and behind the air filter.

6. Air washers should, whenever possible, be installed between the tempering and heating coils to protect them from extreme cold in winter.

7. Filters installed close to an air inlet should be protected from the weather by suitable louvers, in front of which a large mesh wire screen should be provided.

8. Filters should have permanent indicators to give a warning when the filter resistance reaches too high a value.

Safety Requirements

An investigation of safety ordinances should be made by the engineer when the installation of an air cleaner of any considerable size is contemplated. It is possible that combustible filtering media may not be permitted in accordance with some existing local regulations. Combustion of dust and lint on a filtering medium is possible, though the medium itself may not burn.

ADSORPTION OF VAPORS OTHER THAN WATER

Many of the foreign gases in the atmosphere are selectively adsorbed by charcoal. Included are many of the organic gases, such as those emanating from animals and people, some of the gaseous constituents of combustion, alcohols, ketones, esters, and gaseous products of putrefaction.

Charcoals differ widely in their adsorptive capacity. Those which have marked adsorption characteristics, such as properly prepared coconut shell charcoals, are sometimes called *activated charcoals* or *activated carbon.* These materials can adsorb approximately 50 percent of their own weight of many organic gases at 70 F. The charcoal may be used for a long time by reactivation at high temperatures, under which condition it gives up the adsorbed gases. Temperatures of approximately 1000 F are desirable for reactivation. Charcoals for use in air handling systems should be able to stand physical handling, including reactivation, without excessive loss by breakage or dusting.

As applied in air handling systems, the charcoal is placed in perforated metal containers which are grouped in frames and set in the air stream. The percentage removal of an organic gas, such as carbon tetrachloride, is 95 percent or above, when placed in intimate contact with the carbon at 70 F. In commercial apparatus there may be a by-pass effect which depends on the physical arrangement of the charcoal containers. This by-passing reduces the percentage removed in the total gas passing through the adsorber. Resistance to air flow is usually selected within the general range of resistance of impingement filters.

The required quantity of recirculated air to be treated is determined by dividing the requirements for contaminant-free air, minus the outdoor air, by the fraction denoting the percentage removal of the gas in question in the adsorber bank which is to be used.

Adsorbers may be applied to reduce objectionable gases entering through the outdoor air inlet. They may also be used to reduce the odors caused by exhausts from processing. Adsorber beds, in all cases, should be protected from dust, free oil and grease.

PART 2—DUST COLLECTORS

Industrial development and growth of industrial areas have had a cumulative effect upon the problem of dust control. Not only has the atmosphere in many cities become more polluted, but the intensity of pollution at the points of control has increased. Accompanying this increase in pollution there has been a growing consciousness of the need for more effective air cleaning among housewives, store managers, industrialists and legal inspectors, and this has resulted in increasing severity of regulations pertaining to collection of dust and contaminants.

Air cleaning for the supply system is usually accomplished by means of some type of air filter. To prevent escape of industrial dust into the atmosphere, some type of dust collector is required. An industrial air cleaning installation is designed to perform one or more of the following functions:

1. Prevent a nuisance or physical damage to a plant or adjacent property.
2. Prevent re-entry of contaminants to working spaces.
3. Reclaim usable material.
4. Reduce fire, explosion, or other hazards.
5. Permit recirculation of cleaned air to working spaces.

DEGREE OF DUST REMOVAL REQUIRED

Minimum standards of dust removal required of a dust collector may be established by local or state regulations, prepared by municipal smoke abatement or health departments, or by state labor or health departments. Regardless of minimum standards, it is good practice to install the most effective collection equipment available in the light of practical operation features and installation and equipment costs. This is warranted because the required degree of effluent removal is increasing continually. Public nuisance complaints often occur even when the effluent concentration discharged to the atmosphere is below the permissible limits of concentration and visibility. Plant location, contaminants involved and meteorological conditions of the areas must be evaluated in addition to existing regulations or codes of good practice.

Air cleanliness must be of the highest order where cleaned air is recirculated to the workroom. Such recirculation is considered poor practice and is prohibited by many states where toxic materials are involved, except for those cases where discharge to atmosphere is impossible or decidedly impracticable. Where air is recirculated, air contamination must not exceed the established maximum allowable concentrations listed in Chapter 8. Usual requirements are a fraction, often $\frac{1}{3}$ to $\frac{1}{2}$, of this standard, depending on: state involved, air quantities recirculated in relation to the cubical content of working space, and the presence of other exhaust systems discharging to the atmosphere.

FACTORS AFFECTING SELECTION

Selection of a dust collector for a given application requires an evaluation of the following 5 considerations:

1. Dust load and particle size of the contaminant.
2. Degree of removal required.

3. Conditions of air or gas stream with reference to temperature, moisture content, and chemical composition.
4. Characteristics of contaminant whether corrosive, sticky, or packing, and its specific gravity, particle size and shape.
5. Methods of disposal or salvage that meet the conditions imposed by material, process, or plant location.

With the range of variables that must be considered, selection of proper dust collection equipment is based largely on experience, and manufacturers of such equipment should be consulted for their recommendations.

TYPES AND APPLICATION OF DUST COLLECTORS

Dust collectors are available in a multiplicity of designs, frequently involving more than one principle of operation. Basic principles of operation are described in following sections. Manufacturers of various types can be found in the catalog section of THE GUIDE, or in handbooks featuring such equipment.

Electrostatic Dust Precipitators

Electrostatic dust and fume collectors differ materially from the low voltage designs described in Part I of this chapter, although the principles, methods of operation and efficiencies are similar. For dust collector loads, it is obvious that more severe demands are made upon methods of cleaning the collector, disposal of dust accumulations, and servicing practices. Low voltage cleaners to date do not have sufficient inherent dust holding capacity. One exception in the field of exhaust systems is that of the oil mist collector, which functions satisfactorily with the conventional low voltage type electrostatic precipitator.

The *heavy duty dust collectors* employ an assembly of parallel collector electrodes of various constructions, including corrugated plate, rod curtains or perforated plate. Air flow is usually horizontal, although special construction may permit vertical air flow. The earlier high voltage collectors were made in the form of vertical pipes, and are still used for high operating pressures and for *wet collector* designs where water continuously flows downward inside the pipe walls of the collector electrode. The negative discharge electrodes or rods are accurately centered between the usual 9 in. collector electrode spacing, the latter being of positive charge. Precipitation occurs in a single stage wherein ionization and collection are carried on simultaneously throughout the unit, and depend on high potentials of 60,000 to 75,000 volts. The two-stage type which has the added primary stage of ionizer and receiving electrodes, however, fulfills some needs better than the single stage type.

High efficiencies are obtained by allowing suitable time for contact in the collector zone, and by proper ratio of air flow velocity to that of transverse velocity of the negatively charged particles toward the positive charged collector plates. Air velocities vary from 4 fps to 8 fps with negligible pressure drops.

Attention should be directed to the prohibition of air recirculation to occupied spaces. High voltage cleaning equipment produces ozone in excessive quantities and, in addition, develops oxides of nitrogen.

Collector electrode cleaning is conveniently managed by a rapping device, either electrical or pneumatic, without interrupting operation. The dry plate surfaces release the dust readily into hoppers below the plates.

For longer than two decades, electrical precipitators of the heavy duty high voltage type have been used for the control of hazardous materials, and for nuisance abatement. They have been applied to flue gases from cement kilns, smelters and paper plants; to exhaust systems serving crushers, grinders and conveyors; to chemical and metal working plants; and to many pulverized-fuel, fly-ash applications. To date, precipitators have been used principally for elevated temperature installations, and dry products that are free of condensation or dampness.

Fabric Collectors

Dust collectors in this group, often known as *cloth dust arresters* or *cloth filters*, remove dust by passing air at low velocity through a filter material. Cotton cloth is the usual material, although wool, glass, asbestos, and metal screen are sometimes employed. Filter velocities depend on dust concentration, particle size, and permissible vibration interval. Normally at about 3 fpm, they often are reduced to 1 fpm or lower where heavy dust loads of very fine material are involved. Velocities in excess of 6 fpm are seldom used, except in automatically vibrated sectional collectors where velocities as high as 20 fpm are frequently employed.

Collection efficiency is high, even for low micron dust sizes, when the collector is properly maintained. As collected material builds up on filter surfaces, increasing the resistance to air flow, such a system must be stopped at 4 to 8 hour intervals so that the dust load can be vibrated from the filter surfaces to reduce the pressure loss.

Filter cloth is supported in the form of envelopes or bags in a suitable steel housing. Space requirements are quite large, generally necessitating an outdoor location. Pressure drop is normally 2 to 4 in. water column. Material is collected dry. Fabric arresters are limited to applications where air is above the dew-point, as condensation packs the collected material with resultant high pressure drop, and prevents removal by vibration. Temperatures should not exceed 180 F for cotton, or 200 F for wool.

Wet Collectors

In a wet dust collector, the contaminant is brought into contact with a liquid, usually water, for removing the dust from the gas stream. The various available designs represent combinations of methods that make cataloging according to principles, pressure drop, or efficiency, difficult. Wet type dust collectors have the ability to handle high temperature and moisture laden gases. The collection of dust in a wetted form eliminates a secondary dust problem in disposal of collected material. However, the use of water may introduce corrosive conditions within the collector, and freezing protection may be necessary if collectors are located outside in cold climates. Space requirements are nominal. Pressure losses and collection efficiency vary widely with design.

1. *Static Washers.* These units, unlike most *air washers*, are designed to handle heavy concentrations of dust. Both scrubber and eliminator plates (each having flooding nozzles) are employed in addition to the bank of sprays ahead of the scrubber plates, and 2 banks of opposed sprays located ahead of the eliminators. A hopper-bottom tank with recirculating pump completes the assembly. Pressure drop is about ½ to ¾ in. of water. Spray towers can be placed in this same group, as they consist of a tower structure with various nozzle arrangements, and usually include an eliminator section at the top.

Wet glass cell washers have special sprays playing on filter cells filled with fiberglass or other filter media. Flooded eliminator plates are used to remove free moisture from the air stream. Pressure drop is about ¾ in. of water, and the length of the unit is comparable to that of a single stage air washer. Applications are usually limited to low dust concentrations.

2. *Packed Towers.* Collectors in this group are essentially contact beds through which gases and liquids pass, either concurrently, counter-currently, or in cross flow, and are used primarily for nuisance abatement of highly corrosive contaminants. The liquid usually enters at the top of the tower, while the gases may enter at the top, at the bottom, or through an open side.

Water flow rates of 5 to 10 gpm per 1000 cfm (70 F volume) are distributed frequently through V-notched ceramic or plastic weirs. High temperature deterioration is avoided by use of brick linings permitting 1600 F gases direct from furnace flues.

Air flow pressure loss for four-foot beds of irregular shaped materials, such as ceramic saddles or coke, range from 1½ to 3½ in. water, with respective face area velocities of approximately 200 to 300 fpm.

3. *Wet Centrifugal.* A number of designs utilize a combination of centrifugal force and water contact to effect collection. In designs of this group, the collector is cylindrical, either in shape of a tower or with the axis horizontal. Air is introduced tangentially, and frequently directed counter-current to flow of water by baffles or directional plates. Water may be brought into contact with the dust particles by keeping collector surfaces washed by spray nozzles, by induced water picked up by the air, or by fall of water due to gravity. Pressure losses range from 2½ to 6 in. water.

4. *Dynamic Precipitator.* This type uses water sprays within a fan housing, and obtains precipitation of the dust particles on the wetted surfaces of an impeller with special fan blade shape. No external pressure drop is involved, although mechanical efficiency is somewhat lower than the mechanical efficiency of standard exhaust fans.

5. *Orifice Type.* In this type, the air flowing through the collector is brought in contact with a sheet of water in a restricted passage. Water flow may be induced by the velocity of the air stream, or maintained by pumps and weirs. Pressure losses vary from 1 in. or less in water wall spray booth collector designs, to from 3 to 6 in. in most industrial collector arrangements. Pressure losses as high as 20 in. are used, with some collectors designed to collect very small particles.

6. *Disintegrator.* This type of unit generally consists of one or more stages, and is largely used for cleaning producer, blast furnace, or other gases where the gas is to be used in engines, and must be practically free of dirt. The spray is generally in the fan inlet, and elimination is effected largely on the fan blades and also on the surfaces beyond.

A special two-element fan is used; the air with its dust content and water spray enters one side of the wheel and is discharged from the *inlet* of the other wheel. As the air passes through the cyclonic chamber, a high degree of scrubbing action takes place.

Relatively high pressure losses are encountered, with resulting high horsepower requirements.

CENTRIFUGAL DUST COLLECTORS

Centrifugal collector design can be divided into four groups:

1. *Cyclone Collector.* This type is commonly applied for the removal of coarse dusts from an air stream, as a precleaner to more efficient dry or wet dust collectors, or as a separator in product conveying systems using an air stream to transport material. Principal advantages are low cost and low pressure drop (¾ to 1½ in. water), but this type cannot be used for high efficiency collection of fine particles.

2. *High Efficiency Centrifugal Collectors.* These have been developed to obtain higher centrifugal force action on dust particles in a gas stream. Centrifugal force is a function of peripheral velocities and angular acceleration, and improvement in dust separation efficiency has been obtained by (a) increasing velocities through a cyclone shaped collector; (b) utilizing a skimmer or other design feature; (c) using a number of small diameter cyclones in parallel; and (d) in some unusual applications, by placing units in series.

While such collectors do not generally reach an efficiency on small particles equal to that of the electrostatic, fabric, or some wet type units, their effective collection range is extended appreciably beyond that of the conventional cyclone. Pressure losses of collectors in this group range from 3 to 8 in. water column.

3. *Dry Type Dynamic Precipitator*. In this type, dust is precipitated by centrifugal force upon specially shaped blades of an exhauster wheel, and then conveyed through a dust circuit in the fan casing to the dust storage hopper.

4. *Cinder Catching Fan*. This is a special type of collector in which removal of solids is effected by a slotted scroll. It is a fan unit which is often used for the dual purpose of dust removal and induced draft service.

SETTLING CHAMBERS

While it would be possible in theory to remove dust by settling in a large chamber when conveying velocities are reduced to the point where the particles would no longer be held in suspension, such devices have little practical application in dust collecting equipment. Extreme space requirements and the presence of eddy currents may nullify the effective velocity. The settling chamber type of collector can therefore be used only for removal of extremely coarse particles. Pressure drop should be ¼ to ½ in. water column, plus that necessary to accelerate the air motion to the required conveying velocity at discharge of the settling chamber.

SONIC AGGLOMERATORS

Sonic agglomerators, now in an early stage of development, may prove to be valuable accessories in the dust collecting field. High-intensity sound can be utilized to coagulate low-micron and sub-micron sized aerosols such as smoke, fumes, or very fine dusts into resultant particles which are large enough to be collected efficiently with several of the types of dust collecting equipment previously described.

This high-intensity sound can be created by various means such as piezo-electric crystal generators, magneto-striction units, or high-powered sirens which are practical at present for units with greater air flow capacities.

For effective coagulation, a minimum sound level of 155 to 160 decibels[6, 7, 8] is required. The sound frequencies used are dependent on the size of the particles to be coagulated, and may vary from 1000 cycles per second for larger particles to far into the ultrasonic region for smaller particles. Fairly high dust concentrations, on the order of one grain per cubic foot of gas, are required for effective coagulation. Neither operating temperatures in the range of 0 F to over 1000 F, nor electrical properties of the aerosols,[6] have any apparent effect on agglomeration efficiencies

TESTING METHODS

Methods of determining the performance of industrial air cleaning devices will depend upon the nature of the air contaminant, its quantity, the required accuracy of the test, and the type of air cleaning device. The technics used in collecting the samples are the same as those utilized in the field of industrial hygiene.

The tests may be facilitated by feeding, uniformly, a known amount of the material to be removed. The performance efficiency may be calculated from the amount of material introduced, the quantities of air involved, the amount of material intercepted, and the quantity that escapes. When it is necessary to test the device under actual use, the material entering and leaving must be sampled simultaneously over a sufficiently long period of time to collect an adequate amount for analysis. If feasible, the quantity of material removed by the cleaning device during the test period should also be determined.

Unusual care must be exercised in collecting the samples to insure that the sampling areas selected are actually representative of the material entering and leaving the cleaning device. With gases, vapors and fresh fumes, this problem is not great. On the other hand, mists, dusts and aged fumes may present considerable trouble. When the material is confined in a duct system, it is common practice to collect the sample along the center line. The sampling tube is located parallel with, and facing, the flow. The sampling velocity should be approximately the same as the air velocity in the duct. With large ducts it may be desirable to make traverse tests to locate an optimum sampling position.

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CHAPTER 34

SPRAY APPARATUS

Air Washers, Humidification with Air Washers, Dehumidification and Cooling with Air Washers, Well and Water Main Temperatures, Apparatus for Direct Humidification, Unit Humidifiers, Water-Cooling Towers, Water Use and Conservation, Rivers and Lakes, Spray Cooling Ponds, Atmospheric and Mechanical Draft Cooling Towers, Mechanics of Atmospheric Water-Cooling, Design Conditions, Water-Cooling Tower Design, Selection of Water-Cooling Towers, Operation and Maintenance

AIR humidification is effected by the vaporization of water, and always requires heat from some source. This heat may be added to the water prior to the time vaporization occurs, or it may be secured by a transformation of sensible heat of the air being humidified to latent heat as the vapor is added to the air. The thermodynamics of the process are discussed in Chapter 3. The removal of moisture from air may, or may not, involve the removal of heat from the air-vapor mixture. With spray equipment, dehumidification of air always necessitates the removal of heat.

AIR WASHERS

An air washer consists essentially of a chamber or casing in which is provided a spray nozzle system, a tank at the bottom of the chamber for collecting the spray water as it falls, and an eliminator section at the leaving end of the chamber for removal of drops of entrained moisture from the delivered air. Air is drawn through the casing of the washer, where it comes into intimate contact with the spray water. A heat transfer takes place between the air and water, resulting in either humidification or dehumidification of the air, depending upon the method of operation and the relative temperatures of air and spray water.

To prevent backlash of spray ahead of the washer chamber, and to aid in more uniform air distribution, inlet diffusion plates or eliminator baffles, where necessary, are provided in the air entrance end of the air washer. Inlet diffusion plates are used when the air flow and water spray are in the same direction; eliminator baffles of special design are used where one or more of the water sprays opposes the air flow. At the outlet end of the washer suitable flooded eliminator plates are used. These plates, for the removal of entrained moisture, usually cause four to six changes in direction of the air flow.

Figs. 1 and 2 show the essential construction features of conventional air washers. Intimate contact between the air and the water is secured (1) by breaking the water into fine drops; (2) by passing the air over surfaces continuously wetted by water; or (3) by a combination of the two.

The wetted surfaces in an air washer may be of fiber glass, metal or scrubber plate construction. Scrubber plate types of washers are generally used to wash reclaimable products from the air, and are composed of several baffle type plates located across the air stream. Water is supplied at the top of the washer to spray over these plates. In the case of the fiber glass or metal surfaces, the water spray is usually rather coarse and at low pressure. In many cases these sprays are set at an angle with the air flow. Air washers of this type not only perform necessary heat transfer functions, but also are effective removers of dust and dirt from the air stream.

Essential requirements in the air washer operation are: uniform distribution of the air across the chamber section; moderate air velocity of from 250 to 600 fpm in the washer chamber; an adequate amount of spray water broken up into fine droplets throughout the air stream, at pressures of from 15 to 30 psig; sufficient length of travel through the water

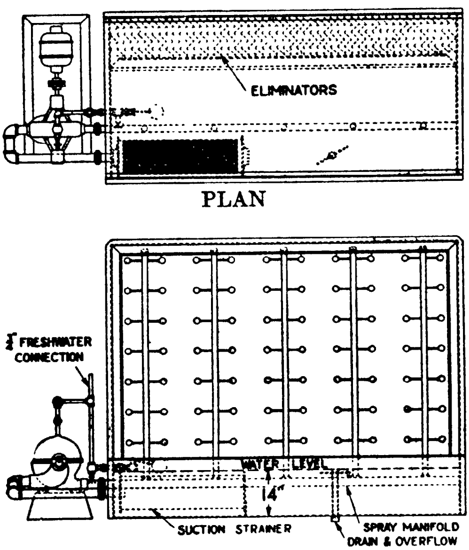

Fig. 1. Typical Single-Bank Air Washer

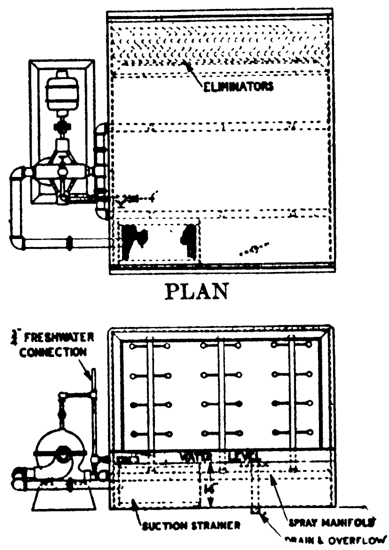

Fig. 2. Typical Two-Bank Air Washer

spray and wetted surfaces; and the elimination of entrained moisture from the outlet air.

Expected performances, physical size, length, number of sprays, etc., vary greatly, depending upon the functions of the installation. In general, the width and height of an air washer are dictated by the space available. Washers of nearly equal height and width are desirable from an air flow and economic standpoint, although not necessary. The length of washers varies considerably. A space of approximately $2\frac{1}{2}$ ft between spray banks is used, and the first and last banks of sprays are located about 1 ft to $1\frac{1}{2}$ ft from the entering or leaving end of the washer. In addition, air washers are very often furnished with cooling coils or heating coils within the washer chamber, and the use of these coils affects the overall length of the washer.

Where increase of overall heat transfer between the air and water is required, multistage washers are used. These washers are equivalent to a number of washers in series, and the water is often pumped from one stage to the other where conditions permit.

The resistance to air flow through an air washer varies with the type of eliminator and wetted surfaces, number of banks of spray and their direction, air velocity, size and type of other resistances such as cooling and heating coils, and other factors such as air density. Resistances vary from as low as ¼ in. to higher than 1 in. water column, and it is therefore necessary that the manufacturer be consulted in regard to the resistance of any particular washer design involved.

HUMIDIFICATION WITH AIR WASHERS

Air humidification can be accomplished in three ways with an air washer. These are: (1) use of recirculated spray water without prior treatment of the air; (2) preheating the air and washing it with recirculated spray water; and (3) using heated spray water. In any air washing installation the air should not enter the washer with a dry-bulb temperature less than 35 F in order to eliminate danger of freezing the spray water.

Method 1. Except for the small amount of energy added from outside by the recirculating pump in the form of shaft work, and for the small amount of heat leak from outside into the apparatus, including the pump and its connecting piping, the process would be strictly adiabatic. Evaporation from the liquid spray would therefore be expected to bring the air immediately in contact with it to saturation adiabatically; and, since the liquid is recirculated, its temperature would be expected to adjust to the thermodynamic wet-bulb temperature of the entering air.

It does not follow from the foregoing reasoning that the whole air stream is brought to complete saturation, but merely that its state point should move along a line of constant thermodynamic wet-bulb temperature as explained in Chapter 3. The extent to which the final temperature approaches the thermodynamic wet-bulb temperature of the entering air, or the extent to which complete saturation is approached, is conveniently expressed by a ratio known as *humidifying effectiveness* or *saturating effectiveness*, and is defined as

$$e_h = \frac{t_1 - t_2}{t_1 - t'} \tag{1}$$

where

e_h = humidifying effectiveness, percent.
t_1 = dry-bulb temperature of the entering air, Fahrenheit degrees.
t_2 = dry-bulb temperature of the leaving air, Fahrenheit degrees.
t' = thermodynamic wet-bulb temperature of the entering air, Fahrenheit degrees.

The following may be taken as representative humidifying or saturating effectiveness of an air washer for the conditions stated:

1 bank—downstream	60–70 percent
1 bank—upstream	65–75 percent
2 banks—downstream	85–90 percent
2 banks—1 upstream and 1 downstream	90–95 percent
2 banks—upstream	90–95 percen

The humidifying or saturating effectiveness of a washer is dependent upon the essential items of design mentioned under Air Washers. Other conditions being the same, low velocity of air flow is more conducive to higher humidification effectiveness.

Method 2. The preheating of the air increases both the dry- and wet-bulb temperatures, lowers the relative humidity, but does not alter the humidity ratio (pound water vapor per pound dry air). At a higher wet-

Table 1. Average Maximum Water Main Temperatures[a]

State	City	Temp. F	State	City	Temp. F	State	City	Temp. F
Ala.	Birmingham	84	Mass.	Boston	80		Tulsa	86
	Mobile	80		Cambridge	70	Ore.	Eugene	60
Ark.	Little Rock	83		Fall River	76		Portland	65
Ariz.	Phoenix	82		Lowell	50	Pa.	Altoona	74
	Tucson	80		Lynn	68		Erie	75
Calif.	Anaheim	60		New Bedford	70		Johnstown	74
	Berkeley	69		Salem	68		McKeesport	82
	Fresno	80		Worcester	76		Philadelphia	85
	Fullerton	75	Mich.	Detroit	77		Pittsburgh	86
	Glendale	68		Flint	70	R. I.	Providence	68
	Los Angeles	80		Grand Rapids	84	S. C.	Charleston	83
	Oakland	69		Highland Park	77		Greenville	81
	Ontario	70		Jackson	56		Spartanburg	78
	Pasadena	82		Kalamazoo	53	S. D.	Rapid City	65
	Pomona	75		Lansing	64	Tenn.	Chattanooga	84
	Riverside	78		Saginaw	82		Knoxville	90
	Sacramento	72	Minn.	Duluth	55		Memphis	85
	San Bernardino	65		Minneapolis	85		Nashville	90
	San Diego	84		St Paul	80	Texas	Amarillo	70
	San Francisco	71	Mo.	Jefferson City	82		Austin	90
	Whittier	75		Kansas City	84		Beaumont	86
Colo.	Denver	75		Springfield	82		Dallas	86
Conn.	Bridgeport	66		St. Joseph	84		El Paso	85
	Hartford	73		St Louis	85		Fort Worth	85
	New Haven	76		Springfield	74		Galveston	90
	Waterbury	72	Nebr.	Lincoln	70		Houston	83
D. C.	Washington	84		Omaha	85		Port Arthur	83
Del.	Wilmington	83	Nev.	Reno	70		San Antonio	78
Fla	Jacksonville	86	N. H.	Manchester	76		Wichita Falls	85
	Miami	82	N. J.	Jersey City	63	Utah	Logan	44
	Tampa	77		Newark	75		Salt Lake City	65
Ga.	Atlanta	85		Paterson	78	Va.	Fredericksburg	75
	Macon	80		Trenton	79		Lynchburg	73
Idaho	Boise	60	N. Y.	Albany	68		Norfolk	80
Ill	Chicago	78		Buffalo	78		Richmond	85
	Cicero	76		Jamaica	56	Wash.	Olympia	58
	Evanston	73		Mt. Vernon	74		Seattle	62
	Moline	83		New Rochelle	75		Spokane	51
	Peoria	67		New York	72		Tacoma	57
	Rockford	59		Rochester	70	W Va.	Charleston	85
	Springfield	82		Schenectady	60		Huntington	78
Ind	Evansville	88		Syracuse	74		Wheeling	78
	Gary	75		Utica	69	Wis.	LaCrosse	54
	Indianapolis	84		Yonkers	70		Madison	58
	South Bend	61	N C.	Asheville	74		Milwaukee	75
	Terre Haute	82		Charlotte	85		Racine	68
Iowa	Cedar Rapids	78		Raleigh	92			
	Des Moines	77		Winston-Salem	82			
	Sioux City	62	N. M.	Albuquerque	65	Province		
Kans.	Concordia	57	Ohio	Akron	76			
	Kansas City	86		Canton	50			
	Topeka	88		Cincinnati	84			
	Wichita	72		Cleveland	77	Alta	Calgary	64
Ky.	Louisville	85		Columbus	84	B. C.	Vancouver	60
La.	Baton Rouge	85		Dayton	60	Ont.	London	50
	New Orleans	90		Lakewood	82		Toronto	63
	Shreveport	88		Springfield	72	P E.I.	Charlottetown	48
Me.	Augusta	60		Toledo	83	Que.	Montreal	78
Md.	Baltimore	75	Okla.	Oklahoma City	82		Quebec	68

[a] These averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown. Some values were supplied by H. E. Degler, Marley Company. Some were obtained from City Water Department records. The highest values given by the various authorities are usually those listed

bulb temperature, but the same humidity ratio, more water can be absorbed per pound of dry air in passing through the washer, assuming that the humidifying effectiveness of the washer is not adversely affected by operation at the higher wet-bulb temperature. The analysis of the process occurring in the washer itself is the same as that explained under *Method 1.* The final desired conditions are secured by adjusting the amount of preheating to give the required wet-bulb temperature at entrance to the washer.

Method 3. Even if heat is added to the spray water, the mixing occurring in the washer itself may still be regarded as adiabatic. The state point of the mixture should move in a direction determined by the specific enthalpy of the heated spray as explained in Chapter 3. It is possible, by elevating the water temperature, to raise the air temperature, both dry-bulb and wet-bulb, above the dry-bulb temperature of the entering air.

In each of the methods, 1, 2 or 3, the air leaving the air washer may require reheating to produce in the conditioned space the required dry-bulb temperature and relative humidity.

DEHUMIDIFICATION AND COOLING WITH AIR WASHERS

Cooling of the wet-bulb temperature of an air vapor mixture can be accomplished by an air washer if the temperature of the spray water is lower than the wet-bulb temperature of the air. Moisture removal is obtained when the spray water temperature is lower than the dew-point of the entering air. In these cases the final dry-bulb temperature and relative humidity of the leaving air are dependent upon the design factors of the air washer.

Both sensible and latent heat are removed in the process of dehumidification by cold spray water. Abstraction of sensible heat occurs during the entire time that the air is in contact with the spray medium. Latent heat removal takes place as condensation occurs. Therefore, the lower the spray temperature, the greater the amount of moisture removal per pound of dry air, all other conditions remaining the same.

Washers with two or more banks of spray are usually selected for dehumidifying installations, whether for comfort or industrial installations. Generally such air washers cool the air to within one or two degrees (Fahrenheit) of the leaving spray water temperature; this differential will increase somewhat when the difference between the entering wet-bulb and leaving dew-point is relatively large.

Where a limited supply of cold water is available, multiple stage washers may be used to great advantage. In such washers the cool water is pumped through the multiple spray systems in series and counterflow to the air flow. Such an arrangement brings the delivery air in contact with the coldest water, securing a maximum amount of cooling and saving water.

When using cold well water or water from city water mains, care should be used to secure accurate data on the water temperatures. Table 1 lists some approximate water main averages which may be used as a guide, but they should be verified from local records. This is particularly true with city water main temperatures. In the case of well water temperatures, Fig. 3 shows the approximate temperatures of water to be expected from wells at depths of 30 to 60 ft.

Air washers for dehumidifying and cooling usually have separate recirculating pumps. These pumps deliver a mixture of cold and recirculated water under the control of a three-way valve. The valve may be actuated either by a thermostat in the washer outlet, or by a humidity or other controller in the space being conditioned.

Air washers for dehumidifying are very often furnished with direct expansion or water cooling coils within the washer space, in which case water for the washer sprays is entirely recirculated.

APPARATUS FOR DIRECT HUMIDIFICATION

Humidifiers may be divided into two general types which are, according to the method of operation: (1) indirect, such as the air washer, which

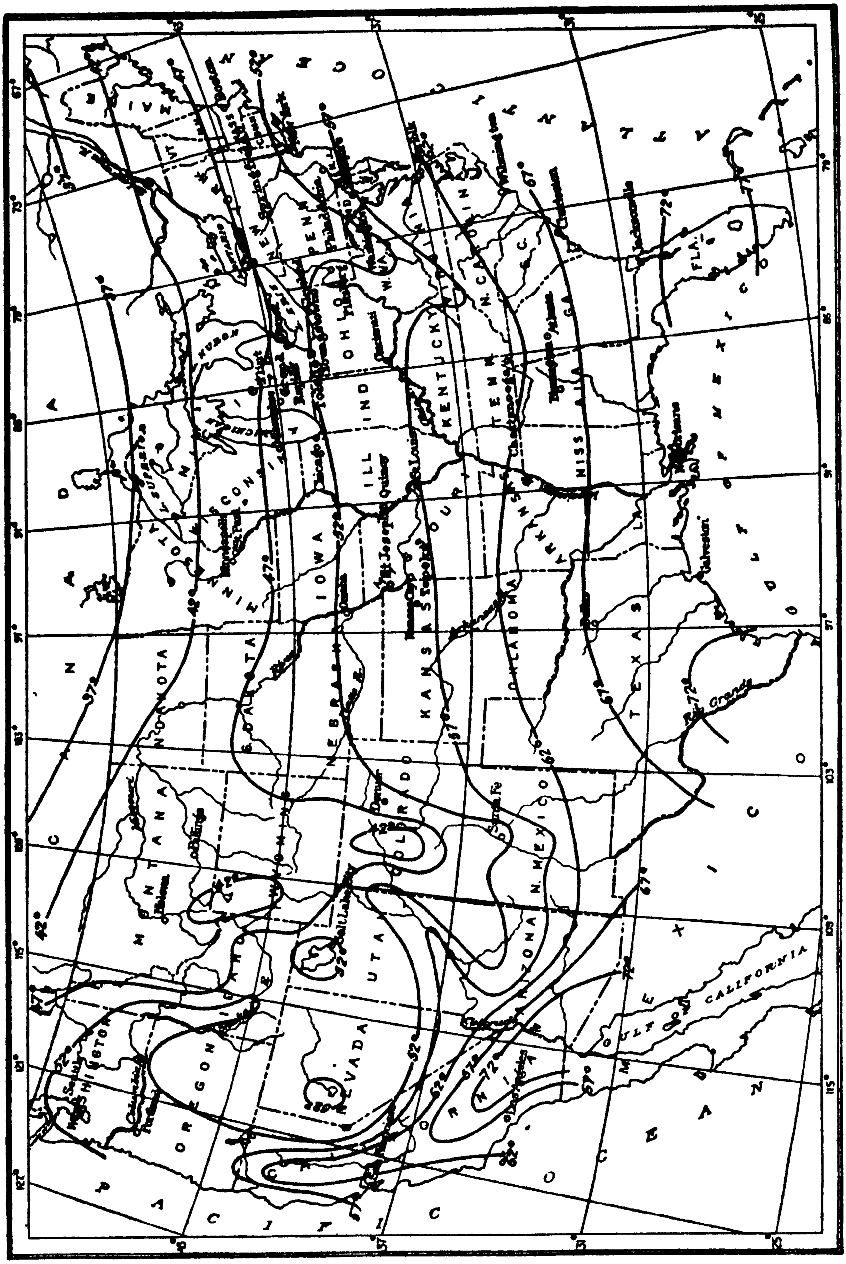

Fig. 3. Approximate Well Water Temperatures at Depths of 30 to 60 Ft[1]

introduces moistened air; and (2) direct, which sprays moisture into the room or introduces moisture by means of steam jets.

As in the cases of humidification by use of an air washer, the heat necessary for the vaporization of the moisture added to the air by direct humidification is secured either from heat stored in the spray water or by a transformation of sensible to latent heat in the air humidified. In the latter case, the enthalpy of the air remains constant, but the dry-bulb temperature of the air is reduced.

Direct humidification is usually preferable where high relative humidities must be maintained, but where there is little cooling or ventilation required. In comfort air conditioning, where both humidification and ventilation are required, the indirect humidifier is preferable. In industrial applications, where the cooling or ventilation load is large and where very high relative humidities must be maintained, a combined system employing both direct and indirect humidifiers is sometimes used.

Spray Generation

Spray generation is obtained by (1) atomization, (2) impact, (3) hydraulic separation, and (4) mechanical separation.

Atomization involves the use of a compressed air jet to reduce the water particles to a fine spray. With the *impact* method, a jet of water under pressure impinges directly on the end of a small round wire. Where *hydraulic separation* is employed, a jet of water enters a cylindrical chamber and escapes through an axial port with a rapid rotation which causes it immediately to separate in a fine cone-shaped spray. In the *mechanical separation* process, water is thrown by centrifugal force from the surface of a rapidly revolving disc and separates into particles sufficiently small to be utilized in certain types of mechanical humidifiers.

Spray Distribution

Spray distribution is obtained by (1) air jet, (2) induction, and (3) fan propulsion.

The air jet which generates the spray in atomizers also carries the spray through a space sufficient for its distribution and evaporation, and this method of distribution is termed *air jet*. Where distribution is obtained by *induction*, the aspirating effect of an impact or centrifugal spray jet is utilized to induce a current of air to flow through a duct or casing, and this air current distributes the spray. *Fan propulsion* obviously consists of the utilization of fans to entrain and distribute the spray.

Industrial type direct humidifiers are commonly classified as (1) atomizing, (2) high-duty, (3) spray and (4) self-contained or centrifugal.

Atomizing Humidifiers. Several types of atomizing humidifiers employ nozzles placed within the room and rely upon compressed air to effect complete atomization of the water which is converted to vapor by the heat of the room air. Some of these nozzles depend upon an aspirating effect to draw the water into the nozzle and atomize it; others operate on a combined air and water pressure. It is usual for nozzles of the water pressure type to be controlled by a diaphragm valve actuated by the pressure of the atomizing air.

High-Duty Humidifiers. Water is supplied under high pressure (usually about 150 psi) through pipe lines from a centrally-located pumping unit. The spray-generating nozzle, which is of the impact type, is located in a cylindrical casing. A drainage pan provides for the collection and return of unevaporated water which flows through a return pipe to a filter tank, from which it is recirculated. A powerful air current is forced through the humidifier by means of a fan mounted above the unit.

The air enters from above, is drawn through the head, charged with moisture, and cooled. It then escapes from the opening below at a high velocity in a complete and nearly horizontal circle. The spray is evaporated and the resulting vapor diffused. This distribution of fine spray over the maximum possible area promotes complete and rapid vaporization.

Spray Humidifiers. This type consists of an impact spray nozzle in a cylindrical casing with a drainage pan below it. The aspirating effect of the nozzle induces a moderate air current through the casing which distributes the entrained spray. The general method of circulating and returning the water is similar to that employed for high-duty humidifiers. A suitable pump and centrally-located filter tank are required.

Self-Contained Humidifiers. The self-contained or centrifugal humidifier has the ability to generate and distribute spray without the use of air compressors, pumps, or other auxiliaries. These may be used either singly or in groups. In large installations, where suitable connections are provided to permit the cleaning and servicing of individual units without affecting the room as a whole, group control of the water and power may be employed.

UNIT HUMIDIFIERS

The term *unit humidifier* denotes an assembly of elements the principal function of which is to humidify. The essential element of a unit humidifier is an atomizer or evaporator. To this may be added a fan, a heater, outlet vanes or diffusers, and a housing to enclose the various parts.

Unit humidifiers fall into four general classifications, depending on the method of causing evaporation. These are as follows: (1) *Nozzle Type*, (2) *Rotary Type*, (3) *Cascade Type*, and (4) *Heater Type*.

In the *nozzle type* of humidifier water is sprayed into the air and evaporation is effected by adiabatic exchange of energy. Units of this type in simplest form, spray a fine mist of water directly into the air in a space. They are used to a great extent in the textile industry.

In the *rotary type* of humidifier the spray is created by rotating vanes or discs which throw the water by centrifugal force, and in so doing break it up into a fine mist. In all other respects, this type of humidifier is similar to the nozzle type. It has the advantage over the nozzle type of being less liable to become clogged.

In the *cascade type* the humidification takes place by water falling in sheets over a series of baffles or trays. This type is usually furnished with a fan, air heater, and air filter all enclosed in a housing.

In the *heater type* of humidifier the water is heated either to the boiling point or to a temperature at which the water vapor readily passes into the air stream. There are many variations of this type of humidifier. In the simplest form the heating element using steam, hot water, gas, oil, or electricity is placed in a pan or vessel of water, and the vapor passes from the surface of the water to the stream of air.

A modification of this type of humidifier is the combination of spray nozzle and heater type in which the water is sprayed over a hot surface and evaporated. It has the disadvantage of accumulating scale on the surfaces of the vessel or heating surface.

WATER-COOLING TOWERS

The removal and dissipation of heat from a compressed refrigerant or from exhaust steam are important factors in the efficient operation of a refrigerating plant or an electric steam-generating station. This heat removal is generally accomplished by first transferring the heat of the *gas* to cooling water in a heat exchanger. The water, if cheap or plentiful, may be wasted to the nearest sewer or open waterway, such as a river or lake. Where water usage is restricted or expensive, or where the available water contains dissolved salts which would form scale on the heat-exchange apparatus, it is necessary to recirculate the water, and to cool it, after each passage through the heat-exchanger, by contact with moving air in some type of water-cooling apparatus.

Water Use and Conservation

Many communities have found that present water systems are not sufficiently large to satisfy the increasing demands of domestic and industrial users. The reasons for such shortages are primarily: (a) inadequate purification and water-distribution systems; (b) inadequate sanitary and storm-sewer disposal systems; or (c) inadequate sources of water.

Even when an adequate supply of water is available from the water mains or private wells, many cities do not have sufficient sanitary or storm sewer facilities to handle increasing demands. The sanitary systems are usually limited because of the capacity of the filtration plants, and therefore many cities restrict the use of the sanitary system to sewage.

Rivers and Lakes

Until the year 1920, large generating stations were usually located on the banks of rivers, lakes, or artificial ponds. The removal and dissipation of the heat from the Diesel cylinder or the exhaust steam of a turbine was accomplished by taking in the circulating water at a considerable distance from the discharge, thus preventing mixing of the heated discharge with the inlet water. The use of water from streams for this purpose has the following disadvantages: the site may be far removed from the fuel source or from power consumers; water supply may limit plant expansion; municipal restrictions on use of water may hamper operation; costly intake structures with screens and sediment basins may be required; drastic flood or drouth conditions, the vagaries of most rivers, upstream pollution, scale-forming constituents, debris, sand, algae, and formation of troublesome ice may cause operating difficulties.

When lakes and cooling ponds have been used as a source of circulating water, the hot water is discharged close to the surface at the shore line. Natural air movement over the surface of the water causes evaporation over that area, thus carrying the heat away at a rate of about 4 Btu per (hr) (sq ft) (F deg temp difference between air and water). Increased density of the water due to loss of heat, causes the cooled water to sink to the bottom of the pond. The suction connection is therefore located as far below the surface as possible, and at as great a distance from the discharge as practicable. The area required by such cooling ponds is about 50 times that of a spray pond, or about 1000 times that of a water-cooling tower to dissipate the same quantity of heat and achieve equal operating costs. If the surfaces of such ponds were below the level of surrounding terrain, and the shore were wind-sheltered by trees or other vegetation, so that natural air movement across the surface of the water would be retarded, the use of a spray pond or water-cooling tower would be indicated.

SPRAY COOLING PONDS

The spray pond consists of a water collecting basin, above which spray nozzles are located in an arrangement such as shown in Fig. 4 to spray the water upwards into the air. Properly designed spray nozzles break the water into small drops, but not into a mist. Since the objective is to cool the pond water, the individual drops must be heavy enough to fall back into the pond and must not float away in the air. The water surface exposed to the air passing over the pond becomes the integrated area of all the small drops. The spray pond requires about one-fiftieth of the space required by the cooling pond to dissipate the same quantity of heat with equal results, due to four factors: (1) the speed with which the drops are propelled into the air and fall back into the water basin; (2) the increased wind velocity at a point above the surrounding obstruction; (3) the increased volume of air delivery due to the greater vertical cross-section of air permissible; and (4) the vastly increased area of contact between water and air.[2]

Spray pond effectiveness is increased by: (1) elevating the nozzles to a higher point above the surface of the water in the basin; (2) increasing the spacing between nozzles of any one capacity; (3) using smaller capacity

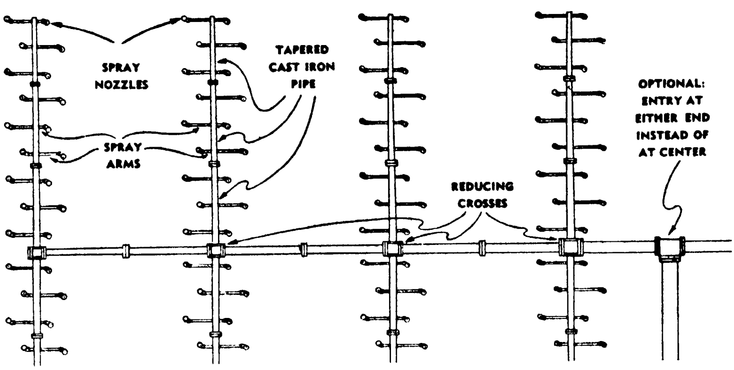

FIG. 4. TYPICAL NOZZLE ARRANGEMENT FOR A SPRAY POND

nozzles to decrease the concentration of water per unit area; and (4) using smaller nozzles and increasing the pressure to maintain the same concentration of water per unit area.

It is usual practice to locate the nozzles from 5 to 12 feet above the surface of the water (dependent also upon depth of water and curb level) with water supply at 5 to 7 psig pressure at the nozzles. Nozzles spray from 25 to 60 gpm each, and the nozzles are spaced so that the average water delivered to the surface varies from 0.1 gpm (small ponds) to 0.4 gpm (large ponds) per square foot. See Table 2 for additional spray pond design data. Best results are obtained by placing the nozzles in a long, relatively narrow area, located broadside to the wind.

Louver fences, to prevent the carrying of entrained water beyond the edge of a spray pond by the air on the leeward side, are required for all roof locations and for ground locations where space is restricted; the outer nozzles should be located at least 20 ft from the edge of the basin. Such fences up to 12 ft in height usually are constructed of horizontal overlapping louvers supported between vertical posts. The air, in passing between these louvers, tends to be freed of the larger drops of water. The louvers also restrict the flow of air, particularly at the higher wind velocities, thus reducing the possibility of water being carried from the spray cloud. The height of an effective fence should be equal to the height of the spray cloud. Algae formations may be a nuisance in a spray pond. Such growths are minimized by the periodic addition of bromine, chlorine, chlorinated lime, copper sulfate, or various blends of chlorophenates (see Chapter 42).

The performance of a spray pond is limited because of space requirements and the probable high cost of piping and pumping. Water-cooling towers, however, allow the designer a wider range of performance within a given space because of the possibility of altering the smaller physical dimensions or varying the water concentration, measured in gallons per (minute) (square foot of tower area). In most cooling towers the water is broken up into drops many times, whereas with the spray pond it is broken up only once and, consequently, in the latter the rate of cooling diminishes rapidly as the temperature of the surface of the drop approaches the wet-bulb temperature of the ambient air.

TABLE 2. SPRAY POND DESIGN DATA
Conventional Up-Spray System

	Units	Standard	Minimum	Maximum
Water capacity per nozzle	gpm	35 to 50	25	60
Nozzles per 12 ft length of pipe		6	4	6
Height of nozzles above water level	ft	6	5	12
Nozzle pressure	psig	6	5	7
Size of nozzles and nozzle arms	in.	2	$1\frac{1}{2}$	2
Distance between spray lateral piping	ft	25	13	38
Distance nozzles from pond side unfenced	ft	25 to 35	20	50
Distance nozzles from pond side fenced	ft	15 to 20	15	25
Height of louver fence	ft	12	12	12
Depth pond basin	ft	4 to 5	2	—
Friction loss allowed per 100 ft pipe	ft	1 to 3	—	—
Design wind velocity	mph	5	3	—

ATMOSPHERIC COOLING TOWERS

Spray-filled atmospheric cooling towers are used for open-area installations because of their dependence upon the velocity and direction of the wind. Operation is not so limited as with spray ponds, but the design is generally based on a 3 mph wind, and the performance falls off rapidly as the ambient air velocity decreases. These towers require less basin area, less piping, and no more mechanical equipment than spray ponds, but these savings may be largely offset by the extra cost of the structure. The drift nuisance is similar to that of spray ponds. The word *tower* used in this connection is a misnomer, as the design simulates a narrow spray pond with length twice the width, or more, having elevated nozzles and a high louver fence. As usually built, the nozzles spray downward from the top of the structure, and the distance from the center of the nozzle system to the louvers on either side is not more than half the distance that the nozzles are elevated above the water-collecting basin. Heights range from 6 to 15 ft, with the total width of the structure usually not greater than the height. Loadings range from 0.6 to 1.5 gpm per sq ft of tower area, and hence require about one-fourth the area of an equivalent spray pond. As the louvers are wetted continuously, they add to the surface of water exposed to the cooling air. The spray-filled atmospheric tower is shown in Fig. 5.

Much of the atmospheric water cooling for refrigeration work during the past 30 years has been done with natural-draft deck type towers, also referred to as *atmospheric deck towers*, see Fig. 6. These towers consist of a sturdy wooden or steel frame 20 to 50 ft high and 8 to 16 ft wide, carrying open horizontal wooden latticework or *decks* at regular intervals from top to bottom. The hot water is distributed over the upper part of the structure by means of troughs, splash heads, or nozzles, and drops from deck to deck enroute to the basin. The purpose of the decks is primarily to arrest the fall of the water, to break and re-break it into drops so as to present the most efficient cooling surface to the air which is passing through the tower transversely to the decks. The wooden decks also add to the area of water surface exposed to the air, but since they offer resistance

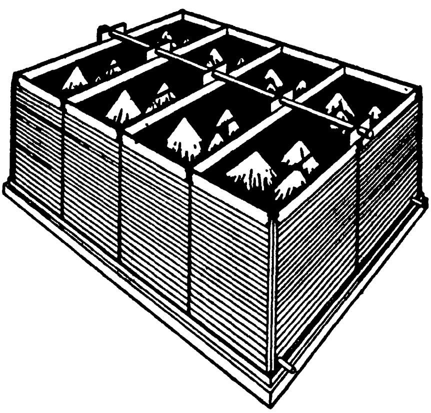

FIG. 5. SPRAY FILLED ATMOSPHERIC COOLING TOWER

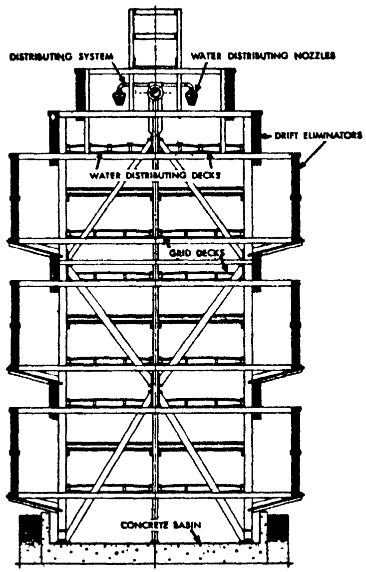

FIG. 6. ATMOSPHERIC DECK TOWER

to the flow of air, the number and arrangement of the decks depend upon basic tests and operating experience.

To prevent loss of water on the leeward side of the tower, wide louvers (drift eliminators) are attached at regular intervals from top to bottom; these louvers extend outward and upward at an angle of 45 to 50 deg. In most designs the top edge of each louver extends above the bottom edge of the one above. These louvers serve the same function as a louver fence around a spray pond, namely, to stop the water drops carried by the air beyond the open area of the tower, and to control the quantity of air permitted to pass through the tower.

The efficiency of a deck tower is improved primarily by increasing length or height, or both, within limits; the length and height increase the area of tower exposed to the wind. The improvement is not directly proportional to the change made in either case. Neither does a certain percentage change of one dimension make an equal improvement in efficiency on two equal towers of different original lengths or heights. Since the range of efficiency varies through wide limits, it is impracticable to attempt to list data here on the area required per unit quantity of water. Improved efficiency, due to added height, is obtained at the expense of additional pumping head and increased weight per unit of area, whereas improvement gained by greater length or width will increase the area and, consequently, the foundation required.

Drift loss in a properly designed deck tower is considerably less than in the spray pond, but the drift nuisance may be considerable, and for this reason atmospheric deck towers are unsuitable for downtown building roofs, locations adjacent to buildings, or near expensive mechanical equipment in industrial plants. They must be located in an open area, broadside to the prevailing wind. They are inefficient with less than 3 mph wind velocity and with wind directions other than broadside. These towers are long and high in proportion to width, and must be securely anchored to prevent uplift or overturning during high winds. High pumping requirements (30 to 60 ft) and total dependence upon atmospheric caprice, especially wind (quantity and direction), are disadvantages.

Due to new uses and growth of demand in recent years, requirements for water-cooling equipment have become increasingly varied and exacting, necessitating refinements and specialized adaptations. The principal demand for large water-cooling systems in recent years has come from the petroleum industry and steam power plants. Refrigeration, air conditioning, and engine-jacket cooling service today employ a large percentage of the medium sized and small water-cooling towers installed.

MECHANICAL DRAFT TOWERS

The mechanical draft tower consists usually of a vertical shell constructed of wood, metal, transite, or masonry. Water is distributed near the top, uniformly over the area, and falls to the collecting basin in the bottom, passing through air which is being circulated in the tower from bottom to top by forced or induced draft fans, or which is circulated horizontally in crossflow towers by induced draft fans.

In vertical towers the air passes counterflow to the water and is in contact with the hottest water just before leaving the tower; hence, a given quantity of air picks up more heat than the average equal quantity of air on natural draft equipment. This permits the water to be cooled with the least quantity of air required by any type of cooling equipment. As movement of air through the towers is obtained by power-consuming fans, it is essential that this air quantity and the draft loss be reduced to a minimum so as to secure low operating cost.

The inside of a mechanical draft tower may be spray filled, *i.e.*, the water surface is presented to the air by filling the entire inside of the structure with water droplets from the spray nozzles, or it may be packed with wood filling, over which the water cascades from top to bottom. In many cases, a combination of the spray-filled and wood-filled design is used.

The *forced draft* type of *tower* (Fig. 7), has the advantages of being suitable for corrosive waters, and having the fan mounted near the ground level on a rigid foundation where it is easily accessible.

The heated air leaves the top of a forced draft tower at a low velocity and may be subject to *recirculation* to the fan inlet, with consequent reduction in performance. This reduction could be as much as 20 percent under certain conditions. During cold weather, recirculation may cause ice formation on adjacent equipment and buildings, as well as in the tower fan ring, with possible resultant fan breakage. Fan sizes are limited to 12 ft or less, and therefore more fans, motors, starters, and wiring are needed than for induced draft towers. Induced draft towers, since fans and motors are not visible, are therefore somewhat more adaptable to architectural treatment.

In the *spray-filled mechanical draft tower*, the area presented to the air is the combined surface area of the small drops present in the tower at any one time. The net free cross-sectional area of the air spaces in a spray-filled tower is greater than that of the wood-filled tower for the same plan area. Before discharging to the atmosphere, the water-laden exhaust air passes through a drift eliminator to remove entrained moisture. This type of tower is particularly applicable for installations in restricted areas where city ordinances require fire-proof construction.

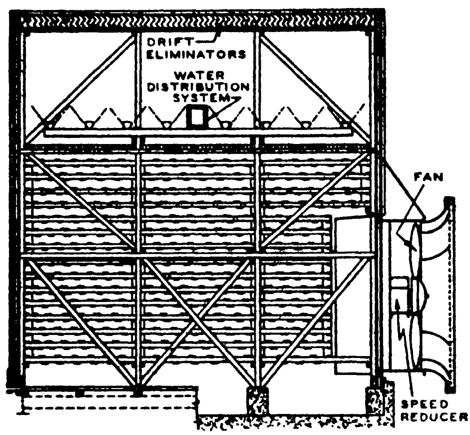

FIG. 7. FORCED DRAFT COOLING TOWER

In the wood-filled tower, lumber of various cross sections is laid horizontally across the space on as close centers, horizontally and vertically, as required, without introducing too great a resistance to air flow. The water is distributed over the top layer by means of spray nozzles, troughs, splash heads, or through evenly spaced nozzles located in the floor of an overhead open-type water distribution basin, and drops from piece to piece of the wood filling as it progresses downward. As the air moves upward or across the wood filling, the latter presents a large wetted surface, repeatedly breaks up the falling drops of water, and continuously provides new drop surfaces whose integrated areas are several times that of the wood-fill area.

The efficiency of a mechanical draft tower is improved by increasing the amount of filling, height, area, or air quantity. Increasing the height increases the length of time the air is in contact with the water, without affecting seriously the fan power required, but increases the pumping power. Increasing the area while maintaining constant fan power increases the air quantity somewhat and, because of lowered velocity, increases the time this air is in contact with the water. The surface area of water in contact with the air is increased in both cases. Increasing the air quantity decreases the time the air is in contact with the water, but since a greater quantity of air is passing through, the average differential between the water temperature and wet-bulb temperature of the air is increased, and this speeds up the heat transfer rate. Increased air quantities are obtained only at the expense of increased fan power, which, for fans of the disc type, increases approximately as the cube of the air handled.

The performance of mechanical draft towers is independent of wind velocity; hence, it is possible to design them for more exacting performance.

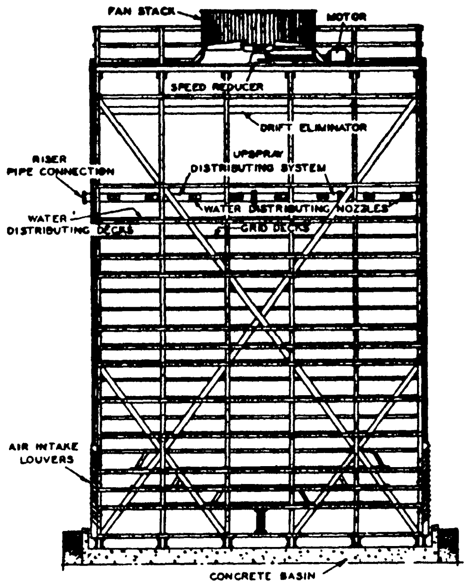

FIG. 8. COUNTERFLOW INDUCED DRAFT COOLING TOWER

They require less space and less piping than atmospheric deck towers, and the pumping head varies from 11 to 26 feet, depending upon the design. Overall plant economy, due to colder water temperature, usually more than offsets the additional operating expense and initial cost as compared with those of atmospheric towers.

The *counterflow* (conventional) type of *induced draft tower* has the fan located at the top, Fig. 8, to provide vertical air movement across the filling. Air is discharged upward at a high velocity to prevent recirculation. Another type, for small requirements, has the induced draft fan in one end (see Fig. 9) to provide horizontal flow.

Another induced draft tower, developed for the purpose of obtaining compactness, larger capacity, increased flexibility and improved performance, is the *crossflow* type. This type of tower employs multiple fans centered along the top, each fan drawing air through two cells paired to the suction chamber which is partitioned midway beneath the fans and fitted with *drift eliminators* that turn the air upward toward the fan outlet. This tower obtains a horizontal air movement as water falls in a cascade of small drops over the filling and across the air stream with less resistance to air flow. The air travel is longer than with the conventional design.

Air velocities through mechanical draft towers vary from 250 to 400 fpm over the gross area of the structure. The air requirements are approximately 300 to 400 cfm of air per ton of mechanical refrigeration, and about 100 to 150 cfm of air per gallon of water passing through the tower. Cooling tower calculations are based upon the fact that mechanical refrigeration requires approximately 30 gallon-deg of cooling water per minute per ton of refrigeration. In atmospheric cooling towers, if 5 gpm were circulated, the water-cooling range would be 6 deg; with mechanical draft towers, 3 or 4 gpm are usually circulated for a desired water-cooling range of 10 or 7½ F. Some designs of mechanical draft towers are limited to 6 or 7 gpm per sq ft because of blanketing effect, while the capacity of the most efficient types ranges up to 9 or 10 gpm.

When an inside cooling tower is required, some adaptation of a spray filled or wood filled induced draft tower is often used, and occasionally an

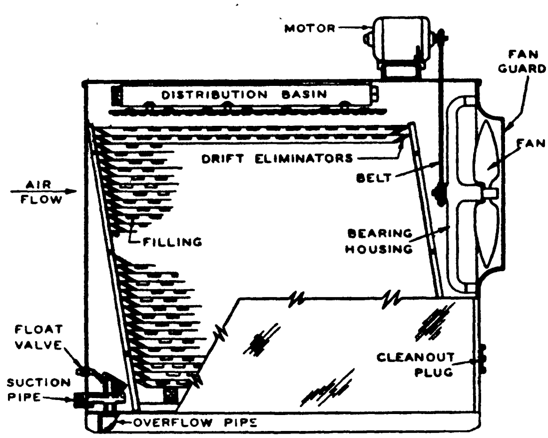

FIG. 9. SMALL HORIZONTAL INDUCED DRAFT COOLING TOWER FOR 3 TO 50-TON REFRIGERATING UNITS

air washer is converted to this service. In this type of application precautions must be taken to prevent the discharged air from short circuiting to the intake.

MECHANICS OF ATMOSPHERIC WATER-COOLING

The heat exchange in atmospheric water-cooling equipment is accomplished partially by a transfer of sensible heat which raises the wet-bulb temperature of the moving air; but most of the cooling is due to an exchange of latent heat resulting from the evaporation of a small part of the water. If all of the water were cooled by evaporation, the rate of evaporation would be approximately one percent for each 10 deg of cooling. In practice, the loss of circulating water by evaporation will approximate 1 percent for 12 to 14 deg of actual cooling due to the additional amount of cooling by sensible heat transfer, and the rate of evaporation will vary from about 0.64 percent of the water circulated in the winter to 0.88 percent in the summer for a water-cooling range of 10 deg.

The lowest temperature to which water may be cooled in atmospheric cooling equipment is the temperature of adiabatic saturation, which is at the wet-bulb temperature of the air. Performance is measured in terms of *approach* (5 to 10 F deg, with 7 F deg average) of the cooled water to the wet-bulb temperature of the ambient air when cooling the water through some desired range. The *water-cooling range* in some installations will vary from 10 to 12 F deg when a spray pond is used, and from 5 to 17 F deg (with 10 F deg average) for a mechanical draft cooling tower.

Heat absorption by the moving air in an atmospheric water-cooling tower continues as long as the wet-bulb temperature of the air is lower than the temperature of the water. The rate of heat transfer depends upon: (1) the area of water in contact with the air; (2) the relative velocity of the air and water during contact; (3) the difference between the wet-bulb temperature of the air and the initial temperature of the water; and (4) the time of contact of the air with the water. The rate of heat dissipation is also influenced by many other lesser factors[3] which further complicate the cooling tower design. Ultimate selection of water-cooling equipment for any specified service depends on overall economic considerations established from correlated performance data. As the enthalpy of the moving air increases, its wet-bulb temperature rises (see Chapter 3). Since it is impracticable to allow the air to be in contact with the water for a long enough time to permit the wet-bulb temperature of the moving air and the temperature of the water to reach equilibrium, atmospheric water-cooling equipment aims to circulate only enough air to cool the water to the desired temperature with least expenditure of power.

DESIGN CONDITIONS FOR WATER-COOLING

The maximum wet-bulb (design) temperature at which the total quantity of circulating water must be cooled through a specified range by water-cooling equipment is never selected as the highest wet-bulb temperature ever known to have occurred for some locality, nor the average wet-bulb temperature over any period of time. The *maximum* basis would require cooling equipment several times larger than normal capacity, and the *average* basis would result, for a large part of the time, in higher condenser temperatures than those for which the plant was designed.

Accepted design practice for water-cooling towers, evaporative condensers, and spray ponds, is to use the maximum hourly outdoor dry-bulb temperature which will be exceeded no more than $2\frac{1}{2}$ percent of the time for the months of June to September; also, to use the maximum hourly wet-bulb temperature which will be exceeded no more than 5 percent of the total hours for the same period. Tabulation of these data has not been completed. The limited portion of such data as are available is given in Table 3, Chapter 12 for airport weather stations; for other localities design dry-bulb and wet-bulb temperatures in use locally are tabulated as a guide to design temperatures. More complete summer weather data, statistics, charts, maps, and technical analysis have been prepared by Albright.[4]

Equipment for steam turbine condensers and internal combustion engines, is usually based upon somewhat lower design temperatures if peak loads occur at night or during winter months when outdoor temperatures are lower.

Knowing the hot water temperature and the wet-bulb temperature for which the equipment must be designed, the cold water temperature must be chosen to place the requirement within the effectiveness range of the type of atmospheric water cooling apparatus to be used. This effectiveness is expressed as the percentage ratio of the actual cooling effect to the maximum possible cooling effect. Since the wet-bulb temperature of the entering air is the equilibrium temperature to which the water could be cooled, the effectiveness of water cooling apparatus can be indicated thus:

$$E_1 = \frac{(\text{hot water temperature} - \text{cold water temperature}) \times 100}{\text{hot water temperature} - \text{wet-bulb temperature of entering air}} \qquad (2)$$

where

E_1 = water cooling effectiveness, percent.

Magnitudes of this effectiveness ratio will vary through wide limits in accordance with construction and conditions of operation. Values indicative of the commercial range of the effectiveness ratio are given in Table 3, although unusual designs may operate outside these ranges.

Example 1: A mechanical refrigeration installation requires 3 gpm of cooling water, per ton of refrigeration, with the hot-water temperature at 95 F and the cold-water at 84 F, with a design wet-bulb temperature of 78 F. Find the water-cooling effectiveness of a mechanical draft tower for the above conditions.

Solution: Substituting known conditions in Equation 2,

$$\text{Water-cooling effectiveness} = \frac{95 - 84}{95 - 78} \times 100 = 64.7 \text{ percent (typical).}$$

From a consideration of the factors which include the water-cooling range and the design wet-bulb temperature of the ambient air, the quantity of water required can be calculated from the amount of heat to be rejected. The average quantities of heat to be removed from various types of mechanical equipment that require cooling are listed in Table 4.

WATER-COOLING TOWER DESIGN

Because of the many variables[3] in water-cooling tower calculations and performance, it is difficult to provide simple handbook equations and tables whereby an engineer can readily select the type and size of unit for a definite requirement. Each manufacturer has a semi-confidential method of sizing a tower, based largely upon research and actual performance correlated with definite requirements; selection of water-cooling equipment for any specified service must ultimately depend upon overall considerations established from reliable design and performance data.

Some of the variables encountered in water-cooling tower work are: continuously changing air and water temperatures throughout the structure; varying moisture content, pressure, and volume of the moving air; caprice of the weather, ambient air changes in temperature, humidity, wind velocity and direction, and the amount of sunshine. Other less important physical properties of the air and water affecting tower performance are: density, specific heat, conductivity, viscosity, vapor pressure, surface tension, latent heat, coefficient of expansion, vapor diffusivity, emissivity, and molecular weight. The air velocity, overall and in different parts of the tower, and also the type of air movement provided by natural draft, forced draft, induced draft, counterflow, or crossflow design have an important effect on heat transfer. Different features of construction will produce dissimilar velocities of water, and will affect its distribution and diffusion as well as the size of drops, jets, sprays, and sheets. The pressure and elevation of the water supply system, as well as the adsorption

TABLE 3. EFFECTIVENESS OF WATER COOLING EQUIPMENT

Cooling Equipment	Water Cooling Effectiveness—Percent		
	Minimum	Typical	Maximum
Spray Ponds	30	40 to 50	60
Spray Filled Atmospheric Towers.	40	45 to 55	60
Atmospheric Deck Towers	50	50 to 60	90
Mechanical Draft Towers	50	55 to 75	93

and interfacial surface tension of the wetted tower areas also affect distribution.

Dissolved gases and other impurities in the water influence the water-cooling process. Additional cooling tower variables affecting its performance include: location (ground, roof, nearby obstructions, wind orientation), dimensions, relative proportions (contour) of tower structure, materials, type and arrangement of interior surfaces; the louver and drift-eliminator designs as they facilitate the air flow to and from the tower; temperature of the structure at different points as influenced by the external and internal conditions. Other cooling tower factors to be considered are: loss of water by entrainment (drift loss), design and location of water-collecting basin, and surface evaporation therefrom; also the noise generated by the air, water, fan, and structure vibration.

Basically, a water-cooling tower is a heat exchanger in which heat flows from the water to the air (1) by a flow of sensible heat from the warm water to the cooler air, and (2) by an exchange of latent heat resulting from the evaporation of a small part of the circulating water to increase the humidity ratio of the air by a corresponding amount. The general principles involved are similar to those encountered in the processes of diffusion in absorption and extraction equipment.[5,6]

Details of the application of the process to water-cooling tower performance have been published by various authorities,[7,8,9,10,11,12] and those interested in the derivation of the various equations should refer to these references, as listed at the end of this chapter. The approach in each case is based on a heat balance in which the total heat given up by the water equals the total heat absorbed by the air. These derivations are also based on certain assumptions, *viz:* that the specific heat of water is unity at the temperatures encountered; that there is no loss in weight of the

TABLE 4. HEAT ABSORBED BY COOLING WATER

Mechanical Equipment	Btu per Min per Ton	Btu per Lb of Steam	Btu per bhp-hr
Refrigeration Compressor	250	—	—
Refrigeration, Absorption System	550	—	—
Steam Turbine Condenser	—	1000	—
Steam Jet Refrigerating Condenser	550	1100	—
Diesel Engine Jacket & Lube Oil:			
Four-cycle, Supercharged	—	—	2600
Four-cycle, Non-supercharged	—	—	3000
Two-cycle, Crank-case Compressor	—	—	2000
Two-cycle, Pump Scavenging, Large Unit.	—	—	2500
Two-cycle, Pump Scavenging, High Speed	—	—	2200
Natural Gas Engine:			
Four-cycle	—	—	4500
Two-cycle	—	—	4000

circulating water as a result of evaporation; that the water suspended in the tower is surrounded by a film of air which is saturated with water vapor and at the temperature of the water surrounded; and that the basic theory of cooling tower operation proposed by Lewis[13] and developed by Merkel[14] is applicable. This theory refers to the fact that the numerical value of the coefficient of sensible heat transfer, when divided by the numerical value of the coefficient of diffusion, equals the specific heat (at constant pressure) of air. The reader should observe that this relationship refers to the numerical values of three distinct constants, the units for each being different. The above relationship makes it possible to simplify the heat transfer equation by combining the two driving forces into one potential represented as the difference between the enthalpy of the air film (at water temperature) surrounding the water, and the enthalpy of the main air stream.

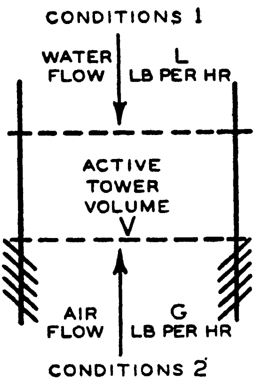

FIG. 10. OPERATIONS IN A TYPICAL WATER COOLING TOWER

Tower Performance Factor

The operations taking place in a typical water-cooling tower are shown in Fig. 10. If the reduction in water flow rate, due to evaporation within the volume, is neglected, and the usual concepts of heat flow and mass heat transfer are applied, the equations typifying cooling tower operation are:

$$\frac{KaV}{G} = \int_2^1 \frac{dh}{h'' - h_a} \tag{3}$$

and

$$\frac{KaV}{L} = \int_2^1 \frac{d\theta}{h'' - h_a} \tag{4}$$

where

a = overall average wetted area (surface of water drops plus wetted tower surface), square feet per cubic foot of active tower volume.
G = weight rate of flow of air, pounds of dry air per hour.
h = enthalpy, Btu per pound of dry air.

h_a = enthalpy of air-vapor mixture, Btu per pound of dry air.
h'' = enthalpy of saturated air-vapor mixture at water temperature, Btu per pound of dry air.
K = overall energy unit conductance, Btu per (hour) (square foot overall average wetted area) (Btu enthalpy difference per pound of dry air).
L = water rate, pounds per hour.
θ = temperature of water in tower, Fahrenheit.
θ_1 = temperature of inlet water, Fahrenheit.
θ_2 = temperature of outlet water, Fahrenheit.
V = active tower volume, cubic feet.

Either term $\frac{KaV}{G}$ or $\frac{KaV}{L}$ may be called the *Tower Performance Factor* or *Number of Tower Units* (*NTU*).

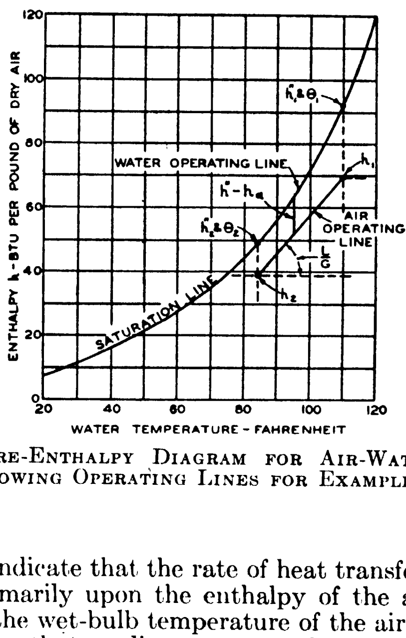

FIG. 11. TEMPERATURE-ENTHALPY DIAGRAM FOR AIR-WATER VAPOR MIXTURE SHOWING OPERATING LINES FOR EXAMPLE 2

These equations indicate that the rate of heat transfer from the water to the air depends primarily upon the enthalpy of the air, the latter being dependent only on the wet-bulb temperature of the air. This explains the common observation that cooling tower performance is independent of inlet dry-bulb air temperature, and that adiabatic conditions exist.

The integration of Equations 3 and 4 must be performed by mechanical or graphical means, because direct mathematical integration would be accurate only within narrow temperature limits. The temperature enthalpy diagram in Fig. 11 represents the conditions for either of the above equations. The water is cooled from the temperature θ_1 to θ_2, and the enthalpy of the air film surrounding it follows the saturation line h''. Air enters the tower at a wet-bulb temperature of t_2', and an enthalpy of h_2. It is heated to an outlet wet-bulb temperature of t_1', with an enthalpy of h_1. Since the heat rejected by the water equals the heat absorbed by the air, the heat absorbed per pound of air is a function of the pounds of water per pound of air going through the tower, and the slope of the air operating line is the L/G ratio.

Example 2: It is desired to cool 150,000 lb of water per hour (about 100 tons of mechanical refrigeration) from 110 to 84 F with 125,000 lb of dry air per hour, with a design wet-bulb air temperature of 75 F. These conditions could prevail with a steam-turbine driven centrifugal refrigeration compressor. Determine the *Tower Performance Factor;* show in tabular form the successive steps for this mechanical integration by selecting two-degree intervals of the water-temperature range.

Solution: The accompanying Table 5 shows the sequence of mechanical integration for the given water and air temperatures. The first column shows the water temperature θ in increments of two degrees ($\Delta\theta = 2$ F deg). Column 2 gives the enthalpies of the saturated air-vapor mixture at the water temperature, Btu per pound of dry air. The enthalpy of air, h_a in column 3, has an original value of 38.61 Btu per lb corresponding to the 75 F entering wet-bulb temperature of the ambient air; this value of h_a increases in equal increments of 1.2 $\left(\frac{L}{G}\text{ratio}\right)$ Btu per F deg, hence, $\Delta h = \Delta\theta \times \frac{L}{G} = 2 \times \frac{150{,}000 \text{ lb}}{125{,}000 \text{ lb}} = 2.4$. The potential for mass heat transfer is $(h'' - h_a)$ as shown in column 4; this is frequently called the tower driving force potential. The values in column 5 for each increment are determined by dividing 2.4 Btu per F deg by the average value of $(h'' - h_a)$; and column 6 is calculated in a similar manner, except that the increments are two degrees instead of 2.4 Btu.

TABLE 5. SEQUENCE OF MECHANICAL INTEGRATION TOWER PERFORMANCE FACTOR

1 WATER TEMP. θ	2 ENTHALPY OF FILM h''	3 ENTHALPY OF AIR h_a	4 ENTHALPY DIFFERENCE $(h'' - h_a)$	5 $\frac{\Delta h}{(h'' - h_a)}$ (avg.)	6 $\frac{\Delta\theta}{(h'' - h_a)}$ (avg.)
84	48.22	38.61	9.61	0.249	0.208
86	50.66	41.01	9.65	0.247	0.206
88	53.23	43.41	9.82	0.241	0.201
90	55.93	45.81	10.12	0.232	0.194
92	58.78	48.21	10.57	0.221	0.184
94	61.77	50.61	11.16	0.208	0.173
96	64.92	53.01	11.91	0.194	0.162
98	68.23	55.41	12.82	0.180	0.150
100	71.73	57.81	13.92	0.165	0.137
102	75.42	60.21	15.21	0.150	0.125
104	79.31	62.61	16.70	0.137	0.114
106	83.42	65.01	18.41	0.124	0.103
108	87.76	67.41	20.35	0.112	0.093
110	92.34	69.81	22.53		
			Tower Performance Factor =	2.460 or	2.050

Hence, the mechanical integration for the above conditions gives two results:

$$\frac{KaV}{G} = \sum \frac{dh}{h'' - h_a} = 2.46, \textit{ Tower Performance Factor}$$

and

$$\frac{KaV}{L} = \sum \frac{d\theta}{h'' - h_a} = 2.05, \textit{ Tower Performance Factor}$$

The results obtained in *Example 2* are designated as the *Tower Performance Factor* (*TPF*) or the *Number of Tower Units* (*NTU*); these figures represent correlated values that are directly proportional to the performance being considered. Similar calculations could be made for other quantities and temperatures of air and water. It should be noted that this factor is not related to the equipment doing the cooling, that any numerical value may represent an infinite number of possible performance conditions, that any cooling tower arrangement may give almost any performance under certain conditions. Also the mechanical integration procedure used above applies only to counterflow apparatus. However, the same principles may be applied to crossflow atmospheric water-cooling towers, although the method is more involved.

The basic mathematical theory for water-cooling towers is now well established and recognized, but each manufacturer relies upon experimental results and practical experience with his own tower designs to establish a system for rating each unit that he builds. The problem of cooling tower design or selection is based on a knowledge of the characteristics of the equipment being considered. The *Tower Performance Factor* is a variable which is a function of the design; it also varies with the water loading and air velocity. Experimental data indicate that it varies with the heat load, although this variation may be due to deviations from the theoretical calculations which become more pronounced at the higher temperatures. The reference literature contains *Tower Performance Factors* which have been reported by various investigators, but the reader should be warned that the use of such factors, without a full understanding of the source, may lead to erroneous results.

SELECTION OF WATER-COOLING TOWERS

The correct type and size of water-cooling equipment for a given service cannot be determined intelligently without considering the characteristics of the various types, together with the many correlated requirement factors. Very few installations are exactly alike in details of requirements, hence, conditions affecting performance and operation of the several types of water-cooling equipment vary widely because of the many diversified applications and wide-spread geographical locations.

Before the characteristics of a specific water-cooling apparatus can be judged desirable or undesirable for a given heat load and wet-bulb temperature, a survey should consider the importance of each of the following items: first cost including all necessary auxiliaries, area, height, weight, effect of wind velocity and direction, rigidity of structure to withstand high winds, safety, conformity to building codes, drift nuisance, make-up water requirements and cost of chemical treatment if needed, total power for pumping (plus fan operation in the case of mechanical draft), maintenance, available locations (with due thought to possible future expansion, wind restrictions, space cost, proximity and accessibility, etc.), appearance, the equipment's operating flexibility for the most economical conformance to varying loads or seasonal changes, and other considerations occurring with regard to a specific application.

For a definite heat-load dissipation, the type and size of a water-cooling tower is primarily affected by the following conditions:

1. Gallons per minute of cooling water.
2. Geographical location of the tower installation.
3. Wet-bulb design temperature of ambient air (see Table 3, Chapter 12).
4. Temperature of the hot water entering the tower at normal rating.
5. Temperature of the cold water leaving the tower at normal rating.
6. Ground, roof, or sub-structure installation.
7. Area available for cooling tower.
8. Proximity to other structures.
9. Surface of water exposed to each unit quantity of air.
10. Time of contact of the air with the water; this depends upon height (or length) of tower, and upon the relative velocity of air and water.

The selection of a proper water-cooling range depends upon: (1) type of service—refrigeration, internal-combustion engine, or steam condenser; (2) wet-bulb air temperature at which the equipment must operate; and (3) type of condenser or heat exchanger employed.

Because the design of an entire plant is usually affected by the quantity and temperature of the cooling water supply, plants should be designed for cooling water conditions which can be most efficiently attained. The first consideration is usually the limiting temperature of the plant. For example, if an ammonia compressor refrigerating plant is to be designed for 185 psig head pressure as a normal maximum, the limiting temperature of the ammonia in the condenser is 96 F. Should the ammonia temperature go above this figure, the head pressure will exceed 185 psig and the power consumption increase. To obtain this head pressure, the temperature of the circulating water leaving the condenser must always be less

TABLE 6. CONDENSER DESIGN DATA

Gas	Desired Pressure in Condenser	Gas Temperature in Condenser, Fahrenheit	Leaving Hot-Water Temperature, Fahrenheit	
			Best Condenser Design	Average Condenser Design
Steam	28 in. vacuum	101.2	97	93
Steam	27 in. vacuum	115.1	110	105
Steam	26 in. vacuum	125.4	120	114
Ammonia	185 psi*	96.0	92	88
Carbon dioxide	1030 psig*	86.0	83	80
Methyl chloride	102 psig*	100.0	96	92
Freon, F-12	117 psig*	100.0	96	93
Freon, F-12	126 psig*	105.0	100	97
Freon, F-12	136 psig*	110.0	104	101

* Head pressure.

than 96 F by an amount depending upon the size and design of the condenser, the quantity of water being circulated, and the refrigerating tonnage being produced. A condenser having a large surface per ton of refrigeration may be designed to operate satisfactorily with the leaving hot-water temperature within 3 or 4 deg of the ammonia temperature corresponding to the head pressure, while a small condenser may require a 10 deg difference.

Table 6 lists several gases with data for the temperatures and pressures for which commercial condensers are designed. Careful evaluation of costs of water and electrical power should be made before deciding to use city water for jacket water and condensers. Economy of operation generally indicates the use of either a water-cooling tower or an evaporative condenser for most refrigeration installations of five tons or more capacity. Refer to Chapter 36, for information on Evaporative Condensers. Internal-combustion engines have limiting hot water temperatures of 140 to 180 F for closed systems, and 110 to 130 F for open systems, depending upon the quality of the cooling water. The cooling of such fluids as milk or wort has variable requirements, and is usually accomplished in counterflow heat-exchangers in which the leaving circulating water is at a much higher temperature than is the leaving fluid.

OPERATION AND MAINTENANCE

Water Treatment. The amount of make-up water required by a cooling tower depends upon evaporation loss, drift loss, and blow-down. Evaporation losses average 0.80 percent of the water circulated for each 10 F deg range. Drift loss is the water carried out of the tower by the air currents in the form of droplets or mist. In properly designed induced draft towers this loss normally approximates one-tenth of one percent, and most cooling tower manufacturers will guarantee a drift loss not to exceed two-tenths of one percent. The amount of blow-down water wasted depends upon the hardness of the circulating water, type of water softening used and the amount of drift loss. Blow-down is normally controlled to maintain the concentration of soluble and scale-forming solids below the point where the formation of scale would occur or would be caused by corrosion.

Algae formations will plug nozzles and prevent proper distribution of the water over the tower filling. This growth may also collect on equipment served by the cooling tower, and thereby reduce the heat transfer rate. Algae should be held at a minimum or eliminated by use of bromine, chlorine, chlorinated lime, copper sulfate, or various blends of chlorophenates (see Chapter 42).

Although some scale-forming materials are found in practically all water, those which cause trouble in water-cooling systems are normally calcium and magnesium carbonates. Scale formation in equipment served also reduces heat transfer rates. Scale can be reduced materially or prevented by softening the make-up water with lime and soda ash, zeolite, or sulfuric acid, or by use of small amounts of sodium hexametaphosphate. Water softening or treatment requires close regulation and control by a competent chemist. Too high a concentration of soluble solids in cooling tower water may raise the temperature of the water leaving the tower, and may cause sludge deposits or corrosion in the system. Concentration of solids is normally controlled by either blowing down or by a continuous overflow to the sewer. Refer also to Chapter 42.

Delignification. The presence of sodium carbonate in the circulating water results in delignification of any wood with which water comes in contact. This chemical dissolves lignin which binds the wood fibers together and leaves the wood surface in a white fibrous condition. Prolonged exposure reduces the structural strength of the wood. Delignification first appears on parts of the tower that are alternately wet and dry, since evaporation at such points rapidly increases the concentration of dissolved solids. The presence of sodium carbonate in harmful amounts is generally indicated by a high pH of 9 to 11. The effect of the sodium carbonate may be neutralized by the use of sulfuric acid. It is desirable to have the pH value of the water at 7 to 7.5 (7.2 pH value is neutral for redwood).

Two-speed Motors. For readily adapting tower performance to temporary or seasonal decreases in heat load, and especially for winter operation, the use of two-speed motors (for fan drives) is recommended. The chief advantage is that when operated at half-speed, fans require only about 15 percent of the power used at full speed. Particularly in multi-fan towers, the ready flexibility provided by two-speed motors results in considerable savings, even though load reductions may sometimes call for only one or a few fans to be operated at half speed.

Cold-weather Operation. Extremely cold water normally does not increase performance to any great extent, but increases operating hazards considerably. Water-cooling towers operated in sub-freezing weather are subject to ice formation on the louvers and the outer portion of the filling.

To prevent icing in cold-weather operation, the cold raw water (tower circulating water) temperature should be maintained as high as practicable, taking into consideration the effect upon the economy of the equipment served. One or more of the following procedures are recommended for induced draft towers: (a) run two-speed motors on low speed, or shut off some of the fans; (b) shut down some cells completely and put all of the water over the remaining cells; (c) reduce water flow to the tower and shut off some of the cells; (d) by-pass the cooling tower with part of the water and shut off some of the fans or cells of the tower.

If ice should form on the louvers and filling, one of the following methods of removal can be used: (a) reversing (for not more than 10 minutes) the rotation of the motor driving the fan and thus blowing the warm air out through the louvers; (b) shutting down fans on some sections temporarily, but not the water. When these cells have thawed out, use the same procedure on other cells.

Where intermittent operation of a system is employed, water in outside basins may cause considerable damage due to freezing. To prevent this, such basins are drained when out of service and therefore in some small roof installations a tank large enough to hold all the water in the system may be installed inside the building.

Maintenance. Well-maintained equipment provides the best operating results and the least overall maintenance cost. A regular schedule should be set up for the structural and mechanical upkeep of water-cooling towers. The life and continued utility of any cooling tower is directly dependent upon its inherent qualities, climatic environment, type of service, severity of operation, and general care and maintenance.

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CHAPTER 35

AIR HEATING AND COOLING COILS

Uses for Coils, Coil Construction and Arrangement, Steam Coils, Water Coils, Direct-Expansion Coils, Flow Arrangement, Applications, Coil Selection, Heat Transfer and Air Flow Resistance, Performance of Heating and Dry Cooling Coils, Overall Coefficient of Heat Transfer, Performance of Dehumidifying Coils, External Film Coefficient, Internal Film Coefficient, Determining Size of Cooling Coil

COILS described in this chapter are used for heating or cooling an air stream under forced convection. Surface coil equipment may be made up of a number of banks assembled in the field, or the entire assembly may be factory constructed. The applications of each type of coil are limited to the field within which it is rated. Other limitations are imposed by code regulations, by proper choice of materials for the fluids used and the condition of the air handled, or by an economic analysis of the possible alternates on each installation.

USES FOR COILS

For *heating service*, coils are used as tempering coils, preheaters, reheaters or booster heaters. The function of the coils is air heating only, but the apparatus assembly may include means for humidification and air cleaning. Steam or hot water are the usual heating media, although others are used in special cases, such as reheating by means of discharge gas from a refrigerating system.

Coils are used for *air cooling* with or without accompanying dehumidification. Examples of cooling applications without dehumidification are precooling coils using well water or other relatively high temperature water to reduce the load on the refrigerating machinery, or water cooled coils removing sensible heat in connection with chemical moisture-absorption apparatus. By proper coil selection it is possible to handle both sensible cooling and dehumidification together. The assembly usually includes air cleaning means to protect the coil from accumulation of dirt, and to keep dust and foreign matter out of the conditioned space. Although cooling and dehumidification are the usual functions, cooling coils are sometimes purposely wetted to aid in air cleaning and odor absorption.

The usual cooling media used in surface coils are cold water or Group I (*ASA* Classification) refrigerants, but others are used in special cases. Brines are seldom required for the range of applications covered by this chapter, although there are cases where low entering air temperatures with large latent heat loads require a refrigerant temperature so low that use of water becomes impracticable. Sometimes, also, brine from an industrial system already installed is the only convenient source of refrigeration.

For combined cooling and dehumidifying, surface coils present an alternate to spray dehumidifiers. For many applications it is possible, by proper selection of apparatus, choice of air velocities, refrigerant temperatures, etc., to perform the same duty with either. In a few cases both sprays and coils are used. The coils may then be installed within the spray chamber, either in series with the sprays or below them. In making the selection between spray and surface dehumidifiers, certain advantages of each should be considered. The fact that a spray dehumidifier is usually designed to deliver nearly saturated air, tends to simplify the control problem. In this case the dry-bulb temperature is also the dew-point, and hence, a dew-point control can be arranged by using a simple duct thermostat. Spray dehumidifiers have an advantage over unwetted coils of obtaining some air cleaning and odor absorption. On the other hand, coils make possible a closed and balanced cooling water circuit, obviating the unbalanced pumping head, the complication of water level control, and danger from possible floods incidental to multiple spray dehumidifiers, especially if located on different levels. The use of coils often makes it possible for the same surface to serve for summer cooling and winter heating by circulating cold water in the one season and hot water in the other, with consequent saving in apparatus and piping. Another advantage is that where the surface coil system can be used with direct expansion of refrigerant, it is comparatively low in initial and operating costs. The safety of the occupant must be kept in mind in comfort conditioning applications. Some localities have refrigeration codes which restrict the use of direct-expansion coils in the air stream, and hence, local codes should be consulted by the engineer before a system employing direct expansion methods is designed. The choice between spray dehumidifiers and coils depends upon the necessities and the economic aspects of each case, and no general rule can be given. There are many installations in which either may be used.

COIL CONSTRUCTION AND ARRANGEMENT

Coils are basically of two types, those consisting of plain tubes or pipe, and those having *extended* surfaces. The former are little used for the applications covered by this chapter, but are often employed where conditions cause frost accumulation, and for cooling within spray dehumidifiers.

The heat transmission from air passing over a tube to a fluid flowing within it is impeded by three resistances. The first is that from the air to the surface of the tube and is usually called the outside surface resistance or air-film resistance. The second is the resistance to the conduction of heat through the metal itself. Finally there is another surface or film resistance to the flow of heat between the inside surface of the metal and the fluid in the tube. For the applications under consideration both the resistance of the metal wall to heat conduction, and the inside surface or film resistance are usually low as compared with the air-side surface resistance. Economy in space, weight and cost makes it advantageous to decrease the external surface resistance, where it is proportionately large, to approach that of the tube wall, and that from the tube to refrigerant. This may be accomplished by increasing the external surface by means of fins. Sometimes water spray is applied to the same type surface as would have been used without it. The overall heat transfer is not necessarily increased much, but the water spray may serve other purposes than to increase the flow of heat, such as air and coil cleaning.

In fin or *extended* surface coils the external surface of the tubes is known as *primary*, and the fin surface is called *secondary*. The primary surface consists generally of round tubes or pipes which may be staggered, or in some cases placed in line with respect to the air flow. The staggered arrangement is usually preferred because it obtains a somewhat higher heat transfer value. Numerous types of fin arrangement are used, the most common of which are spiral, flat and flat-crinkled or corrugated, all as shown in Fig. 1. While the spiral fin surrounds each tube individually in all cases, the flat types may be continuous (including several rows of tubes), or they may be round or square, with individual fins for each tube. All of these, as well as other less common types, are in use, the selection for a particular installation being based on economic considerations, space requirements and resistances of individual designs of coils. A most important factor in the performance of extended surface coils is the bond between the fin and the tube. An intimate contact between the tube and the fin must be maintained permanently in order to assure a continuing rated performance after the heating units have been in service for a period of time. In some coils, fins are wound on the tubes under pressure, in order to upset the metal slightly at the fin root, and then are given a coating of solder while the fin and tube are still revolving, for the purpose of assuring a uniform coating of solder. In other types, the spiral fin may be knurled into a shallow groove on the exterior of the tube. The tube may be ex-

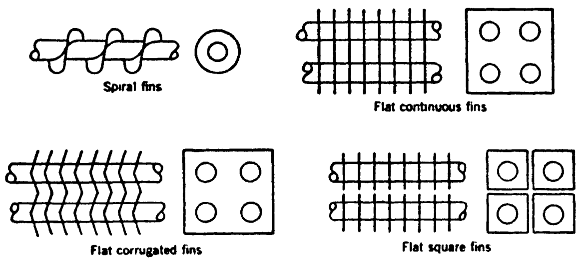

FIG. 1 TYPES OF FIN COIL ARRANGEMENT

panded after the fins are assembled, or the tube hole flanges of a flat or corrugated fin may be made to override those in the preceding fin and so compress them upon the tube. There are also types of construction where the fin is formed out of the material of the tube itself.

For *heating coils*, materials most generally used are copper and aluminum. Steel is occasionally used where sodium or calcium chloride brine is circulated in the tubes. Aluminum fins on copper tubes are a common construction. Generally speaking, brass does not serve as a satisfactory fin material because of corrosion difficulties. *Cooling coils* for water or for volatile refrigerants most frequently have copper fins and tubes, although aluminum fins on copper tubes are also used. There are many makes of heating and cooling coils of the light weight extended surface type for both heating and cooling with tubes commonly $\frac{1}{2}$, $\frac{5}{8}$, $\frac{3}{4}$, and 1 in. outside diameter, and with fins spaced three per inch up to eight per inch. The tube spacing generally varies from about $1\frac{1}{8}$ to $2\frac{1}{2}$ in. on centers, depending upon the width of individual fins and on other considerations of performance. Fin spacing should be chosen for the duty to be performed, with special attention being paid to lint accumulation and, especially in dehumidifying, the consideration of frost accumulation.

Steam Coils

For proper performance of steam heating coils, condensate and air must be continuously eliminated and the steam must be evenly distributed to the individual tubes. This distribution is usually accomplished by individual orifices in the tubes, by distributing plates and orifices in the steam header, or by perforated internal steam-distributing pipes extending into the individual tubes. The latter arrangement has the advantage of distributing the steam throughout the length of each tube, and is conducive to uniform temperature of delivered air. The tendency of condensate to freeze at the bottom of the coil with cold entering air and light heating loads, is also minimized. This is especially valuable for outside air preheaters.

Water Coils

The performance of water coils, for heating or cooling, depends on the elimination of air from the system and proper distribution of water. Air

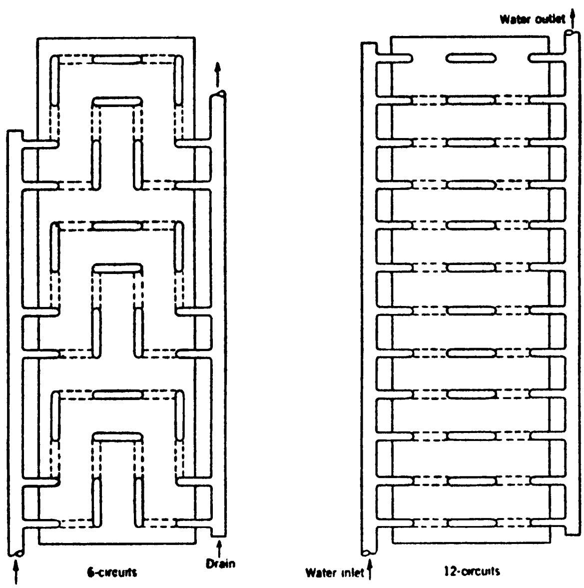

FIG. 2. VARIOUS WATER CIRCUIT ARRANGEMENTS

elimination is taken care of in the system piping as described in Chapter 21. To assure a pressure drop sufficient for adequate distribution, but at the same time to provide against excessive pumping head where large water quantities are handled, water coils are provided with various water circuit arrangements. For instance, a typical coil 18 tubes high and 6 tubes deep in the direction of air flow can be arranged for 6, 9, 18, 24, or 36 parallel water circuits, as conditions may require. Orifices in individual tubes are occasionally employed, but are usually unnecessary as the resistance of individual water circuits is generally sufficient to effect a satisfactory distribution. In precooling coils using well water, where there may be considerable sand and other foreign matter in the water, provision for cleaning of individual tubes is of advantage. It is important to arrange water coils for complete drainage (see Fig. 2). The drains are usually provided in the water piping at the coil header.

Direct-Expansion Coils

Coils for volatile refrigerants present more complex problems of fluid distribution than do water, brine or steam. It is desirable that the coil be effectively and uniformly cooled throughout, and necessary that the compressor be protected from entrained, unevaporated refrigerant. There are two types, namely, flooded systems, and thermal expansion valve systems, as shown in Figs. 3 and 4. In a flooded coil, the circulation is similar to that in a water tube boiler. The liquid is maintained at the proper level by the action of a float regulator as shown in Fig. 3. The thermal expansion valve system depends upon the thermal valve automatically feeding just as much liquid to the coils as is required to maintain the superheat at the coil suction outlet within predetermined limits, which vary from about 6 to 10 deg. The thermal valve arrangement is in common use for the type of coils covered by this chapter, while the flooded system is rarely used.

With the flooded system the refrigerant distribution through the tubes depends on properly selecting the length of the feeds, and the head of liquid imposed upon the liquid inlets. No auxiliary distributing devices are required. With the thermal valve system, there are two factors to consider.

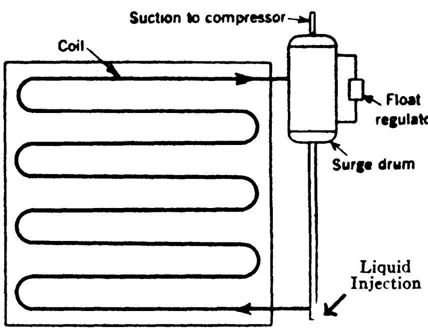

FIG. 3. DIRECT-EXPANSION COIL WITH FLOODED SYSTEM

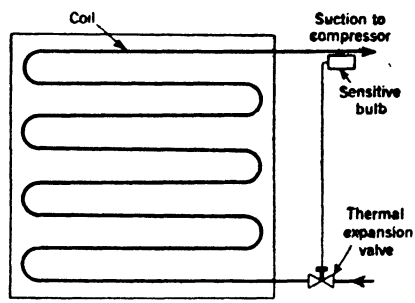

FIG. 4. DIRECT-EXPANSION COIL WITH THERMAL VALVE SYSTEM

There must be, generally, more than one refrigerant feed through the coil per thermal valve to keep the pressure drop through the refrigerant circuit within practical limits, and to reduce the corresponding penalty in increased evaporating temperature. At the same time the coil must be so arranged that the required suction superheat can be attained with a minimum sacrifice in the performance of the coil as a whole. It is general practice to attain this superheat within the coil itself, and not by the use of external heat exchangers or other auxiliary devices.

With thermal expansion valves it is advantageous to keep the pressure drop through the refrigerant feeds as low as possible. The feeds are laid out to expose each to the same mean temperature difference so that it handles the same refrigerating load. Here, a distributing means is imposed between valve and coil liquid inlets to divide the refrigerant equally among the feeds. Such a distributor must be effective for distributing both liquid and vapor, because the entering refrigerant is a mixture of the two. Fig. 5 shows three typical types of distributors. In distributor A the liquid and gas mixture from the thermal valve is led tangentially into a chamber. The coil feed connections extend outward radially at the top of this chamber. In distributor B the refrigerant is discharged at a high velocity through a central jet against the end plate, forming a uniform mixture of gas and liquid within the distributor, from which individual connections are led as shown. In type C the refrigerant enters at high velocity from the thermal valve and is discharged against the end plug in which the individual liquid feeds are closely arranged. These distributors can be used in either vertical or horizontal position. There are also other types of headers such as the centrifugal and weir type. The individual liquid connections from the distributor to the coil inlet are commonly made of small diameter tubing, and are all of the same length and diameter in order to impose the same friction between the distributor and the coil. Since the thermal valves act in response to the superheat at the coil outlet, this superheat should be produced with the least possible sacrifice of active evaporating surface. Sometimes a single thermal valve is used per coil. In other cases, multiple valves are used, with the coil divided across the air flow or parallel to the air flow as shown in Fig. 6. The arrangement of Fig. 7

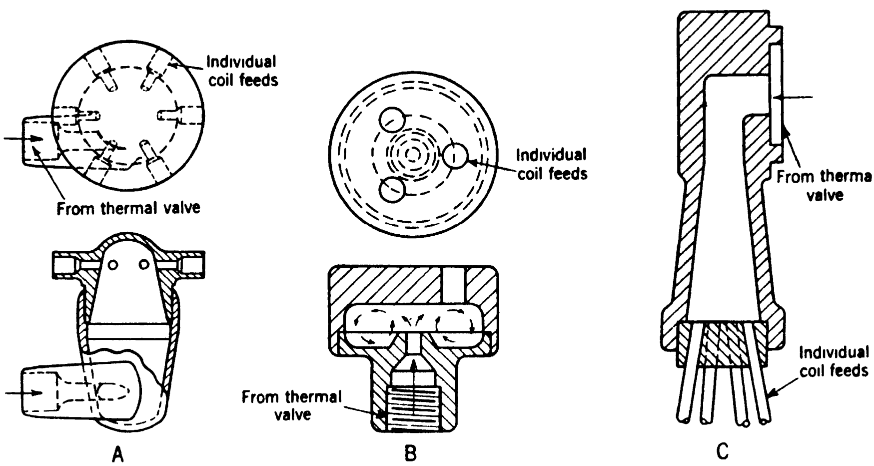

Fig. 5 Types of Refrigerant Feed Distributing Heads

should be avoided, since it offers the disadvantage of unequal load on the two parallel circuits.

Flow Arrangement

The relative directions of flow of the air outside the tubes and the medium within them, influence the performance of the surface. There are three types of relative flow in common use. Fig. 8A shows parallel-flow in which the air and the medium in the tubes proceed through the coil in the same direction. Fig. 8B shows counter-flow in which the medium in the tubes proceeds in a direction opposite to the flow of air. Fig. 8C shows cross-flow in which the air and the medium in the tubes pass at right angles to each other. The counter-flow arrangement is almost universally used in brine or water coils to take advantage of the highest possible mean temperature difference for given entering water and air temperatures. It is also commonly used, in coils fed with volatile refrigerant, to take advantage of the higher air temperature for superheating the leaving gas. In deep coils, however, it is sometimes advantageous to use parallel flow from second row to last row, and then to complete the circuit by passing through the first row to take advantage of the higher air temperature for superheating. Complete evaporation and superheating of the refrigerant are essential to proper operation of the thermal expansion valve. Cross-flow is common in steam heating coils, the temperature within the tubes being substantially uniform, and the mean temperature difference the same whatever the direction of flow, relative to the air. Cross-flow is to be avoided in coils with volatile refrigerants, because of unequal loading of parallel circuits, and the danger of short circuiting of liquid refrigerant which disturbs proper functioning of the thermal expansion valve.

Applications

Heating coils in field assembled banks are used for a number of purposes as described in Chapter 29. They may be arranged with the air flow

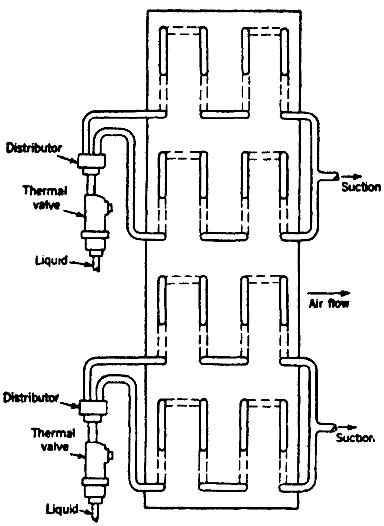

Fig. 6. Arrangement for Face Control

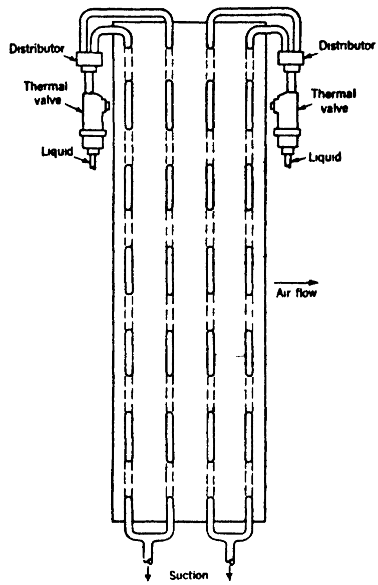

Fig. 7. Arrangement for Depth Control

vertical or horizontal, although the latter is more common. For steam heating, the coils may be set with the tubes vertical or horizontal. In the latter case the coil should be sloped to provide for condensate drainage. Because of the multi-circuit feed arrangement and the necessity for avoiding air and water pockets, water heating coils are generally arranged with the tubes horizontal. Certain precautions must be taken against freezing. Where steam coils are used with entering air below freezing temperature, throttling the steam supply may cause freezing of the condensate in the bottom of the coil, if the tubes are of the variety not provided with internal distributing pipes or an equivalent arrangement.

There are coils available having inner distributing tubes, and having the supply and return headers cast in one piece. In this type of coil the condensate that forms in the outer tube has resulted from steam fed from the inner tube orifices. This condensation flowing back along the warm inner tube is prevented from freezing. A wide range of modulation at very low temperatures without danger of freezing, is therefore obtained. As an added precaution, with both steam and water coils, the outside air inlet dampers are often closed automatically when the fan is stopped to avoid trouble caused by very cold outside air drifting in during *off* periods.

A typical arrangement of cooling coils is shown in Fig. 9. Some means should be provided to filter all the entering air to keep dirt and foreign matter from accumulating on the coils. The assembly is provided with a drip-pan to catch the condensate during summer dehumidifying duty, and to collect the non-evaporated water from the humidifying sprays in winter. The drip connection should be made ample in size and liberally provided with cleanout fittings. It should not be exposed to freezing temperatures in winter if the apparatus is used on winter humidifying duty. Access doors should be provided for servicing filters, humidifying nozzles, and fan bearings, and for cleaning the coils. When coils are used for dehumidifying, eliminators must be used beyond the coil to catch any water which may be blown into the air stream. It is customary to include these eliminators when the air velocity exceeds about 450 fpm. Where a number of coil

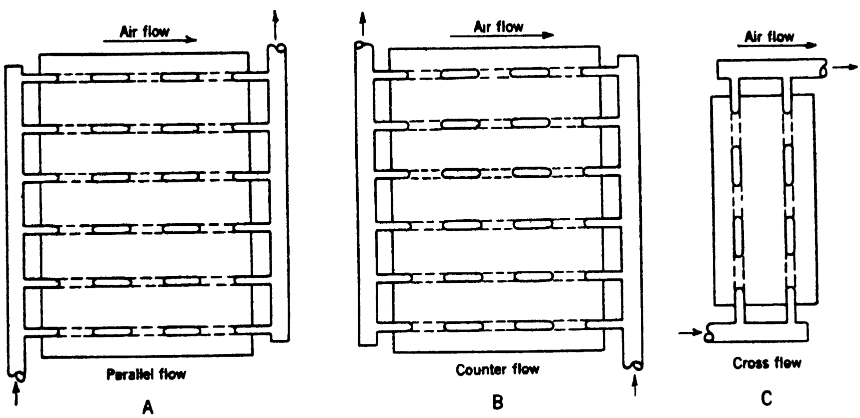

Fig. 8. Flow of Media in Tubes in Relation to Air Flow

sections are stacked one upon another, and where the velocities are low, so that eliminators need not be used, occasional trouble results when water splashes down from one coil to the next and blows out into the air stream. In such cases drip troughs as shown in Fig. 10 are used to collect this water and conduct it to the condensate pan.

Sometimes finned surface coils on summer cooling and dehumidifying duty are provided with water sprays. These sprays are of two types. In the first type, a set of spray nozzles is arranged for intermittent cleaning. These sprays are not operative when the system is in use, and no recirculating pump is provided. The second arrangement requires a collecting tank and a recirculating pump. The water is in circulation whenever the apparatus is in operation, and assists in keeping the coil clean and in absorbing odors. Fig. 11 illustrates such an arrangement. Wherever air by-passes are used around a coil on summer duty for control purposes, it is advantageous to direct only return air through the by-pass rather than a mixture of return and outside air. The casing should be arranged accordingly. To maintain the air quantity handled by the fan reasonably constant, and to assure the required design quantity of by-passed air when the by-pass damper is open, cooling coil banks are frequently furnished with both face and by-pass dampers as shown in Fig. 9.

Although both heating and cooling coils are made of sufficient strength to take up expansion and contraction arising within themselves, care should be taken to avoid imposing strains from the piping on the coil connections. (See Chapter 20.)

COIL SELECTION

In the selection of a coil it is necessary to consider several factors:

1. The duty required—heating, cooling, dehumidifying.
2. Temperature of entering air—dry-bulb only if there is no dehumidification; dry- and wet-bulb if moisture is to be removed.
3. Available heating and cooling media.
4. Space and dimensional limitations.
5. Air quantity limitations.
6. Allowable resistances in air circuit and through tubes.

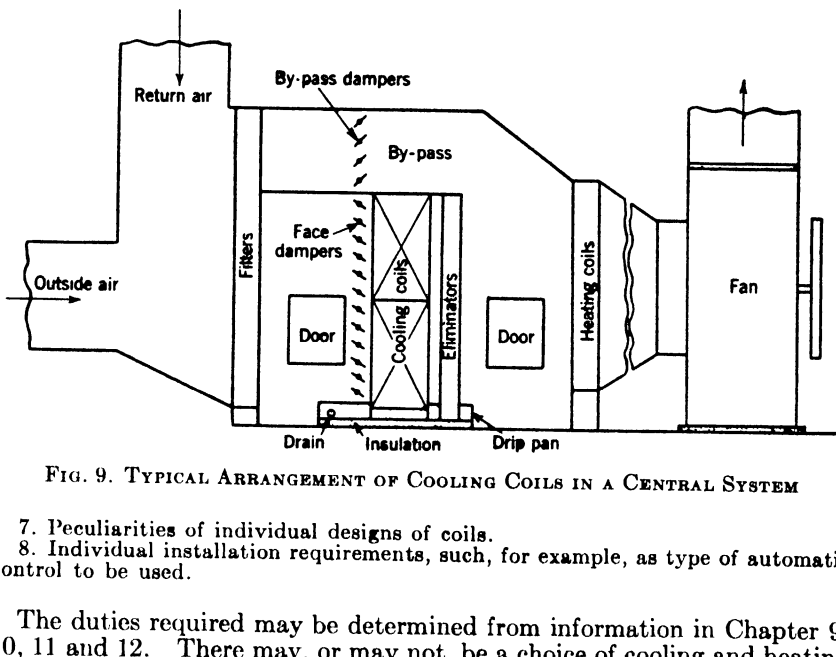

FIG. 9. TYPICAL ARRANGEMENT OF COOLING COILS IN A CENTRAL SYSTEM

7. Peculiarities of individual designs of coils.
8. Individual installation requirements, such, for example, as type of automatic control to be used.

The duties required may be determined from information in Chapter 9, 10, 11 and 12. There may, or may not, be a choice of cooling and heating media, as well as temperatures available, depending upon whether the installation is new or is in combination with present sources of heating or cooling. Space limitations are dictated by the requirements of individual cases. The air quantity is influenced by a number of considerations. The air quantity through heating coils is often made the same as that necessary to handle the summer cooling load. The air handled may be fixed by the use of old ventilating ducts as the air distribution system for new air conditioning apparatus, or may be dictated by requirements of satisfactory air distribution or ventilation. The resistance through the air circuit influences the fan horsepower and speed. This resistance may be limited to allow the use of a given size of fan motor, or to keep the operating expense low, or it may be limited by the maximum fan peripheral velocity which requirement of quietness may permit. The friction through the water or brine circuit may be dictated by the head available from a given size of pump and pump motor. As the fan and pump motor inputs represent a refrigerating load on cooling installations, it is economical to keep them low.

Proper performance of a surface heating or cooling coil depends upon correct choice of the original equipment, and upon certain other factors. The usual coil ratings are based on a uniform face velocity of air. If the air is brought in at odd angles or if the fan is located so as to block part of the air flow, the performance as given in the manufacturer's ratings cannot usually be obtained. To obtain rated performance it is necessary that the air quantity be adjusted on the job to that used in determining the coil selection, and that it be kept at this value. The most common causes of a reduction of air quantity are the fouling of the filters and collection of dirt in the coils. These difficulties can be avoided by proper design and

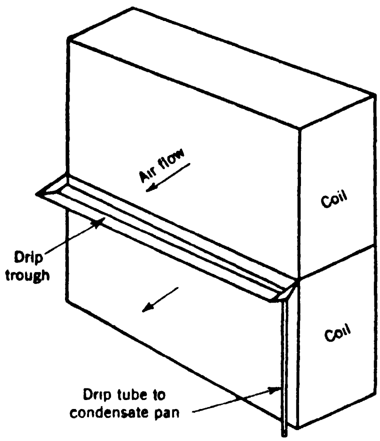

Fig. 10. Coil Arranged with Drip Trough

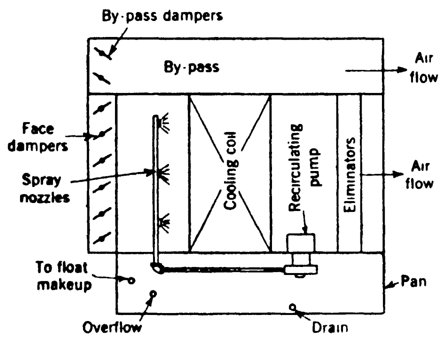

Fig. 11. Recirculating Spray System for Cleaning Coils

proper servicing. There are a number of ways in which coils may be cleaned. A common method is to wash them off with water. They can sometimes be brushed and cleaned with a vacuum cleaner. In bad cases of neglect, especially on restaurant jobs where grease and dirt have accumulated, it is sometimes necessary to remove the coils and wash off the accumulation with steam, compressed air and water, or hot water. The most satisfactory solution, however, is to keep the filters serviced, and thus make the cleaning of the coils unnecessary.

The proper selection of coils requires an understanding of the requirements of each case, and should be based on an economic analysis of the plant design as a whole. No general rule can, therefore, be laid down for the selection of heating or cooling coils. It is possible, however, to point out the limits of usual practice and to indicate the influence of the variables involved in the coil selection.

Heating Coils

Steam and *hot water heating coils* are usually *rated* within these limits:

Air Face Velocity—200 to 1200 fpm, sometimes up to 1500 fpm.
Steam Pressure—2 to 200 psig, sometimes up to 350 psig.

Hot Water Temperature—150 to 225 F.
Water Velocity—2 to 6 fps.

Individual cases may deviate widely, but the tabulation given herewith will serve as a guide to usual *heating installation practice:*

Air Face Velocity—500 to 800 fpm face, 500 being a common figure.

Delivered Air Temperature—varies from about 72 F for ventilation only, to about 150 F for complete heating.

Steam Pressure—2 to 10 psig, 5 psig being common.

Hot Water Temperature—150 to 225 F.

Water Velocity—2 to 6 fps.

Water Quantity—Based on about 20 deg temperature drop through a hot-water coil.

Air Resistance—The total resistance through heating coils is usually limited to from 3/8 to 5/8 in. of water gage for public buildings, to about 1 in. for factories.

The selection of heating coils is relatively simple as it involves dry-bulb temperatures and sensible heat only, without the complication of simultaneous latent heat loads, as in cooling coils. For a given duty, entering air temperature, and steam pressure, it is possible to select several arrangements of the same design of coil depending upon the relative importance of space, cross-sectional area, and air resistance.

Cooling Coils

Cooling and *dehumidifying coils* are usually *rated* within these limits:

Entering Air Dry-Bulb—60 to 100 F.

Entering Air Wet-Bulb—50 to 80 F.

Air Face Velocities—300 to 800 fpm (sometimes as low as 200 and as high as 1200).

Volatile Refrigerant Temperatures—25 to 55 F, at coil suction outlet.

Water Temperatures—40 to 65 F.

Water Quantities—2 to 6 gpm per ton, or equivalent to a water temperature rise of from 4 to 12 deg.

Water Velocity—2 to 6 fps.

The ratio of total to sensible heat removed varies in practice from 1.00 to about 1.65, *i.e.*, sensible heat is from 60 to 100 percent of total, depending on the application. (See Chapter 29.) Since required ratios may demand wide variations in air velocities, refrigerant temperatures, and coil depth, general rules as to their values may be misleading. On usual comfort installations air face velocities between 400 and 600 fpm are frequent, 500 being a common value. Refrigerant temperatures ordinarily vary between 40 and 50 F where cooling is accompanied by dehumidification. Water velocities range from 2 to about 6 fps.

When no dehumidification is desired, for which condition the dew-point of the entering air is equal to or lower than the cooling coil surface temperature, the coil selection is made on the basis of dry-bulb temperatures and sensible heat transfer only, the same as with heating coils. It is possible also to choose various arrangements of face area, depth, air velocity, etc., for the same duty.

Dehumidifying Coils

The selection of coils for combined cooling and dehumidifying duty is more involved than for heating or sensible cooling, and requires consideration of both dry- and wet-bulb air temperatures. It is further complicated by the fact that the proportional amount of dehumidification required is also highly variable. The methods outlined in the section, Heat Transfer and Resistance, may be used to determine whether it is possible for a coil to perform the duty required. If entering and leaving air conditions are

TABLE 1. VARIOUS COOLING COIL ARRANGEMENTS

SELECTION	1	2	3	4
Total cooling capacity, tons	100	100	100	100
Sensible cooling capacity, tons	69	69	69	69
Latent cooling capacity, tons	31	31	31	31
Ratio total to sensible heat	1.45	1.45	1.45	1.45
Air quantity, cfm	47,800	41,700	37,100	46,800
Cfm per total ton	478	417	371	468
Face velocity, fpm	325	423	500	600
Resistance, in. water	0.11	0.27	0.51	0.37
Coil face area, sq ft	147	99.0	74.2	78.1
Coil rows deep	4	6	8	4
Coil evaporator temp. F deg	45	45	45	38

arbitrarily specified, the corresponding duty sometimes cannot be obtained at all without the use of reheat. As with heating and sensible cooling coils, there are combinations of face areas, depth, air velocity and refrigerant temperatures which will give the required performance. This is illustrated in Table 1.

It is possible, as shown in Table 1, to perform approximately the same duty at a given refrigerant temperature with small face area and large thickness or vice versa. The large face area coil gives low air velocity and resistance, but high air quantities per ton. The coil of small face area and great depth requires small air quantities per ton of refrigeration, high resistance and high air velocities. As shown also in Table 1 the same sensible, latent and total cooling capacity may be obtained with various refrigerant temperatures by proper choice of coil. This makes it possible to keep the evaporating temperature high enough to carry the load with a chosen size of condensing unit. High evaporating temperatures with correspondingly small compressor operating expense can be attained, but at the expense of coil surface, air quantity or both. The choice will be determined by the necessities of individual installations.

For a given quantity and condition of entering air, the evaporating temperature of a volatile refrigerant coil is determined by a balance between the condensing unit and the coil. The total, sensible and latent cooling capacity can then be determined from the coil rating information. If the condensing unit and cooling coil have been properly balanced for the required load and, due to miscalculated duct resistance or improper choice of fan speed, the air quantity is reduced, the total cooling capacity will also be reduced. The decrease generally affects the sensible capacity. This is also true when the air by-pass or volume control is used.

It is necessary that not only the total capacity, but also that both sensible and latent cooling requirements be met. The installation of an excess of coil results in an increase in total capacity, but not in proportion to gain in latent heat capacity. On installations controlled from dry-bulb temperature, the operating time is shortened because of the added sensible cooling capacity. This results in less moisture pick-up and higher relative humidity than calculated. If an oversize condensing unit is installed, the opposite situation occurs. Generally, this is not a disadvantage, except that it results in a load from outside air greater than calculated, as well as in increased power consumption. If oversize equipment is furnished, a balance should be made to assure that the ratio of total to sensible capacity is the same as in the estimated load.

Sometimes, arbitrary air quantities are specified for ventilation or other

TABLE 2. CAPACITY BALANCES FOR MAXIMUM AND MINIMUM LOAD CONDITIONS

Conditions	Capacity in Tons			Ratio $\frac{\text{Total}}{\text{Sensible}}$
	Total	Sensible	Latent	
Required at peak load conditions	10.90	7.90	3.00	1.38
Required at minimum load conditions	6.62	3.36	3.26	1.98
Peak load equipment balance	10.90	7.90	3.00	1.38
Same equipment balanced at minimum load conditions	9.85	6.58	3.26	1.50
Same equipment balanced at maximum load conditions with 40 per cent by-pass	8.38	5.05	3.33	1.66
Same equipment balanced at minimum load conditions with 38,800 Btu per hour reheat	6.62	3.36	3.26	1.98

reasons independent of the selection of the cooling coil. As shown in Table 1, the coil selection can be altered to take care of various air quantities for the same duty.

Where coil and condensing unit are selected for the peak load condition, and the sensible load partially disappears due to fall of outside temperature or other cause, the condensing unit and coil will rebalance. This may obtain more sensible and less latent capacity than required at the light load condition, with an increased relative humidity in the conditioned space. Such a condition is shown in Table 2. If approximately 40 percent of the total air is by-passed, the condition is improved as indicated. The situation may be entirely avoided by using reheat, where it is possible to handle any ratio of sensible and latent loads and maintain the design temperature and humidity.[1]

Care should be taken to avoid freezing at light loads. In general, freezing occurs when the coil surface temperature falls to 32 F. With usual coils for comfort installations, this does not occur unless the evaporating temperature at the coil outlet is about 20 to 25 F. The exact value depends on the design of the coil and the amount of loading. Although it is not customary to choose coil and condensing units to balance at low temperatures at peak loads, there is danger of this occurring when the load decreases. This is further aggravated if a by-pass is used so that less air is passed through the coil at light loads. It may be even worse if the control is arranged for decrease of inside temperature with fall of that outside. Freezing can be avoided by making the full load balance a high evaporating temperature, and checking the balance at the minimum load.

Care should be exercised in the design of humidity control to minimize the cycling of the refrigerating compressor because of re-evaporation of moisture from the fins. It is sometimes necessary to by-pass air around a coil when the compressor is not operating.

HEAT TRANSFER AND AIR FLOW RESISTANCE

The transfer of heat between the heating or cooling medium and the air stream is influenced by several variables:

1. The temperature difference.
2. The design and surface arrangement of the coil.
3. The velocity and character of the air stream.
4. The velocity and character of the medium in the tubes.

The driving force is usually taken as the logarithmic mean temperature difference for heating or cooling without dehumidification. For combined cooling and dehumidification, the logarithmic difference does not apply strictly, and such problems should be handled as described in a later section on Performance of Dehumidifying Coils. With volatile refrigerants there is often an appreciable pressure drop and corresponding change in evaporating temperature through the refrigerant circuit. The problem is further complicated by the fact that the refrigerant is evaporating in part of the circuit, and superheating in the remainder. In spite of this, heat transfer and ratings for coils using volatile refrigerants are usually based on a refrigerant temperature corresponding to the average pressure in the coil.

The design and surface arrangement of the coil include such items as materials, type, thickness, height and spacing of the fins, and the ratio of this surface to that of the tube, the use of the staggered or in-line tube arrangement, and provisions to increase the air turbulence such as the use of corrugated as against flat fins. Staggered tubes increase the total heat transfer, as against the in-line arrangement, and corrugated fins may be more effective than flat. This design and surface arrangement has a large effect on the air film heat transfer resistance.

The velocity of the air usually considered is the coil face velocity. This bears a *varied relation to the actual velocity* over the surface, depending upon the individual coil design. As long as a fixed design of coil is under consideration face velocities may be used, but they may be unsatisfactory in comparing different designs, as it is the *actual surface velocity that is significant.* The air volume is often based on standard air at 70 F and a barometric pressure of 29.92 in. Hg. The use of air volume in coil *rating* information may be misleading. *The significant value is mass velocity in pounds per (minute) (square foot of face area) and not cubic feet per minute,* because for a fixed volume the corresponding weight may vary widely, depending upon the temperature and barometric pressure.

At the same mass air velocity, varying performance can be obtained depending upon the turbulence of the air flow into the coil, and upon the uniformity of distribution of air over the coil face. The latter is very important in obtaining reliable test ratings, and in realizing rated performance in practical installations. The resistance through the coils will assist in distributing the air properly, but where the inlet duct connections are brought in at sharp angles to the coil face, the effect is frequently bad and there may even be reverse air currents through the coils. This reduces the capacity, but can be avoided by proper layout or by the use of directing baffles.

Heat transfer depends also upon the velocity of the medium in the tubes and upon its character, whether flowing water, condensing steam or evaporating volatile refrigerant. Heat transfer rates expressed as Btu per (square foot of internal surface) (degree logarithmic mean effective temperature difference between the fluid and tube wall) are, for example: about 150 to 300 for evaporating dichlorodifluoromethane, about 350 to 1200 for water at 2 and 6 fps, and about 1200 for condensing steam. The influence of the medium in the tubes on the overall heat transfer rate is therefore apparent.

Because of these variables, reliable rating and performance information for any design of coil must be based on actual tests on that coil under the expected conditions of operation. A comparison between the performance of two designs, unless based on such tests on each, may lead to entirely erroneous conclusions. Details on coil calculation and performance follow.

PERFORMANCE OF HEATING AND DRY COOLING COILS

The performance of heating and dry cooling coils depends in general upon:

1. The overall coefficient of heat transfer from the fluid within the coil to the air it heats or cools.
2. The mean temperature difference between the fluid within the coil and the air flowing over the coil.
3. The physical dimensions of the coil.

Thus, for any one definite operating condition, the heating or cooling capacity of a given coil is expressed by the following basic formula:

$$q_t = U \times (\Delta t_m) \times A \times N \qquad (1)$$

where

q_t = total heat transferred by the coil, Btu per (hour) (square foot of coil face area).

U = overall coefficient of heat transfer, Btu per (hour) (square foot of external coil surface) (Fahrenheit degree temperature difference between the fluid within the coil and the air flowing over the coil).

Δt_m = mean temperature difference, Fahrenheit degrees, between the fluid within the coil and the air passing over it. (This is commonly taken as the logarithmic mean temperature difference).

A = external surface area of the given coil, square feet per (square foot of coil face area) (row of coil depth).

N = number of rows of coil depth.

Overall Coefficient of Heat Transfer

Of all factors affecting the performance of heating or dry cooling coils, the overall coefficient of heat transfer is the most difficult to determine, as it is influenced by several factors which depend upon coil design and conditions of operation.

Considering any coil, whether of bare pipe or of finned type, the overall heat transfer coefficient for a given size and design of coil can always be considered as a combined effect of three individual heat transfer coefficients, namely:

1. The film coefficient of heat transfer between air and the external surface of the coil, usually given in Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference).
2. The coefficient of heat transfer through the coil material—tube wall, fins, ribs, etc.
3. The film coefficient of heat transfer between the internal surface of the coil and the fluid flowing within the coil, usually given in Btu per (hour) (square foot internal surface) (Fahrenheit degree mean temperature difference).

These three individual coefficients acting in series result in an overall coefficient of heat transfer in accordance with the basic laws given in Chapters 5 and 9. For a *bare pipe* coil the overall coefficient of heat transfer, whether for heating or for cooling (without dehumidification), can be expressed by a simplified basic formula as follows:

$$U = \frac{1}{\frac{R}{f_i} + \frac{L}{k} + \frac{1}{f_o}} \qquad (2)$$

where

U = overall coefficient of heat transfer, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between air and fluid within the coil).

f_i = film coefficient of heat transfer between the internal surface of the coil and the fluid flowing within the coil, Btu per (hour) (square foot internal surface) (Fahrenheit degree mean temperature difference between that surface and the average fluid temperature).

f_o = film coefficient of heat transfer between air and the external surface of the coil, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between the mass of air and the external surface).

k = conductivity of material from which the bare pipe is constructed, Btu per (hour) (square foot) (Fahrenheit degree per inch thickness).

L = thickness of tube wall, inches.

R = ratio between external and internal surface of the bare tube, usually varying from 1.03 to 1.15 for the tube used in typical heating or cooling coils. This ratio R is inserted in the formula in order to place internal fluid coefficient of heat transfer on the basis of external surface.

Frequently, when pipe or tube walls are thin and of material having high conductivity (as is the case in construction of typical heating and cooling coils) the term L/k in Equation 2 becomes negligible and is generally disregarded. (The effect of the term L/k in typical bare pipe heating or cooling coils seldom exceeds 1 to 2 percent of the overall coefficient). Thus, in its simplest form, for bare pipe:

$$U = \frac{1}{\frac{R_i}{f} + \frac{1}{f_o}} \qquad (3)$$

For finned coils the formula[2] for the overall coefficient of heat transfer can be conveniently written:

$$U = \frac{1}{\frac{R}{f_i} + \frac{1}{\eta f_o}} \qquad (4)$$

in which the term η, called the *fin efficiency*, is introduced to allow for the resistance to heat flow encountered in the fins.

The term R, in this case, is the ratio of *total* external surface to internal surface. For typical designs of finned coils for heating or cooling, this ratio varies from 10 to 30. Term R is again introduced to place the internal surface coefficient of heat transfer on a basis of external surface. In the discussions which follow, coefficients f_i and ηf_o will be considered separately, and also various ways of combining them will be outlined.

The performances of all heating and dry cooling coils are influenced by these same factors. But, when cooling coils operate wet or act as dehumidifying coils, the performance cannot be predicted on the basis of overall coefficients, and an analysis must be made on the basis of individual film coefficients as will be explained.

PERFORMANCE OF DEHUMIDIFYING COILS

When a cooling coil operates with a surface temperature which is below the dew-point of the air entering the coil, moisture is condensed and the air leaves the coil with a humidity ratio lower than it had when it entered the coil. To understand the performance of surface coils under such conditions, assume that air enters a cooling coil at conditions corresponding to point 1 in Fig. 12. As long as the surface temperature of the coil is above the dew-point, the air is cooled without dehumidification, and its condition leaving the coil will be somewhere on line 1-A. Its exact position on this line depends on the air velocity and the external film coefficient, as well as upon the surface temperature. When the surface temperature just equals the dew-point, the air leaves with conditions represented by point A. If the surface temperature is below the dew-point, condensation takes place, and the air has a final condition somewhere along the line A-2-3 which is a

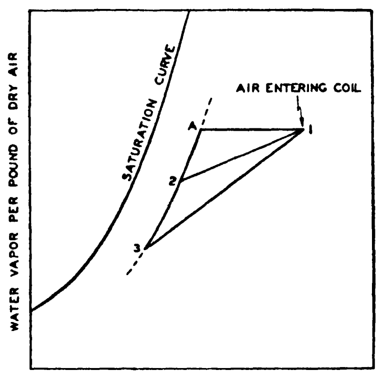

FIG. 12. PERFORMANCE OF DEHUMIDIFYING COIL

line at a constant horizontal distance from the saturation curve. It should be understood that the line 1-A-2-3 is not intended to represent the path of the condition of the air as it passes through the coil from row to row. It is simply the path traced by the exit air conditions as the surface temperature is gradually reduced, with other conditions remaining constant.[3]

In the process of dehumidification, since heat is being transferred to the coil surface by two different mechanisms (convection and condensation), it is evident that an overall coefficient of heat transfer cannot be determined by the same method used for heating and for dry cooling coils. However, if it is assumed that the sensible heat transfer of a dehumidifying coil is unaffected by the presence of moisture on its surface, Equation 5 may be obtained to express this part of the heat transfer in terms of the external film coefficient and the surface temperature.

$$q_s = f_o \times A \times N \times (\Delta t_o) \quad (5)$$

where

q_s = sensible heat transferred, Btu per (hour) (square foot of coil face area).
t_1 = dry-bulb temperature of air entering coil, Fahrenheit degrees.
t_2 = dry-bulb temperature of air leaving coil, Fahrenheit degrees.
t_s = average temperature of coil external surface, Fahrenheit degrees.

Δt_o = logarithmic mean temperature difference between air and coil surface =

$$\frac{t_1 - t_2}{\log_e \dfrac{t_1 - t_s}{t_2 - t_s}}$$

If Equation 5 is combined with another equation expressing sensible heat transfer in terms of mass velocity and temperature difference, the variables may be arranged in the following form (which is useful for the solution of dehumidification problems and for the determination of f_o from test data):

$$\frac{f_o AN(t_1 - t_2)}{\log_e \dfrac{t_1 - t_s}{t_2 - t_s}} = 0.243G(t_1 - t_2)$$

or,

$$\frac{f_o AN}{0.243G} = \log_e \frac{t_1 - t_s}{t_2 - t_s} \tag{6}$$

where

0.243 = specific heat of humid air, Btu per (pound) (Fahrenheit degree).
G = air mass velocity, pounds per (hour) (square foot of coil face area).

An examination of Fig. 12 will reveal that when t_s is at the dew-point of the entering air:

$$\frac{t_1 - t_s}{t_2 - t_s} = \frac{t_1 - t_{dp1}}{t_a - t_{dp1}}$$

and when t_s is below the dew-point:

$$\frac{t_1 - t_s}{t_2 - t_s} = \frac{t_1 - t_{dp1}}{t_2 - t_{dp2}}$$

Therefore, Equation 6 may be written in its most useful form as

$$\frac{h_a AN}{0.243G} = \log_e \frac{t_1 - t_{dp1}}{t_2 - t_{dp2}} = \log_e \frac{t_1 - t_s}{t_2 - t_s} \tag{7}$$

where

t_s = minimum dry-bulb possible without dehumidification, Fahrenheit degrees.
t_{dp1} = dew-point of air entering coil, Fahrenheit degrees.
t_{dp2} = dew-point of air leaving coil, Fahrenheit degrees.

This equation may be used to establish a line, as A-2-3, for a given coil if f_o is known for the coil, or it *may be used to determine f_o from test data for the purpose of rating coils.* The use of this equation for coil selection is illustrated in Example 1 at the end of the chapter. Equation 7 is also important as a means of determining the external film coefficient.

External Film Coefficient

While formulas have been developed expressing the film coefficient f_o for air passing parallel to a plane surface, they cannot be used directly for fins on tubes because of air turbulence, and because of the temperature gradient prevalent from the edge of a fin to its center. It is therefore necessary to make tests to evaluate the combined term ηf_o. The term, ηf_o, will be written merely f_o in this discussion, as there is no necessity for separately evaluating η, and because values of f_o are usually applied only to the particular coils for which tests are made.

The air side coefficient, f_o, of a coil of particular dimensions is an exponential function of the mass velocity of the air:

$$f_o = Z\,G^n \tag{8}$$

where

f_o = film coefficient of heat transfer, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between air and average surface temperature).

G = air mass velocity, pounds per (hour) (square foot of coil face area).

Z and n = constants which depend upon both air turbulence and surface arrangement.

Evaluation of constants Z and n may be accomplished through the use of test data in Equation 7 which gives values of h_s directly from the results of any *wet coil test*. If f_o, calculated in this manner, is plotted against values of G which prevailed during the tests, a straight line should result on logarithmic coordinates. The slope of this line is the value of n. The value of Z may then be determined by direct substitution in Equation 8.

For finned coils of different designs, values of Z and n are extremely variable, depending on the particular design and arrangement of the coil surface. Therefore, it is desirable that these constants be determined directly from test data for each type of coil surface.

Internal Film Coefficient

The internal film coefficient, f_i which appears in Equation 3, is evaluated in various ways, depending upon the nature of the fluid, and whether the fluid is changing state.

When evaporating refrigerants are used in tubes, the temperature of the fluid is fairly constant, being affected principally by pressure drop through the tubes, by superheat of the evaporated refrigerant, and by the presence of oil in solution. To obtain maximum coil capacity it is necessary to keep the pressure drop through the tubes at a minimum, to keep the superheat as low as possible without carrying liquid back to the compressor, and to arrange for good separation and return of oil to the compressor. Another important factor is the removal of gas to keep the tube surface flooded with liquids as much as possible. The internal film coefficient is markedly increased by heavy heat loads, because the increased turbulence and gas velocity cause good contact of the liquid with the tubes. Values of f_i usually lie between 150 and 450. For rating of dehumidifying coils, satisfactory results are obtainable by first determining the average external surface temperature from Equation 7, and then using the difference between the external film temperature and the refrigerant for evaluating f_i in Equation 9.

$$f_i = \frac{q_t}{\frac{AN}{R}\,(t_s - t_r)} \tag{9}$$

where

f_i = internal film coefficient of heat transfer, Btu per (hour) (square foot of internal tube surface) (Fahrenheit degree).

t_r = average refrigerant temperature, Fahrenheit degrees.

To evaluate f_i by this method the same tests that were required to determine f_o may be used.

When water is the cooling medium in tubes, the rate of heat transfer is a function of its velocity, which influences the number of contacts of the water molecules with the tube surface, per unit of time. Increased water velocity and reduced tube diameter cause increased heat transfer. Heat transfer is also greater at higher temperatures of the water. The basic formula for the film coefficient of heat transfer for flow of water in smooth tubes is as follows:

$$f_i = 1.5(t + 100) \frac{V^{0.8}}{D^{0.2}} \tag{10}$$

where

V = water velocity, feet per second.

D = internal diameter of tube, inches.

t = average water temperature, Fahrenheit degrees.

Equation 10 should not be used when Reynolds Number is less than 2000.

Since, in the case of finned tubes using water as a refrigerant, test values of f_i based on the calculated surface temperature for the entire coil may be lower than those obtained by use of Equation 10, actual test results are preferred if available.

When saturated steam is condensed in the tubes of coils, the film coefficient f_i varies from 1000 to 2000, depending on freedom from air in the steam, and upon good drainage of the tubes. The coefficient is fairly constant for a particular coil, giving values of $(t_s - t_r)$ that are directly proportional to q_t. However, if water coil test results are analyzed on a row-by-row basis good agreement with Equation 10 will result.[4]

The use of turbulence promoters increases the value of f_i for liquids in tubes at the expense of pressure drop. The increase obtained depends upon the type of turbulence promoter and the rate of flow. No general statement can be made regarding their use, and it is best to refer to detailed papers on this subject for further information.[4, 5]

Determining Size of Cooling Coil

To illustrate the use of individual film coefficients in coil calculations, the procedure for selecting the proper size cooling coil and for determining exit air condition, coil surface temperature, total coil load and refrigerant temperature, is outlined in Example 1.

Example 1: An industrial application requires the cooling of a certain quantity of air from a condition of 102 F dry-bulb and 85 F wet-bulb to a final condition of 80.5 F dry-bulb and 73 F wet-bulb. The air velocity across the coil is to be 400 fpm and coil data are as follows: f_o = 10.7 at 400 fpm, f_i = 325, external surface area = 15 sq ft per (square foot of face) (row of coil depth), ratio of external surface area to internal surface area = 15.

Solution: (1) Lay out the problem psychrometry as indicated in Fig. 13 and note that the minimum horizontal distance between the load ratio line and the saturation curve is 1.8 F dry-bulb at point A Fig. 13. This means that $t_2 - t_{dp2}$ in Equation 7 must not be less than 1.8. Therefore, Equation 7 should be solved for N to determine the proper number of rows to be used for the coil.

$$\frac{f_o AN}{0.243G} = \log_e \frac{t_1 - t_{dp1}}{t_2 - t_{dp2}} = \log_e \frac{102 - 80}{1.8} = \log_e 12.22 = 2.5$$

Then substituting values for f_o, A, and G, N may be found as follows:

$$\frac{10.7 \times 15N}{0.243 \times 1740} = 2.5 \text{ from which, } N = 6.58$$

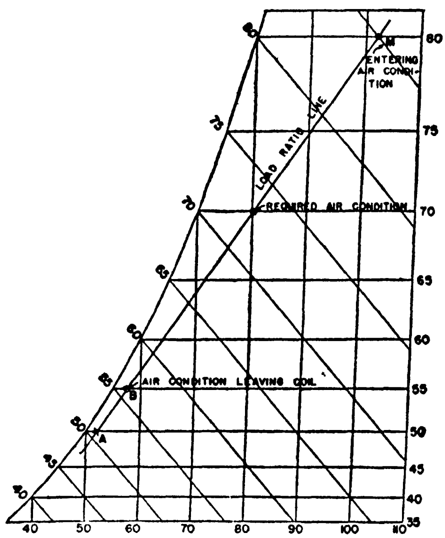

FIG. 13. PSYCHROMETRIC LAYOUT FOR COIL SELECTION

(2) This establishes the maximum whole number of coil rows that can be used as 6, and it is now possible to determine the actual location of the exit air conditions from Equation 7 by solving for the actual value of $t_2 - t_{dp2}$ for a 6 row coil.

$$\frac{10.7 \times 15 \times 6}{0.243 \times 1740} = \log_e \frac{102 - 80}{t_2 - t_{dp2}} = 2.275$$

This establishes values of 9.78 for $\frac{t_1 - t_{dp1}}{t_2 - t_{dp2}} = \text{B}$ and 2.25 for $t_2 - t_{dp2}$.

(3) Next, the exit air condition at 57.3 F dry-bulb and 56 F wet-bulb as shown at B, is found by locating a point on the load ratio line at a horizontal distance of 2.25 dry-bulb degrees from the saturation curve.

(4) The surface temperature may now be found from Equation 7 which may also be written as

$$t_s = \frac{Bt_2 - t_1}{B - 1}$$

where

$$B = \frac{t_1 - t_{dp1}}{t_2 - t_{dp2}}$$

$$t_s = \frac{9.78 \times 57.3 - 102}{8.78} = 52.3$$

(5) The total coil load may be calculated from the enthalpy difference across the coil and the air quantity using the weight of dry air instead of the weight of the mixture.

$$q_t = G_a\ (h_1 - h_2) = 1700\ (49.24 - 23.77)$$

$$= 43{,}200 \text{ Btu per (hr) (sq ft of face area)}$$

where

G_a = weight of dry air per (hour) (square foot of coil face area).

h_1 = enthalpy of air vapor mixture entering coil, Btu per pound of dry air.

h_2 = enthalpy of air vapor mixture leaving coil, Btu per pound of dry air.

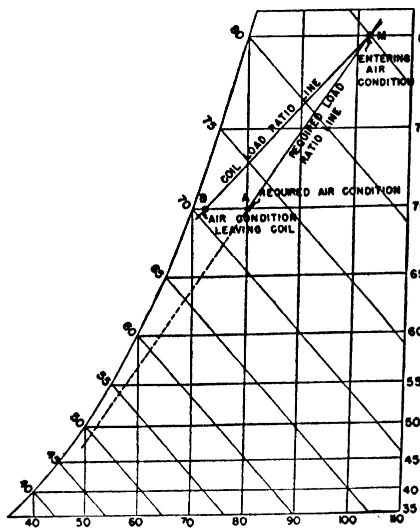

FIG. 14. PSYCHROMETRIC LAYOUT FOR COIL SELECTION USING REHEAT

(6) The refrigerant temperature may be found from Equation 9

$$\frac{43{,}200}{15 \times 6 \times \frac{325}{15}} = (t_s - t_r) = 22.1$$

Therefore, $t_r = (52.3 - 22.1) = 30.2$.

Thus a coil 6 rows deep, operating at a refrigerant temperature of 30.2 F and a face velocity of 400 fpm, is required; and it will carry a total load of 43,200 Btu per (hour) (square foot of face area). The air conditions leaving the coil are too low for the conditions of the problem and therefore it is necessary to by-pass air at the entering condition to obtain the desired result of 80.5 F dry-bulb and 73 F wet-bulb.

Although the preceding solution is satisfactory, it may be more desirable in some cases to use a higher refrigerant temperature and employ reheat to obtain the desired load ratio. Such a solution is shown in Fig. 14. In this case, the coil load ratio line intersects the saturation curve and, therefore, a coil of any depth may be selected.

If a coil depth of 6 rows is maintained, the exit air conditions for the coil are indicated at point B Fig. 14 as 72.3 F dry-bulb and 70.8 F wet-bulb, and the surface temperature will be

$$t_s = \frac{9.78 \times 72.3 - 102}{8.78} = 69.0$$

The coil load will be: $q_t = 1700\,(49.24 - 34.66) = 24{,}800$ Btu per (hour) (square foot of face area) and the refrigerant temperature will be found from Equation 9:

$$\frac{24{,}800}{15 \times 6 \times \frac{325}{15}} = (t_s - t_r) = 12.7$$

Therefore, $t_r = 69.0 - 12.7 = 56.3$.

Thus, for the case where reheat is used, a coil 6 rows deep operating at a refrigerant temperature of 56.3 F is required. The total coil load will be 24,800 Btu per (hour) (square foot of face area) but the actual effective load will be less by the amount of reheat required. Therefore, for a given load, a larger coil and more refrigerating capacity are required when reheat is used.

LETTER SYMBOLS USED IN CHAPTER 35

η = fin efficiency.

A = external area of coil, square feet per (square foot of coil face area) (row of coil depth).

$B = \dfrac{t_1 - t_{dp1}}{t_2 - t_{dp2}}$.

D = internal diameter of tube, inches.

G = air mass velocity, pounds per (hour) (square foot of coil face area).

G_a = dry air mass velocity, pounds dry air per (hour) (square foot of coil face area).

f_i = film coefficient of heat transfer between fluid and internal coil surface, Btu per (hour) (square foot internal surface) (Fahrenheit degree mean temperature between fluid and surface).

f_o = film coefficient of heat transfer between air and external coil surface, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between air and coil).

h_1 = enthalpy of air-vapor mixture entering coil, Btu per pound of dry air.

h_2 = enthalpy of air-vapor mixture leaving coil, Btu per pound of dry air.

k = conductivity of pipe or tube material, Btu (square foot) (hour) (Fahrenheit degree per inch thickness).

L = thickness of tube wall, inches.

N = number of rows of coil depth.

n = a constant, exponent of G in Equation 8, obtained by plotting, on logarithmic coordinates, G against values of f_o. The value of n is the slope of the line.

q_s = sensible heat transferred, Btu per (hour) (square foot of coil face area).

q_t = total heat transferred by coil, Btu per (hour) (square foot of face area).

R = ratio between external and internal surface of tube.

t = average water temperature, Fahrenheit degrees.

t_1 = dry-bulb temperature of air entering coil, Fahrenheit degrees.

t_2 = dry-bulb temperature of air leaving coil, Fahrenheit degrees.

t_a = minimum dry-bulb temperature possible without dehumidification, Fahrenheit degrees.

t_{dp1} = dew-point of air entering coil, Fahrenheit degrees.

t_{dp2} = dew-point of air leaving coil, Fahrenheit degrees.
t_r = average refrigerant temperature, Fahrenheit degrees.
t_s = average temperature of external surface of coil, Fahrenheit degrees.
Δt_m = mean temperature difference between fluid in coil and air passing over coil, Fahrenheit degrees.
NOTE: Δt_m is usually the logarithmic mean.
Δt_o = logarithmic mean temperature difference between air and coil surface.
U = overall coefficient of heat transfer, Btu per (hour) (square foot of external coil surface) (Fahrenheit degrees temperature difference between fluid in coil and air flowing over coil).
V = water velocity, feet per second.
Z = a constant for use in Equation 8 obtained by plotting on logarithmic coordinates G against values of f_o.
NOTE: Numerical subscripts refer to condition entering and leaving, respectively.

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CHAPTER 36

REFRIGERATION

Refrigeration Theory: Definitions and Basic Concepts, Refrigerants, Vapor Compression Refrigeration Cycles, Clearance and Volumetric Efficiency, Complex Refrigeration Cycles, Air Cycle, Steam Jet, Absorption and Ice Systems, Heat Pump; Basic Refrigeration Equipment: Compression Machines, Condensers, Evaporators and Coolers; Refrigeration Controls, Piping and Accessories; Equipment Characteristics and Selection

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WITH the increasing use of all-year comfort air conditioning installations, the importance of refrigeration to the air conditioning engineer has been greatly magnified. The details of equipment operation, maintenance and design remain problems for the refrigeration engineer, but the air conditioning engineer does retain a responsibility to the customer which requires on his part some knowledge of the different refrigeration cycles and the relative merits of each. In order to assist in meeting this need, the present chapter has been divided into four parts, the first covering the fundamental technical relationships which govern the selection and analysis of an operating cycle, the next two presenting brief discussions of basic refrigerating equipment and auxiliaries, and the last, information on selection criteria.

REFRIGERATION THEORY

Definitions and Basic Concepts

The *ton of refrigeration* is a quantity unit which originated in the days when harvested ice was the principal source of summer cooling. By definition the *ton* is the cooling effect realized when one ton of 32 F ice melts to water at 32 F; since the latent heat of fusion of ice is 144 Btu per pound, the *ton* represents a unit cooling effect of 144 × 2,000 = 288,000 Btu. In common practice the ton is usually considered a rate (rather than quantity) unit, and is taken as 288,000 Btu per day (24 hours), or 12,000 Btu per hour, or 200 Btu per minute. Thus for air conditioning calculations, the size of the requisite refrigeration machine, expressed in tons, can be obtained by dividing the heat gain of the structure, expressed in Btu per hour, by 12,000. In equation form:

$$H_t = (\text{Btu per hour heat gain}) \div 12{,}000 \tag{1}$$

where

H_t = load in tons.

The *working substance*, or *refrigerant*, is the fluid which carries heat through the refrigeration cycle from the evaporator, where heat enters the refrigerant, to the condenser where the heat is discharged to some cooling medium. The great majority of modern refrigeration systems use a liquefiable vapor as the working substance. By altering the pressure of the refrigerant its boiling temperature is changed, allowing the material to *boil* in the evaporator at a temperature sufficiently lower than that of the conditioned space, to insure maintenance of an effective heat transfer rate from the space (or in some cases from a secondary cooling fluid such as brine or cold water) to the refrigerant. The vapor formed in the evaporator is then raised in pressure (by a compressor, or by the absorber-generator combination of the absorption system) until its new boiling temperature exceeds the temperature of the available cooling medium. Under these conditions, heat transfer is established from the refrigerant vapor to the cooling medium with resultant condensation of the refrigerant. When condensed, the high-pressure liquid refrigerant is reduced in pressure and again allowed to *boil* in the evaporator.

In order to permit evaluation of the effectiveness with which any given cycle operates, some term is desirable which would be comparable to the *efficiency* that is used for heat engines. In refrigeration the desired effect is heat extraction, and the cost of achieving this extraction is the amount of energy which must be supplied as shaft work. Thus the ratio of refrigerating effect to the heat equivalent of the compressor work is used as a measure of effectiveness, and is defined as the coefficient of performance.

If the desired effect is the rejection of heat through the condenser instead of heat extraction through the evaporator, the refrigeration system is then termed a heat pump. In this case the coefficient of performance is the ratio of the heat rejected from the condenser to the heat equivalent of the compressor work. The coefficient of performance for the heat pump is greater than that for a system operating as a refrigerating machine, because all mechanical shaft work required to operate the compressor is dissipated as useful heat through the condenser.

The Carnot cycle, an ideal, thermodynamically reversible cycle consisting of an adiabatic expansion and an isothermal expansion, followed by an adiabatic compression and an isothermal compression to form a closed cycle, may be shown to be a measure of the maximum possible conversion of heat energy into mechanical energy. In its reversed form it is a measure of the maximum performance possible for any refrigeration cycle operating either as a refrigerator or as a heat pump. Although it cannot be applied in an actual machine because of the impossibility of obtaining complete reversibility, it is, nevertheless, extremely valuable as a criterion of inherent limitations. The coefficient of performance (CP) of a reversed Carnot cycle system operating as a refrigeration system is

$$(\mathrm{CP}) = \frac{T_s}{T_c - T_s} \tag{2}$$

where

T_s = evaporator temperature, Fahrenheit degrees, absolute.
T_c = condenser temperature, Fahrenheit degrees, absolute.

With the ideal Carnot cycle operating as a heat pump, the coefficient of performance is

$$(\mathrm{CP}) = \frac{T_c}{T_c - T_s} \tag{3}$$

The Carnot cycle coefficient of performance for both a refrigerating machine and a heat pump increases as the spread between the evaporator and the condenser temperatures decreases. In general, the same is true for an actual system operating as either a refrigerating machine or a heat pump.

Refrigerants

A desirable refrigerant should possess chemical, physical, and thermodynamic properties which permit its efficient application in refrigerating systems. In addition, when the volume of the charge is large, there should be little or no danger to health or to property in case of its escape.

Thermodynamically, a material for use as a refrigerant should have a large latent heat of vaporization since it is this heat quantity—subject to minor variations—which constitutes the working effectiveness of the refrigerant. Further, since the work required to compress a vapor increases rapidly with the pressure ratio, the thermodynamic characteristics of the fluid should be such that the required low-to-high temperature range can be achieved with only a moderate change in ratio. A further consideration, from the standpoint of practical operating effectiveness, is that the suction pressure should not be below atmospheric (to prevent leakage of air into the refrigerant lines) nor should the condenser pressure be excessively high (to prevent need for extra-heavy construction). The specific volume-specific enthalpy relationship is also important because some materials would have such low density, when in vapor form, that impractical compressor displacements would be needed to handle the suction vapor.

Properties of refrigerants are usually given either in tabular or graphic form. In contrast to the temperature-entropy plotting which is used almost exclusively in steam-power work, refrigeration problems are usually referred to a pressure-enthalpy chart. The advantage of pressure-enthalpy plotting is that linear distances on the chart correspond to energy gains or losses, and the two types of processes, constant-pressure and constant-enthalpy, which occur most frequently in refrigeration cycles, can both be represented by straight vertical or horizontal lines. Fig. 1 presents a pressure-enthalpy chart for dichlorodifluoromethane (Freon-12). Although tabular arrangements of refrigerant properties require interpolation between values, they have the advantage of an accuracy greater than that obtainable from a chart. Tables 1, 2, and 3 give the thermodynamic properties of three of the more common refrigerants used in air conditioning installations: dichlorodifluoromethane (Freon-12), monochlorodifluoromethane (Freon-22) and monofluorotrichloromethane (Freon-11); the first two of these are commonly used in reciprocating compressors; the last refrigerant is used in centrifugal machines.

Referring to Table 1, the first column gives the range of saturation temperatures likely to occur in practice. The second column gives the saturation pressure expressed in pounds per square inch absolute corresponding to a given temperature, while the next six columns give the three fundamental specific properties, volume, enthalpy, and entropy, of the saturated liquid and saturated vapor, respectively. The last four columns give values of enthalpy and entropy for gases with 25 deg and with 50 deg of superheat; note particularly that the column heading *50 F superheat* means, not that the gas is at a temperature of 50 F, but that its temperature exceeds by 50 deg the saturation temperature corresponding to its actual pressure. Thus, F-12 vapor at 38.0 psig and 91 F possesses 50 deg of superheat, since its saturation temperature corresponding to 52.7 psia is 41 F.

The tabular arrangements of refrigerant properties are literally for saturated or superheated materials only. In many cases, however, the engineer must work with sub-cooled liquids. With an accuracy sufficient for all practical purposes, the specific volume and the enthalpy of any sub-cooled refrigerant can be taken as equal to the values read from the tables for a saturated liquid at the same temperature. Thus, if F-12 at 121 psia and

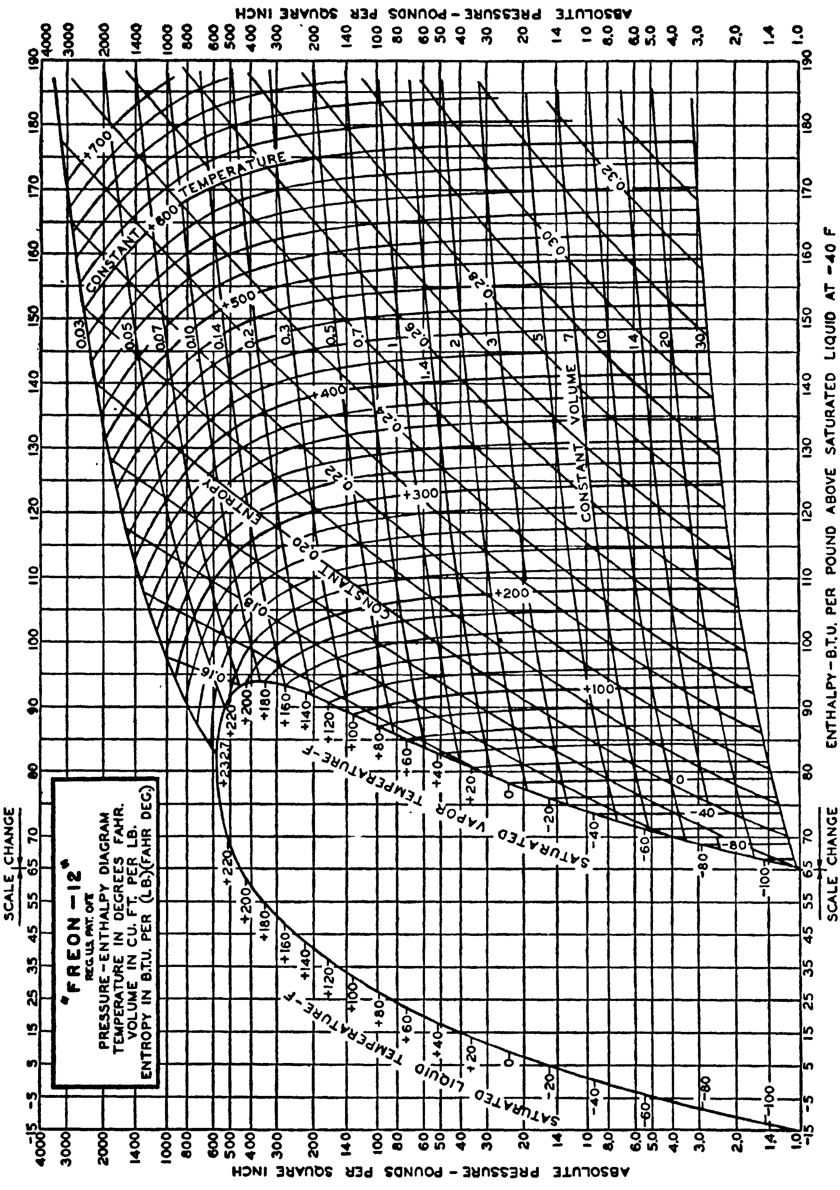

FIG. 1. PRESSURE-ENTHALPY DIAGRAM FOR FREON-12

40 F is passing through a pipe, its volume and enthalpy can be determined from Table 1 as 0.0116 cu ft per pound and 17.0 Btu per pound.

Frequently it is necessary to determine the properties of a *wet* vapor or of a mixture of liquid with some added vapor, such as is found at discharge from an expansion valve. This can be done from the tables by noting that the specific enthalpy of the mixture must be equal to that of the saturated liquid, plus a fraction of the latent heat of vaporization equal to the fraction

TABLE 1. PROPERTIES OF DICHLORODIFLUOROMETHANE (F-12)

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Enthalpy and Entropy Taken From −40 F							
				Enthalpy		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Enthalpy	Entropy	Enthalpy	Entropy
0	23.87	0.0110	1.637	8.25	78.21	0.01869	0.17091	81.71	0.17829	85.26	0.18547
2	24.89	0.0110	1.574	8.67	78.44	0.01961	0.17075	81.94	0.17812	85.51	0.18529
4	25.96	0.0111	1.514	9.10	78.67	0.02052	0.17060	82.17	0.17795	85.76	0.18511
5	26.51	0.0111	1.485	9.32	78.79	0.02097	0.17052	82.29	0.17786	85.89	0.18502
6	27.05	0.0111	1.457	9.53	78.90	0.02143	0.17045	82.41	0.17778	86.01	0.18494
8	28.18	0.0111	1.403	9.96	79.13	0.02235	0.17030	82.66	0.17763	86.26	0.18477
10	29.35	0.0112	1.351	10.39	79.36	0.02328	0.17015	82.90	0.17747	86.51	0.18460
12	30.56	0.0112	1.301	10.82	79.59	0.02419	0.17001	83.14	0.17733	86.76	0.18444
14	31.80	0.0112	1.253	11.26	79.82	0.02510	0.16987	83.38	0.17720	87.01	0.18429
16	33.08	0.0112	1.207	11.70	80.05	0.02601	0.16974	83.61	0.17706	87.26	0.18413
18	34.40	0.0113	1.163	12.12	80.27	0.02692	0.16961	83.85	0.17693	87.51	0.18397
20	35.75	0.0113	1.121	12.55	80.49	0.02783	0.16949	84.09	0.17679	87.76	0.18382
22	37.15	0.0113	1.081	13.00	80.72	0.02873	0.16938	84.32	0.17666	88.00	0.18369
24	38.58	0.0113	1.043	13.44	80.95	0.02963	0.16926	84.55	0.17652	88.24	0.18355
26	40.07	0.0114	1.007	13.88	81.17	0.03053	0.16913	84.79	0.17639	88.49	0.18342
28	41.59	0.0114	0.973	14.32	81.39	0.03143	0.16900	85.02	0.17625	88.73	0.18328
30	43.16	0.0115	0.939	14.76	81.61	0.03233	0.16887	85.25	0.17612	88.97	0.18315
32	44.77	0.0115	0.908	15.21	81.83	0.03323	0.16876	85.48	0.17600	89.21	0.18303
34	46.42	0.0115	0.877	15.65	82.05	0.03413	0.16865	85.71	0.17589	89.45	0.18291
36	48.13	0.0116	0.848	16.10	82.27	0.03502	0.16854	85.95	0.17577	89.68	0.18280
38	49.88	0.0116	0.819	16.55	82.49	0.03591	0.16843	86.18	0.17566	89.92	0.18268
39	50.78	0.0116	0.806	16.77	82.60	0.03635	0.16838	86.29	0.17560	90.04	0.18262
40	51.68	0.0116	0.792	17.00	82.71	0.03680	0.16833	86.41	0.17554	90.16	0.18256
41	52.70	0.0116	0.779	17.23	82.82	0.03725	0.16828	86.52	0.17549	90.28	0.18251
42	53.51	0.0116	0.767	17.46	82.93	0.03770	0.16823	86.64	0.17544	90.40	0.18245
44	55.40	0.0117	0.742	17.91	83.15	0.03859	0.16813	86.86	0.17534	90.65	0.18235
46	57.35	0.0117	0.718	18.36	83.36	0.03948	0.16803	87.09	0.17525	90.89	0.18224
48	59.35	0.0117	0.695	18.82	83.57	0.04037	0.16794	87.31	0.17515	91.14	0.18214
50	61.39	0.0118	0.673	19.27	83.78	0.04126	0.16785	87.54	0.17505	91.38	0.18203
52	63.49	0.0118	0.652	19.72	83.99	0.04215	0.16776	87.76	0.17496	91.61	0.18193
54	65.63	0.0118	0.632	20.18	84.20	0.04304	0.16767	87.98	0.17486	91.83	0.18184
56	67.84	0.0119	0.612	20.64	84.41	0.04392	0.16758	88.20	0.17477	92.06	0.18174
58	70.10	0.0119	0.593	21.11	84.62	0.04480	0.16749	88.42	0.17467	92.28	0.18165
60	72.41	0.0119	0.575	21.57	84.82	0.04568	0.16741	88.64	0.17458	92.51	0.18155
62	74.77	0.0120	0.557	22.03	85.02	0.04657	0.16733	88.86	0.17450	92.74	0.18147
64	77.20	0.0120	0.540	22.49	85.22	0.04745	0.16725	89.07	0.17442	92.97	0.18139
66	79.67	0.0120	0.524	22.95	85.42	0.04833	0.16717	89.29	0.17433	93.20	0.18130
68	82.24	0.0121	0.508	23.42	85.62	0.04921	0.16709	89.50	0.17425	93.43	0.18122
70	84.82	0.0121	0.493	23.90	85.82	0.05009	0.16701	89.72	0.17417	93.66	0.18114
72	87.50	0.0121	0.479	24.37	86.02	0.05097	0.16693	89.93	0.17409	93.99	0.18106
74	90.20	0.0122	0.464	24.84	86.22	0.05185	0.16685	90.14	0.17402	94.12	0.18098
76	93.00	0.0122	0.451	25.32	86.42	0.05272	0.16677	90.36	0.17394	94.34	0.18091
78	95.85	0.0123	0.438	25.80	86.61	0.05359	0.16669	90.57	0.17387	94.57	0.18083
80	98.76	0.0123	0.425	26.28	86.80	0.05446	0.16662	90.78	0.17379	94.80	0.18075
82	101.70	0.0123	0.413	26.76	86.99	0.05534	0.16655	90.98	0.17372	95.01	0.18068
84	104.8	0.0124	0.401	27.24	87.18	0.05621	0.16648	91.18	0.17365	95.22	0.18061
86	107.9	0.0124	0.389	27.72	87.37	0.05708	0.16640	91.37	0.17358	95.44	0.18054
88	111.1	0.0124	0.378	28.21	87.56	0.05795	0.16632	91.57	0.17351	95.65	0.18047
90	114.3	0.0125	0.368	28.70	87.74	0.05882	0.16624	91.77	0.17344	95.86	0.18040
92	117.7	0.0125	0.357	29.19	87.92	0.05969	0.16616	91.97	0.17337	96.07	0.18033
94	121.0	0.0126	0.347	29.68	88.10	0.06056	0.16608	92.16	0.17330	96.28	0.18026
96	124.5	0.0126	0.338	30.18	88.28	0.06143	0.16600	92.36	0.17322	96.50	0.18018
98	128.0	0.0126	0.328	30.67	88.45	0.06230	0.16592	92.55	0.17315	96.71	0.18011
100	131.6	0.0127	0.319	31.16	88.62	0.06316	0.16584	92.75	0.17308	96.92	0.18004
102	135.3	0.0127	0.310	31.65	88.79	0.06403	0.16576	92.93	0.17301	97.12	0.17998
104	139.0	0.0128	0.302	32.15	88.95	0.06490	0.16568	93.11	0.17294	97.32	0.17993
106	142.8	0.0128	0.293	32.65	89.11	0.06577	0.16560	93.30	0.17288	97.53	0.17987
108	146.8	0.0129	0.285	33.15	89.27	0.06663	0.16551	93.48	0.17281	97.73	0.17982
110	150.7	0.0129	0.277	33.65	89.43	0.06749	0.16542	93.66	0.17274	97.93	0.17976
112	154.8	0.0130	0.269	34.15	89.58	0.06836	0.16533	93.82	0.17266	98.11	0.17969
114	158.9	0.0130	0.262	34.65	89.73	0.06922	0.16524	93.98	0.17258	98.29	0.17961
116	163.1	0.0131	0.254	35.15	89.87	0.07008	0.16515	94.15	0.17249	98.48	0.17954
118	167.4	0.0131	0.247	35.65	90.01	0.07094	0.16505	94.31	0.17241	98.66	0.17946
120	171.8	0.0132	0.240	36.16	90.15	0.07180	0.16495	94.47	0.17233	98.84	0.17939
122	176.2	0.0132	0.233	36.66	90.28	0.07266	0.16484	94.63	0.17224	99.01	0.17931
124	180.8	0.0133	0.227	37.16	90.40	0.07352	0.16473	94.78	0.17215	99.18	0.17922
126	185.4	0.0133	0.220	37.67	90.52	0.07437	0.16462	94.94	0.17206	99.35	0.17914
128	190.1	0.0134	0.214	38.18	90.64	0.07522	0.16450	95.09	0.17196	99.53	0.17906
130	194.9	0.0134	0.208	38.69	90.76	0.07607	0.16438	95.25	0.17186	99.70	0.17897
132	199.8	0.0135	0.202	39.19	90.86	0.07691	0.16425	95.41	0.17176	99.87	0.17889
134	204.8	0.0135	0.196	39.70	90.96	0.07775	0.16411	95.56	0.17166	100.04	0.17881
136	209.9	0.0136	0.191	40.21	91.06	0.07858	0.16396	95.72	0.17156	100.22	0.17873
138	215.0	0.0137	0.185	40.72	91.15	0.07941	0.16380	95.87	0.17145	100.39	0.17864
140	220.2	0.0138	0.180	41.24	91.24	0.08024	0.16363	96.03	0.17134	100.56	0.17856

of refrigerant which is present in vapor form. Consider, for example, F-12 with a quality (the percent in vapor form) of 30 percent; the enthalpy of this material would be equal to

$$h_m = h_f + 0.30\,(h_v - h_f) \tag{4}$$

where

h_m = specific enthalpy of the mixture.
h_f = specific enthalpy of the liquid.
h_v = specific enthalpy of the saturated vapor.

Values of h_f and h_v are obtained from Table 1 for the actual pressure of the mixture.

By a reversal of this same procedure the tabular data can be used to determine the state of a mixture leaving an expansion valve. Consider a valve to which saturated liquid at pressure p_s is admitted, and a mixture of saturated liquid and vapor at pressure p_d is discharged. The quality of the material at discharge is then determined by making use of the fact that the expansion process is completely irreversible, is a throttling process, and hence, occurs without change in enthalpy. Thus, the enthalpy of the mixture, h_m, is equal to the enthalpy of the saturated liquid at the entrance state, h_{fs}, and can therefore be read from the table.
Thus,

$$h_{fs} = h_m = h_{vd} - (1 - x)\,(h_{vd} - h_{fd}) \tag{5}$$

or,

$$x = (h_m - h_{fd}) \div (h_{vd} - h_{fd}) \tag{6}$$

where

h_{fs} = enthalpy of saturated liquid at entrance to expansion valve.
h_m = enthalpy of mixture.
h_{vd} = enthalpy of saturated vapor at discharge.
h_{fd} = enthalpy of liquid at discharge.
x = proportion of liquid in the mixture, decimal.

Vapor Compression Refrigeration Cycle

Simple Cycle. The refrigerant *cycle* is the series of state changes (which occur in the conditioning processes) needed to restore the refrigerant to a condition in which it will possess the ability to extract heat from the space to be cooled. For all compression-type systems the cycle consists of four processes: heat gain in the evaporator; pressure rise in the compressor; heat loss in the condenser; pressure loss in the expansion valve. The compression process is accomplished at the expense of energy added to the compressor in the form of shaft work, and the expansion process could be carried out, if the economics of the system would permit, in an expanding engine with consequent release of energy as shaft work. In ordinary systems, however, the additional first cost and maintenance costs of an expanding engine so greatly exceed the advantage resulting from the work realized, that such engines are not used, and the pressure reduction is allowed to occur irreversibly in an expansion valve. Basically, then, a refrigeration cycle consists of two heat transfer processes and two pressure change processes, no work entering into the heat transfer processes and—in the simple cycle—no heat transfer occurring during the pressure-change processes.

TABLE 2. PROPERTIES OF MONOCHLORODIFLUOROMETHANE (F-22)

Sat Temp F	Abs Press Lb per Sq In.	Volume		Enthalpy and Entropy Taken from −40 F							
				Enthalpy		Entropy		50 Deg Superheat		100 Deg Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Enthalpy	Entropy	Enthalpy	Entropy
0	38.79	0.01192	1.373	10.63	105.02	0.0240	0.2293	112.35	0.2446	120.00	0.2590
2	40.43	0.01195	1.320	11.17	105.24	0.0251	0.2289	112.59	0.2442	120.26	0.2586
4	42.14	0.01198	1.270	11.70	105.45	0.0262	0.2285	112.83	0.2438	120.52	0.2581
5	43.02	0.01200	1.246	11.97	105.56	0.0268	0.2283	112.95	0.2436	120.65	0.2579
6	43.91	0.01201	1.221	12.23	105.66	0.0274	0.2280	113.07	0.2434	120.78	0.2577
8	45.74	0.01205	1.175	12.76	105.87	0.0285	0.2276	113.31	0.2430	121.04	0.2572
10	47.63	0.01208	1.130	13.29	106.08	0.0296	0.2272	113.55	0.2426	121.30	0.2568
12	49.58	0.01211	1.088	13.82	106.29	0.0307	0.2268	113.79	0.2422	121.56	0.2564
14	51.59	0.01215	1.048	14.36	106.50	0.0319	0.2264	114.02	0.2418	121.82	0.2560
16	53.66	0.01218	1.009	14.90	106.71	0.0330	0.2260	114.25	0.2414	122.08	0.2556
18	55.79	0.01222	0.9721	15.44	106.92	0.0341	0.2257	114.48	0.2410	122.33	0.2552
20	57.98	0.01225	0.9369	15.98	107.13	0.0352	0.2253	114.71	0.2406	122.59	0.2548
22	60.23	0.01229	0.9032	16.52	107.33	0.0364	0.2249	114.94	0.2402	122.84	0.2544
24	62.55	0.01232	0.8707	17.06	107.53	0.0375	0.2246	115.17	0.2398	123.10	0.2540
26	64.94	0.01236	0.8398	17.61	107.73	0.0379	0.2242	115.40	0.2395	123.35	0.2537
28	67.40	0.01239	0.8100	18.17	107.93	0.0398	0.2239	115.62	0.2391	123.60	0.2533
30	69.93	0.01243	0.7816	18.74	108.13	0.0409	0.2235	115.84	0.2387	123.85	0.2529
32	72.53	0.01247	0.7543	19.32	108.33	0.0421	0.2232	116.07	0.2383	124.10	0.2525
34	75.21	0.01250	0.7283	19.90	108.52	0.0433	0.2228	116.29	0.2380	124.35	0.2522
36	77.97	0.01254	0.7032	20.49	108.71	0.0445	0.2225	116.52	0.2376	124.59	0.2518
38	80.81	0.01258	0.6791	21.09	108.90	0.0457	0.2222	116.74	0.2373	124.84	0.2515
40	83.72	0.01262	0.6559	21.70	109.09	0.0469	0.2218	116.96	0.2369	125.08	0.2511
42	86.69	0.01266	0.6339	22.29	109.27	0.0481	0.2215	117.18	0.2366	125.32	0.2508
44	89.74	0.01270	0.6126	22.90	109.45	0.0493	0.2211	117.40	0.2363	125.56	0.2504
46	92.88	0.01274	0.5922	23.50	109.63	0.0505	0.2208	117.61	0.2359	125.80	0.2501
48	96.10	0.01278	0.5726	24.11	109.80	0.0516	0.2205	117.82	0.2356	126.01	0.2497
50	99.40	0.01282	0.5537	24.73	109.98	0.0528	0.2201	118.02	0.2353	126.27	0.2494
52	102.8	0.01286	0.5355	25.34	110.14	0.0540	0.2198	118.22	0.2350	126.50	0.2491
54	106.2	0.01290	0.5184	25.95	110.30	0.0552	0.2194	118.42	0.2347	126.73	0.2488
56	109.8	0.01294	0.5014	26.58	110.47	0.0564	0.2191	118.62	0.2343	126.96	0.2484
58	113.5	0.01299	0.4849	27.22	110.63	0.0576	0.2188	118.82	0.2340	127.19	0.2481
60	117.2	0.01303	0.4695	27.83	110.78	0.0588	0.2185	119.01	0.2337	127.42	0.2478
62	121.0	0.01307	0.4546	28.46	110.93	0.0600	0.2181	119.21	0.2334	127.65	0.2475
64	124.9	0.01312	0.4403	29.09	111.08	0.0612	0.2178	119.40	0.2331	127.87	0.2472
66	128.9	0.01316	0.4264	29.72	111.22	0.0624	0.2175	119.59	0.2327	128.10	0.2469
68	133.0	0.01320	0.4129	30.35	111.35	0.0636	0.2172	119.77	0.2324	128.32	0.2466
70	137.2	0.01325	0.4000	30.99	111.49	0.0648	0.2168	119.96	0.2321	128.54	0.2463
72	141.5	0.01330	0.3875	31.65	111.63	0.0661	0.2165	120.15	0.2318	128.76	0.2460
74	145.9	0.01334	0.3754	32.29	111.75	0.0673	0.2162	120.32	0.2315	128.97	0.2457
76	150.4	0.01339	0.3638	32.94	111.88	0.0684	0.2158	120.50	0.2312	129.19	0.2455
78	155.0	0.01344	0.3526	33.61	112.01	0.0696	0.2155	120.67	0.2309	129.40	0.2452
80	159.7	0.01349	0.3417	34.27	112.13	0.0708	0.2151	120.85	0.2306	129.61	0.2449
82	164.5	0.01353	0.3313	34.92	112.24	0.0720	0.2148	121.02	0.2303	129.82	0.2446
84	169.4	0.01358	0.3212	35.60	112.36	0.0732	0.2144	121.18	0.2300	130.02	0.2443
86	174.5	0.01363	0.3113	36.28	112.47	0.0744	0.2140	121.34	0.2297	130.23	0.2441
88	179.6	0.01368	0.3019	36.94	112.57	0.0756	0.2137	121.50	0.2294	130.43	0.2438
90	184.8	0.01374	0.2928	37.61	112.67	0.0768	0.2133	121.66	0.2291	130.63	0.2435
92	190.1	0.01379	0.2841	38.28	112.76	0.0780	0.2130	121.82	0.2288	130.83	0.2432
94	195.6	0.01384	0.2755	38.97	112.85	0.0792	0.2126	121.97	0.2285	131.03	0.2429
96	201.2	0.01390	0.2672	39.65	112.93	0.0803	0.2122	122.12	0.2282	131.23	0.2427
98	206.8	0.01396	0.2594	40.32	113.00	0.0815	0.2119	122.26	0.2279	131.42	0.2424
100	212.6	0.01402	0.2517	40.98	113.06	0.0827	0.2115	122.40	0.2276	131.61	0.2421
102	218.5	0.01408	0.2443	41.65	113.12	0.0839	0.2111	122.53	0.2273	131.80	0.2418
104	224.6	0.01414	0.2370	42.32	113.16	0.0851	0.2107	122.66	0.2270	131.99	0.2416
106	230.7	0.01420	0.2301	42.98	113.20	0.0862	0.2104	122.79	0.2267	132.17	0.2413
108	237.0	0.01426	0.2233	43.66	113.24	0.0874	0.2100	122.92	0.2264	132.35	0.2411
110	243.4	0.01433	0.2167	44.35	113.29	0.0886	0.2096	123.04	0.2261	132.53	0.2408
112	249.9	0.01440	0.2104	45.04	113.34	0.0898	0.2093	123.16	0.2258	132.71	0.2405
114	256.6	0.01447	0.2043	45.74	113.38	0.0909	0.2089	123.28	0.2255	132.88	0.2403
116	263.4	0.01454	0.1983	46.44	113.42	0.0921	0.2085	123.40	0.2253	133.05	0.2400
118	270.3	0.01461	0.1926	47.14	113.46	0.0933	0.2081	123.51	0.2250	133.22	0.2398
120	277.3	0.01469	0.1871	47.85	113.52	0.0945	0.2078	123.62	0.2247	133.39	0.2395

Data from Kinetic Chemicals, Inc., 1945

The most common and least complicated type of refrigeration cycle is called the simple saturation cycle, and is shown diagrammatically in Fig. 2 and plotted upon pressure-enthalpy coordinates in Fig. 3. For this system, saturated vapor flows without gain or loss of heat from the evaporator to the suction of the compressor. During passage through the compressor the energy added as shaft work goes entirely to increase the enthalpy of the refrigerant, and the compression process, which is assumed to occur irreversibly and without external heat transfer, is characterized by constant entropy. Thus, the state of the superheated vapor leaving the compressor can be determined from the tables of thermodynamic properties by noting the discharge pressure and fixing, also, the entropy of the saturated vapor at entrance to the compressor.

TABLE 3. PROPERTIES OF MONOFLUOROTRICHLOROMETHANE (F-11)

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Enthalpy and Entropy Taken From −40 F							
				Enthalpy		Entropy		25 F Superheat		50 F Superheat	
		Liquid*	Vapor	Liquid	Vapor	Liquid	Vapor	Enthalpy	Entropy	Enthalpy	Entropy
0	2.59	0.01020	13.700	7.81	90.4	0.0178	0.1975	93.9	0.2049	97.4	0.2120
5	2.96	0.01024	12.100	8.81	91.2	0.0200	0.1974	94.7	0.2047	98.2	0.2117
10	3.38	0.01028	10.700	9.82	92.0	0.0222	0.1973	95.5	0.2045	99.0	0.2114
15	3.85	0.01032	9.530	10.80	92.8	0.0243	0.1971	96.3	0.2043	99.8	0.2111
20	4.36	0.01036	8.490	11.90	93.7	0.0264	0.1970	97.2	0.2041	100.7	0.2109
25	4.94	0.01040	7.580	12.90	94.5	0.0286	0.1969	98.0	0.2039	101.5	0.2107
30	5.57	0.01045	6.770	13.90	95.3	0.0307	0.1969	98.8	0.2038	102.3	0.2105
35	6.27	0.01049	6.080	14.90	96.1	0.0328	0.1968	99.6	0.2037	103.1	0.2103
40	7.03	0.01053	5.460	16.00	96.8	0.0349	0.1968	100.3	0.2036	103.8	0.2101
45	7.88	0.01057	4.920	17.00	97.6	0.0370	0.1967	101.1	0.2035	104.6	0.2099
50	8.79	0.01062	4.440	18.10	98.4	0.0391	0.1967	101.9	0.2034	105.4	0.2098
55	9.80	0.01066	4.020	19.10	99.2	0.0412	0.1967	102.7	0.2033	106.2	0.2097
60	10.90	0.01071	3.640	20.20	100.0	0.0432	0.1967	103.5	0.2033	107.0	0.2096
65	12.10	0.01076	3.300	21.30	100.8	0.0453	0.1967	104.3	0.2032	107.8	0.2094
70	13.40	0.01081	3.000	22.40	101.5	0.0473	0.1967	105.0	0.2032	108.5	0.2093
75	14.80	0.01086	2.740	23.50	102.2	0.0493	0.1967	105.7	0.2031	109.2	0.2092
80	16.30	0.01091	2.500	24.50	102.9	0.0513	0.1966	106.4	0.2030	109.9	0.2090
85	17.90	0.01096	2.280	25.60	103.6	0.0533	0.1966	107.1	0.2029	110.6	0.2089
90	19.70	0.01101	2.090	26.70	104.4	0.0553	0.1966	107.9	0.2028	111.4	0.2088
95	21.60	0.01106	1.918	27.80	105.1	0.0573	0.1966	108.6	0.2028	112.1	0.2087
100	23.60	0.01111	1.761	28.90	105.7	0.0593	0.1965	109.2	0.2027	112.7	0.2085
105	25.90	0.01116	1.620	30.10	106.4	0.0613	0.1965	109.9	0.2026	113.4	0.2084

Superheated vapor from the compressor flows to the condenser where de-superheating and condensation take place. From the condenser the refrigerant flows to the expansion valve, undergoes a constant-enthalpy pressure reduction, and returns to the evaporator where it again removes a quantity of undesired heat. When the evaporator is arranged to permit direct cooling of room air by the refrigerant, the system is said to be of the *direct expansion* type, while a system in which the evaporating refrigerant cools water or brine, which in turn cools the air, is said to be *indirect*. Although many differences exist between most actual systems and that of the simple saturation cycle, this latter is, nonetheless, of great value in that it provides an extremely simple method of rapidly achieving an approximate analysis of probable power requirements, compressor size, etc. Further, the equations used in analysis of a simple saturation cycle form the basis of the more complex treatments required for compound refrigeration cycles. For these reasons a typical simple saturation problem will be worked in detail.

Example 1: A simple saturation cycle carries a 7 ton load when operating between suction and discharge pressure of 52.7 psia and 121 psia with F-12 as the refrigerant. Determine: (*a*) the cooling effect provided by each pound of refrigerant; (*b*) the refrigerant circulating rate; (*c*) the horsepower required; (*d*) the quantity of heat to be dissipated from the condenser; (*e*) the required condenser cooling water, in gallons per minute, if temperature rise of water passing through the condenser is 8 deg; (*f*) the bore and stroke of a double acting cylinder (neglecting the effect of the piston rod) if speed of compressor is 500 revolutions per minute; (*g*) coefficient of performance.

Solution: (*a*) Saturated liquid F-12 at 121 psia leaves the condenser and enters the expansion valve. The enthalpy of this material (from Table 1) is 29.68 Btu per pound, and this must also be its enthalpy at entrance to the evaporator. Leaving the evaporator as a saturated vapor at 52.7 psia, its enthalpy is 82.82, so the refrigerating effect must be 82.82 − 29.68 = 53.14 Btu per pound.

(*b*) The refrigerant circulating rate is equal to the total heat to be picked up in unit time, divided by the pick-up per pound of refrigerant or,

$$W_r = (7 \text{ ton} \times 200) \div 53.14 = 26.3 \text{ lb per minute.}$$

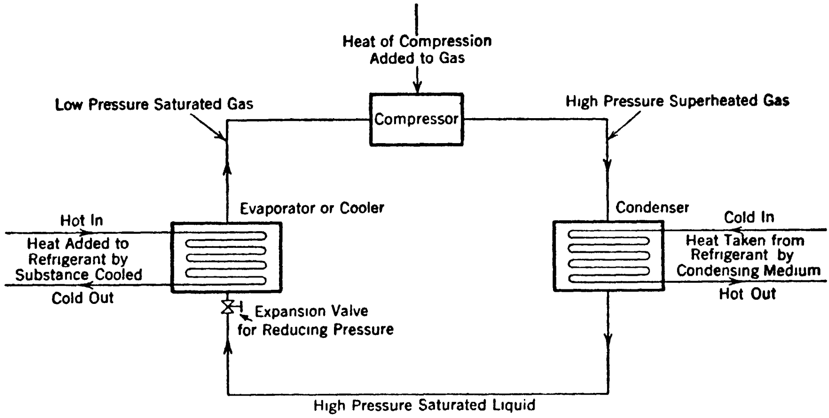

FIG. 2. MECHANICAL REFRIGERATION SYSTEM

(*c*) The horsepower required is equal to the increase in energy of the refrigerant passing through the compressor (expressed in Btu per minute) divided by the conversion factor 42.42, which is the number of Btu per minute corresponding to 1 hp,

$$(\text{hp}) = W_r\,(h_d - h_{vs}) \div 42.42 \qquad (7)$$

where

hp = horsepower.
W_r = refrigerant circulating rate in pounds per minute.
h_d = enthalpy of vapor at condition of discharge from compressor.
h_{vs} = enthalpy of saturated vapor entering compressor.

W_r is known from (*b*) and h_{vs} is the enthalpy of refrigerant as it enters the compressor in a saturated vapor state at 52.7 psia; thus h_{vs} = 82.82.

In order to determine h_d, the state of the refrigerant must first be determined at the compressor discharge. At the known suction state the entropy (from Table 1 for saturated vapor at 52.7 psia) is 0.16828 and, since the compression is assumed to occur isentropically, it therefore follows that the discharge stage must have the same entropy at 121 psia. From the table the entropy of vapor superheated 25 deg is

0.17330, so the superheat, t_{sd}, possessed by the actual gas discharged from this compressor can be obtained by interpolation as,

$$\frac{t_{sd}}{25} = \frac{0.16828 - 0.16608}{0.17330 - 0.16608}$$

from which $t_{sd} = 7.6$ deg.

As the saturation temperature at 121 psia is 94 F the actual temperature, t_d, of the vapor leaving the compressor is, $t_d = 94 + t_{sd} = 94 + 7.6 = 101.6$ F. By the same kind of interpolation the enthalpy of the discharged vapor can be determined from the enthalpies given for vapor superheated 25 F and for saturated vapor,

$$\frac{(h_d - 88.10)}{(92.16 - 88.10)} = \frac{(0.16828 - 0.16608)}{(0.17330 - 0.16608)}$$

from which, $h_d = 89.34$ Btu per pound.

Then substituting in Equation 7,

$$(\mathrm{hp}) = 26.3\ (89.34 - 82.82) \div 42.42 = 4.03.$$

(*d*) The rate of heat loss from the condenser, Q_c, must be equal to the sum of the energies picked up by the refrigerant in the evaporator and the compressor,

$Q_c = 53.14 + (89.34 - 82.82) = 53.14 + 6.52 = 59.66$ Btu per pound or $26.3 \times 59.66 = 1569$ Btu per minute. This same figure can, of course, be determined more

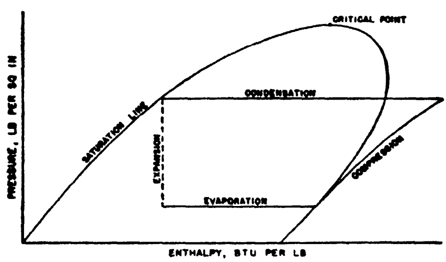

FIG. 3. PRESSURE-ENTHALPY DIAGRAM FOR SIMPLE SATURATION CYCLE

directly by subtraction of the enthalpy of liquid leaving the condenser from the enthalpy of superheated vapor going into it, thus,

$$Q_c = 26.3\ (89.34 - 29.68) = 1569 \text{ Btu per minute.}$$

(*e*) The cooling water rate (based on a gallon as 8.34 lb) is $1569 \div (8 \times 8.34) = 23.5$ gpm.

(*f*) The compressor size is fixed by the volume of gas which must be drawn into the machine per unit time. Saturated vapor at 52.7 psia has a specific volume, from Table 1, of 0.779 cu ft per pound, hence $26.3 \times 0.779 = 20.49$ cfm of gas must be handled. Assuming a volumetric efficiency of 90 percent, the compressor must then displace $20.49 \div 0.9 = 22.8$ cfm. The speed is given as 500 rpm and, as the unit is known to be double-acting, the displacement is therefore $(22.8 \times 1728) \div (2 \times 500) = 39.4$ cu in. If the unit were designed so that bore d and stroke were the same,

$$(\pi d^3) \div 4 = 39.4 \qquad d = 3.69 \text{ in.}$$

$$(g)\ (\mathrm{CP}) = (h_{vs} - h_{fc}) \div (h_d - h_{vs})$$

$$= (82.82 - 29.68) \div (89.34 - 82.82) = 8.17$$

where h_{fc} is the specific enthalpy of liquid at discharge from the condenser.

The coefficient of performance of *Example* 1 may be compared with that of an ideal system operating on the Carnot cycle between the same temperature limits. Then T_e = 501 F (which is 41 F + 460) and T_c = 554 F (which is 94 F + 460) and,

$$(\mathrm{CP}) = \frac{501}{554 - 501} = 9.6$$

The actual cycle is therefore 8.17 ÷ 9.6 or 85 percent as effective as a Carnot cycle between the same temperature limits.

Influence of Suction Pressure

Brief consideration of the analytical procedure used in discussion of the simple saturation cycle will bring out the need for maintaining the suction pressure on any refrigeration system as high as the load will permit. As the suction pressure increases, for fixed discharge pressure, the enthalpy of refrigerant entering the evaporator remains unchanged, but the leaving enthalpy increases and, hence, the refrigerating effect increases. Further, compressor energy input is reduced not merely because of the greater enthalpy of the gas at suction, but also because of a reduction in the enthalpy of the superheated gas at discharge. Since the refrigerating effect is greater and the work less, it is obvious that there will be a substantial gain in the coefficient of the performance.

The actual value of suction pressure on any system is obviously determined by the required temperature which must be maintained in the conditioned space. For a direct expansion system the evaporator can be held at a temperature not much less than that of the conditioned enclosure, except in cases where lower temperatures may be needed in order to establish a desired ratio of dehumidifying to cooling load. When dehumidification requirements dictate the use of unusually low evaporator temperatures, the increased operating cost should properly be charged against the dehumidification rather than the sensible cooling.

Influence of Discharge Pressure

In contrast to the suction pressure, the compressor discharge pressure should be kept as low as operating conditions will allow. This pressure must be high enough to provide a saturation temperature of refrigerant within the condenser which is greater than the exit temperature of the cooling water. The discharge pressure therefore is a direct function of the temperature of the cooling fluid, and will automatically rise whenever the temperature of cooling water (or air) rises; it will also rise when the flow rate of the cooling medium is decreased.

Increase in discharge pressure (for fixed suction pressure) raises the enthalpy of the gas leaving the compressor; hence, increases the work of compression. Further, as the enthalpy of saturated liquid leaving the condenser increases with pressure, the refrigerating effect must decrease. Thus the effect of such a pressure rise is to require more work per pound of refrigerant handled, and at the same time to necessitate an increase in the refrigerant flow rate.

Influence of Water Jacket

The preceding discussion has, in every case, assumed isentropic compression. Where exact performance data are not available, this assumption is a desirable one since it leads to a conservatively large determination of the power required. In most actual systems, the compression process departs from isentropic due to irreversible heat transfers which occur between the vapor in the cylinder and the cylinder wall, and also because of intentional heat dissipation from the outside of the cylinder walls to the surroundings, or to a cooling fluid passing through a water jacket around the cylinder. Compressor cooling is highly desirable as a method of reducing power consumption.

Influence of Superheating and Subcooling

The most common departure from conditions of the simple saturation cycle is that resulting from admission of superheated vapor to the compressor. Thermodynamically, superheat is undesirable because the enthalpy increase required to compress a vapor through a given pressure range increases with superheat. Further, superheated vapor leaving an evaporator is usually an indication that the suction pressure is lower than necessary. Under practical operating conditions, however, superheat is almost universally used as a means of assuring complete vaporization of the refrigerant going to the compressor. With modern compressors operating at high speed, and with relatively small clearance space, it is

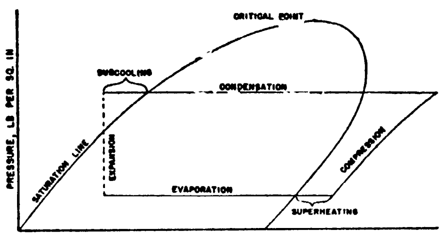

FIG. 4. PRESSURE-ENTHALPY DIAGRAM FOR REFRIGERATION CYCLE WITH SUBCOOLING AND SUPERHEATING

particularly necessary to avoid admission through the suction valves of liquid refrigerant.

Another common departure of actual systems from the simple saturation cycle occurs because of subcooling of refrigerant in the condenser. Thermodynamically, such subcooling is advantageous since it increases the refrigerating effect without affecting the unit energy requirements of the compressor. Further, it can be shown that for a fixed ratio of condenser cooling water to refrigerant circulating rate, the total compressor power requirements will be greater when operating at simple saturation than when operating with maximum sub-cooling. What is even more surprising is that condenser pressure may be lower for the sub-cooling cycle than for the saturation cycle. This condition results from the fact that, for the same capacity on a heavily loaded condenser, the refrigerant flow rate is less when there is sub-cooling.

Because of the advantages attendant upon the use of sub-cooling, many methods are in use for obtaining some sub-cooling effect outside of the condenser. One common procedure is to use the cold vapor leaving the evaporator to cool the liquid flowing from condenser to expansion valve.

Another somewhat unusual sub-cooling cycle allows cold refrigerant from the downstream side of the expansion valve to cool liquid refrigerant from the condenser down to the evaporator temperature. Fig. 4 shows the pressure-enthalpy diagram for a typical refrigeration cycle operating with both sub-cooling of the refrigerant from the condenser and superheating of the refrigerant leaving the evaporator.

Clearance and Volumetric Efficiency

Clearance, like displacement, is a characteristic—usually fixed—of a given compressor. In some cases clearance pockets are provided which place within the operator's control the ability to alter the clearance of the machine, but most moderate size compressors are built with fixed clearance. By definition, the clearance is the percentage of the volume swept by the piston, which is represented by spaces in the end of the cylinder (including valve spaces, etc.) when the piston is at the end of its stroke.

Because of the trapping of high pressure vapor in the clearance space, and its subsequent re-expansion, the suction valves of the compressor do not open until the piston has completed part of its stroke. Hence, the volume of fresh vapor introduced into the compressor per stroke is less than the volume swept by the piston. The ratio of actual volume of fresh gas to swept volume is, by definition, the clearance volumetric efficiency, CVE. In equation form,

$$(\mathrm{CVE}) = 100 - V_c\left[\frac{v_s}{v_d} - 1\right] \tag{8}$$

where

CVE = clearance volumetric efficiency.
V_c = clearance, percent of volume swept by piston, which is contained in spaces at end of cylinder when piston is at end of stroke (clearance includes valve spaces, etc.).
v_s = specific volume of gas at compressor inlet.
v_d = specific volume of gas at compressor discharge.

Values of v_s and v_d can be obtained directly or by calculation from the tables of properties of refrigerants.

In addition to clearance, there are several other factors which tend to reduce the volumetric efficiency. The suction gases from the evaporator are heated and expanded upon contact with the hot cylinder walls during the suction stroke. This results in a reduction of the actual charge drawn into the cylinder. Wire-drawing through the suction and discharge valves reduces the suction pressure in the cylinder below that in the evaporator, and increases the discharge pressure above that in the condenser. Leakage of gases around the pistons also decreases the volumetric efficiency. The total volumetric efficiency (TVE) includes all of these factors and is reliably obtained only by laboratory measurements. It is too difficult to predict the effects of these factors to any degree of accuracy comparable to actual tests.

Complex Refrigeration Cycles

The preceding sections have dealt only with refrigeration systems in which there is but one evaporator; compression is accomplished through but a single stage, and expansion proceeds through a single expansion valve. In large systems or in low temperature systems in which the compression ratio is high, the compression process can be carried out in stages, with the refrigerant passing through several cylinders arranged for operation in series. The thermodynamic advantage of such compound compression arises from the fact that intercoolers can be placed between the stages of compression to extract heat from the vapor, and thereby cause the overall compression process to approach more closely the ideal condition of isothermal compression. Essentially, such intercoolers serve the same purpose as a cooling jacket, but with greater effectiveness because of the more satisfactory heat transfer conditions.

In the simple saturation cycle the saturated liquid entering the expansion valve commences to vaporize as soon as its pressure starts to drop. The vapor produced during the expansion process has no further use, in terms of refrigerating effect, since it has already picked up its latent heat of vaporization as a result of heat which it has extracted from the unvaporized residue. Thus the instant such vapor forms, its usefulness is at an end, and to allow such material to undergo a further drop in pressure is uneconomical. With compound compression, there is at least one intermediate pressure at which flash vapor can be extracted. In such cases several expansion valves can be utilized with all of the refrigerant from the condenser passed through a first expansion valve to the higher suction pressure, and the flash vapor then extracted and returned to the condenser through the high compression stage. The remaining refrigerant can then pass through a second expansion valve where the pressure is dropped to that corresponding to the low-pressure evaporator. The number of expansion valves is limited by the number of stages of compression.

Further cycle complications may arise if more than one evaporator is to be operated with a single compressor, and particularly, if the pressures in these evaporators are to differ. The most common solution is to operate the compressor at the suction pressure of the lowest pressure evaporator, and to equip all other evaporators with back-pressure regulating valves or throttling devices between the evaporator and the compressor suction. This permits these evaporators to operate at higher pressures and, therefore, higher temperatures than those corresponding to the compressor suction conditions. However, this is accomplished only with a loss of power, since all of the refrigerant from all of the evaporators must be compressed through the maximum lift from the lowest pressure in the system.

The Air Cycle System

Air cycle refrigeration, one of the earliest forms of cooling, became obsolete for many years because of its low coefficient of performance and high operating costs. Recently, however, it has been applied with success to aircraft cooling systems where, with low equipment weight, it can utilize a portion of the cabin air supercharger capacity. It is unique among refrigeration systems in that the refrigerant remains in the gaseous phase throughout the cycle.

Fundamentally, the air cycle is essentially the same as the vapor cycle. Compression is accomplished by a reciprocating or centrifugal compressor, and, since there is no change of phase of the refrigerant upon expansion, an *air cooler* replaces the condenser, and a *refrigerator*, the evaporator. Although some cooling would result from the expansion of the gas through an ordinary expansion valve, a much greater drop in air temperature is accomplished if the expansion is controlled to approach the isentropic by replacing the valve with an expansion engine or turbine. Furthermore, the work recovered by such an expansion engine can be utilized to supply part of the work of compression or to drive other devices.

It is a common misconception that aircraft flown at high altitudes do not require comfort cooling. With pressurized cabins the work of compression results in an air temperature increase which, when added to the heat supplied by ram effect, solar radiation, electrical and mechanical equipment and the occupants of the plane, may make the conditions intolerable without comfort cooling. At 600 mph, the ram temperature effect of stopping the air relative to the plane will result in an entering air temperature of 164 F when the ambient air is at the standard Army summer sea level design temperature of 100 F. At 1000 mph, the entering air temperature is almost 280 F.

Air cycle systems are used in practically every jet fighter and many modern commercial passenger planes flying today. In comparative studies[1] made during the design of the cooling system for one large commercial airliner, it was shown that an air cycle system was much lower in both weight and space requirements than either a vapor compression or dry ice system. It had the further advantages of ease of repair and the use of a completely non-toxic refrigerant. The weight, for example, was reduced from approximately 60 lb per ton for a vapor compression system, or an initial weight of 130 lb per ton for a dry ice system, to approximately 25 lb per ton for an air cycle system. The usual disadvantage of high power requirements for the operation of the air cycle system was shown to be more than offset by the reduction in fuel requirements for transporting the bulk and weight of the cooling system through the air. Quite possibly, with continued development and further experience, air cycle refrigeration systems may be used economically for other applications, particularly in the transportation field.

The Steam Jet System

The *steam jet* system, under certain circumstances, is desirable for use in air conditioning.[2] Steam supplies directly the power used for compressing the refrigerant, thus eliminating the losses connected with other methods of supplying energy. As the compression ratio between the evaporator and condenser under normal circumstances is large, the mechanical efficiency of the equipment is somewhat lower than that of the positive mechanical type compressor. The condensing water requirements are considerably greater, as both the refrigerant and the impelling steam must be condensed.

The steam jet system functions on the principle that water under high vacuum will vaporize at low temperatures. Steam jet boosters or compressors of the type commonly used in power plants for various processes, will produce the necessary low absolute pressure to cause evaporation of the water.

A diagrammatic representation of a typical steam ejector water cooling system is shown in Fig. 5. The figures correspond to an average representative system. The water to be cooled enters the evaporator and is cooled to a temperature corresponding to the vacuum maintained. Because of the high vacuum, a small amount of the water introduced in the evaporator is flashed into steam. As this requires heat, and the only source of heat is the rest of the water in the evaporator tank, this other water is almost instantly cooled to a temperature corresponding to the boiling point determined by the vacuum maintained. The amount of water flashed into steam is a small percentage of the total water circulated through the evaporator, amounting to approximately 11 lb per (hr) (ton) of refrigeration developed. The remainder of the water at the desired low temperature is pumped out of the evaporator and used at the point where it is required.

The ejector compresses the vapor which has been flashed in the evaporator, plus any entrained air taken from the circulated water, to a somewhat higher absolute pressure. The vapor and air mix with the impelling steam on the discharge side of the jet, and the total mixture then passes from the ejector into the condenser.

The slight amount of air which may be entrained in the cooled water is removed by a small secondary ejector which raises the pressure sufficiently so that the air can be discharged to the atmosphere. A secondary condenser is then necessary to condense the steam in the secondary jet.

While a single booster of smaller than 15 tons capacity is difficult to build, steam jet vacuum cooling units have been built for as small as 5 to 6 tons capacity. They can readily be built for steam pressures of from 5 to 200 psig, and condenser water temperatures as high as 90 F. The

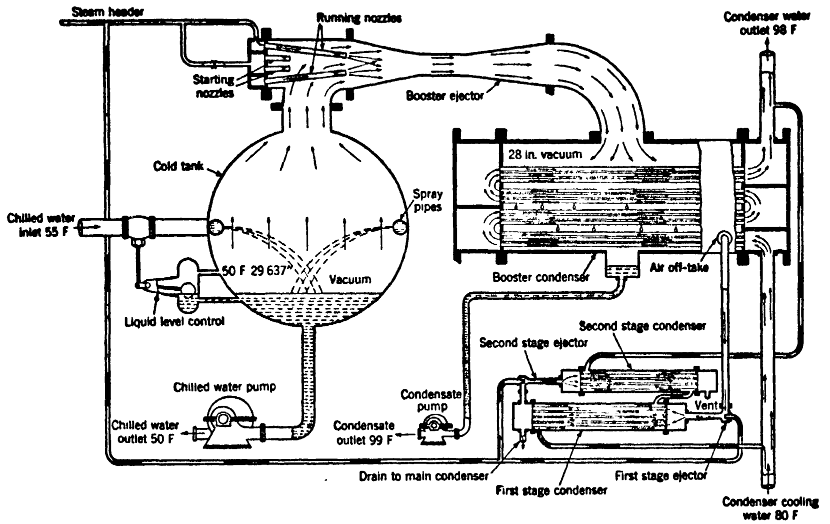

FIG. 5. DIAGRAMMATIC ARRANGEMENT OF STEAM JET VACUUM COOLING UNIT

steam consumption in pounds per hour per ton of refrigeration increases rapidly as the booster steam pressure is lowered. For example, the lowering of the booster steam pressure from 200 to 90 psig results in an increase in steam consumption of approximately 5 percent, whereas a further decrease in booster steam pressure to 10 psig increases the steam consumption by approximately 72 percent over that required at 200 psig.

The capacity of a steam jet system is usually controlled by controlling the number of boosters in use since the unit usually has several boosters operating on the same evaporator. Usually one booster is automatically controlled, whereas the others are manually operated. The capacity is dependent, as for all compressors, upon the evaporator temperature, or in other words, the suction pressure. For example, the capacity is lowered approximately 17 percent if the evaporator or chilled water temperature is lowered from 50 to 45 F. The capacity therefore can be controlled to some extent by regulating the evaporator temperature.

The Absorption System

The absorption and compression refrigeration cycles differ only with respect to the method of compression. Each cycle requires a condenser, expansion valve, and evaporator, but the absorption cycle utilizes three major equipments in place of the mechanical compressor; these equipments are the *absorber*, the *pump*, and the *generator*. Vapor from the evaporator is absorbed by a low temperature absorbent fluid which is then pumped to the generator where heat is supplied to boil off the refrigerant. The absorbent is now cooled and readmitted, through a pressure-reducing valve, to the absorber.

In addition to the three primary equipments of the absorption cycle it is necessary to provide auxiliary equipment, usually an analyzer and a rectifier, to remove from the refrigerant leaving the generator, insofar as is possible, the absorbent which vaporizes and leaves the generator with the refrigerant. Removal of this material is of great importance to effective operation of the system, since even a small concentration of absorbent

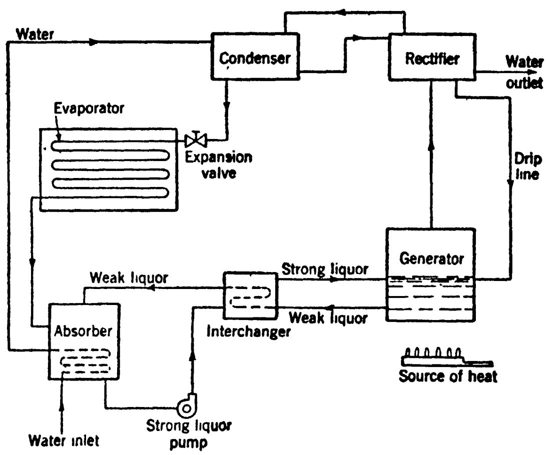

Fig. 6. Closed Absorption System

in the refrigerant will suffice to reduce greatly the evaporator pressure required for maintenance of a given evaporator temperature. Thermodynamic analysis of absorption cycles is relatively complex, and requires the use either of tables or graphs showing the equilibrium relationships and thermodynamic properties of the refrigerant-absorbent combination. Data of this kind are available in bibliography item A, and a discussion of various absorbents is given in bibliography item F. Thermodynamically, the effectiveness of a refrigerant-absorbent combination increases directly with its negative deviation from Raoult's Law.

Fig. 6 shows a typical absorption cycle flow diagram. Cooling water first goes through the absorber (where it extracts the heat of absorption which is liberated by the refrigerant vapor as it goes into solution), then through the condenser, and finally through the rectifier. Refrigerant from the evaporator enters the absorber where it goes into solution in the absorbent; the high concentration solution is then pumped to the generator where heat is supplied; the refrigerant (with some absorbent vapor) leaves for the rectifier and the warm low concentration solution is returned to the absorber. In the rectifier selective condensation occurs, the concentration of the absorbent in the condensate being much greater than its concentration in the entering vapor mixture; rectifier condensate is dripped back to the generator.

The ratio of refrigerating effect to heat input (the performance ratio or commonly used efficiency measure of absorption machines) is only 40 to 45 percent with the ordinary ammonia absorption system and, aside from the inherent disadvantages involved in the use of a toxic and explosive refrigerant, this is not sufficiently high to make it competitive with other types of systems when used in air conditioning applications. Therefore, recently, several absorption systems using hygroscopic brines of salts such as lithium chloride or lithium bromide[3] (solids in the pure state) as absorbents and water as the refrigerant have been developed. Such systems are limited to higher temperature applications but, thermodynamically, have

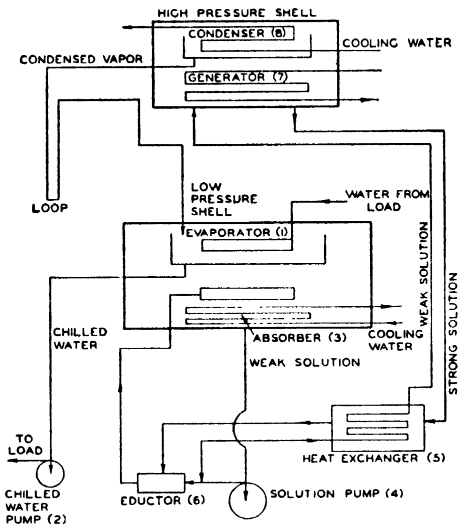

FIG. 7. DIAGRAM OF LITHIUM BROMIDE WATER ABSORPTION SYSTEM

the advantages of a refrigerant with a high latent heat of vaporization and nonvolatile absorbents with a large negative deviation from Raoult's Law. None of the absorbent is carried off with the refrigerant from the generator and the performance ratio ranges as high as 75 percent. Both the refrigerant and the absorbent are non-toxic and non-explosive and the performance ratio does not vary greatly between 20 percent of capacity and full load. This increased efficiency places operating costs in competition with other forms of refrigeration in many high temperature applications such as air conditioning.

One form of lithium-bromide-water absorption system is shown schematically in Fig. 7 with the generator and condenser shown located in a high pressure shell and the evaporator and absorber in a low pressure shell. The water to be cooled flows to the evaporator 1 from the load where a small portion of it is flashed into vapor thus cooling the remaining water which is then returned by pump 2 to the load. The pressure in the low side shell and, therefore, the temperature of the water passing through the shell is controlled by the temperature and concentration of the lithium bromide brine sprayed over absorber coil 3. The water vapor flashed in the evaporator chamber is absorbed by the strong salt solution to form a weak solution which drains from the low pressure chamber and passes through solution pump 4. From pump 4 a portion of the weak brine is delivered through the heat exchanger 5 to the high pressure shell and the remainder is mixed with the strong solution through eductor 6 and delivered back to the low pressure shell. Heat applied at the generator 7 boils off the water vapor earlier condensed in the absorber and returns the brine to its original concentration. The condenser 8, also located in the high pressure shell, liquifies the water boiled off by the generator and this condensate is returned through a liquid loop to the evaporator. The re-concentrated solution is returned from the high pressure shell through the heat exchanger 5 to the eductor 6 where it is mixed with a portion of the weak solution and pumped to the absorber with the mixture at a still relatively high concentration.

Ice Systems

Cold water systems using ice as the cooling agent have been installed in some theaters, restaurants, funeral homes, churches and other places where short hours of operation and high peaks of cooling demand make this type of system desirable. A comparatively small quantity of ice in the water cooling tank of such a system can release refrigeration at a relatively rapid rate. For instance, neighborhood theaters having a peak demand of 1,200,000 Btu per hr (100 tons refrigeration) have found 8 ton capacity ice bunkers satisfactory.

In operation, the water in the air conditioning system is circulated over ice placed in an insulated box, and is cooled to the 38 or 40 deg range or higher, if desired. This cold water is pumped from the ice bunker to air cooling coils or spray type air washers. The blowers, coils, air washer or air handling sections are the same as those parts in any system employing cold water as a refrigerant.

The ice water cooler or ice bunker is usually built at the installation in a location where it can easily be iced. It can be constructed of any desired material such as concrete, steel, or wood with an adequate amount of insulation to save the ice from one period of use to the next. The basic requirement is that the tank be durable and water-tight.

The temperature of the water is controlled at a predetermined point by a thermostat in the supply line. If the temperature drops too low, a part of the return water is by-passed directly to the sump and is not cooled over the ice. In the larger systems it is customary to install an overflow control which, as the ice melts, discards the excess water through an economizer coil, the surface of which is large in relation to the flow so that the water is warmed to 60 F or more as it is discharged from the system.

In an attempt to lower initial equipment cost and operating expense, or increase the refrigeration capacity of an existing air conditioning system, storage refrigeration has been utilized in a few applications. Some of the methods which have been adopted include the storage of refrigeration in the form of chilled water, chilled brine, ice on evaporator coils[4] and the accumulation of thin sheets of ice on copper plates in a steel tank.[5] If the peak load factor is low as compared with a long period of operation, such as in a restaurant, or if the hours of operation are short but the usage factor high, as in a church, then it is possible to consider storage refrigeration. This method of accumulating refrigeration frequently makes it possible to use low cost off-peak electric power. Power costs may also be reduced by

TABLE 4. HEAT PUMP HEAT SOURCES AND SINKS

HEAT SOURCE	AIR	CITY WATER	WELL WATER	SURFACE WATER	WASTE WATER	EARTH	SOLAR
Source Classification	Primary	Primary or auxiliary	Primary	Primary	Primary or auxiliary	Primary or auxiliary	Auxiliary
Suitability as Heat Sink	Good	Good	Good	Good	Variable with source	Usually poor	None
Availability (Location)	Universal	Cities	Uncertain	Rare	Limited	Extensive	Universal
Availability (Time)	Continuous	Continuous—except local shortages	Continuous—unless well runs dry	Continuous	Variable	Continuous, temperature level drops from a maximum to a minimum as heat is removed, slowly rises when pump is inoperative	Intermittent. Unpredictable, except over extended time
Expense (Original)	Low, less than earth and water sources except city	Usually lowest	High, result of drilling well	Low	Variable	High, usually less than the cost of drilling a well	Unexplored
Expense (Operating)	Relatively low	High, usually prohibitive	Low to moderate	Relatively low	Relatively low	Relatively moderate	Unexplored. Promising as auxiliary for reducing operating cost
Temperature (Level)	Favorable 75-95% of time in most of U. S.	Usually satisfactory	Satisfactory	Satisfactory	Usually good	Initially good—drops with time and rate of heat withdrawal	Excellent
Temperature (Variation)	Extreme	Variable with location (10 to 25 F Deg)	Small	Moderate	Usually moderate	Large—less than for air, however	Extreme
Design Information	Inadequate	Usually adequate	Usually adequate	Usually adequate	Adequate if source is constant in supply and temperature	Inadequate	Practically available
Size of Equipment	Particularly bulky	Moderate	Moderate (except well)	Moderate	Variable (usually moderate)	Moderate (except ground coils)	Probably bulky
Adaptability to Mass Production	Excellent, can be factory assembled and tested	Excellent	Excellent, (except well)	Excellent	Poor	Fair	Fair
Sources it may Augment		Air, earth				Air, water	Air, earth, water
Special Problems	Least heat available when demand greatest. Coils may become frosted requiring extra capacity, alternate source, or standby heat. May require duct work. Variable air temperature makes control difficult.	Scale on coils. Local use restrictions during shortages. Disposal. Water temperature may become too low to permit further heat removal.	Corrosion, scale may form on heat transfer surface. Disposal. Water location, temperature, composition usually unknown until well drilled Well may run dry	Water may cause scale, corrosion, and algae fouling	Usually scale forming or corrosive. Often insufficient supply Very limited application, hence requires individual design	Limited by local geology and climate. Installation costs difficult to estimate. Requires considerable ground area, may damage lawns, gardens	Probably will require heat storage equipment at either evaporator or condenser side.

installing a smaller refrigeration plant, augmented by a storage system, and by operating it for longer periods.

The Heat Pump

It has been almost 100 years since Prof. William Thomson (Lord Kelvin) first proposed the use of a compressor as a "warming engine" and as a means of heating buildings to replace equipment for direct burning of fuels. Several early working models were constructed, but the device has remained essentially of laboratory interest until the last 20 years.

Although frequently referred to incorrectly as the reverse cycle system, the heat pump cycle is identical with the ordinary refrigeration cycle, and differs only in the sense that the desired effect is rejection of the heat from the condenser rather than absorption of heat in the evaporator. A discussion of the coefficient of performance for the heat pump is found earlier in this chapter.

The first actual residential heat pump installation was probably made in Scotland in 1927 and since that time, a number of commercial and residential systems have been made in this country. Both progress and growth of interest have been particularly rapid during the past four years and, consequently, at the present time there are several hundred residential installations and probably a greater number of commercial systems. However much research is needed before the residential heat pump installation can successfully emerge to compete economically and with equal reliability with the more common forms of heating and fuels.

From an analysis of the equation for the coefficient of performance, it is evident that the economical adaption of the heat pump as a practical means of heating, requires that the temperature of the source from which the heat is extracted be as high as possible, and that the temperature of the sink to which the heat is rejected for heating purposes, be as low as possible. Thus, with a small temperature spread between the evaporator and the condenser, six or more times as much heat may be obtained theoretically (and three to five times practically) as the heat equivalent of the work necessary to operate the system. There are a number of limitations, however, the most serious of which is the lack of ready availability of a practical source of heat.

One of the major problems in the development of the heat pump involves research on, and the compilation of reliable design data for, the various heat sources and sinks available. The four principal potential sources of heat are air, water, earth, and solar energy. Of these, the first three are primary sources of heat which may be used alone. The fourth, solar energy, while of tremendous potentiality, will probably be developed in most localities as auxiliary to the other three. In addition, there are other minor sources such as process waste heat, sewage, etc., which may be used under special circumstance.

There are also a number of industrial applications of heat pumps, for purposes other than space heating, which are practical largely through economic considerations of the particular process involved. Table 4 presents a summary of the advantages and disadvantages of each of these major heat sources.

By reference to Table 4, it will be seen that, to date, the most satisfactory heat sources are air, water, and earth, and that air and water are the most satisfactory heat sinks. There are, therefore, six possible combinations of source and sink in application: air to air, air to water, water to air, water to water, earth to air, and earth to water. In addition, it should be recognized that heat storage devices may be used with any of these systems involving either a single or a dual heat source. One promising possibility involves the utilization of a storage pit or cistern operating upon a heat of fusion cycle and supplied by supplementary heat from air or solar sources. Heat of fusion may, for example, also be utilized with city water and sewage disposal to increase the practicability of these sources.

When heat is to be obtained from a ground coil it should be emphasized that, in general, the heat extracted must be replaced by heat from the sun, received by radiation to the ground or by heat carried into the ground by rain.

Combinations of heat sources, such as air and water, air and ground, air and solar energy, or ground and solar energy, may also be used and will frequently improve the coefficient of performance over an entire heating season; but they will probably result in considerably greater initial cost.

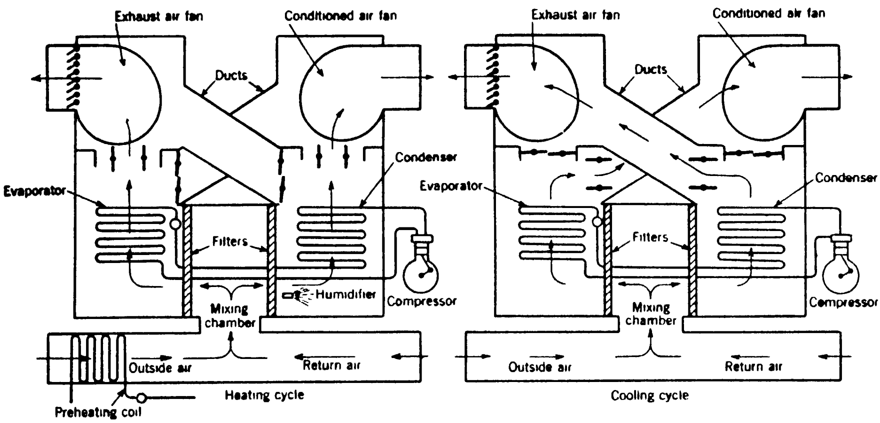

FIG. 8. SCHEMATIC OPERATION OF REVERSED CYCLE CONDITIONING SYSTEM

A typical arrangement of a heat pump system with air as the heat source is shown in Fig. 8.

Both water and air are practical media to which the condenser heat may be rejected; but the generation of steam requires too high a condenser temperature. Practical operation, therefore, dictates that the heat pump be used in conjunction with either an air or water heating system with actual distribution of the heat to the rooms through either air ducts, convection radiators, or panels.

In general, it is believed that the future of this device is very promising, and it is recognized that there are a number of practical and economical systems in operation at the present time. However, design data for the majority of types of systems are inadequate, and it is therefore recommended that the enthusiasm and the interest which have greeted the emergence of the heat pump from the category of a scientific toy to one of practical application, should be tempered with caution, since it will probably be several years before proper development and design information will enable the widespread satisfactory application of the heat pump. It is for this reason that no design information is presented at this time in THE GUIDE. A wealth of literature concerning individual installations is available, and reference to the many articles which have appeared in current technical magazines should form sufficient information for those interested in the development of this type of heating.

BASIC REFRIGERATION EQUIPMENT

Compression Refrigeration Machines

Compression of the refrigerant gas drawn from the evaporator may be accomplished by one of several means. Positive displacement may be used as in the reciprocating, rotary or gear types of compressors; centrifugal force may be applied as in the centrifugal compressor; an ejector may be used as in the steam jet refrigeration cycle; or absorption of a low pressure refrigerant gas in a secondary fluid, followed by the absorbent's release upon application of heat, may be utilized. A detailed discussion of the equipment required for each of these types of systems is beyond the scope of this chapter. For a more comprehensive treatment, reference may be made to the bibliography. The present discussion is limited to positive displacement reciprocating compressors, rotary compressors and centrifugal compressors.

Reciprocating Compressors

Reciprocating compressors may be classified according to (a) cylinder design, (b) compressor drive, (c) valves, and (d) lubrication and cooling.

Cylinder Design. Cylinder design may vary as to number, arrangement, and action (*i.e.*, single-acting or double-acting). Single-acting compressors usually have their cylinders arranged vertically, radially, or in a V or W shaped arrangement. Double-acting compressors, with refrigerant gas drawn in, and compressed on both the head and crank ends of the cylinder, are usually arranged horizontally. Reciprocating units are available with from one to sixteen cylinders with the V, W, or radial arrangements best adapted to the greatest numbers. The present trend is toward higher operating speeds with a low displacement per cylinder, together with an increase in the number of cylinders. Whereas the original reciprocating compressors were slow speed (50 to 55 rpm) steam driven devices, modern electric motor-driven compressors range up to 3500 rpm. Cylinder heads are usually bolted tight to the cylinders, but in some large compressors where there is danger of wet compression or of foreign materials entering the compressor space, a secondary head known as a safety head, may be seated at the end of the cylinder and held in position with heavy springs. Normally, this head remains stationary, but excessive pressures in the clearance space are relieved by movement of the safety head, and thus prevent damage to the cylinder.

Compressor Drives. Reciprocating compressors may be subdivided on the basis of source of motive power, and whether they are open or hermetic. Practically all modern compressors are electric motor-driven, although a few large, steam-driven compressors are still being installed where steam forms the most economical source of energy. In a few cases, as with truck transportation, the compressor may be driven by an internal-combustion engine.

The division of compressors into open or closed types is dependent upon whether the motive power is received from an external source, or whether the motor is direct drive and sealed within the housing. In the open type, power is received from an external source with one end of the compressor crankshaft extending through the crankcase, and usually V-belt driven. The point of emergence of the shaft from the crankcase forms a weak point of refrigerant leakage, and is most frequently sealed with a bellows type crankshaft seal. Horizontal double-acting compressors operate with a sliding piston rod, moving back and forth through a stuffing box. If the motor is direct-drive and enclosed within the compressor housing, the compressor is classified as closed or hermetic. This eliminates the necessity of any shaft seal, and not only prevents refrigerant leakage at this point, but reduces operating noise. One disadvantage is the inaccessibility of moving parts for repairs, but lubrication is greatly simplified since both the motor and compressor operate in a sealed space with the lubricating oil.

Compressor Valves. All refrigeration compressor valves are dependent for their operation upon a difference in pressure between the inside of the cylinder and the suction or discharge line. Although mechanically-operated valves might have some advantage, they have proved unsatisfactory because each change in the evaporator or condenser-operating pressures requires a change of valve setting. The pressure differentials required for operation of the valves depend upon the valve design and the compressor speed. The suction and discharge valves may be arranged with both located in the compressor head, or with the suction valve on the top of the piston and the discharge valve in the compressor head (uniflow arrangement). The valves themselves are usually classified as either poppet, ring-plate or flexing.

Lubrication and Cooling. Lubrication of modern compressors is accomplished by either splash lubrication or forced lubrication. The latter is used on large compressors, while simple splash lubrication is used in the smaller units.

Large compressors are usually water cooled with the water jacket either cooling the cylinder walls, or both the cylinder walls and the compressor head. Small compressors are either water cooled or air cooled with extended finned surfaces cast on

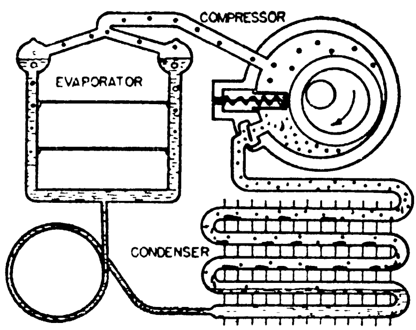

Fig. 9. Diagrammatic View of Rotary Compressor with Flooded Evaporator and Capillary Tube

the exterior of the cylinder. In a few cases small compressors may be found in which there is no attempt to add any purposive cooling other than through non-finned surfaces to the lower temperature air.

Water cooling is more effective than air cooling, but even under the best conditions cylinder cooling removes only a portion of the superheat in the refrigerant gas. This removal of heat from the cylinder results in some decrease in the work of compression, as well as reduction in condenser load.

Rotary Compressors

In recent years rotary compressors, usually hermetically sealed, have become quite popular for fractional tonnage applications and are being designed in increasingly larger sizes. Of the various designs attempted, the single blade rotary compressor, shown diagrammatically in Fig. 9, is the most popular. An eccentric driven rotor revolves within a housing in which the suction and discharge passages are separated by means of a sealing blade. When the rotating eccentric first passes this blade and the suction opening, the compressor suction space is very small. As the eccentric rotates, this crescent-shaped space becomes increasingly larger, thereby drawing in a charge of suction gas. When the eccentric again passes the blade, the gas charge is cut off from the suction inlet, compressed, and discharged from the compressor. Such rotary compressors are quiet in operation and reasonably free from vibration. In common with other types of hermetically sealed units, they have the advantages of comparatively low loss of refrigerant and sealed-in lubrication.

Centrifugal Compressors

Centrifugal compressors are used with very low pressure refrigerants; usually both evaporator and condenser work below atmospheric pressure. Water and monofluorotrichloromethane (F-11) are the refrigerants commonly used in centrifugal machines.

Compression of the refrigerant is accomplished by means of centrifugal force; therefore, this type of compressor is inherently suitable for large volumes of refrigerant at low pressure differentials. Two or more stages are usually required and high speeds are necessary to obtain good efficiency.

The evaporator is usually constructed as an integral part of the centrifugal type condensing unit, to chill water which is then circulated to the

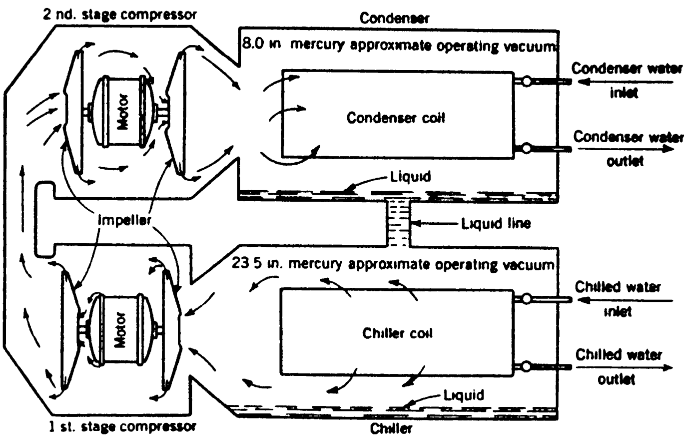

FIG. 10. ENCLOSED TYPE CENTRIFUGAL CONDENSING UNIT

air conditioning system. This is done because it would not be economical to pipe these large volumes of refrigerant any distance.

Centrifugal compressors, like reciprocating compressors, can be divided into two general types, open and enclosed. In general, the open type compressor is geared to the driving mechanism, and operates at higher speed than the driving motor or turbine. A modern, completely enclosed, direct-driven centrifugal compressor is illustrated in Fig. 10.

Centrifugal refrigeration compressors are particularly well suited to direct steam turbine drive because of their high operating speed. Water cooling equipment of one design is operated between 3500 and 4000 rpm for units developing 1000 to 2000 tons capacity, and from 7000 to 8000 rpm for units developing 100 to 200 tons capacity. However, a great many applications, particularly in the smaller sizes, are electric motor-driven and equipped with standard gear-type speed increasers. Centrifugal systems are particularly well adapted to large capacities (up to 3000 tons) although it is also possible to secure units as low as 50 tons in rating. Because centrifugal units operate best with refrigerants possessing a high specific volume, and because of the simplification of lubrication difficulties, they are frequently used for extremely low temperature applications. They are adaptable to a wide range of temperatures from −130 F to 50 F. One important advantage is their flexibility under varying loads, since units may be de-

signed to operate with reasonable efficiency at capacities as low as 20 percent of normal load.

Condensers

Condensers used for liquefying the refrigerant are of three general designs: (1) *air cooled,* (2) *water cooled,* and (3) *evaporative* (combination air and water).

1. *Air cooled condensers* are seldom used for capacities above 3 tons of refrigeration, unless an adequate water supply is extremely difficult to obtain, as, for instance, in railway air conditioning. Even on fractional tonnage installations, air is used as the condensing medium only where water is expensive, or where simplicity of installation warrants the higher condensing pressure and consequent power costs higher than would be obtained using water as the condensing medium.

The conventional air cooled condenser consists of an extended surface coil across which air is blown by a fan. The hot discharge gas enters the coil at the top and, as it is condensed, flows to a receiver located below the condenser. Air cooled condensers should always be located in a well ventilated space so that the heated air may escape and be replaced by cooled air.

The principal disadvantages of air cooled condensers are the power required to move the air, and the reduction of capacity on hot days. This loss of capacity, due to high condensing pressures on hot days, requires that equipment of increased capacity be selected to meet the peak load. Thus at normal loads the equipment is oversized. The principal advantages are low installation costs and simplicity, and for these reasons they are frequently used in small self-contained units.

2. *Water cooled condensers* are commonly used with compressors of one horsepower or larger in size, and they are found almost exclusively on large installations. They usually prove to be the most economical choice if an adequate water supply and means for its disposal are available. Although water cooled condensers may be of many designs, the shell and coil and the shell and tube are most commonly found in present day practice.

The amount and temperature of the condensing water determine the condensing temperature and pressure, and indirectly the power required for compression. It is therefore necessary to determine a balance so that the quantity of water insures economical compressor operation.

Because there is a decided tendency to conserve the water in city mains, and because most large cities are restricting the use of water for air conditioning and refrigeration equipment, it is often necessary to install cooling towers or evaporative condensers. Cooling towers, unfortunately, produce the warmest condensing water at the time when the load on the system is greatest, so that the refrigeration equipment must be designed to meet the maximum load at abnormal condensing water temperatures. If properly designed, this makes little difference in the efficiency of operation throughout the year, except at those times when the condensing water temperature is highest. As this occurs only for 5 percent of the entire cooling period, it can be disregarded as a factor in establishing yearly operating costs. For further information on cooling towers, reference may be made to Chapter 34.

3. *Evaporative condensers* were developed to alleviate the over-burdened water supply and drainage facilities of communities where many small air conditioning systems using water cooled condensers were applied. The adaptation of cooling towers to small installations is not practicable. The evaporative condenser combines the functions of the two by using a minimum amount of water on a finned surface, cooling it to approximately the wet-bulb temperature of the surrounding atmosphere.

The end view of a typical evaporative condenser is shown in Fig. 11. The fan draws the air over a finned tube condenser which is kept wet by a water spray. The discharge refrigerant gas from the compressor enters the top of the condenser coil, and the liquid refrigerant is drained from the bottom of the coil into a liquid receiver, and then circulates through the remaining portion of the system in the usual way.

The water is circulated through the spray nozzles, and the level is maintained in the sump by means of a float valve. The eliminator plates are placed in the path of the water-air mixture so as to remove the entrained water. The air leaving the unit is almost completely saturated, so that care must be taken in locating discharge ducts to prevent condensation.

Evaporative condensers are available in sizes up to 100 tons or more. These units use only a small portion of the water required for a water cooled condenser. The water is vaporized by the heat of the refrigerant so that each pound of water used extracts approximately 1000 Btu from the refrigerant, whereas under standard rating conditions where the water temperature rise is 20 F, each pound of water extracts only 20 Btu from the refrigerant. Including the water lost by entrainment

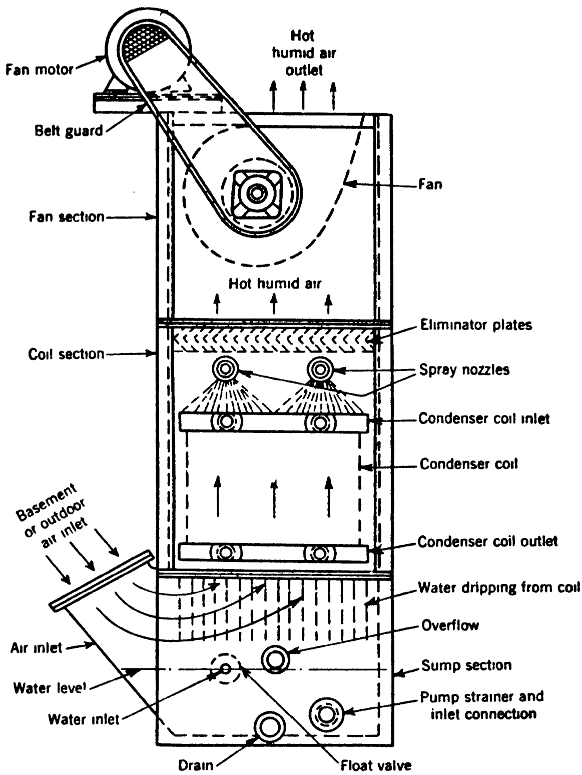

FIG. 11. SCHEMATIC VIEW OF AN EVAPORATIVE CONDENSER

in the discharge air, by overflow and stand-by evaporation, the water used is about 3 to 5 percent of the amount that would be required for a water cooled condenser.

The evaporative condenser requires more maintenance, occupies greater space (must be located where air is available), and has a higher first cost than the water cooled condenser, but where the use of water is restricted or expensive, the evaporative condenser has become widely accepted. Compared with a water cooled condenser and cooling tower, which combination uses about the same quantity of water, the evaporative condenser has the advantage of lower cost and smaller space requirements.

Evaporators and Coolers

Refrigeration evaporators must be designed for efficient removal of heat from the medium being cooled, as well as effective boiling of the refrigerant and a minimum drop of pressure through the coil. There are two general types of evaporators, dry and flooded. In the dry evaporator the refrigerant enters in the liquid state, and the design provides for complete evaporation with the vapors leaving slightly superheated. In flooded evaporators not all of the refrigerant is evaporated, the liquid-vapor mixture leaving the evaporator flows into a surge drum from which the vapors are drawn into the compressor suction line, and the liquid is recirculated through the evaporator.

The types of coolers used in connection with air conditioning work fall into three general groups: (1) direct water coolers, (2) direct air coolers, and (3) brine coolers for circulation of the brine in a closed system, and thus cooling indirectly either water or air.

1. *Water coolers.* One method of the direct cooling of water is to install direct expansion coils in the spray chamber so that the water sprayed into the air comes in direct contact with the cooling coils. Another common and efficient method of cooling spray water is to use a Baudelot type of heat absorber where the water flows over direct expansion coils at a rate sufficiently high to give efficient heat transfer from water to refrigerant.

Another type of spray water cooler is the shell and tube heat exchanger in which the refrigerant is expanded into a shell enclosing the tubes through which the water flows. The velocity of the water in the tubes affects the rate of heat transfer, and as the refrigerant is in the shell completely surrounding the tubes at all times, good contact and a high rate of heat transfer are insured. The disadvantage of such a system is that with the falling off of load on the compressor, the suction temperature or the temperature in the evaporator drops, and there is a possibility of freezing the water in the tubes, which, of course, might split the tubes and allow the refrigerant to escape into the water passage. This danger can be eliminated by automatic safety devices.

Another system of cooling spray water is to submerge coils in the spray collecting tank, or in a separate tank used for storage. The heat transmission through the walls of the coils, however, is low and a great deal more surface is required than for any other type of cooler. However, with large storage tanks this type of cooling can be utilized to advantage.

2. *Air coolers.* When direct cooling of air is employed, the refrigerant is inside the coil and the air passes over it. Cooling depends upon convection and conduction for removing the heat from the air. The type of coil used can be either smooth or finned, the finned coil being more economical in space requirement than the smooth coil. The fins, however, must be far enough apart so as not to retain the moisture which condenses out of the air.

When refrigeration evaporators are used for cooling air or other gases by forced convection, they are usually termed blast coils or unit coolers. A blast coil may be placed in a duct or in an assembled unit, and the air forced across the coil and discharged through distributing ducts or directly into the space to be conditioned. Unit coolers, designed much like unit heaters, consist of a finned coil, propeller fan, and controls suspended directly in the space to be cooled.

3. *Indirect brine coolers.* The indirect cooler, where brine is cooled by the refrigerant and the resulting cold brine is used to cool either air or water, introduces several other considerations. It is not the most economical from a power consumption standpoint, as it is necessary to cool the brine to a temperature sufficiently low so that there is an appreciable difference between the average brine temperature and that of the substance being cooled. This requires that the temperature of the refrigerant must be still lower, and consequently the amount of power required to produce a given amount of refrigeration increases due to the higher compression ratio. There are other considerations which make such a system desirable. In the first place, where a toxic refrigerant is undesirable or cannot be used because of fire or other risks, especially in densely populated areas, the brine can be cooled in an isolated room or building and can then be circulated through the air conditioning equipment. This arrangement eliminates any possibility of direct contact between the air and refrigerant.

REFRIGERATION CONTROL

In addition to the compressor, evaporator, and condenser, several auxiliaries are required for proper operation of a refrigeration system. Some device must be supplied for the controlled expansion of the refrigerant from the high condenser pressure to the low evaporator pressure; controls are required for the on-off operation of the compressor, the flow of the condensing medium, and for safety devices; proper piping is required for connecting the various portions of the systems. Where refrigerating apparatus is used for the cooling of rooms, additional controls are required.

Expansion Devices

Some form of expansion device must be provided to control the rate of flow of the liquid refrigerant between the high and low side pressures of the system. This device is usually an expansion valve and may be either manual or automatic; however, with few exceptions, manual valves are obsolete and no longer used.

Automatic Expansion Valves. An automatic or pressure controlled expansion valve operates to maintain a constant pressure in the evaporator. The liquid refrigerant passes through an orifice, the opening size of which is controlled by means

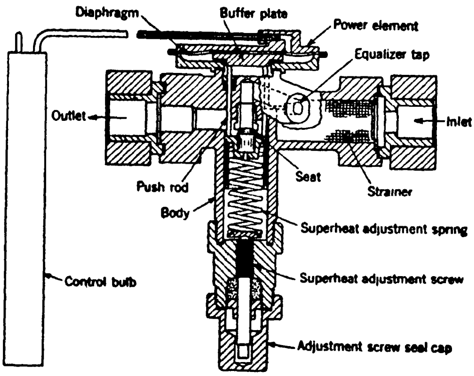

FIG. 12. TYPICAL THERMOSTATIC EXPANSION VALVE

of a needle valve connected to a flexible bellows. This bellows expands or contracts with variations in the evaporator pressure transmitted to the expansion chamber through the refrigerant outlet from the evaporator. The position of this needle valve is controlled by the degree of compression in an adjustable spring, balanced against the bellows, and these two forces operate to maintain a constant pressure in the evaporator by increasing or decreasing the flow of liquid refrigerant. Such an expansion valve is usually applied to evaporators of the direct expansion type, but is not satisfactory for fluctuating loads such as are encountered in air conditioning installations.

Thermostatic Expansion Valves. A thermostatic expansion valve controls the flow of liquid refrigerant to the evaporator so as to maintain the entire coil filled with evaporating refrigerant, and to keep a constant superheat in the refrigerant gas leaving the coil. The construction of such a valve is shown in Fig. 12 and is similar to that for an automatic expansion valve but incorporates, in addition, a power element responsive to changes in the degree of superheat of the refrigerant gas leaving the coil. This power element consists of a bellows connected by means of a capillary tube to a feeler bulb fastened to the suction line from the evaporator. The bulb, bellows, and tube are usually charged with the same liquid refrigerant used in the evaporator itself. A starved condition in the evaporator results in a greater superheat in the gas leaving the evaporator, and this in turn operates through the power element to increase the flow of liquid refrigerant. A flooded evaporator reduces the discharge superheat, and thus tends to reduce the flow of liquid refrigerant. Such an expansion valve is satisfactory for operation with fluctuating loads since this type of control tends to keep the evaporator filled with refrigerant at all times.

Low-Side Float Valves. A liquid refrigerant control of the low-side float valve type consists of a ball float located in either a receiver or the evaporator itself on the low pressure side of the system. A needle valve, operated through a simple lever mechanism attached to the float, permits the passage of more or less refrigerant, as the level in the receiver or the evaporator fluctuates. Such a control must be used in conjunction with a flooded evaporator, and has been applied extensively to household refrigerators and, to some extent, in commercial and industrial installations.

High-Side Float Valves. A high-side float valve differs from a low-side float valve in that the float is located in a receiver or container on the high pressure side of the system. Proper operation again depends upon metering of the refrigerant through a controlled opening, depending upon the level of the liquid refrigerant in the container. Such a control has the disadvantage that the evaporator must be placed directly adjacent to the float container, or some intermediate pressure device must be applied to prevent flashing of the refrigerant upon pressure drop.

Capillary Tubes. A capillary tube may be used as a liquid refrigerant expanding device. Such a device consists of an extremely small bore tube (in the order of 0.04 inch in diameter) of five to twenty feet in length. Although such a restricting device operates as a very simple means of expanding the liquid refrigerant, it has the disadvantage that no modifications are possible to adjust the rate of expansion under various operating conditions. The bore and length of the tube, as well as the proportions of the rest of the system, are critical. It is for these reasons that its application has been limited to factory assembled domestic and commercial units.

Refrigeration Control Devices

In addition to automatic control of expansion of the liquid refrigerant, a completely automatic refrigeration system requires (1) some means for on-off operation of the compressor motor, (2) control of the flow of the condensing medium, and (3) safety devices for prevention of damage to the equipment. In addition, special controls designed for specific applications are frequently required. The various types of devices used to accomplish these purposes are so numerous that it would be impossible to describe all of their modifications. Only the general purposes and operating characteristics of the more typical mechanisms are here discussed.

Compressor Motor Controls. Two types of controls are used for intermittently starting and stopping compressors. The first of these is a pressure motor control responsive to the evaporator pressure, and the second a thermostatic motor control responsive to the temperature of the load surrounding the evaporators. In the first case the compressor operation is indirectly dependent upon the temperature of the load, and is controlled by the refrigerant pressure at the point of control location. The second type is dependent upon the temperature of the load being cooled.

With the pressure actuated device, the control is frequently located directly on the condensing unit, and the low pressure in the suction line or the crankcase of the compressor is used to control motor operation. Such a control usually consists of a low pressure bellows, connected through tubing directly to the low pressure control source, and an electrical switch operated through linkage by the movement of the bellows. The electrical circuit is closed on rising pressure, and opened on falling pressure. The thermostatic type of motor control is usually similar in construction to the pressure control, with the exception that a temperature bulb and capillary tube replace the pressure line, and the temperature bulb is located adjacent to the evaporator itself. In this case motor control is directly responsive to changes in the temperature of the load surrounding the evaporator. Frequently, a high pressure safety cutout switch is combined with the motor control, and operates to cut off the power from the motor in case the high side pressure exceeds a predetermined limit.

Solenoid Valves. Solenoid or magnetic stop valves are frequently used in refrigeration systems for control of gas and liquid flow. When applied as liquid stop valves, they are placed in the liquid line between the condenser or receiver and the evaporator, and the line is open to passage of the refrigerant only when the compressor is in operation. When the compressor is not in operation, leakage of liquid refrigerant in the evaporator is prevented. In some cases such a magnetic stop valve is operated directly by a thermostat located at the point of the load, and the compressor motor operation is controlled independently by a low pressure switch.

Magnetic liquid stop valves are also widely used for the control of the refrigerant flow to individual evaporators in a multiple evaporator system operated by one compressor. In some installations magnetic liquid line and suction line valves are used to isolate completely an evaporator for defrosting purposes. A magnetic valve may be installed in a by-pass around one or more cylinders of a multiple cylinder compressor, and thereby be used to unload a compressor during starting to reduce load. Additional applications are found in control of the circulation of brine in a secondary refrigeration system.

Suction Pressure Valves. Suction pressure control valves, frequently called back pressure regulators or two-temperature valves, are sometimes placed in the suction line to prevent the evaporator pressure and temperature from dropping below a predetermined level. Typical applications of such controls occur in water cooling or milk cooling systems, where freezing and other damage would result if the evaporator pressure dropped too low, or in multiple systems where several evaporators are supplied by one condensing unit. Thus, different evaporators may be kept at different temperatures by maintaining a pressure drop between the evaporator and the suction line.

Condensing Water Control. The majority of the refrigeration systems, other than fractional horsepower, use water cooled rather than air cooled condensers, since the lower condensing temperatures result in more economical operation. Automatic control of the water flow to the condenser must be maintained if water wastage is to be eliminated. Such control may be provided through the use of either an electric solenoid water valve or by means of a pressure control valve. With a solenoid valve, the flow is two-position, either *off* or *on*, and its operation is simultaneous with starting and stopping of the compressor motor. With a pressure operated valve, the flow is modulated and is dependent entirely upon condenser pressure rather than condensing unit operation. Similar water valves controlled thermostatically by the temperature of the water discharged from the condenser are also available.

Safety Controls. Many controls are designed not to aid in proper operation of the system, but to prevent damage in case of improper operation. One such safety control is the high-pressure cutoff frequently combined with the low-pressure motor control as previously described. Another safety control often used is a low voltage cut-off which shuts down the system automatically in case the line voltage drops below a minimum value. High pressure relief valves are used for safety purposes to prevent damage in case excessive condensing pressures are encountered.

Refrigeration Control for Air Conditioning Equipment

When refrigerating equipment is used for space cooling, two major control problems exist: one is control of the temperature and the other, control of the humidity. In some applications the amount of latent heat to be removed is small compared with the sensible heat. In such cases, sufficient dehumidification will usually occur without any special provisions. In other cases, such as theaters, where the latent load is relatively high, the air must be cooled below its dew-point temperature, and sometimes re-warmed to return it to the comfort range. Chapter 38, Automatic Control, discusses, among other topics, control systems for unit coolers, well water and ice cooling systems, central fan systems, and all-year air conditioning systems.

REFRIGERATION PIPING

The pressure drop which occurs during passage of the refrigerant through connecting piping is similar in effect to that which occurs through suction and discharge valves of the compressor. Thus, the effect of the pressure drop in the suction line between evaporator and compressor requires that a lower pressure be maintained inside the compressor during suction than is maintained in the evaporator. The pressure drop through the connecting piping between the compressor and the condenser requires that a higher pressure be maintained inside the compressor during discharge than in the condenser. These losses result in a greater compression ratio, and therefore greater power requirements, as well as a lower volumetric efficiency and higher displacement requirements. Pressure losses in the liquid line between condenser or receiver and the expansion valve may result in some flashing of the liquid refrigerant, unless the liquid is subcooled. In all cases, frictional losses should be kept to a minimum, and piping should be selected which will give the smallest loss consistent with overall economy in the system.

Refrigerant liquid lines from the receiver to the expansion valve should be preferably designed with a pressure drop of less than 5 psi, and with 10 psi as the maximum. A velocity of 100 to 250 fpm is recommended to prevent a pressure drop great enough to cause vaporization of the refrigerant ahead of the expansion valve. If the evaporator is to be located at a higher elevation than the condenser or receiver, account should be taken

Table 5. Freon-12 Liquid Lines, Tons Capacity per 100 Ft Equivalent Length

Line Size, Inches	Pressure Drop per 100 Ft Equivalent Length, Psi			
	3	5	10	20
3/8 OD	0.88	1.14	1.80	2.58
1/2 OD	2.89	3.64	5.56	8.50
1/2 IPS	4.86	6.81	10.2	15.8
5/8 OD	4.86	6.81	10.2	15.8
3/4 IPS	9.73	12.6	18.5	27.0
7/8 OD	10.5	14.1	21.8	33.0
1 IPS	21.4	28.2	41.3	60.8
1 1/8 OD	21.4	28.2	41.3	60.8
1 1/4 IPS	36.9	48.1	70.5	101.
1 3/8 OD	36.9	48.1	70.5	101.
1 1/2 IPS	62.0	80.2	114.	160.
1 5/8 OD	62.0	80.2	114.	160.
2 IPS	124.	161.	231.	328.
2 1/2 IPS	230.	297.	426.	607.
3 IPS	364.	469.	676.	972.
3 1/2 IPS	539.	704.	1005.	1430.
4 IPS	753.	972.	1385.	1945.

Note: Tonnage values above those underlined give velocities of 300 fpm or less.

of the pressure drop for each foot of static liquid lift. Approximate values are 0.26 psi per foot for ammonia, 0.57 psi per foot for Freon-12, 0.51 psi per foot for Freon-22, and 0.64 psi per foot for Freon-11. Where there is a possibility of vaporization of some of the liquid before reaching the expansion valves, means for subcooling should be provided.

Since a reduction of suction pressure at the compressor results in an appreciable reduction in capacity and more power input per ton of refrigeration, great care should be given to the proper sizing of suction lines between the evaporator and the compressor. Although comparatively high velocities, 500 to 5000 fpm, may be used, the optimum value will depend upon the refrigerant and the operating pressure range. Since return of the oil to the compressor must be considered in the case of Freon and methyl chloride, for these refrigerants the minimum velocity should be 500 fpm for horizontal runs and 1000 fpm for vertical runs. For the Freons, the usual design velocities range between 1000 and 2000 fpm. Too high velocities create noise problems and excessive pressure drops. The total

TABLE 6. MAXIMUM TONS OF COMPRESSOR CAPACITY FOR FREON-12 LINES
(Only for temperatures indicated)

Line Size, Inches	Suction Lines Based on 105 F Condensing Temperature — Psi Pressure Drop per 100 Ft Equivalent Length at 40 F Saturation						Discharge Lines — Condensing Temperature	
	½	1	2	3	4	5	115 F	90 F
½ OD	0.14	0.20	0.28	0.35	0.41	0.45		
⅜ IPS	0.17	0.24	0.34	0.42	0.49	0.54		
⅝ OD	0.25	0.35	0.51	0.62	0.73	0.81	1.43	1.15
½ IPS	0.35	0.45	0.65	0.79	0.93	1.03	1.87	1.50
⅞ OD	0.55	0.76	1.10	1.34	1.58	1.75	2.97	2.38
¾ IPS	0.68	0.94	1.35	1.65	1.92	2.12	3.26	2.62
1⅛ OD	1.26	1.80	2.57	3.17	3.76	4.15	5.05	4.05
1 IPS	1.43	2.01	2.89	3.54	4.17	4.60	5.29	4.25
1⅜ OD	2.21	3.12	4.45	5.50	6.38	7.05	7.72	6.19
1¼ IPS	2.70	3.82	5.37	6.72	7.68	8.48	9.16	7.35
1⅝ OD	3.40	4.78	6.79	8.42	9.77	10.8	10.92	8.75
1½ IPS	4.05	5.75	8.10	10.12	11.6	12.8	12.5	10.0
2⅛ OD	6.12	8.60	12.1	15.1	17.4	19 2	19.2	15.3
2 IPS	7.66	10.9	15.3	19.2	32.2	24.5	20.6	16.5
2⅝ OD	12.0	17.1	24.0	30.1	34.6	38.2	32.2	25.9
2½ IPS	12.0	17.1	24.0	30.1	34.6	38.2	32.2	25.9
3⅛ OD	19.1	27.2	38.2	47.8	55.0	60.7	51.5	39.8
3 IPS	20.9	29.4	42.3	51.8	60.0	66.2	54.5	43.8
3⅝ OD	27.8	39.7	55.7	69.8	80.3	88.7	72.0	57.6
3½ IPS	30.2	43.2	61.0	76.1	87.0	96.0	78.8	63.3
4⅛ OD	38.6	55.2	78.0	97.3	111	123	95.8	77.1
4 IPS	40.7	58.6	83.0	103	118	130	101.6	81.6
5 IPS	71.3	100	141	176	203	224	171.5	137.8
6 IPS	126	183	257	322	366	403	266	214
8 IPS	211	297	422	523	602	664	461	370
10 IPS	352	503	712	887	1024	1130	725	582
12 IPS	550	780	1106	1373	1582	1748	1041	836

pressure drop in the suction line should be between one and two psi, if the velocity can be kept to within the specified limits.

Compressor discharge or hot gas lines may be designed with velocities from 1000 to 5000 fpm, except for dense gases such as carbon dioxide, where noise considerations will reduce the upper limit. A pressure drop of 2 to 4 psi is recommended for the discharge lines. Extensive tables are available in the literature for the determination of pressure drops through refrigerant lines with various refrigerants. The capacities listed in Tables 5, 6 and 7 are published in ACRMA Equipment Standards-1946, of the *Air Conditioning and Refrigerating Machinery Association*, and are used by permis-

TABLE 7. APPROXIMATE SUCTION-LINE CAPACITY FACTORS FOR EQUAL PRESSURE DROP OF FREON-12

Saturated Suction Temperature, F.	50	40	30	20	10	0	−10	−20
Factor	1.09	1.00	0.92	0.86	0.80	0.74	0.66	0.56

sion. Table 5 shows the tonnage capacity normally allowed for Freon-12 liquid lines per foot equivalent length of pipe, and Table 6 the maximum tonnage for suction and discharge Freon-12 lines. Table 7 presents suction-line capacity factors for equal pressure drop.

ACCESSORIES

Dehydrators, oil separators, strainers, vibration eliminators, sight glasses, and various types of valves are accessories frequently needed for the proper

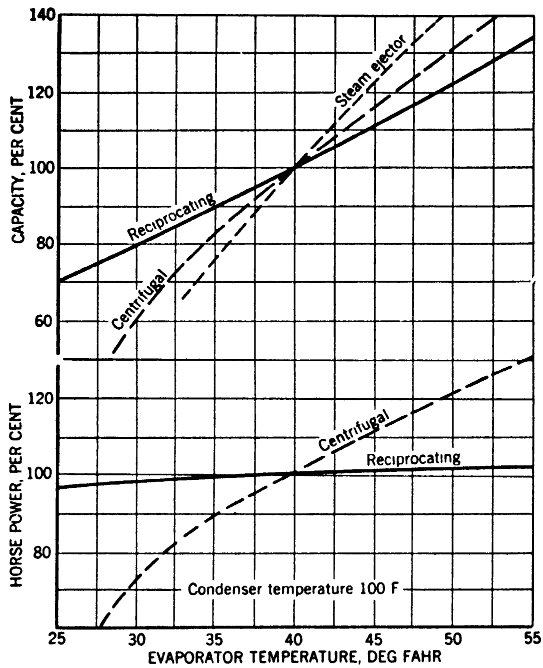

FIG. 13. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

installation and operation of refrigeration systems. Refrigerant-line dehydrators or dryers usually consist of copper containers fitted with tubing connections at either end, and contain a desiccant such as silica gel, activated alumina, or calcium chloride. The liquid refrigerant is circulated through the dryer during operation of the system, and the moisture content of the refrigerant charge is thus kept to a minimum.

Oil separators are installed between the compressor and condenser to prevent excessive oil removal from the compressor crankcase and its passage into the condenser and evaporator. The oil is separated from the gaseous refrigerant by gravity during its passage through a chamber of sufficient size to reduce the velocity. A float-operated valve maintains a maximum oil level in the separator, and additional oil is forced by pressure difference through a line back to the crankcase.

Screen strainers are frequently installed in the liquid line piping before solenoid valves and expansion valves, as well as before regulating valves

in water lines leading to water cooled condensers. Sight glasses which permit visual inspection of the condition of the refrigerant are sometimes installed on factory assembled commercial unit systems. It is particularly advisable to place such a fitting before the expansion valve, if the evaporator is located above the condenser.

Flexible vibration eliminators, usually consisting of a bellows design covered with woven copper wire, are sometimes installed in copper lines where units such as compressors are installed on flexible mountings, or where vibration is otherwise a problem. Packed or packless shut-off valves are necessary where it may be required to isolate portions of a system.

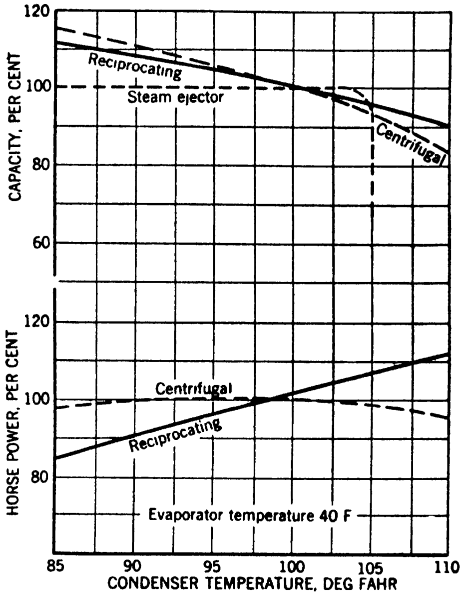

FIG. 14. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

EQUIPMENT CHARACTERISTICS AND SELECTION

The various types of compression systems have quite different characteristics of capacity and power with varying evaporator and condenser temperatures, as may be noted from curves in Figs. 13 and 14.

From Fig. 13 it may be observed that power requirements for the centrifugal compressor increase much more rapidly than for the reciprocating compressor, with increase in evaporator temperature. Similarly, the capacities of the steam ejector and centrifugal compressors increase more rapidly than those of the reciprocating compressor with increase in evaporator temperature. Thus, both the steam jet and centrifugal machines tend to be more self-regulating than the reciprocating. It is also evident from Fig. 13 that the steam jet equipment is best suited for operation at high evaporator temperatures.

The effect of condenser temperature upon the power and capacity of the different types of compressors is shown in Fig. 14. It may be noted

that the power required by the reciprocating compressor increases rapidly with increase in condenser temperature, while the power curve for the centrifugal compressor is relatively flat. It is also evident that the capacity of the steam jet compressor is independent of condenser temperature until a certain point is reached, where it drops to zero. As previously stated, steam jet equipment requires more condensing water than other types of compression systems. Consequently, steam jet systems are well suited to those applications where condensing water is cheap, or where condensing water is rather high in temperature.

The selection of proper refrigeration equipment for any air conditioning

TABLE 8. BASIS OF EQUIPMENT SELECTION

CAPACITY TONS	MAJORITY USED	SOME USED	FEW USED
0 to 5	Unit systems in conditioned space.	Unit central systems using duct distribution.	Built up central systems.
5 to 25	Built up central systems using reciprocating compressors.	Unit central systems using duct distribution.	Unit systems in conditioned space. Built up systems using absorption and adsorption systems.
25 to 50	Built up central systems using reciprocating compressors.	Built up central systems using centrifugal compressors.	Central systems using adsorption systems.
50 to 400	Built up central systems using reciprocating compressors.	Built up central systems using steam jet and centrifugal compressors.	
400 and Over	Built up central systems using centrifugal compressors.	Built up central systems using steam jet.	

job is of utmost importance for satisfactory results. The most important factors in the selection of the equipment are:

1. Loads (as determined by the conditions of the space to be cooled).
2. Economics (both initial and operating costs).
3. Codes (local safety codes must be adhered to and influence the type of system to be used).

A broad division of equipment to be used for a particular installation or application may be made on the basis of the magnitude of the load. Current general practice is outlined in Table 8.

Unit or *packaged* systems, consisting of a reciprocating compressor, condenser, evaporator, and fans, are generally used in the smaller sized jobs where electric power is available, as they are manufactured complete, ready to install, and are the most economical (see Chapter 25).

The reciprocating compressor in the built-up central system (see

Chapter 29) covers the widest range of application since it is applicable to either the direct expansion or indirect systems, and can be driven by steam or gas engines, or by electric motors. The quantity of condensing cooling medium required is also less than for any other system, with the exception of the centrifugal compressor, which uses the same amount.

Centrifugal compressors are used for large installations, and usually where the indirect system is required. The driving mechanism can be a steam turbine or electric motor. The steam jet system is used where steam is available and cooling water can be had in large quantities.

It will be noted by referring to Fig. 13 that all systems using compressors have a common characteristic, namely, that the capacity varies with the evaporating temperature. Not only can the equipment be selected to

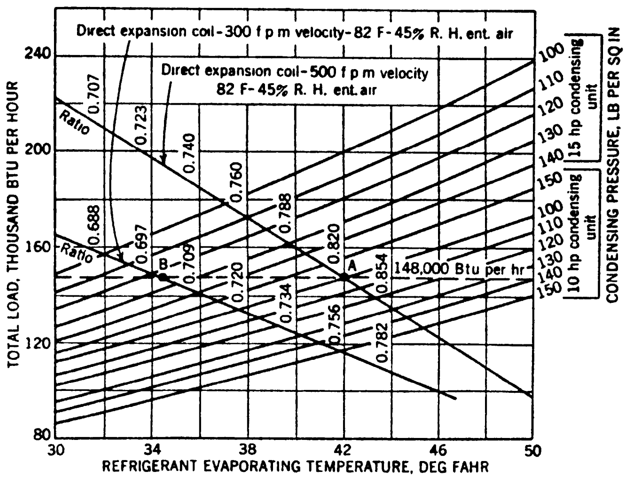

Fig. 15. Compressor and Coil Performance

produce a given result, but the performance can be predicted under varying load conditions by the simple expedient of using the variable of evaporating temperature as the abscissa, and the load or capacity as the ordinate in a series of curves.

Manufacturers of compressors and cooling coils furnish performance data for apparatus that can be plotted in the form of curves similar to those shown in Fig. 15. The performance of a compressor is plotted as a series of curves, each curve being drawn for a given condensing pressure. The performance of a direct expansion coil at two different air velocities is plotted on the same graph. The operating point will be, of course, where the two curves cross.

Data given in Table 9 illustrate two types of conditioned enclosures having the same total load of 148,000 Btu per hour, but with two different ratios of sensible to total heat. In the case of the office with a ratio of 82 percent sensible to total heat, the operating point *A* in Fig. 15 is found to be 42.2 F evaporating temperature, with a face velocity of 500 fpm. In the case of the restaurant, with a ratio of 69.5 percent sensible to total heat, the air velocity is lowered to 300 fpm, and the evaporating tempera-

TABLE 9. TYPICAL OPERATING CONDITIONS FOR TWO TYPES OF LOAD

Type of Enclosure	Load, Btu per Hour			Ratio Sensible to Total	Air Entering Coil		Operating Balance Point		
	Sensible	Latent	Total		F Deg	Per Cent R.H.	Evaporator Temp F Deg	Condenser Pressure Lb per Sq In.	Per Cent Sensible Heat
Restaurant....	103,000	45,000	148,000	0.695	82	45	34.4	123	69.9
Office . .	121,000	27,000	148,000	0.820	82	45	42.2	100	82.1

ture is lowered to 34.4 F as shown in point *B* of Fig. 15. In order to obtain the same capacity, a larger condensing unit is used. This illustration assumes zero pressure drop through the suction line. The pressure drop can be taken into account by shifting the compressor performance curves by the amount of pressure drop expressed in Fahrenheit degrees.

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CHAPTER 37

DEHUMIDIFICATION BY SORBENT MATERIALS

Definitions and Principles, Adsorbents, Dehumidification by Solid Adsorbents, Dehumidification Equipment Using Solid Adsorbents, Absorbents, Dehumidification by Liquid Absorbents, Dehumidification Equipment Using Liquid Absorbents, Calculation of Moisture Load, Vapor Transfer to Dehumidified Space

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DEHUMIDIFICATION as used herein is the reduction of the water vapor content of a given volume of air or other gas. The term thus describes a special case of dehydration which covers the removal of moisture in any form from matter. The degree of dehumidification required varies greatly with different applications, and is one of the prime considerations influencing the choice of a method. Dehumidification may be accomplished by latent heat removal, together with sensible heat removal, as described in Chapters 29, 34, and 35, or by the use of sorbents.

Sorbents are substances which have the property of extracting and holding other substances (usually gases or vapors, *e.g.*, water vapor), brought into contact with them. All materials are sorbents to a greater or lesser degree. The weight of water held by a substance will increase or decrease, depending upon whether the vapor pressure of the water held by the substance is less or greater, respectively, than the partial pressure of water vapor in the surrounding atmosphere. As generally used, however, the term sorbents refers to those materials having a capacity for moisture which is large compared to their volume and weight. Such materials are divided into two general classifications:

1. *Adsorbent*—A sorbent which does not change physically or chemically during the sorption process. Certain solid materials, such as activated alumina, silica gel, activated bauxites, and activated charcoal have this property. The action of adsorbents, most of which adsorb some gases and condensible vapors besides water vapor, is selective. Thus, in the case of a mixture containing both water and organic vapors, silica gel would remove the water vapor in preference to the organic vapors, while the reverse would be true in the case of activated carbon. The selective property of adsorbents is made use of in some instances for the removal of objectionable and contaminating vapors from an air or gas mixture. (See Chapter 8.)
2. *Absorbent*—A sorbent which changes either physically, chemically, or both, during the sorption process. Calcium chloride is an example of a solid absorbent, while liquid absorbents include solutions of lithium chloride, calcium chloride, lithium bromide, and the ethylene glycols.

ADSORBENTS

The ability of an adsorbent to remove water vapor from a gas is explained by the fact that the vapor pressure of the water in the adsorbent (when in the reactivated condition) is less than the partial pressure of the water vapor in the surrounding atmosphere. For instance, when an active adsorbent is brought into contact with a gas of high humidity, there is a tendency for the vapor pressure of the water in the adsorbent to reach equilibrium with the partial pressure of the water in the surrounding gas, with the result that water is extracted by the adsorbent and its weight increased, while the moisture content of the gas is correspondingly reduced. (The adsorbent is said to be saturated for a given condition when equilibrium is attained.) The weight of water a given adsorbent will extract is dependent upon the relative humidity (ratio of the partial pressure in the gas to the saturation pressure at a given temperature) and the temperature of the adsorbent. The process is reversible; if the temperature of the adsorbent is raised until the vapor pressure of the adsorbed water becomes greater than the partial pressure of the vapor in the surrounding atmosphere, water will be released by the adsorbent. After the adsorbent cools to room temperature, for instance, the vapor pressure of the water in the adsorbent falls below the partial pressure of the vapor in the atmosphere, and the adsorbent will again start extracting water. The elimination of water by the addition of heat is known as reactivation, and is a means of regenerating the adsorbent so that it may be used repeatedly.

Adsorption is proportional to the amount of surface (internal and external) of the sorbent. The materials that are used commercially as solid adsorbents have a porous structure of sub-microscopic dimensions, which gives them extensive surface area. An adsorbent should meet the following requirements in order to be satisfactory for dehumidification purposes:

1. Have a high adsorptive capacity under normal atmospheric conditions.
2. Be chemically stable, resisting contamination from impurities.
3. Be physically rugged to resist breakdown from handling and use.
4. Be capable of reactivation at temperatures generally obtainable.
5. Be heat-stable at reactivation temperatures.
6. Have a weight per unit volume such as to avoid excessive bulk.
7. Be available at reasonable cost.

Activated Alumina

Activated alumina is a granular porous material which removes by adsorption substantially 100 percent of the moisture from gases, vapors, and certain liquids. Regeneration or reactivation may be accomplished by employing a heating medium at temperatures ranging from 350 to 600 F. After many cycles of adsorption and reactivation it is substantially as effective as originally, and retains its original size and shape.

Activated alumina is produced by chemically controlled precipitation from a sodium aluminate solution resulting from the extraction of alumina from bauxite by the Bayer process. By subsequent processes this precipitate is converted into a highly porous adsorptive material. It is low in iron and silica, each normally less than 1/10 of 1 percent. Commercially produced material is uniform in analysis and physical form.

Activated alumina is a partially dehydrated aluminum trihydrate containing about 7 percent water and small amounts of soda, oxides of iron, silicon, and titanium, as well as very minor amounts of other elements indicated spectrographically. Substantially all of the soda is combined with silica and alumina as an insoluble constituent.

Activated alumina is inert chemically to most gases and vapors, is nontoxic, and will not soften, swell or disintegrate when immersed in water. High resistance to shock and abrasion is one of its more important physical characteristics. Commercial sizes range from a powder passing through 300 mesh screen to particles 1 in. in diameter. The sizes commonly used are 8–14 mesh and $\frac{1}{4}$ in. to 8 mesh. The average weight for most forms is

50 lb per cubic foot. Its high degree of purity warrants classification among commercially pure chemicals.

Silica Gel

Silica gel is a prepared form of silicon dioxide (silica) having an extremely porous structure which makes it an efficient adsorbent. It is made by mixing predetermined concentrations of an acid, such as sulfuric acid, and a soluble silicate, usually sodium silicate, and allowing the mixture to set to a jelly-like mass called hydrogel. The product takes its name from its condition as a colloid at this stage of its manufacture. After setting, the hydrogel is broken into small lumps, washed, dried, crushed, and screened to the desired particle sizes and then given a final heat treatment or activation. The surface area of silica gell has been found to be in excess of 50,000 sq ft per cubic inch of product. Silica gel has high adsorptive capacity per unit weight, and may be reactivated repeatedly at temperatures up to 600 F. Reactivation is generally accomplished by blowing gases through the silica gel at approximately 300 F, or by heating in a well vented oven maintained at this temperature until no more moisture is given off. Silica gel is a high purity, rugged, non-toxic, heat-stable material, having a specific heat of 0.2, and is most inert. There is no change in the size or shape of the particles as it becomes saturated, and no corrosive or injurious compounds are given off. It is available commercially in a number of grades, ranging in particle size from a 3 to 8 mesh product to an impalpable powder passing through a 325 mesh screen. The product generally used for dehumidification applications has a particle size of 6 to 12 mesh, and a bulk density of between 40 and 45 lb per cubic foot.

Activated Bauxite

Activated bauxites are certain natural products which, after controlled heat treatment, have properties which make them suitable for use as solid adsorbents. They are marketed under different trade names by several processors. The activated bauxites consist primarily of Al_2O_3, Fe_2O_3, SiO_2, TiO_2, and H_2O in varying percentages. The surface area, adsorptive capacity, and other properties of the several products differ to some extent, depending upon the source of the original material and its subsequent treatment. The available activated bauxites are durable products having a specific heat of about 0.24, and can be regenerated at temperatures between 300 F and 500 F. They are usually supplied in a number of particle sizes, and have a bulk density of between 55 and 65 lb per cubic foot.

There are other solid substances having marked adsorbent properties, but details concerning them are not available.

DEHUMIDIFICATION BY SOLID ADSORBENTS

Since adsorption is primarily a condensation process, heat, equivalent to the latent heat of evaporation of the vapor, plus the heat of wetting (which is an additional amount of heat, depending upon the vapor being adsorbed and the adsorbent used) is liberated. The sum of the latent heat of evaporation and the heat of wetting is known as the heat of adsorption. During adsorption it might be said that latent heat is transformed into sensible heat, which is dissipated into the adsorbent, into the metal of the adsorbent container, and into the gas mixture, resulting in a rise in temperature. Fig. 1 shows the relationship between temperature, vapor pressure, and moisture content of a solid adsorbent. These curves indicate the general performance of solid adsorbents, although the exact values vary for the different adsorbents, and may vary even for different types of the same compound. The effects of vapor pressure and temperature upon the moisture content of an adsorbent may be observed by referring to Fig. 1. When the given type of solid adsorbent is in equilibrium with air having a dry-bulb temperature of 70 F and 70 percent relative humidity, that is, having a dew-point of 60 F or a water vapor pressure of 13.2 mm Hg, the water content of the adsorbent is 33 percent. With air having the same dry-bulb

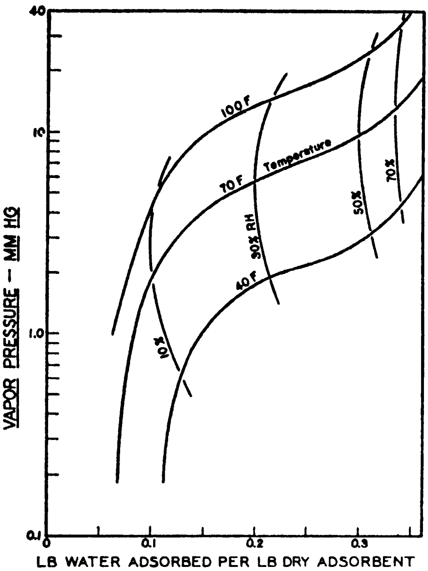

Fig. 1. Temperature—Vapor Pressure—Concentration Characteristics for a Typical Solid Adsorbent

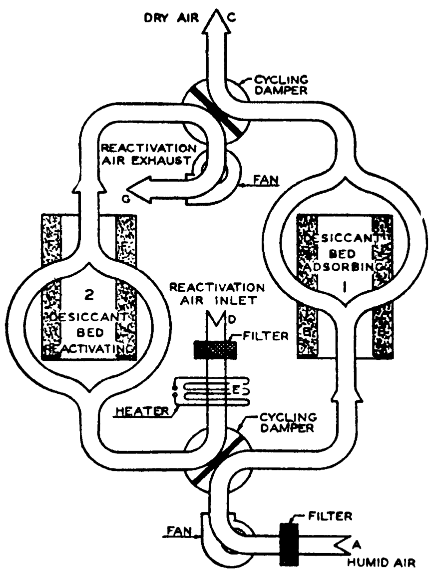

Fig. 2. A Typical Solid Adsorbent Dehumidification Unit Air Flow Diagram

temperature and a dew-point of 37 F, or a vapor pressure of 5.6 mm Hg, the water content is 20 percent. The increase in weight for an activated solid adsorbent, after it reaches equilibrium with a gas of any given water vapor content, may be found by subtracting the residual water content (for example 6 percent) from the equilibrium value. In the case of the two examples cited, the actual water gain would be 27 percent and 14 percent, respectively. The effect of temperature upon the adsorptive capacity may be observed by following the 5.6 mm Hg vapor pressure line. At a temperature of 70 F, the moisture content of the adsorbent is 20 percent, while at 100 F, the equilibrium water content is 11 percent.

In practice, the temperature rise in the dehumidified air caused by the adsorption heat, is approximately 10 deg F for each grain of moisture removed per cubic foot of air at atmospheric pressure. This temperature rise occurs progressively through the adsorbent bed, and is an important consideration in predetermining the performance of a given design of apparatus. Data such as these, together with information covering other characteristics such as specific heat, resistance to air flow, etc., are of value in the basic design of adsorption apparatus. In the solution of air conditioning problems, however, reference must be made to performance data on established apparatus designs.

DEHUMIDIFICATION EQUIPMENT USING SOLID ADSORBENTS

A typical solid adsorbent dehumidification unit air flow diagram is shown in Fig. 2. The apparatus consists of two adsorbent containers (adsorbers) with necessary interconnecting piping, valves, and auxiliaries consisting of filters, fans, activation air heater, controls, and, in some instances, a cooler for the dehumidified air. Before entering the adsorber, the air to be dehumidified is drawn into a filter to remove dust and other impurities. In passing through the adsorbent bed, the moisture content of the air is reduced and the dehumidified air is then introduced into the space or process requiring it. While the first adsorber is dehumidifying the air, the second adsorber is being reactivated by means of outside air drawn through a filter and heater in which its temperature is raised to 300 F. The heat may be supplied by electric heating elements, steam coils, the direct products of combustion of gas, oil, waste heat, or any other convenient source. In passing through the adsorbent bed, the hot gases supply the necessary heat for releasing adsorbed water from the adsorbent, and then carry it out of the adsorber to the activation gas outlet, where it is exhausted to the outside atmosphere. In some instances a thermostat placed in the activation outlet connection shuts off the activation fan and heater when the adsorbent is completely reactivated, as indicated by a rapid rise in the temperature of the outlet activation gas. The length of the adsorption period may be controlled by a timing device which changes the valves or dampers from the adsorbing to the activating position, or by a humidistat located in the dehumidified air connection or in the dehumidified space. The majority of commercial units are time controlled.

In applications where a continuous stream of dehumidified air is not required, a single adsorber type unit may be used, while in other cases where a continuous stream of dehumidified air is required, a multiple number of adsorbers, or even a continuously rotating system, may be used.

If the air to be dehumidified is very warm, and especially where exceedingly low dew-point dehumidified air is required, it is advantageous to install a pre-cooler to reduce the temperature of the inlet air. In this way the working temperature in the adsorber is lowered, and the overall performance appreciably increased. Some equipment manufacturers install cooling coils in the adsorbent beds for the same purpose, while others divide the adsorbent bed and install coolers between the sections.

Dehumidification equipment is employed to the best advantage where the air conditioning problem is primarily one of obtaining low relative humidity control rather than temperature control. This requirement is found in the case of the preservation of inactive naval vessels where the interior of the ship must be kept at a relative humidity below 30 percent to avoid corrosion, mold, mildew, and other moisture damage that occurs at humidities substantially in excess of this figure. Other advantageous applications for dehumidification systems are found in industrial processes where low relative humidity atmospheres are required during the manufacture, as well as for preservation of the finished products. Dehumidification with cooling may be used to advantage in work-rooms or other spaces occupied by humans, where the moisture load is high in comparison to the sensible heat load. In many instances, where independent control of temperature and humidity is important, dehumidification is used to advantage in conjunction with cooling.

ABSORBENTS

Any absorbent substance may be used as a dehumidifying agent if it has a vapor pressure with respect to water lower than the partial pressure of the water vapor in the mixture from which the moisture is to be removed.

Solid Absorbents. The substances used are generally the solid forms of the liquid absorbents. At present they are used principally in small desiccating chambers and in small dryers of the cartridge type, through which air is forced under presure. Calcium chloride is frequently used because of low cost.

Liquid Absorbents. These are primarily water solutions of materials in which the vapor pressure is reduced to a suitable level by controlling the concentration and temperature of the dehumidifying solution. Water solutions of the chlorides or bromides of various inorganic elements and certain organic compounds, are the liquid absorbents used in air conditioning.

In addition to having suitable water vapor pressure characteristics, an absorbent, to be satisfactory, should also meet the following requirements:

1. Be widely available at low cost.
2. Be non-corrosive, odorless, non-toxic, and non-inflammable.
3. Be chemically inert against any impurities in the air stream.
4. Be stable over the range of use.
5. Must not precipitate at the lowest temperature to which the apparatus is exposed.
6. Have low viscosity, and be capable of being economically regenerated or concentrated after having been diluted by the moisture absorbed.

DEHUMIDIFICATION BY LIQUID ABSORBENTS

In liquid absorption systems the air-vapor stream is brought into intimate contact with the absorbent solution by passing the air stream through a tower into which the brine is introduced as a finely divided spray, or by passing the air through a tower or contactor which is continuously sprayed with the brine, thereby presenting a large surface of absorbent to the air to be dehumidified. The difference in the partial pressure of the water in the concentrated brine and the partial pressure of the water vapor in the air, causes the water vapor to be given up by the air to the brine until equilibrium is approached. The water vapor is condensed during this operation, and its addition to the absorbent solution results in a decrease in the concentration of the solution. As the water vapor condenses, the latent heat of condensation is released in the absorbent solution. An additional, frequently appreciable, quantity of heat known as the heat of solution or heat of mixing, is also released. The heat released as the result of condensation and mixing, is directly transferred to the brine, to the equipment, and to the air being dehumidified, thereby causing a rise in temperature.

A modified system includes means for removing heat from the absorbent solution, either within the contactor or externally. Thus the temperature of the solution may be higher than, equal to, or lower than that of the air, depending on the chemical employed and the ultimate use of the dehumidified air.

Fig. 3 shows the relationship between temperature, vapor pressure and moisture content of a typical absorbent of the *inorganic* type. These curves indicate the general performance of such absorbents, although the exact values vary for the different compounds.

When the given absorbent is in equilibrium with air having a dry-bulb temperature of 70 F and a relative humidity of 70 percent, *i.e.*, having a dew-point of 60 F, or a water vapor pressure of 13.2 mm Hg, the water content of the solution is 4.8 lb water per pound of anhydrous absorbent. With air having the same dry-bulb temperature and a dew-point of 37 F, or a vapor pressure of 5.6 mm Hg, the water content is 2.1 lb per pound of absorbent. Therefore, when a solution of 2.1 lb water per pound of absorbent is exposed to an atmosphere of 70 F and 70 percent relative humidity, it will absorb an additional 2.7 lb of water in reaching equilibrium.

The effect of temperature on the absorptive capacity may be observed by following a constant vapor pressure line in Fig. 3. At a temperature of 70 F and a vapor pressure of 13.2 mm Hg, the moisture content is 4.8 lb

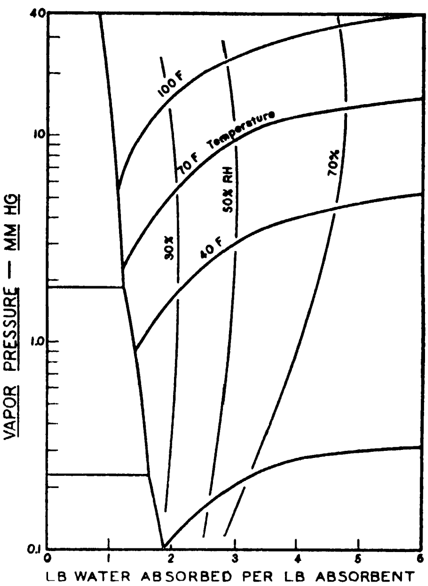

Fig. 3. Temperature—Vapor Pressure—Concentration Characteristics for a Typical Inorganic Absorbent (Halogen Salt Group)

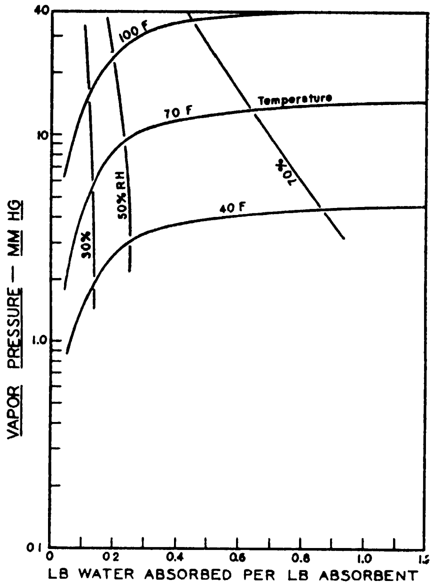

Fig. 4. Temperature—Vapor Pressure—Concentration Characteristics for a Typical Organic Liquid Absorbent

water per pound of absorbent. At 100 F, the moisture content is 1.8 lb water per pound of absorbent.

Fig. 4 shows the relationship between temperature, vapor pressure, and moisture content of a typical liquid absorbent of the *organic* type.

DEHUMIDIFICATION EQUIPMENT USING LIQUID ABSORBENTS

One type of system utilizing liquid absorbents includes an external interchanger having essential parts consisting of a liquid contactor, a solution concentrator, a solution heater and a cooling coil, all as shown in Fig. 5. The contactor and cooling coil are located in the wet air stream. The air to be conditioned is brought into contact with an aqueous brine solution having a vapor pressure below that of the entering air, resulting in a transfer of moisture from the air to the brine solution. This results in a conversion of latent heat to sensible heat, which raises the solution temperature and consequently, the air temperature. The temperature change of the air being processed is determined by the cooling water temperature and the amount of moisture removed in the equipment. Control of leaving air temperature may be obtained by precooling the absorbent solution in a suitable surface cooler, by tap, well, or chilled water.

The excess water of condensation, which dilutes the brine, is removed in the solution concentrator. This is a low pressure steam heat exchanger which over-concentrates a portion of the weak liquor, and returns it to the main brine reservoir for re-cycling. The concentrator operates in the manner of an evaporative condenser, whereby moisture is evaporated from the brine by the heating coils into a stream of regeneration air taken from, and rejected to, the outside atmosphere. Low pressure steam is normally used for heating the brine. When it is desirable or necessary to use gas or electricity, an auxiliary low pressure steam boiler is usually added to the equipment. Concentrators operating on a simple boiler principle have not as yet been commercially practical.

It should be noted that the solution concentration phase is the reverse

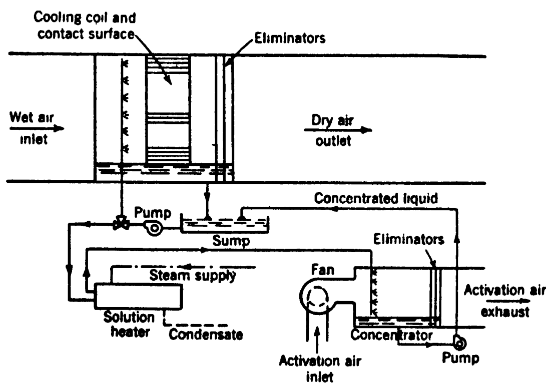

Fig. 5. Liquid Absorbent Equipment in Which Solution Cooler and Contractor are Combined

of the absorption process. During concentration, the aqueous vapor pressure of the solution is greater than that of the surrounding air, while during dehumidification, the reverse is the case. Utilization of this principle permits winter humidification by heating (instead of cooling) the solution pumped to the contactor. Water is thereby evaporated into, instead of being condensed out of, the conditioned air stream. This requires dilution of the brine externally to the contactor, rather than concentration.

CALCULATION OF MOISTURE LOAD

Calculation of the dehumidification required to maintain lower than normal moisture content in a given room begins with determination of the rate of moisture gain in the room from all sources. It is common practice, when maintaining a low humidity ratio, to recirculate a large percentage of the air in the room through the dehumidifier, and to add only enough outside air to meet the needs of the problem. The humidity ratio of the mixture of outside and recirculated air and the dehumidifier performance data can be used to calculate the humidity ratio of the air leaving the dehumidifier. The difference between the humidity ratio of the air in the room and that of the dehumidified air entering the room represents the effective dehumidification per pound of air. The rate of internal moisture gain in grains per minute, divided by the effective dehumidification in grains per pound of air, equals the air quantity required in pounds per minute. The following typical example using arbitrary values shows a general method of determining the dehumidifying requirements. Sensible heat determination considerations are discussed in other chapters, and are purposely omitted here.

Example 1: A solid absorbent dehumidifier having performance characteristics as shown in Fig. 6 is to be used to maintain inside conditions of 73 F and 20 percent relative humidity, *i.e.*, 24.1 grains per pound of dry air, 30 F dew-point, in a room 20 ft x 30 ft x 10 ft high, having a total wall, ceiling, and floor surface area of 2200 sq ft. Outside design conditions are 72 F dew-point (118.4 grains per pound).

Internal sources of moisture are: 4 occupants; an open natural gas burner using 15 cu ft of natural gas per hour; an open top water tank, having an area of 2 sq ft exposed surface, in which water is maintained at 87 F, with air movement over the water surface being 100 fpm. Determine the quantity and condition of the dehumidified air to be supplied to the room.

Solution: The internal moisture gain consists of items 1 to 5.

		Grains per Minute
1. From occupants:	$4 \times 1800/60 =$	120
1800 grains per person per hour is obtained from Fig. 7, Chapter 6, by interpolation between curves C and D.		
2. From burned gas:	$15 \times 650/60 =$	162
1 cu ft natural gas produces approximately 650 grains of moisture.		
3. From exposed water surface:	$2 \times 20 =$	40
Evaporation from water surface is assumed to be 20 grains per (minute) (square foot) at 87 F water with air movement of 100 fpm.		
4. From infiltration:	$\frac{6000}{60 \times 13.56} \times (118.4 - 24.1) =$	696
One air change, 6000 cu ft, assumed per hour (see Chapter 10).		
5. Moisture transmitted through room surface:	$\frac{2200}{60} \times 3 \times (0.783 - 0.176) =$	67
Permeability assumed to be 3 grains per (square foot) (hour) (inch Hg vapor pressure difference on two sides of wall).		
Total moisture gain from internal sources		1085

Let q be the air delivered to the room, pounds per minute. Let it be assumed for this problem that 85 percent of the air is recirculated and 15 percent is outside air. Enough air must be supplied to replace leakage from the system or to satisfy normal ventilating requirements for the occupants of the room as given in Chapter 6, whichever is greater. The amount is estimated from experience or obtained by test.

The humidity ratio of the mixture of recirculated and outside air entering the dehumidifier is then:

$$\frac{0.85q(24.1) + 0.15q(118.4)}{0.85q + 0.15q} = 38.3 \text{ grains per pound entering dehumidifier.}$$

For the dehumidifier whose performance is shown in Fig. 6, for 38.3 grains per pound in entering air, the leaving humidity ratio will be 6.5 grains per pound.

Effective dehumidification in the room is 24.1 − 6.5 or 17.6 grains per pound of supply air.

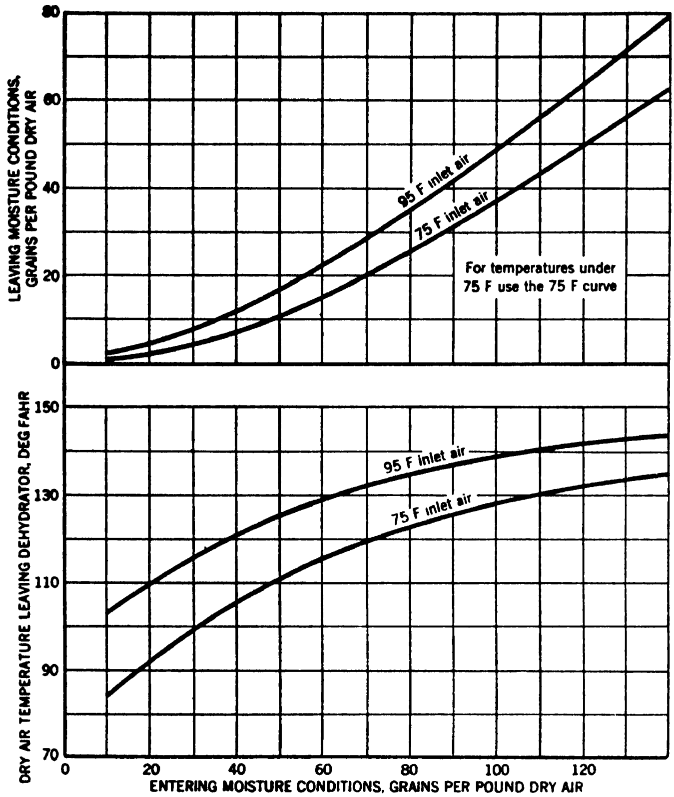

Fig. 6. Performance Data for Typical Commercial Solid Adsorbent Dehumidifier

Then $q = \dfrac{1085 \text{ grains per minute}}{17.6 \text{ grains per pound}} = 61.6$ lb air per minute *minimum* that must be supplied to the room to maintain 30 F dew-point.

Note that this figure represents the minimum requirement for the arbitrary conditions set forth and that in practice, safety margins should be added to the outside air percentage figure and to the calculated internal moisture gain.

VAPOR TRANSFER TO DEHUMIDIFIED SPACE

The walls enclosing a dehumidified space are subjected to a vapor pressure differential. The pressure of the vapor outside the walls tends to force moisture through the walls into the dehumidified zone of relatively low vapor pressure. As this process can be an unnecessary load on the dehumidifying equipment, provisions should be made for keeping the vapor transfer to a minimum. Also, if the space is cooled below the ambient dew-point, there is a possibility that condensation may occur within the walls, unless vapor transfer is controlled. For these reasons a vapor barrier should be located within the wall construction as near to the high vapor pressure side as feasible. To be effective, a barrier must be continuous and should be so located within the structure that it will be protected from rupture. (See section on Water Vapor and Condensation, Chapter 9.)

CHAPTER 38

AUTOMATIC CONTROL

Basic Types of Control, Types of Controllers, Actuating Devices, Actuated Controls, Residential Control Systems, Zone Control, Automatic Control Application, Control for Central Fan Systems, District Heating Control, Panel Heating Control, Indicating and Recording Equipment

THE function of *automatic control*, as applied to the heating, ventilating and air conditioning industry, may be broadly subdivided into the maintenance of *temperature*, *humidity*, and *pressure*, within pre-determined ranges. It automatically coordinates the operation of the various controlled devices in proper sequence to produce the desired result.

BASIC TYPES OF CONTROL

Available automatic control equipment may be divided into four main groups depending on the primary source of power:

1. *A self-actuated regulator* is one in which all the energy necessary to operate a valve or damper motor is supplied by the responsive element or bulb. Temperature changes at the bulb result in pressure changes of an enclosed fluid which transmits them directly to the valve or damper motor. Instruments of this type are available either with a rigid bulb or with flexible tubing from the bulb to the operating motor. The flexible tubing may be furnished in varying lengths, and is generally protected by a flexible metal armor.

2. *Electrically operated* equipment utilizes electric current as a primary source of energy, its flow being regulated as required to operate motors, relays, or other controlled items.

Electrical controls may be divided into two classes: (1) those wherein the primary measuring device utilizes contacts to regulate the flow of the electric current, and (2) those wherein the primary measuring device is a resistance wire component of an electronic circuit; these are known as *electronic controls*.

3. In *pneumatically operated* equipment the primary source of energy is compressed air usually at a pressure of 15 to 25 psig. The flow of this air is proportioned as required to operate valves, dampers, relays, or other controlled devices

4. In *hydraulically operated* equipment the primary source of power is a suitable liquid at a pressure of 15 to 25 psig or higher, which is handled in the same manner as compressed air.

TYPES OF CONTROLLERS

The basic types of controllers and their operations are:

1. *Two-position* or *on-off controllers* are the simplest type and are clearly described by the name. With controllers of this type the valve or damper motor can assume only two positions, either open or shut.

2. *Proportional* or *gradual acting controllers* function to re-position the controlled device, by small increments of travel, to regulate the flow as the controller senses a slight change in the controller condition.

3. *Floating controllers* act to produce valve or damper movement whenever there is a deviation from the control point. Whenever the condition to be controlled is above the control point, the valve closes at a constant rate, and continues to close until the temperature returns to the control point. Below the control point the valve reverses its action and moves in the other direction until the control point is again reached.

4. *Automatic reset* (*or proportional plus reset*) *controllers* function to reposition a valve or damper by small increments of travel, as in a proportioning controller. In addition, a mechanical device in the controller automatically and constantly resets the instrument to offset the normal drift (inherent in a proportional controller) be-

tween maximum and minimum load. The rate of reset is manually adjustable and must be set to meet the load requirements of the individual system.

Controllers may also be designated by types as: a *non-indicating controller,* when it does not indicate the controlled condition and performs the control function only; an *indicating controller,* when fitted with a pointer, thermometer, or gage which indicates the controlled condition; a *recording controller,* when it is combined with a clock mechanism and chart which records the controlled condition.

ACTUATING DEVICES

The starting point of any control system is the thermostat, hygrostat, pressure regulator, or other mechanism which is sensitive to a change and responds in the desired manner.

Thermostats are usually of the room, duct, or immersion types. Various types of thermostats found in common use are defined in the following paragraphs.

1. A *thermostat* is an instrument which is responsive to changes in temperature, and initiates a force that repositions valves, dampers, etc., to maintain selected temperatures.

2. A *room thermostat* is usually mounted on the wall of the space to be controlled with the measuring element arranged so that it is affected by the room temperature.

3. A *duct thermostat* is provided with fittings suitable for installation in duct work. The *insertion* type is equipped with a rigid bulb and is arranged so that the temperature responsive element or bulb extends through the wall of the duct. The *remote bulb* type is arranged so that the bulb and instrument head are connected by means of a flexible tube of the desired length. The bulb is inserted in the duct, and the head is located where it is accessible for adjustment and inspection.

4. An *immersion thermostat* is provided with fittings suitable for installation in a pipe line or tank where a fluid tight connection is required. Both insertion and remote bulb types are available. A union connection and separable socket, when used, permit removal of the bulb without draining the line or tank. The sockets may be of copper, stainless steel, or other materials.

5. A *day-night* or *two-temperature thermostat* controls a heating or cooling source to maintain either of two selected temperatures. They may be indexed (set at desired control temperature) individually or in groups from a remote point by means of a manual or time switch.

6. A *summer-winter* or *heating-cooling thermostat* is similar to the day-night type, except that both the temperature setting and action are changed by the indexing means. Such a thermostat could open a volume damper on a rise in temperature in summer, and close the same damper on a rise in temperature in winter.

7. A *submaster thermostat* has its temperature setting raised or lowered a predetermined amount for a given change in some other variable. For example, the water temperature on a heating system may be raised as the outdoor temperature drops. A *master instrument* is used to reset a submaster thermostat and may be a switch, pressure controller, thermostat, or similar device. In the foregoing example the master thermostat would be located where it would respond to outdoor temperature, and the submaster thermostat would be located in the pipe line of the heating system.

A *hygrostat* is a controller which is sensitive to changes in relative humidity, and is available in room and duct types. Where the controlled condition is below 20 percent or above 80 percent, or the temperature is above 100 F, selection of a suitable type and kind of hygroscopic element is essential.

A *pressure regulator* is a device which is sensitive to changes in pressure. It may be of the type which controls a *single* pressure or of the *differential* type which maintains a predetermined difference between two pressures. For pressures in duct work, *static pressure regulators* are available in the *differential* type. They are sensitive to changes of 1/100 in. of water.

ACTUATED CONTROLS

Thermostats, hygrostats, pressure regulators and other actuating devices obtain control of heating and cooling mediums, fuels, liquids, etc., by actuating various control devices such as control valves, dampers, damper motors, relays, or controllers defined in the following paragraphs.

A *control valve* is designed to control the flow of fluids, and may be considered as a variable orifice which is repositioned by a motor operator, as directed by a thermostat or other controlling device.

1. A *normally open* or *direct acting valve* will assume an open position when all operating power is removed.
2. A *normally closed* or *reverse acting valve* will assume a closed position when all operating power is removed.
3. *Single seated valves* are designed for *tight* shut-off using appropriate disc materials for various pressure ranges.
4. *Pilot piston valves* serve a similar function on high pressure installations.
5. *Double seated* or *balanced valves* are designed for applications where tight shut-off is not required. They are not affected by varying inlet pressures or pressure differentials, and thus are widely used where these conditions exist.
6. A *three-way valve* is fitted with a double faced disc, operating between two ports, and functioning to close one port as the other is opened. Depending upon how it is installed, it may be used as:
   a. A *three-way mixing valve* to mix as required two fluids entering the two inlet ports and leaving through the common outlet port.
   b. A *three-way diverting valve* to divert the flow from the inlet port to either of the outlet ports.

Valve discs, poppets, and seats are available in various shapes to meet any desired flow characteristics with various materials as required by service conditions.

A *damper* is designed to control the flow of air or gases, and is similar to a valve in this respect. *Single blade dampers* are generally restricted in size because of the difficulty of securing proper operation with high velocity air. *Multi-blade* or *louver dampers* can be furnished so that adjacent blades move in the same direction or in opposite directions. The *opposed blade type* gives better directional and flow characteristics than the parallel blade type.

For long life and trouble free operation, dampers should be constructed with heavy metal frames, blades of iron adequately braced, and ample bearing surfaces of non-corrosive materials. When fairly tight closing is desired, felt may be glued and riveted on the edges and ends of the blades. Other materials for blades and frames are also used for special services.

A *damper motor* is repositioned by a controlling instrument, and is connected to the damper blades as required to give the desired movement. It can be mounted on the damper frames or mounted outside the duct and connected to an extended shaft on one or more damper blades. Suitable brackets are available for floor, wall, or duct mounting of the motor.

A *relay* is a device which uses an auxiliary source of energy to amplify or convert the force of a controller into available energy at a valve or damper motor. Various types of relays are designated as follows:

1. An *electro-pneumatic relay*, when electrically energized, starts or stops the flow of air as required.
2. A *pneumatic-electric relay*, when affected by different air pressures, starts or stops the flow of electrical energy as required.
3. A *switching relay* or *pilot valve* may be used to switch the operation of a con-

trolled device from one controller to another, or to reverse the action of a controlled device in response to an impulse from a controller.

4. An *averaging relay* is affected by the forces from two or more controllers, and the resulting flow of energy is in accordance with the average of these forces.

5. A *positioning relay* has a direct connection to a valve or damper motor lever and is affected by both valve or damper position and controller demand. It is repositioned by a thermostat or other controlling device, and is arranged to give a definite motor position for a given force from the thermostat without regard for motor hysteresis, friction, or pressure variations of the controlled fluid.

6. An *electronic amplifier* is used only in *electronic control* circuits to amplify the micro-currents of electronic controllers to usable voltages required by standard electric actuating devices. They may be either two-position or proportioning relays.

A *sequence controller* is used to operate two or more devices in a prearranged sequence. It is generally used in connection with refrigeration compressors, and may be arranged to prevent simultaneous starting in the event of temporary electrical shutdown or control medium failure.

Manual switches are available in the two-position or proportional types. *Two-position switches* change the flow of energy from one line to one or more other lines; or from one pair of lines to another pair of lines. *Proportioning switches* vary the flow of energy as determined by the manual setting of the switch.

RESIDENTIAL CONTROL SYSTEMS

The control equipment function in a residence may vary from the regulation of a coal-fired heating plant to the completely automatic control of an all-year air conditioning system. Regardless of the type of heating or air conditioning system used, the control system should be selected carefully to insure safety and comfort of the occupants, and also economy of operation.

Heating Unit Controls

Typical controls for the appliances used to supply heat in residences are as follows:

1. *Hand Fired Coal Burners.* The control of a hand fired coal burner for a boiler or furnace normally consists of a room thermostat operating a two-position electric control motor, which in turn opens the draft damper and closes the check damper on a demand for heat. The motor then closes the draft damper and opens the check damper when the thermostat is satisfied. A limit control on the boiler or furnace should be connected to the motor so that it may check the fire whenever a predetermined temperature or pressure has been exceeded. A manually operated basement switch is usually included on the motor so that the draft may be opened and the check closed when the boiler or furnace is being filled with coal.

2. *Coal Fired Stokers.* Domestic stokers are usually controlled by a room thermostat, a limit control, and a stoker relay. When the thermostat calls for heat, the relay causes the stoker motor to increase the flow of fuel and air to the burner to its maximum rate. When the thermostat is satisfied, the relay provides for a minimum flow of fuel and air to the burner to maintain the fire at its minimum rate. The limit control prevents the continuance of the maximum fuel rate if the temperature or pressure in the boiler or furnace exceeds a predetermined value, and also stops the feeding of fuel if the fire goes out. Automatic ignition usually is not available and the firing of a stoker is normally on or off.

3. *Automatic Oil Burners.* Automatic oil burner controls normally consist of a room thermostat, a limit control, a combustion safety control, and a control relay. On a call for heat by the thermostat, the relay starts the oil burner motor which supplies oil and air to the burner. An ignition device consisting of an electric spark or a gas flame ignites the oil automatically. If for any reason the oil and gas mixture does not ignite, a time delay mechanism in the relay is operated by the combustion safety control after a predetermined length of time to cause the oil and air

supply to be shut off. If the oil and air mixture ignites properly the buerner continues to run until the thermostat is satisfied, or until the limit control affected by the temperature or pressure in the boiler or furnace stops the burner.

4. *Automatic Gas Burners*. The controls for an automatic gas burner usually include a room thermostat, a limit control, a safety pilot, gas pressure regulator and a gas valve (solenoid, motorized or diaphragm type). Upon a demand for heat at the thermostat, the gas valve is opened, admitting gas to the burner. The safety pilot ignites the gas which continues to burn until the thermostat is satisfied, or until the limit control shuts off the gas valve. The limit control may also reduce (throttle) the gas flame as required to maintain a desired temperature or pressure of the heating medium. If the pilot flame is extinguished for any reason, either before or after the main gas valve is turned on, the safety pilot closes the gas valve, thus eliminating the danger of delivering gas to the burner without ignition.

5. *Electric Heating*. Electric heating has become popular in those areas where electric power is plentiful and inexpensive. The electric heating elements may be located in each individual room and turned on and off by thermostats in each room, or the heat may be supplied by a central heating system. In the case of a central heating system, the control is usually of the proportioning type which energizes from five to ten heating elements in sequence, according to the demand for heat, by means of a sequence controller consisting of a series of switches operated by a proportioning motor. A limit switch recycles the sequence controller if the furnace or boiler exceeds a predetermined temperature.

Limit Controls

A *high limit* control for steam consists of a pressure control, having bellows responsive to the boiler pressure, which breaks an electric contact when the steam pressure exceeds a predetermined point, thereby preventing the burner from delivering additional heat to the boiler. A low water cut-off should also be used to stop the burner if the water in the boiler drops to a dangerous level.

A high limit control for a hot water boiler consists of an immersion thermostat (usually equipped with a bi-metal helix) inserted in a well in the boiler. This control stops the burner when a predetermined water temperature has been reached in the boiler.

In a warm air system the high limit control is a thermostat including a bi-metal helix inserted in the bonnet of the furnace. It will shut off the source of heat when a predetermined furnace temperature is exceeded.

Room Thermostats

Room thermostats are of three types, depending on the temperature sensitive element which they employ. These are: (1) the bi-metallic type which distorts with temperature changes; (2) the vapor filled bellows type which expands or contracts with temperature changes; and (3) the electronic type which employs resistance wires and microcurrents which vary with temperature or humidity changes. The first two types employ either electric currents or compressed air to amplify their effect. In the electronic type the micro-currents are amplified by means of electronic amplifier relays. The amplified effect in each instance is used to actuate valves, damper motors, stokers, oil burners, gas burners, etc. Room thermostats may be of the plain or single temperature type, or of the day-night type providing for automatic night lowering and morning increase of the control point. The automatic setback type usually includes a clock mechanism which accomplishes this result. Opinions vary regarding the amount of fuel that can be saved by automatic setback. Tests made both in the Warm-Air Heating Research Residence and the I = B = R Research Home at the University of Illinois indicate that, on thermostatically controlled systems, a possible fuel saving of from 7 to 10 percent may be obtained by reducing the house temperature 6 to 10 deg from about 10:00 p.m. to 5:30 a.m.*

In locating a room thermostat the following rules should be observed:

1. It should always be located towards the center of a relatively open room on the coolest rather than the warmest side of the building.
2. It should never be mounted on an outside wall or other cold surface, or where it is exposed to cold drafts from an outside door.
3. It should never be mounted where it will be affected by direct rays of the sun; by heat from a nearby warm surface such as chimneys, pipes or ducts in a wall, or radiators; or by direct currents from a warm air register.
4. It should never be located where normal circulation of air is impeded by furniture or an opened door.
5. It should be located where it is safe from mechanical injury.

In a typical home, a satisfactory location for the thermostat can usually be found on an inside wall of the living room or dining room.

System Control

There are several types of system control in common use for residential applications. They are usually of the two-position (on-off) type, or of the proportioning type.

1. *Two-Position (On-Off) Control.* The most simple type of domestic control is the type in which the room thermostat starts the burner or other source of heat when the temperature of the air at the thermostat falls below the thermostat setting, and stops the source of heat when the air temperature rises above the setting. If forced warm air or forced hot water is used, the fan or circulator may be turned on and off at approximately the same time as the source of heat.

2. *Proportioning Control.* When a proportioning type of control is used, the flame of the burner may be varied, or the burner may be cycled (started and stopped) frequently to provide for time modulation so that the heat input to the home is proportioned continuously to the heat loss from the home. The fan of a forced warm air system or the circulator of a forced hot water system may be run almost continuously, thereby providing for the constant flow of heat into the home. Such operation minimizes the cold drafts on the floor as caused by cold air dropping from cool walls and windows during the off period of an on-off system.

Air-Conditioning Systems

Year 'round residential air-conditioning systems which provide for heating in winter and cooling in summer should be given the same consideration in selecting the control system as required for commercial air-conditioning systems described later in this chapter, since the basic principles are the same and the final results must provide for the comfort of the occupants. Economy in first cost may result in both lack of economical operation and discomfort.

ZONE CONTROL

In residential heating, it is often desirable to divide the house into two or more zones for greater accuracy of control and comfort. Each zone may then be maintained individually at the desired temperature level. The division by zones should be based upon exposure and occupancy; the most common division is usually found to be:

1. Living section such as living room, dining room, den.
2. Sleeping section.

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* Save Fuel for Victory, University of Illinois, Engineering Experiment Station Circular Series No. 47, p. 31.

3. Service section such as kitchen, pantry, servant's quarters.
4. Recreational areas.

Zone control for steam and hot water heating systems is employed where it is desired to control the heating effect of a multiplicity of radiators or convectors, located in various heated spaces, through the use of a single regulator. Under certain conditions, particularly in buildings of limited size, it is possible to consider the entire building as a single heating zone. In such cases, the zone regulator, or *master controller*, may operate, directly, the automatic firing equipment of the boiler or the reducing valve in the street steam main. In large buildings the demands of satisfactory temperature control, however, will make it necessary to sub-divide the heating system into suitable zones.

There are a number of factors to be considered in zoning, in order that heating requirements in a single zone will be approximately consistent throughout its extent.

1. Exposure may be a factor to be considered, with particular reference to prevailing winds, sun effect, and the shelter afforded by surrounding structures and topographical features.

2. Occupancy may be a determining factor, in that the indoor temperature requirements for the activities carried on in various portions of the building may vary, and the hours of occupancy likewise may differ.

3. The physical characteristics of the building will enter into the sub-dividing of the heating system into zones by reason of the fact that satisfactory temperature conditions throughout a single zone of given extent may not be enjoyed equally in buildings of dissimilar types of construction. Also, the height of the building and its horizontal extent and form are considerations which must be borne in mind.

4. The cost of the zone control equipment for such additional zones as might seem otherwise desirable, often will influence the decision as to the final number of zones to be employed. In buildings of considerable size, accepted practice dictates that there shall be at least one zone for each exposure, although each exposure very possibly should be sub-divided vertically into two or more zones, for the higher structures. Also, the presence of two or more wings, having the same general exposure, may suggest the desirability of more restrictive zoning. In smaller buildings, and in those of larger extent where cost and other conditions limit the number of zones, a common compromise is to combine the North and West exposures in one zone, and the East and South exposures in a second zone. Frequently, when the street floor level is given over to public spaces or to activities which are markedly different from those carried on in the remainder of the building, it is advisable to provide a separate steam main, with conventional room thermostats in each individual area.

For steam heating systems, the radiator output may be varied proportionately to the changes in outdoor temperature, by any of a number of general methods. Those in most common use are:

1. Turning the steam on and off at appropriate intervals, as dictated by temperature or time considerations, proportioned to the need for heating.

2. Varying the pressure of steam in the system in accordance with the demand for heat.

3. Throttling the steam pressure, at the demand of the controller, to allow flow through orifices in proportion to the heating requirements.

In hot water heating, the accepted practice is (1) to vary the temperature of the hot water supplied to the system, or (2) to vary the flow of hot water; both being varied proportionately to the heating requirements.

The regulator for each zone usually is of a type which in some manner responds to the outdoor temperature and effect of sun and wind for the zone. Many of the available zone controllers are arranged in such a way as to be affected also by the temperature of the heating system and the indoor temperature in the zone. Whatever the mechanical features of the regulator, its function is to dictate the flow impulse, or rate of flow, in such a way as to maintain the desired indoor temperature in the zone, regardless of the fluctuations in the outdoor temperature. Provisions also may be made to maintain a predetermined low economy temperature in each zone during periods of non-occupancy; to facilitate quick warming-up following such periods; and to follow those portions of the daily cycle of control with normal heating effect during the occupancy period.

A control panel, at a central location, may be arranged so that manual switches for each zone may raise or lower the operating temperature. Time switches, if desired, may be provided for obtaining, automatically, any day-night or other predetermined control program which the operators of the building may desire. The characteristics of the regulators which are operated by the zone controllers depend upon which of the basic systems of zone control is employed in a given installation. Shut-off, mixing or throttling valves, and various forms of devices to reset or pilot the action of reducing valves and to control firing means, are some of the more common regulators which are used to control the flow of steam or hot water, under the command of zone controllers. The characteristics of these regulators usually are determined by the manufacturer of the type of controller which is selected.

AUTOMATIC CONTROL APPLICATION

Some of the considerations affecting the selection of automatic controls for applications are given in the following paragraphs which describe controls and operation for various types of units.

Unit Heater Control

Two-position (on-off) control by means of a room thermostat is the standard method of control for unit heaters. A limit control should be incorporated to prevent operation of the unit heater fan motor when steam or hot water is not available. The limit control can be a surface thermostat or pressure control. Where there is no possibility of drafts, and continuous air circulation is required, the unit heater fan motor may operate continuously. In this case, a room thermostat controls a valve (two-position or proportioning) in the stream or hot water supply line.

Unit Ventilator Control

Various makes of unit ventilators are designed for different control cycles Selection of automatic temperature control for unit ventilators is largely determined by the design of the particular unit. The choice of control cycle may also be determined by local and state ventilating codes, particularly where units are installed in school rooms. It is desirable to coordinate the selection of control cycle with the unit ventilator manufacturer since, in many cases, modifications of the unit are required for the installation of the control equipment; and, in some cases, it is desirable to have the equipment factory-mounted. Two typical control cycles are: (1) *Variable Outdoor Air with Fixed Minimum*, and (2) *Variable Outdoor Air without Fixed Minimum.*

Control Cycle No. 1. In full heating position, the outdoor air damper is closed, the recirculating air damper is open, and the supply valve is open.

In full cooling position the outdoor air damper is open, the recirculating air damper is closed, and the supply valve is closed or at a minimum position to maintain a minimum discharge air temperature. The sequence of control operations is: On call for cooling, under control of a room thermostat, outdoor air damper opens to minimum setting. The supply valve then closes gradually, after which the outdoor air damper gradually opens to the maximum position and, simultaneously, the recirculating air damper closes. A low limit thermostat mounted above the coil prevents the discharge temperature from dropping below a predetermined minimum.

Control Cycle No. 2. In full heating position the outdoor air damper is closed, the recirculating air damper is open and the supply valve is open.

In full cooling position the supply valve is closed, the outdoor air damper is open and recirculating air damper is closed or at any position required to maintain a minimum discharge air temperature. The sequence of control operations is: On call for cooling, under control of a room thermostat, the valve gradually closes. As the valve leaves full-open position, control of the recirculated air and outdoor air dampers is transferred to a thermostat installed ahead of the heating coil to maintain a minimum air temperature. On continued call for cooling, the valve gradually closes.

If direct radiation is used, it is desirable that it be controlled in sequence with whatever control cycle is adapted for the unit ventilators.

Unit Coolers

Although most unit coolers can be adapted to any control cycle, continuous fan operation is recommended to avoid stratification and wide fluctuations in space temperature. Should the unit be completely self-contained, control of the direct expansion refrigeration unit may be obtained from the temperature of the recirculated air and from suction pressure. In the case of multiple unit systems supplied with refrigerant or chilled water from a central source, a valve in the supply to each cooling coil may be controlled thermostatically from space temperature.

Refrigeration and Dehumidification Equipment

Typical control equipment and its functions are described in the following paragraphs.

Well Water. Where well water is used directly in air washers or cooling coils, control of temperature or humidity is usually obtained by thermostat- or humidistat-operated valves (two-position or proportioning). The two-position valve will provide better dehumidification since a lower coil temperature will be maintained, but the temperature of the discharge air will fluctuate. With proportioning control, better control of discharge air temperatures will be maintained. In both cases the sensible-latent heat ratio is basically a matter of coil design rather than automatic control. Proportioning three-way valves may be used as mixing or diverting valves for better pump performance, and may also be applied to an air washer used with a recirculating pump to control water temperature rather than volume.

Ice Bunkers. Where water is sprayed over the ice in bunkers and circulated to air washers or cooling coils, control is obtained by a thermostat in the water line from the bunker. The thermostat proportions a three-way valve to by-pass enough return water around the ice bunker to maintain a constant discharge temperature.

Compressors. Compressors may supply refrigerant to direct expansion cooling coils in air conditioning units, or to direct expansion coils in water-chilling units. In either case, the compressor motor may be started and stopped directly by a room or duct thermostat, or a pressure controller may be used to regulate the suction pressure of the compressor. In the latter case, a room or duct thermostat may be used to control a solenoid valve in the refrigerant supply line to the cooling coil. A high and low pressure cut-out is standard safety equipment on most compressor installations. Reduced capacity of the refrigerating unit may be obtained by means of temperature or pressure controlled unloading devices which vary the capacity of the compressor in some proportion to variations of cooling load. Program or step controllers, actuated by temperature or pressure, are commonly used in multiple compressor installations. It is desirable in such installations to return the program or step controller to the *off* position when the system is shut down to prevent the full electrical load of multiple compressors from being thrown across the line at the same time. Thermostatic control of water supply to water-cooled condensers may be achieved by means of self-contained controllers or valve and thermostat application.

Steam Jet. A steam jet refrigeration system is commonly controlled by means of a thermostat in the chilled (secondary) water. The thermostat operates a two-position valve in the steam line to the jet. In the case of multiple jet units, program or step control can be achieved by controlling the jets in sequence. As in the case of direct expansion refrigeration units, a system of control is advisable for the water-cooled-condensing unit.

Centrifugal Units. The control of centrifugal refrigeration units or other types of vacuum systems is customarily achieved by means of a thermostat in the chilled water to control the operating cycle of the equipment at full or reduced capacity.

Adsorption Units. Control of adsorption units consists of a damper control which, in response to humidity, controls the air flow through or around the activated bed of adsorption material. Standard controls for cooling are used to reduce the dehumidified air to a desired dry-bulb.

Absorption Units. Since at constant density, the absorption solution will extract water from the treated air in an amount proportional to the solution temperature, the moisture content of the air leaving an absorption unit is regulated by solution temperature. Solution density is held constant by a combination of float control and steam valve controlling the solution regenerator. Two basic methods of control are standard:

1. Constant solution temperature where the solution temperature is set so that, at the full load for which the unit is designed, the discharge air will have the desired moisture content. Proportioning control of the water and two-position control of the steam are recommended to maintain constant temperature.
2. Control by varying the solution temperature so that the moisture content of the discharge air remains constant regardless of load variation. Basically, this control is similar to the constant solution temperature control, with the addition of a hygrostat or wet-bulb controller controlling the water valve from space conditions. In order to secure a constant discharge dry-bulb temperature, a coil is provided with proportioning valve controlled by a proportioning thermostat in the unit discharge.

CONTROL FOR CENTRAL FAN SYSTEMS

Automatic temperature control for central fan heating, cooling, ventilating, and air conditioning systems involves the proper application of various types of controlling instruments and associated regulators such as valves, dampers, and damper operators, relays and other auxiliary equipment which are described in earlier sections of this chapter. In central fan systems the conditions required dictate the type of built-up control system to be used. Otherwise, some arrangement of available *package* equipment probably would be used. It is impossible to state in detail the control apparatus which will be required in even the most representative applications.

In general, in so far as automatic temperature and humidity control equipment is concerned, central fan systems may be divided into certain broad classifications, as follows:

1. Heating.
2. Humidifying.
3. Ventilating and atmospheric cooling.
4. Cooling and dehumidifying.
5. Control of zone temperatures.
6. Year 'round air conditioning with automatic change-over.
7. Constant temperature and humidity.

The apparatus which enters into the automatic maintenance of temperatures and humidities for central fan systems which are designed to produce each of the effects listed in items 1 to 7, is indicated in the following paragraphs:

1. In *heating control*, there are three considerations to be borne in mind: (1) to control space temperature; (2) to prevent drafts; (3) to guard against freezing.

Usually in central fan systems, suitable thermostats operate valves in the steam or hot water supply to heating coils, or face dampers across such coils and by-pass dampers around them. If the heating coils are sub-divided into two or more groups, such as preheaters and reheaters, a duct thermostat (following the preheaters, and located in the entrance to the chamber between the groups of coils) controls the preheaters; but it is essential that the preheater coils be of the steam distributing type. Similarly, a duct thermostat in the fan discharge, where any effect of stratification has been dissipated, operates the valves and dampers associated with the reheaters. In some cases, where there are more than one bank of preheaters, the practice is to place a freeze protection thermostat in the outdoor air intake to control the valve on the first bank of heaters, which is designed so that the heat-rise through it will not cause overheating. A room thermostat in the heated space may serve as the controlling instrument, with a thermostat in the fan discharge serving to prevent the delivery of air at a temperature which might cause drafts. If desired, similar action may be obtained from a thermostat in the return air connection. When there is only one heating coil, a limit thermostat in the fan discharge accomplishes the control, in conjunction with a thermostat in the heated space or in the return air.

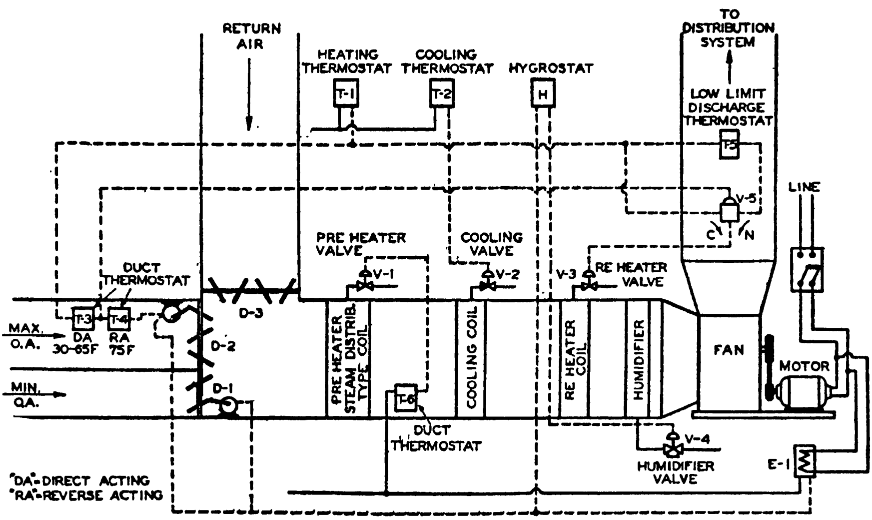

FIG. 1. CONTROL DIAGRAM FOR YEAR 'ROUND AIR CONDITIONING SYSTEM

2. *Humidity control* may be obtained by means of a hygrostat, usually located in the conditioned space or in the return air. Such controlling instruments may operate steam supply valves to humidifiers, mixing valves to control the temperature of the water to sprays, or a system of dampers to regulate the quantity of air passed through the humidifying chamber or by-passed around it. The control of humidity, according to dew-point temperature, is sometimes accomplished by means of a thermostat in the outlet of the humidifying chamber. However, the setting of the dew-point thermostat may have to be changed as the humidity in the conditioned space varies.

3. The *control of ventilating* and *cooling* by the use of outdoor air consists of an arrangement of dampers, usually determining the relative quantities of outdoor air and return air which are to be delivered to the conditioned space. The damper positions are regulated by proper types of thermostats, located in the minimum outdoor air intake, the fan discharge, the conditioned space or the return air. Instruments are available which, with an adequate arrangement of dampers, will cause a maximum quantity of outdoor air to be handled until it becomes more economical to utilize return air.

4. *Cooling and dehumidifying* may be controlled by means of thermostats, and hygrostats or dew-point thermostats, which regulate dampers and mixing valves to maintain air of the proper temperature and humidity in the discharge from the central fan plant. Such controlling instruments normally are located in the fan

discharge or in the return air, or both, and they may be associated with thermostats or hygrostats in the conditioned spaces.

5. Where a *separate duct serves each zone* of an area with which a central fan system is associated, a room thermostat in each zone may operate mixing dampers in the inlet to each zone duct, determining the quantity of warm air which is required from that portion of a plenum chamber into which heated air is delivered, and the quantity of cool air which should be taken from the other portion of the double plenum chamber. In many instances, separate zone heating and zone cooling coils are employed, instead of mixing dampers.

6. The control hook-up for a typical *year 'round air conditioning system*, including automatic change-over from heating to cooling, is indicated in Fig. 1 and described as follows:

Whenever the fan is started, solenoid air valve or relay E-1, actuated by the fan motor starter, opens minimum outdoor air damper D-1, places hygrostat H in service, and allows duct thermostats T-3 and T-4 to control the maximum outdoor air damper D-2 and the return air damper D-3.

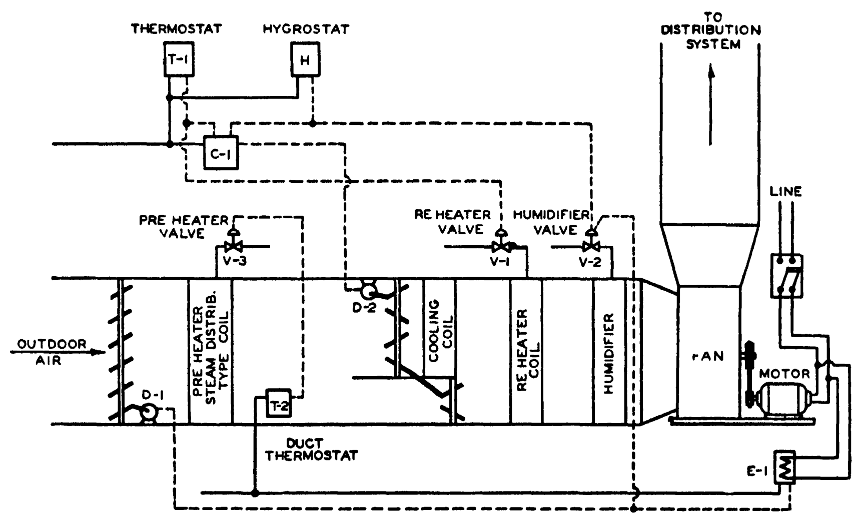

FIG. 2. CONSTANT TEMPERATURE AND HUMIDITY CONTROL FOR YEAR 'ROUND SYSTEM USING 100 PERCENT OUTDOOR AIR

When the fans stop, E-1 is de-energized to close the outdoor air dampers and also to close humidifier valve V-4.

Thermostat T-1 positions steam valve V-3, on the reheater coil, to maintain a constant space temperature. As the space temperature rises, T-1 positions reheater valve V-3 to a closed or to a minimum open position, as determined by low limit discharge thermostat T-5. Duct thermostat T-6, in the preheater discharge, positions preheater coil valve V-1 to maintain a constant preheater discharge temperature.

On rising outdoor temperature, between 30 F and 65 F, duct thermostat T-3 located in the outdoor air intake, moves maximum outdoor air damper D-2 toward the open position. At 65 F outdoor, D-2 will be fully open and return air damper D-3 will be fully closed.

As the outdoor air temperature rises above 65 F, duct thermostat T-3 positions V-5 in such a way as to by-pass low limit thermostat T-5, so that reheater coil valve V-3 is operated directly from thermostat T-1. As outdoor air temperature rises from 65 F to 75 F, duct thermostat T-4 gradually closes maximum outdoor air damper D-2 and opens return air damper D-3.

Cooling thermostat T-2 positions cooling coil valve V-2 to admit more chilled water as the space temperature rises.

Hygrostat H positions humidifier valve V-4 to maintain the desired humidity in the conditioned space.

7. The arrangement of automatic control for a *constant temperature and constant humidity air conditioning system*, using 100 percent outdoor air, is shown in Fig. 2, and the control description follows:

Whenever the fan is running, relay or solenoid air valve E-1, actuated by the fan motor circuit, is energized, opens outdoor air damper D-1, and also permits hygrostat H, in the conditioned space, to control humidifier valve V-2.

When the fan stops, E-1 closes outdoor air damper D-1 and humidifier valve V-2.

Remote bulb thermostat T-2, with bulb located in preheater discharge, operates valve V-3 on the preheater coil to maintain a constant preheater discharge temperature.

On rising temperature, thermostat T-1, in the conditioned space, closes reheater valve V-1 and, through relay C-1, opens face damper D-2 for cooling. On rising humidity in the conditioned space, hygrostat H closes humidifier valve V-2, and likewise, through C-1 may open face damper D-2 for dehumidification.

For closer control, the face and by-pass dampers should be eliminated, and cooling means continuously provided whenever the outdoor dew-point rises above a predetermined maximum. Reheating and humidifying may be required to provide the desired conditions. However, such a system will be less economical in operation.

DISTRICT HEATING CONTROL

There are three general means of obtaining centralized control of heat output of radiators in district heating systems.

1. *Controlling the rate of steam flow* into the radiators. This is accomplished by equipping the radiator inlets with orifices, and controlling the flow of steam through them into the radiator by controlling the difference in pressure between the supply and return.

2. *Controlling the temperature of steam* in the radiators by varying its pressure. This involves the use of high vacuums to obtain low steam temperatures. This must be supplemented by some other type of control for low heat output.

3. *Controlling the length of time steam flows* into the radiators by admitting steam to a heating system intermittently and varying the length of the *on* and *off* periods. Two types of controls are used. (1) A clock control providing *on* and *off* settings of various lengths, which can be changed in accordance with outside temperatures. In most cases these changes are made automatically by means of a thermostatic bulb, placed outdoors. (2) A control, having an outdoor bulb and a bulb attached to the radiator, which varies the length and frequency of the *on* intervals in such a way that the radiator temperature is varied according to the outside temperature. In some cases heat supply is controlled by combinations of the three methods.

Before installing any type of modern temperature control equipment, it is necessary to see that the heating system is put in good operating condition. In general, the heating system in a building is not given the attention that other mechanical equipment is given, because it will continue to function, after a fashion, even though changes in piping, location of radiation, settlement of piping, and the normal wear and tear or other changes have taken place. Because of this depreciation of the system, operation becomes more and more costly, and parts of the building have to be greatly overheated in order to prevent underheating in other parts. Vents, traps, vacuum pumps, and valves should be given a careful inspection and replaced or repaired, if required. The piping should be of adequate size and graded properly. The return piping should be inspected, and any pockets or lifts removed and properly vented. These inspections and repairs are not costly, and may prevent a much greater outlay in future years. In most cities district heating companies will be willing to make a survey of heating systems, and offer recommendations in regard to operation and changes in piping layout. The selection of control equipment depends upon the type and size of building, and the degree of saving which may be obtainable.

PANEL HEATING CONTROL

Automatic controls for radiant and convective heating differ somewhat due to the thermal inertia characteristics of the panel heating surface, and the increase in the mean radiant temperature within the space under increasing loads for panel heating.

Effect of Inertia of Panel

If a panel has considerable heat storage capacity (as compared with a convector or conventional radiator) it will continue to emit heat for some time after the room thermostat has become satisfied and has shut off the

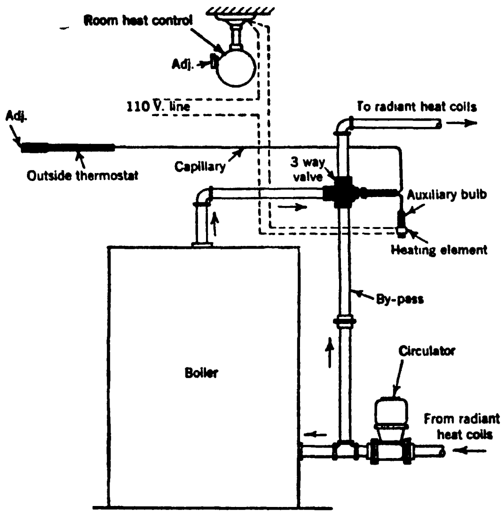

FIG. 3. PANEL HEATING CONTROL SYSTEM

supply of heating medium. This will cause uncomfortably warm conditions to exist in a space. Also, there will be a considerable delay between the time the thermostat calls for heat and the time heat is actually delivered to the space (because of the large part of the heat that must first be stored in the thermally *heavy* radiant surface). Whenever inertia exists in the source of heat supply, uncomfortable cycling of space conditions will result unless means of anticipating load changes *before* they occur in the space, or means of setting the basic energy supply rate from load conditions, are provided.

If a thermally heavy radiant surface is used, the primary control should be actuated by outdoor temperature (load) to determine the basic temperature of the heating medium supplied to the radiant surface. To allow for variations in internal load, an inside thermostat should be used as a high limit to reduce further the heat input, if necessary. If a thermally *light* radiant surface is used, controls may be applied in the same manner as for typical convection heating.

The terms *thermally heavy* and *thermally light*, referring to capacity for heat storage, are comparative and descriptive rather than exact. For example, a concrete floor panel in a frame structure without insulation would represent a *heavy* panel in a *light* structure. A frame type (metal lath and plaster) panel in a concrete structure would represent a *light* panel in a *heavy* structure. As indicated previously, a *heavy* panel in a *heavy* structure provides comfortable conditions if outside controls are used in addition to the inside thermostat. But if a *heavy* panel is used in a *light* structure, rapid changes in outdoor conditions may cause discomfort in spite of outdoor controls, because the structure reacts so much more rapidly than the radiant heating surface.

Since thermally *heavy* radiant surfaces introduce considerable lag in the heating system, it is desirable that a control system be capable of maintaining the lag at a minimum. One method of reducing the lag in heat output is to utilize design water temperature in the panel, thus causing maximum rate of heat from the water in the pipe coils through the concrete slab to the surface of the panel, and into the space when heat is needed. Satisfactory control can be obtained only if the control system is sensitive

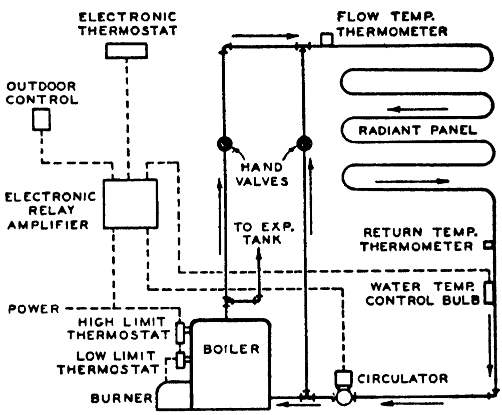

FIG. 4. ELECTRONIC CONTROL SYSTEM FOR PANEL HEATING

enough to regulate the heat output of the panels with sufficient precision to prevent over-heating, and to anticipate heat demand as affected by outside weather conditions.

Two of the many methods used to achieve satisfactory control are shown in Figs. 3 and 4.

Fig. 3 shows a control system operated by an outside thermostat in conjunction with a room heat control instrument. The outside thermostat modulates the temperature of the circulating water in the coils by mixing some of the hot water leaving the boiler with a proportionate amount of return water which is diverted to the three-way valve.

One type of room instrument consists of a blackened copper sphere of 6 or 8 in. in diameter, in which a cylindrical sump contains a volatile liquid. A small electric heating coil creates in the sphere a vapor pressure which remains constant as long as the total heat loss from the sphere is at the desired rate. If the Operative Temperature becomes too high for comfort, a greater vapor pressure results from the smaller heat loss from the sphere. This acts on a diaphragm and reduces the supply of heat to the room. With too low an Operative Temperature, the reverse action occurs. A similar instrument, which has an electric heating element for warming the air inside the sphere and the thermostat-operated switch, is also used for controlling room conditions.

Fig. 4 shows a typical *electronic control* system. The electronic control system operates the circulator in cycles. It varies the flow of water to the panel in proportion to the heat demand, as measured from room, outdoor, and panel conditions. A low limit immersion thermostat, set to maintain desired water temperature in the boiler, starts the burner. A high limit immersion thermostat is set to stop the burner at maximum desired boiler water temperature. A manual by-pass, as shown, is desirable to prevent the flow of water into the panel at temperatures in excess of design conditions.

Circuit Balance

In addition to a thermostatically controlled device for modulating the temperature of the circulating water, it is advantageous to insert in each circuit a locked flow control or adjustable resistance to give uniform conditions throughout all rooms. Owing to unforeseen difficulties with varying frictional losses in pipes, emission factor, and exposures, it is an advantage to be able to regulate permanently the flow through each circuit by means of a key operated valve as indicated in Fig. 3.

Compensation for Increase of MRT

Due to the increase in MRT (mean radiant temperature) within a panel heated space, which necessarily takes place as the heating load increases, the air temperature should theoretically be lowered to maintain comfort. In ordinary structures, with normal infiltration loads, the required reduction in air temperature is not great and a conventional fixed control point room thermostat may be used. If a large infiltration load exists, or if untempered mechanical ventilation is employed, a thermostat with variable control point should be considered. Because of the relationship between MRT and air temperature in the space (and the variable MRT from point to point in the space) a conventional type of room thermostat (either fixed or variable control point as previously determined) measuring principally air temperature, should provide simple and satisfactory control.

Lowered Night Temperature

In general, lowered night temperature control is not recommended with *heavy* panels, though it may be satisfactory with *light* panels.

INDICATING AND RECORDING EQUIPMENT

In addition to the automatic control of temperature and humidity conditions, visual indication and permanent chart records of the variables involved, are desirable. They provide an accurate check on the performance of the system, both from the standpoint of conditions maintained, and cost of operation. Instruments are available to provide accurate records of these variables such as pressure, temperature, humidity, flow, and CO_2, which go to make up a complete heating or air conditioning system. In some cases the control equipment is provided with indicating or recording mechanisms, by means of which the performance of the controls may be observed or recorded, and in other cases, separate instruments are used for the purpose.

CHAPTER 39

MOTORS AND MOTOR CONTROLS

Fundamentals of Motor Selection; Alternating Current Motors, Types and Control Equipment; Motor Rating; Functions of Motor Control Equipment; Glossary of Motor Terms, Enclosures, Speed Classification and Mounting

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THE electric motor, available in many different types suitable for various services, is now the most widely used form of prime mover. The equipment for starting, controlling and protecting these motors varies with the type and with the functions it is desired to attain. Motors are divided into two general classifications, *alternating-current* or *direct-current*, depending on the power source to be used.

FUNDAMENTALS OF MOTOR SELECTION

The following characteristics of the *power supply* should be determined: (1) whether current is alternating or direct, (2) voltage, (3) alternating current phase, (4) alternating current frequency, (5) voltage regulation, (6) continuity of power.

1. *A-C vs. D-C Systems.* For most applications, a-c supply is satisfactory since suitable performance can usually be obtained with a-c motors and control. Where special characteristics, such as an extra wide speed range and severe accelerating or reversing duty are involved, conversion by means of motor generator sets, by rectifiers, or in special cases by converters, may be justified.

2. *Voltage.* Standard conditions of voltage and frequency are the values listed on the name plate of the motor. Reasonable horsepower design limits are given in Table 1. Power lines are often given voltage ratings known as *nominal system voltages* which are numerically slightly different from the standardized motor voltages.

3. *Phases.* Three-phase power supply is most desirable, but only single phase is offered for most residential and rural districts.

4. *Frequency.* Sixty-cycle systems predominate in the United States. In foreign countries, 50-cycle systems are common and nominal system voltages are frequently different.

5. *Voltage Regulation.* The voltage regulation of the power supply should be known in order to select motors which will deliver sufficient torque even with the probable drop in voltage, to start and carry the load. All induction motor torques and synchronous-motor starting and pull-in torques vary as the square of the voltage.

6. *Continuity of Power.* Dips in voltage from switching or other line disturbances may necessitate time-delay undervoltage protection, and, in case of synchronous-motors, high torque designs and resynchronizing control. Sustained low voltage may necessitate higher torque motors.

The following characteristics of the *driven machine* should be determined: (1) mechanical arrangement including position of motor and shaft, portability desired, drive connection, and space limitations; (2) speed range desired; (3) horsepower requirement; (4) torque; (5) inertia; (6) frequency of starting.

1. *Mechanical Arrangement.* Arrangement of the driven machine usually determines whether a *horizontal* or *vertical* motor is needed. Horizontal motors are more generally available and less expensive; most grease-lubricated ball-bearing motors will operate in either position. Fractional-horsepower waste-packed sleeve-bearing

Table 1. Reasonable Horsepower Design Limits for Standard Motor Voltages

Power Supply	Standard Motor Voltage	Suggested Minimum Horsepower	Suggested Maximum Horsepower
Alternating 1-phase	115 230	None None	1 15
Alternating 3-phase	110 220 440–550 2300 4000 4600 6600	None None 1 50 100 250 400	15 200 1000 6000 7500 8000 None

motors are satisfactory for short periods of vertical operation where no thrust is involved.

If shaft is *tilted* for momentary operation, special construction of bearing housings will be required for oil-ring-lubricated sleeve-bearing motors, to avoid loss of lubricant. In case of long periods of tilted operation, bearings suitable for end thrust may be necessary. Ball-bearing motors with grease lubrication are suitable for tilted operation.

Most motors are suitable for mounting with base above a horizontal shaft or to one side of the shaft, provided the end shields are rearranged. If, during operation, the angle of the motor (with regard to the horizontal shaft) changes more than 10 or 12 deg, a ball-bearing motor will usually be required. Sleeve-bearing motors are also applicable within the angle given if modified oil gages are provided.

On portable machines, motors of greater compactness and less weight than standard may be required, and special bearing construction may be needed, except for ball-bearing motors. Direct connection should always be considered where machine speed coincides with available motor speed.

Maintenance, efficiency, power factor, space and initial cost, will determine the choice between *direct connection* and other methods, such as *belt, chain* or *gear* drive. When direct connection is possible (where parts of the driven machine, such as shaft or bearings, are common with the motor structure) a *built-in* construction may be advantageous.

Belt Drive. Diameters and widths of pulleys or sheaves and center distances are factors in determining motor-bearing pressures and shaft deflection. Flat belts should not run at greater speeds than about 5000 fpm. Application of flat belting to vertical-shaft motors is difficult.

Chain Drive. The chain manufacturer should be consulted so that the best drive on a basis of quietness and economy of operation may be selected.

Gear Drive. Compactness and arrangement of drive often indicate *gear motors*, which are obtainable in a variety of mechanical constructions with speed ratios of 3 to 1 upwards, and are generally limited to about 75 hp maximum. Where the pinion of ordinary spur gearing is mounted on the motor shaft, two-bearing motors

Table 2. Speed Ranges for Various Types of Motors

Power Supply	Type	Speed Range
Single Phase a-c	*Brush-shifting repulsion motor *Capacitor-motor with tapped winding Multi-speed capacitor-motor	3:1 2:1 2 or 3 fixed speeds
Poly-phase a-c	Multi-speed squirrel-cage *Wound-rotor motor *2-speed wound-rotor motor Brush-shifting shunt motor *Brush-shifting series motor Squirrel-cage motors with variable frequency supply Motor-Generator Set—D-c Drive Motor Rectifiers—D-c Drive Motor	2, 3 or 4 fixed speeds 2:1 4:1 20:1 3:1 Very wide range Very wide range Very wide range
d-c	Shunt-wound standard *constant-speed* motor with field control *D-c motor with armature control Adjustable-speed motor Shunt motor with adjustable voltage supply	2:1 in some cases Wide From 3:1 to 6:1 Very wide

* Speed regulation relatively wide. Unsuitable for some loads.

are limited in horsepower ratings. Maximum pitch-line speed with steel pinions is about 1400 fpm.

The selection of the motor part of a *gear-motor* is the same as for a conventional motor.

Space limitations may affect the choice of motor and require (a) built-in construction; (b) a gear-motor; (c) forced ventilation using an external blower; or (d) a small frame with Class B insulation permitting higher temperature rise.

2. *Speed Range.* Where more than one speed or a range of speeds is required, one of the motor types listed in Table 2 may be applicable, depending upon the power supply and the speed range required.

3. *Horsepower Requirement.* The horsepower required by the driven machine determines the motor rating. Where the load varies with time, a horsepower vs. time curve will permit determination of the peak horsepower required. The calculation of the root-mean-square (rms) horsepower indicates the proper motor rating from a heating standpoint. In case of extremely large variations in load, or where shut-down, accelerating, or decelerating periods constitute a large portion of the cycle, the rms horsepower may not give a true indication of the equivalent continuous load, and the motor manufacturer should therefore be consulted.

Where the load is maintained at a constant value for an extended period (varying from 15 min to 2 hr, depending on the size), the horsepower rating required will usually not be less than this constant value, regardless of other parts of the cycle.

If the driven machine is to operate at more than one speed, the horsepower required at each speed must be determined.

4. *Torque.* The torque required to operate the driven machine at every moment between initial breakaway and final shutdown is important in determining the type of motor. A torque-speed curve is desirable and sometimes essential.

The *starting torque* or breakaway torque required by the driven machine may be as low as 10 percent, as in the case of medium-sized centrifugal pumps, or as high as 225 to 250 percent of full-load torque, as in the case of a loaded reciprocating two-cylinder compressor. The breakaway torque may vary greatly at different times because of frequency of start, temperature changes, type and amount of lubricant, etc. The motor torque available at the shaft must be well above the torque required by the driven machine, taking into consideration these variables as well as the possibility of low voltage and the type of starter used.

The torque required after breakaway for *acceleration to full speed* varies with different driven machines, remaining at a rather high value throughout acceleration for such machines as loaded compressors and plunger pumps. The torque delivered by the motor must at all points, up to full speed, be in excess of the torque required by the driven machine. The greater this excess torque, the faster will be the acceleration. The approximate time required for acceleration from rest to full speed is:

$$\text{Time in seconds} = (\text{rpm}) \times WR^2 \div (T \times 308) \tag{1}$$

where

(rpm) = full-load speed in revolutions per minute.
T = average torque available for acceleration, foot-pound.
WR^2 = inertia of rotating parts, pound-foot square.

If the time to accelerate is greater than about 20 sec, special motors or starters may be required to avoid overheating.

5. *Inertia of Driven Machine.* The inertia or flywheel effect WR^2 of the rotating parts of the driven machine affects the accelerating time and, therefore, the heating of motors and control, particularly where reversing duty or frequent starting is involved.

Where synchronous motors are applied, the WR^2 must be known, since the pull-in torque required of this motor varies approximately as the square root of the total WR^2 of motor and load.

The WR^2 of a rotating member of the driven machine which operates at a speed different from that of the motor may be converted to an equivalent value at the motor shaft by multiplying by

$$[(\text{rpm of rotating member}) \div (\text{rpm of motor})]^2 \tag{2}$$

6. *Frequency of Starting.* The frequency of starting the driven machine affects the motor and control by increasing their heating, particularly where accelerating

Table 3. Classification of Motors

	Type	Speed Characteristics	Full Voltage		HP Range	Application See Footnotes (a) to (e)
			Starting Torque	Starting Current		
	Constant Speed Drives					
Polyphase a-c	Squirrel-cage general purpose Design A	Constant	Normal 1-2.5 times	High 6-8 times	All	(a) Fans and (c) centrifugal pumps and centrifugal compressors
Polyphase a-c	Squirrel-cage Design B	Constant	Normal 1-2.5 times	Normal 5-6 times	Medium Small	(a) Fans and centrifugal pumps and centrifugal compressors
Polyphase a-c	Squirrel-cage Design C	Constant	High 2-2.5 times	Normal 5-6 times	Medium Small	(b) Reciprocating pumps (e) and compressors started loaded
Polyphase a-c	Wound rotor	Constant or variable	High 1-2.5 times (with secondary control)	Low 1-3 times	All	(a) Hoists (b) reciprocating pumps and compressors (c) and frequent (e) or hard start
Polyphase a-c	Synchronous high speed	Exactly constant	Normal 0.75-1.75 times	Normal 5-7 times	Medium Large	(a) Fans and centrifugal pumps and centrifugal compressors
Polyphase a-c	Synchronous low speed	Exactly constant	Low 0.3-0.4 times	Low 3-4 times	Medium Large	(a) Reciprocating compressors starting unloaded
Single phase a-c	Two value capacitor	Constant	High	Normal	Small	(b) Pumps and compressors
Single phase a-c	Permanent split capacitor	Constant	Low	Normal	Fractional	(a) Fans, Blowers
Single phase a-c	Capacitor start	Constant	Moderate	Normal	Small Fractional	(a) Fans and pumps
Single phase a-c	Repulsion Induction	Constant	High	Normal	Medium Small	(a) Fans (b) pumps and compressors
Single phase a-c	Split phase	Constant and adjustable	Normal	Normal	Fractional	(a) Fans (b) pumps and compressors (d) fans—direct
	Adjustable Speed Drives					
Polyphase a-c	Squirrel-cage high slip. Transformer adjustment	Variable	Normal	Normal	Medium Small	(a) Fans
Polyphase a-c	Squirrel-cage separate winding or regrouped poles	Constant multi-speed	Normal or high	Normal or low	All	(a) Fans (b) pumps and (c) compressors
Polyphase a-c	Wound rotor	Variable	High (with secondary control)	Low	All	(a) Fans (b) centrifugal pumps and compressors
Single phase a-c	Repulsion	Variable	High	Normal	Low and Fractional	(a) Fans, centrifugal pumps (b) compressors
Single phase a-c	Capacitor low torque tapped winding	Variable two speed	Low	Normal	Fractional	(d) Fans, direct
Single phase a-c	Capacitor low torque transformer adjustment	Variable	Low	Low	Fractional	(d) Fans
Single phase a-c	Split phase regrouped poles	Constant	Normal	Normal	Fractional	(d) Fans

a. Drives having medium or low starting torque and inertia (WR^2) such as fans and centrifugal pumps or reciprocating pumps and compressors started unloaded.
b. Drives having high starting torques, such as reciprocating pumps and compressors started loaded.
c. Similar to (*a*) except where frequent or hard starting (large WR^2) requires a higher starting and accelerating torque. *d.* Fans direct connected. *e.* Stoker drives.

time is prolonged by high WR^2 and high load torques. In general, driven machines starting more than 4 to 6 times per hour may require special motors and control.

ALTERNATING CURRENT MOTORS

Alternating current motors are divided into two main classifications: *polyphase* and *single phase* (see Table 3), according to the type of power supply. They are further subdivided by type of motor winding.

When polyphase power is available it is usually found more economical to apply polyphase motors in preference to single phase motors. A typical 5 hp, 1200 rpm capacitor start-induction run single phase motor, for instance, will cost approximately twice as much as the corresponding three phase *Design B* squirrel-cage motor. In addition, the polyphase motor has the advantages of higher power factor and higher efficiency.

TABLE 4. LOCKED-ROTOR CURRENT OF THREE-PHASE, 60-CYCLE MOTORS AT 220 VOLTS[a, b]

HP	Design B, C, and D Amperes	Design F Amperes	HP	Design B, C and D Amperes	Design F Amperes
1 or less	24[c]	.	30	435	270
1½	35	.	40	580	360
2	45		50	725	450
3	60		60	870	540
5	90		75	1085	675
7½	120		100	1450	900
10	150		125	1815	1125
15	220		150	2170	1350
20	290		200	2900	1800
25	365				

[a] The locked-rotor current of three-phase, 60-cycle, constant-speed, induction motors, measured with rated voltage and frequency impressed and with rotor locked, shall not exceed the tabulated values.
[b] Locked-rotor current at other voltages shall be inversely proportional to the voltage.
[c] For 1 hp or less the value is given per hp.

Polyphase Motors

The three types of polyphase motors are: *squirrel-cage induction* motors, *wound rotor induction* motors, and *synchronous* motors.

Squirrel-cage motors are specified by *NEMA* standards providing a variety of speed and torque characteristics. *Design A* motors provide normal starting torque at starting current in excess of *Design B* motors, and are suitable for constant speed application to equipment such as fans and blowers. *Design B* motors provide normal starting torque with *NEMA* starting current values shown in Table 4, which are acceptable by many power companies for full voltage starting. They are used for the same type of application as *Design A*. *Design C* motors provide high starting torque with starting current same as *Design B*, and are used on compressors started without unloaders, and on reciprocating pumps. *Design D* motors have high slip* and are used with flywheels for widely pulsating loads on equipment such as reciprocating compressors and pumps where other motors would draw high peak currents.

Figs. 1 and 2 illustrate the characteristics of squirrel-cage motors. Both power factor and efficiency are improved if the motors are operating

* Refer to Glossary at end of chapter.

as near rated load as possible. In addition, as shown in Fig. 2, power factor and efficiency are better for higher speed motors.

Wound Rotor motors are used for applications requiring high starting torque at low starting current, because a wound rotor motor with its controller and resistance can develop full load torque when starting with about full load current. For comparison, a squirrel-cage motor would require from 3 to 5 times as much current to develop full load torque at starting. The wound rotor motor is also used for varying speed service to drive fans, blowers, and other continuous duty apparatus.

The addition of resistance to the secondary winding of the wound rotor motor changes the speed torque characteristics. The motor speed, with

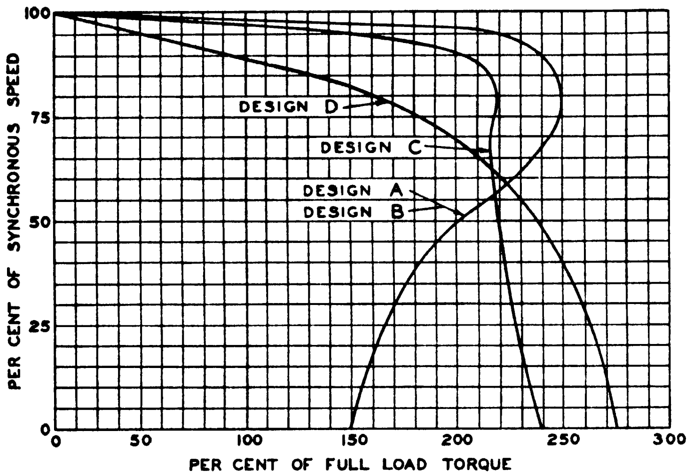

FIG. 1. SPEED TORQUE CHARACTERISTICS OF SQUIRREL-CAGE MOTORS

the resistance added, is dependent on load, and consequently, the motor has very poor speed regulation when secondary resistance is added to reduce the speed to values below 50 percent.

Synchronous motors are used for continuous duty applications at constant speed where efficiency and power factor are important. Another advantage of these motors is that of lower initial cost in large sizes and for low speeds when compared with squirrel-cage type motors.

The outstanding advantage of the synchronous motor is that its power factor can be changed to compensate for the low power factor of other drives in the same location. Lagging power factor is an inherent characteristic of all induction apparatus, such as induction motors and neon signs. Unless synchronous motors or capacitors are installed, the plant power factor may be comparatively low. This does not necessarily mean that corrective equipment must always be installed, but in most cases it is desirable to determine what advantages may be gained by improving the power factor. With purchased power, if the rates include a clause embodying a penalty for low power factor, or a bonus for high power factor, the saving in power costs may often make a very good return on the investment required for the corrective equipment.

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* Refer to Glossary at end of chapter.

Synchronous motors are used to drive fans, blowers, pumps, compressors and other applications. Compressor applications having a high peak torque require the use of flywheels to smooth out power peaks, and should always be referred to the electrical manufacturer for recommendations.

Synchronous motors are provided with built-in damper windings on the rotor and operate during the starting period similarly to squirrel-cage motors. After the motor is nearly up to speed, field excitation is applied and the motor draws into step at synchronous speed. After excitation is applied, the motor runs at exactly constant speed and will remain at this

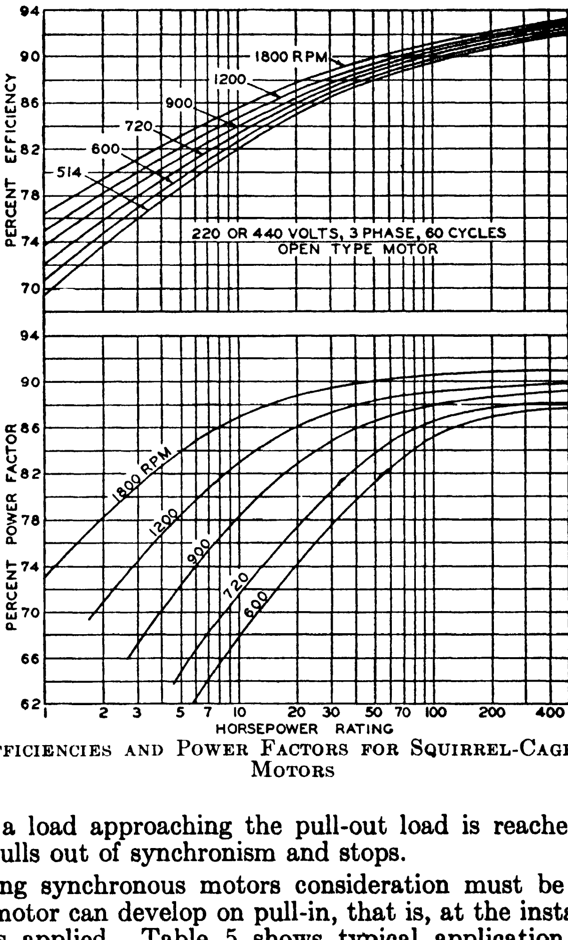

FIG. 2. EFFICIENCIES AND POWER FACTORS FOR SQUIRREL-CAGE INDUCTION MOTORS

speed until a load approaching the pull-out load is reached, whereupon the motor pulls out of synchronism and stops.

In applying synchronous motors consideration must be given to the torque the motor can develop on pull-in, that is, at the instant when field excitation is applied. Table 5 shows typical application requirements of synchronous motor drives, listing starting, pull-in, and pull-out torques.

Multi-Speed motors provide flexibility in many types of drives. Synchronous motors can be furnished only with a 2 to 1 ratio in speed, single winding. Squirrel-cage induction motors may be 2, 3 or 4 speed. Two-speed induction motors are usually of single winding type, having a 2 to 1 speed ratio such as 600 rpm and 1200 rpm, or may be double winding. Three-speed induction motors are always two winding, and four-speed motors are usually two winding with a 2 to 1 speed ratio in each winding. Motors can be provided in constant torque varying torque, or constant horsepower ratings. The constant horsepower type of motor is considerably larger than the constant torque motor, due to the fact that the same horsepower must be developed at either reduced speed or high speed.

In selecting two-speed motors for fan, pump, blower, or compressor applications, it is usually found that two winding motors are more expensive than the single winding type. The control cost for two-speed, two winding motors, however, is more economical, and therefore the combined price of both motor and control for the two winding motor is only slightly higher. Because of the improved performance of the two winding motors, and because of the factor of safety provided by two independent windings, the increased cost is frequently worth the difference.

TABLE 5. TYPICAL APPLICATION REQUIREMENTS OF SYNCHRONOUS MOTOR DRIVES SHOWING STARTING, PULL-IN AND PULL-OUT TORQUES

	Application	Method of Connecting Motor to Load	Starting Conditions	Torques			Remarks
				Starting	Pull-in	Pull-Out	
Fans	Exhaust and ventilating	Coupled or belted	Usually loaded	50	60–125	150	WR^2 of fan must be considered
Blowers	Cycloidal positive	Coupled or engine type	Unloaded	40–60	40–60	150	Two-speed motors sometimes used
Blowers	Blowing engines reciprocating	Engine type	Unloaded	40	40–60	150	
Blowers	Turbo high speed	Direct connected or step up gear	Unloaded (intake closed)	30	50	150	WR^2 of blower must be considered
Compressors	Air	Engine type	Unloaded	40	30	150	Flywheel effect important
Compressors	Ammonia and ammonia booster	High speed—belted Low speed—engine type occasionally coupled	Unloaded (by by-pass)	40	30	150	Flywheel effect important
Compressors	Freon	High speed—belted Low speed—engine	Unloaded (by by-pass)	45	50	150	Flywheel effect important
Compressors	Gas reciprocating	High speed—belted Low speed—engine	Unloaded (by by-pass)	40	30	150	Flywheel effect important

Single Phase Motors

Single phase induction motors have auxiliary windings or devices for starting, and are classified by the method used.

Capacitor start motors develop high starting torque in fractional horsepower ratings, and moderate starting torque in larger ratings. They are used for constant speed drive such as fans, blowers and centrifugal pumps. During the starting period, a winding with a capacitor in series is connected to the motor circuit and when the motor comes up to speed, a centrifugal switch cuts the capacitor and second winding out of the circuit.

Two-value capacitor motors develop high starting torque employing a starting capacitor and a running capacitor. The starting capacitor gives high starting ability, but is suited for short time operation only, and is cut out for the running condition by a centrifugal switch. The running capacitor gives high efficiency at full speed. These motors are used on compressors, reciprocating pumps and similar equipment which may start under heavy load.

Permanent split capacitor motors have low starting torque and are ideally suited for small fan drives. Operation is similar to the capacitor start motor, except that the capacitor is not cut out when running.

Repulsion-Induction motors develop high starting torque. The motors have two rotor windings—a squirrel cage for running and a wound rotor connected to a commutator for starting. No switching device is required to change from starting to running winding as this is accomplished by a gradual shift with speed in the magnetic flux path, so that near rated speed the motor operates completely on the squirrel cage winding.

Repulsion Start-Induction Run motors are similar to the repulsion-

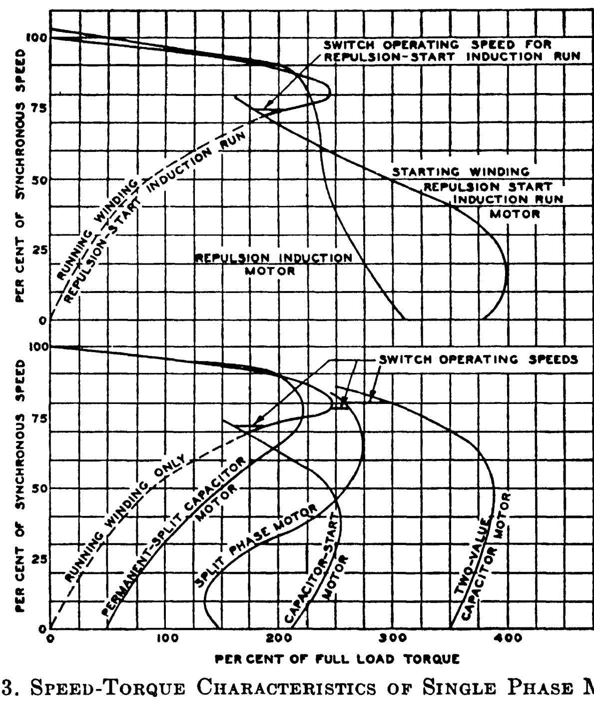

FIG. 3. SPEED-TORQUE CHARACTERISTICS OF SINGLE PHASE MOTORS

induction motor, but they have only the commutator winding. They are supplied with a centrifugal short circuiting switch which shorts the commutator bars when the motor comes up to speed to obtain a winding approximately like the squirrel cage in its function.

Repulsion-Induction and *Repulsion-Start-Induction Run* motors are suitable for applications, such as industrial compressors, requiring high breakaway torque, and where commutator and brush noise are not factors.

Split Phase motors have a high resistance auxiliary winding which is in the circuit during starting, but is disconnected through the action of a centrifugal switch as the motor comes up to speed. Under running conditions it operates as a single phase induction motor with one winding in the circuit. These units are available for the small horsepower ratings, winding circuit. The relay then keeps these contacts open because there is sufficient voltage induced in the starting winding, when the motor is running, to hold the relay in the open contact position.

CONTROL FOR ALTERNATING CURRENT MOTORS

Squirrel-Cage motors are usually linestarted where power company limitations permit. In sizes up to 5 hp at 220 volts, or 7-½ hp at 440 volts, polyphase motors may be started by means of manual switches having overload current elements for motor protection. In larger ratings a linestarter is usually provided with either an additional safety switch or circuit breaker for disconnecting and short circuit protection. Reduced voltage starting may be either of manual or push button controlled magnetic type. In specifying this type of starter, consideration should be given to the fact that starting torque of squirrel-cage motors varies as the square of the applied voltage. For example, a motor developing 100 lb-ft starting

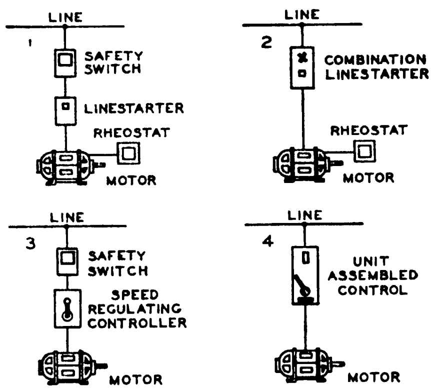

Fig. 5. Recommended Controls for Wound Rotor Motors

torque on full voltage would produce only 25 lb-ft torque on starting on half rated voltage. Fig. 4 illustrates recommended control practice for squirrel-cage motors.

Wound Rotor motors require control of both primary and secondary circuits. The primary* control may be the same as for squirrel-cage motors, manual or magnetic, at full voltage. Secondary* control provides means of varying secondary resistance for starting and speed control. The secondary controller should be specified for starting duty only, or for speed regulating duty. If the secondary controller is to be used for speed regulating duty, the percent speed reduction, the number of speed control points, and the type of load (variable or constant torque) should be specified. Fig. 5 illustrates recommended control practice for wound rotor motors.

Synchronous motor starters should provide pull-out protection, automatic synchronization or automatic stopping of the motor after pull-out, and insurance of complete starting sequence, as well as overload and low voltage protection. The control may be either magnetic or semi-magnetic at full or reduced voltage. Semi-magnetic starters provide automatic field control, but require hand operation for closing the line contactors to start and transfer to full voltage.

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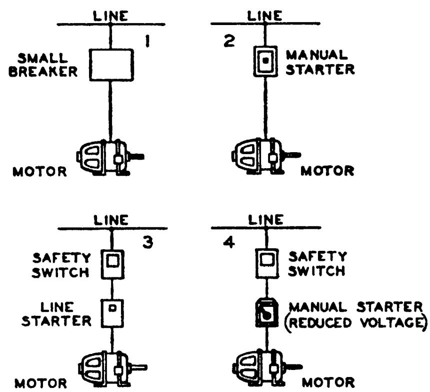
* Refer to Glossary at end of chapter.

In applying reduced voltage starters to synchronous motors it should be remembered that, since these motors are started on damper windings and function during the acceleration period similarly to squirrel-cage motors, the starting torque varies as the square of the applied voltage. Consideration should be given to insure development of sufficient motor torque to accelerate the load.

Multi-Speed control may be either manual or magnetic, and at full or reduced voltage. When using automatic magnetic control with two-, three-, and four-speed separate winding or consequent pole motors, control may be obtained from a remote point by means of a push button master switch. The various speeds of the motor are obtained from the master switch by simply depressing the correct push button. This is known as selective speed control. It is commonly used in the smaller theater installations where the fan and motor are located backstage and the speed control is located in the lobby.



Arrangements 2, 3 and 4 are optional for motors up to 7½ hp, 220 volts.

FIG. 6. RECOMMENDED CONTROLS FOR SINGLE PHASE MOTORS

Multi-speed motor controllers may be provided with compelling relays which make it necessary for the operator to press the first speed button before regulating the motor to the desired speed. This insures that the motor is always started at low speed before adjusting to a higher speed.

Timing relays which provide for automatic acceleration may be used for control. With this feature the motor will always start at low speed and automatically accelerate to the desired speed. Decelerating relays may be used to reduce the shock effect of the braking action on the motor and drive when the speed is reduced from a higher to a lower speed.

Single Phase motor control usually consists only of a linestarter, either manual or magnetic. In some cases it is desirable also to provide a disconnect switch. Fig. 6 illustrates the recommended controls.

MOTOR RATING

The rating of an electric motor depends upon the total temperature which the motor attains under operating conditions. This total temperature depends on both the ambient temperature and the temperature rise of the motor. As motor temperature rise is in turn determined by the ability of the motor to dissipate heat, circulation to the motor should not be restricted. Improper selection of motors with regard to temperature ratings may result in high motor operating temperatures and accompanying reduction in motor life.

In general, the electrical insulation is the portion of the motor most susceptible to injury from high operating temperatures. Of the several types of insulation which are available, the most common type, specified as Class A by the *National Electrical Manufacturers Association*, consists of cotton, felt, paper or similar organic materials, and permits a 55 C rise in temperature over a 40 C ambient temperature for totally enclosed motors. Class B insulation consists of mica, asbestos, fiber glass, or similar inorganic materials, and permits a 75 C rise in temperature over the 40 C ambient for totally enclosed motors. Other types of insulation, such as silicone resin, are available and permit much higher operating temperatures.

The mechanical construction of the different types of motor enclosures, and the rise in temperature with Class A insulation for each type, are enumerated in the glossary at the end of this chapter. Since the difference between the hottest spot and the maximum observable temperature, as measured by a thermometer, is greater for an open machine than for an enclosed machine, the permissible temperature rise is 50 C for an open motor.

FUNCTIONS OF CONTROL EQUIPMENT FOR MOTORS

In general, control equipment for all types of motors should provide (1) means of disconnecting the motor from the power supply, (2) means for starting the motor, (3) overload protection for the motor, (4) protection against low voltage, and (5) means for varying the motor speed.

Full voltage starting for motors is preferable because of its lower first cost and simplicity of control. Except for d-c machines, most motors are mechanically and electrically designed for full voltage starting. The starting inrush current, however, is limited in many cases by regulations of power companies because of the voltage fluctuations which may be caused by heavy current surges. It is therefore often necessary to reduce the starting current below that obtained by across-the-line starting. The power supplier should be consulted to determine the allowable inrush current for any given location.

The choice between full voltage and reduced voltage starting is governed almost entirely by inrush current limitations. The starting torque of all motors varies with the starting current, and it is therefore necessary to insure that the motor is supplied with sufficient current to develop enough torque to accelerate the load.

In present practice overload protection of motors is obtained by use of thermal overload inverse time limit type protection. The usual setting of such protection devices is not to exceed 125 percent of rated full load current for open 40 C rise motors, and not to exceed 115 percent of rated full load current for all other motors, the element tripping after a definite interval of time. The *National Electrical Code* requires the addition of fuses or circuit breakers to protect the overload elements from severe short circuit currents.

Two types of protection are available against low voltage at the motor terminals. One type, called low voltage release, permits the motor line contactor to drop out on low voltage and to close again when the voltage returns to normal, thereby restarting the motor when the abnormal condition is ended. The second type, called low voltage protection, causes the motor line contactor to drop out on low voltage, but prevents restarting when the voltage returns to normal except by the action of an operator. This latter type of protection is desirable where it is necessary for the operator to make initial starting adjustments on the machine.

Manual control for an alternating or a direct current motor is usually located near the motor. When so located an operator must be present to start and stop or change the speed of the motor by operating the control mechanism. Manual control is sometimes employed only as a device to give overload protection, and another device is employed to start and stop the motor. Manual control is used particularly on small motors which operate unit heaters, small blowers, and room coolers in an air conditioning system. In other cases manual control in the form of drums, when used with multi-speed motors, is used only as a speed setting device, while the starting and stopping functions operate automatically through thermostats and pressure switches.

Because of the increasing complexity of air conditioning systems, the equipment is operated preferably by automatic control, and less dependence is placed on manual operation and regulation.

Automatic control of motor starters may be accomplished by the use of remote push button stations, by a thermostat, float switch, pressure regulator, or other similar pilot devices. An added advantage of automatic control is that the main wiring for the starter may be installed near the motor, while the starter may be operated by a remote control device.

GLOSSARY

General Definitions

NEMA is the abbreviation for the *National Electrical Manufacturers Association.*
AEIC is the abbreviation for the *Association of Edison Illuminating Companies.*
EEI is the abbreviation for the *Edison Electric Institute.*

Speed Regulation (d-c motors) is the change in speed between no-load and full-load, expressed in percent of full-load speed; for example, a motor having a no-load speed of 1200 rpm and a full-load speed of 1140 rpm would have a speed regulation of 5.26 percent.

Slip (a-c induction motors) is the difference between the motor speed and synchronous speed expressed in percent of synchronous speed, *e.g.*, a 1200 rpm motor operating at 1140 rpm would have a slip of 5 percent.

Torque is an expression of the turning effort developed by the motor at the shaft, and is usually expressed in ounce-feet for fractional horsepower motors, and in pound-feet for motors of larger ratings.

Primary is the term usually applied to the high voltage or line side of a transformer or motor. In the case of the wound rotor motor the primary is the stator winding.

Secondary is the term usually applied to the low voltage or load side of a transformer or motor. In the case of the wound rotor motor the secondary is the rotor winding.

NEMA Classification of Motor Enclosures

Open motors (40 C rise, rated load, 50 C rise, service factor load) are self-ventilated machines having no restriction to ventilation other than that necessitated by mechanical construction.

Protected motors (50 C rise) have all ventilating openings in the frame protected by perforated covers.

Semi-Protected motors (50 C rise) have the ventilating openings in the top half of the frame only protected by perforated covers.

Drip Proof motors (50 C rise) are so constructed that drops of liquid or solid particles falling on the machine at any angle not greater than 15 deg from the vertical, cannot enter the machine either directly or by striking and running along a horizontal or inclined surface.

Splash Proof motors (50 C rise) are so constructed that drops of liquid or solid particles falling on the machine or coming toward it in a straight line at any angle not greater than 100 deg from the vertical, cannot enter the machine either directly or by striking and running along the surface.

Totally Enclosed Non-Ventilated motors (55 C rise) are so constructed as to prevent exchange of air between inside and outside of the case, but are not air tight and are not equipped with external cooling means.

Totally Enclosed Fan-Cooled motors (55 C rise) are similar to totally enclosed, non-ventilated machines, except that exterior cooling is provided by means of a fan or fans integral with the machine.

Explosion Proof motors (55 C rise) have an enclosing case designed to withstand an explosion of a specified gas or vapor which may occur within it, and to prevent the ignition of the gas or vapor surrounding the motor by sparks, flashes, or explosions of the gas or vapor which may occur within the machine casing.

Dust Explosion Proof motors (55 C rise) have an enclosing case designed and constructed so as not to cause the ignition or explosion of an atmosphere of the specific dust, or to cause ignition of dust on or around the machine. (Proper overload protection and cleanliness are required for successful operation).

Water Proof motors (55 C rise) are so constructed as to exclude water applied in the form of a stream from a hose.

Dust Tight motors (55 C rise) are so constructed that the enclosing case will exclude dust.

Motor Speed Classifications

A Constant Speed Motor is one in which the speed remains practically constant with changes in load; *e.g.*, a d-c shunt wound motor or a-c squirrel-cage motor with low slip.

A Varying Speed Motor is one in which the speed varies with the load, usually decreasing when the load increases; *e.g.*, a d-c series motor or an induction motor with large slip.

An Adjustable Varying Speed Motor is one in which the speed can be adjusted gradually, but when once adjusted for a given load will vary in considerable degree with change in load; *e.g.*, a shunt wound d-c motor adjusted by armature resistance control.

An Adjustable Speed Motor is one in which the speed can be varied gradually over a considerable range, but when once adjusted remains practically unaffected by the load; *e.g.*, a d-c shunt motor with field resistance control. The standard ratings for open type, adjustable speed motors, having a speed range of 3 to 1 and greater are in accordance with the following:

(1) A standard continuous horsepower rating at 150 percent of minimum speed with a temperature rise of 40 C.

(2) The next higher standard continuous horsepower rating at 3 times minimum speed with a temperature rise of 40 C.

(3) Between 150 percent of minimum speed and 3 times minimum speed, the standard continuous horsepower rating with a temperature rise of 40 C will vary with the speed along a straight line connecting these two horsepower ratings. No further increase in horsepower is recognized above 3 times minimum speed.

(4) Below 150 percent of minimum speed the lower continuous horsepower rating (see preceding item 1) will apply with a temperature rise of 50 C.

Example: 20/25 hp, 400 to 1600 rpm. This motor may be rated 20 hp, 40 C at 600 rpm and 25 hp, 40 C from 1200 to 1600 rpm. Beteen 600 and 1200 rpm the rated horsepower increases directly with speed from 20 to 25 hp.

(5) Motors may also be rated 1 hour with temperature rise of 50 C with the higher horsepower rating (see preceding item 2) throughout the entire speed range.

Example: 20/25 hp, 400 to 1600 rpm. This motor may be rated 25 hp, 50 C 400/1600 rpm; 1 hour.

Mechanical Modifications

Vertical Mountings are available for such applications as pumps and agitators. This type of application may require a special umbrella-type hood to protect against dripping liquids.

Flanged Mountings are available for use where motors are built in as part of machines. Motors may also be supplied with flush plate mountings, suitable for close coupled pump and similar applications.

CHAPTER 40

SOUND CONTROL

Unit of Noise Measurement, Apparatus for Measuring Sound, General Problem, Kinds of Noise, Noise Transmitted Through Ducts, Design Room Noise Level, Noise Generated by Fans, Natural Attenuation of Duct System, Duct Sound Absorbers, Air Supply Noises, Cross Transmission Between Rooms, Controlling Vibration from Machine Mountings

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IN ventilating and air conditioning a building or a room, consideration must be given to the effect of the mechanical system on the acoustics of the space conditioned. It is important to consider also that the use of air conditioning often permits keeping the windows closed, thus giving relief from certain external noises, but at the same time increasing the necessity of providing adequate sound control.

It is assumed that in a given space the architect and acoustical engineer have produced a room or rooms which are satisfactory for speech, music, or other uses. The ventilating engineer's sole function is to ventilate and air condition these rooms properly so that they will be physically comfortable without adding any acoustical hazards.

UNIT OF NOISE MEASUREMENT

According to an international standard, the *decibel* (*db*) is the unit for expressing sound pressure levels. The *sound pressure level*, in decibels, is given by the relation:

$$db = 20 \log_{10} \left(\frac{P}{0.0002} \right) \tag{1}$$

where P = the sound pressure in dynes per square centimeter.

The reference pressure (0.0002 dynes per square centimeter) is a sound pressure which is slightly less than the threshold of audibility, at the frequency of 1000 cycles per second, for the person of average hearing. This reference point is approximately the minimum sound pressure that would be audible in a very quiet room to an observer having acute hearing. The ear is essentially a pressure operated device; hence, the sensation of loudness, or magnitude of sound is governed by the sound pressure existing at the point of reception.

The sound level meter measures sound level pressure, the measurement being expressed as *sound level* on a decibel scale with zero corresponding to the reference pressure of 0.0002 dynes per sq cm at 1000 cycles. In the higher ranges of the decibel scale, approaching 120 db, the sensation is one of feeling, and at higher levels the sensation becomes painful.

Associated with sound pressure level is the *sound intensity level*, expressed in decibels above a standard reference intensity. Sound intensity is the average rate of sound energy transmitted through a unit area normal to the direction of propagation, commonly expressed in watts per square centimeter. The *sound intensity level*, in decibels, is given by the relation:

$$db = 10 \log_{10} \left(\frac{I}{10^{-16}} \right) \qquad (2)$$

where I = the sound intensity in watts per square centimeter.

The reference intensity is 10^{-16} watts per square centimeter, coinciding with the reference pressure of 0.0002 dynes per sq cm or 2×10^{-4} microbar. A microbar is the unit of pressure commonly used in acoustics, one microbar being equal to one dyne per sq cm.

The relationship of the decibel scale to sound pressure and sound intensity is shown in Table 1. A stated sound level in decibels, under standardized procedure, will thus be related to a threshold of 0.0002 dynes per square centimeter, or to a threshold of 10^{-16} watts per sq cm. The standardization upon terminology, procedures and reference levels may be found in Standards[1] published by the *American Standards Association.*

APPARATUS FOR MEASURING SOUND

The measurement of sound or noise is conventionally made by means of a sound-level meter[2] consisting of a microphone, an amplifier, a variable attenuator, weighting networks, and an indicating meter which reads directly in decibels. The approved sound-level meter must comply with the specifications of the American Standard for Sound Level Meters for Measurement of Noise and Other Sounds, Z24.3-1944, approved and published by the *American Standards Association.* The meter is designed to indicate sound level above the standard reference level. Three measuring networks are generally provided: (1) flat response, (2) 70 db network and (3) 40 db network. The various networks are approximations of the equal-loudness contours relating intensity and frequency sensation response of the normal human ear.[1] Where there are no specific codes which specify the particular network to be used, general practice would indicate use of the 40 db network for sound levels up to about 55 db, the 70 db network for sound levels from about 55 db to 85 db, and the flat response network for higher levels. When sound level measurements are stated, the specific weighting network used, *i.e.*, 40 db, 70 db or flat response, should always be reported. Complexity in design and calibration, and variations in component parts of the sound level meter impose some deviation from design objective response. Allowable deviations in response or acceptable tolerations recognized in the Standard, vary from ±2 db in the 1000 cycle range to ±5 db, or more, below 100 cycles and above 1200 cycles per second.

GENERAL PROBLEM OF SOUND CONTROL

The problem confronting the air conditioning engineer is to design a system which will operate without increasing the noise level in the conditioned space. It is therefore necessary:

1. To determine the noise level existing without the equipment.
2. To ascertain the noise level which would exist if the equipment were installed without sound control.
3. To provide as a part of the installation, sufficient sound control appliances and treatment to reduce the sound level due to the installation to a sound level at least three decibels, and preferably five decibels, below that found in Item 1.

To accomplish this the engineer should have information of three kinds:

1. A knowledge of the noise levels currently considered acceptable in various rooms, in order that he may have a basis on which to proceed.

TABLE 1. DECIBEL SCALE VS. SOUND PRESSURES AND SOUND INTENSITIES

DECIBEL LEVEL	PRESSURE dynes per sq cm	INTENSITY watts per sq cm	DECIBEL LEVEL	PRESSURE dynes per sq cm	INTENSITY watts per sq cm
0	0.000200	1.000×10^{-16}	40	0.0200	1.000×10^{-12}
1	0.000224	1.259×10^{-16}	50	0.0631	1.000×10^{-11}
2	0.000252	$1\ 585 \times 10^{-16}$	60	0.200	1.000×10^{-10}
3	0.000282	$2\ 000 \times 10^{-16}$	70	0.631	1.000×10^{-9}
4	0.000317	2.520×10^{-16}	80	2.00	1.000×10^{-8}
6	0.000399	4.000×10^{-16}	90	6.31	1.000×10^{-7}
8	0.000503	6.310×10^{-16}	100	20.0	1.000×10^{-6}
10	0.000631	1.000×10^{-15}	110	63.1	1.000×10^{-5}
20	0.00200	1.000×10^{-14}	120	200.0	1.000×10^{-4}
30	0.00631	1.000×10^{-13}			

2. A knowledge of the nature and intensity of the noise created by the various parts of the equipment.

3. A knowledge of how, when necessary, to vary and control the noise level beween the equipment and the conditioned space.

In addition, the engineer should have sufficient information to predict the levels produced by noises which may be transmitted by the duct system from one conditioned space to another, or from an outside space to the conditioned space. In either case, the designer must know the probable noise level at the point where the noise originates. From this he can compute the attenuation or transmission loss required in order to bring this level down to that required in the conditioned space. If there is likelihood of direct transmission through a duct, the attenuation required may be computed as shown in section Noise Transmitted Through Ducts. If the transmission is through dividing walls, it will be necessary to refer to published data on losses through standard building constructions.[3]

Information concerning the sound levels created by ventilating and air conditioning equipment such as fans, motors, air washers and similar items, has not yet been completely established. However, numerous manufacturers are in a position to supply such data upon many of their products. Additional information is being collected. Uniformity in method of test and presentation of sound measurement data for fans, has been standardized in the Sound Measurement Test Code for Centrifugal and Axial Fans, developed by the *National Association of Fan Manufacturers*. The Code prescribes that the sound level shall be measured by the flat response network of the sound-level meter. Readings on the 40 db and 70 db networks may also be taken and reported, but the flat response reading is required to comply with the Code requirement. General practice is to use the *slow* or damped needle reading of the meter. The *fast* or undamped needle of the indicating meter generally reads one to two db lower than the *slow* or damped meter needle. The same Code prescribes a method of determining the sound level reading at each of seven stations, spaced at 5 ft from the outside of the fan housing, and located in a horizontal plane passing through the fan shaft. The sound level of the fan is the average of the seven readings. The level so determined is valuable primarily for comparative purposes rather than absolute values. Of more value to the design engineer would be the sound level at the fan outlet and at the beginning of the distribution duct system.

The technique of sound measurement in a moving air stream of ap-

preciable velocity has not yet been mastered, although it is a subject of current investigation. Development tests indicate that the sound level at the fan outlet is in the order of 10 to 15 db higher than the average value determined by the seven-station traverse around the fan. Where sound treatment of a distribution duct is required, the initial level should be taken as approximately 12 decibels higher than the reported Code rating sound level.

KINDS OF NOISE

In solving a sound problem, it is desirable to consider, separately, the several means by which noise reaches the room. This avoids to some extent the necessity of knowing the noise level at the source, and instead, places the emphasis on ascertaining the level at the point where the sound enters the room.

The noise introduced into a room or building by ventilating or air conditioning equipment may be divided into two general kinds, depending on how it reaches the room:

1. Noise transmitted through the ducts.
   a. From equipment such as fans, motors, pumps, sprays, etc.
   b. From outside, and transmitted through duct walls into air stream.
   c. From duct wall vibrations, transmitted into air stream.
   d. From air currents, including eddying noises.
   e. Cross talk and cross noises between rooms connected by the same duct system.
   f. Noise produced by the grilles.
2. Noise transmitted through the building construction.
   a. From machine mountings as vibration.
   b. From equipment through room wall surfaces.

The next step in the solution of this problem is to present data and discuss methods whereby solutions of the noise problem can be obtained when the allowable room noise level, and the path through which the noise reaches the room, are known.

NOISE TRANSMITTED BY AIR THROUGH DUCTS

Operation of an air distribution system results in the generation of noise which may be transmitted by air through the ducts to the ventilated or conditioned room. The transmission of this noise may be controlled by the proper application of sound absorptive material within the ducts. The application of the absorptive material is a problem in balancing the room noise level requirements against the intensity of the noise generated. The four steps in the problem are:

1. Determination of acceptable room noise level resulting from the operation of the equipment.
2. Determination of noise level generated by the equipment.

Add 5 decibels to the difference between items 1 and 2 to obtain the overall noise reduction required between the equipment and the room. In the discussion which follows, reduction of noise will be referred to as *attenuation* of noise.

3. Determination of the natural attenuation of the duct system.
4. Selection of the proper sound treatment for the duct system.

The difference in decibels between the overall attenuation required and the natural attenuation (3) is the additional sound attenuation to be provided by absorptive materials installed in the duct system, or by special constructions designed to absorb sound. Experience has shown, for example, that where ventilating requirements permit, introduction of an expansion chamber or a change in area in the duct will frequently provide further reduction in low frequency noise.

TABLE 2. TYPICAL SOUND LEVELS[a]
Weighted Network Response

Rooms	Sound Level in Decibels to be Anticipated		
	Min.	Representative	Max.
Sound Film Studios	10	14	20
Radio Broadcasting Studios	10	14	20
Planetarium	15	20	25
Residence, Apartments, etc	33	40	48
Theaters, Legitimate	25	30	35
Theaters, Motion Picture	30	35	40
Auditoriums, Concert Halls, etc	25	30	40
Churches	25	30	35
Executive Offices, Acoustically Treated Private Offices	30	38	45
Private Offices, Acoustically Untreated	35	43	50
General Offices	50	60	70
Hospitals	25	40	55
Class Rooms	30	35	45
Libraries, Museums, Art Galleries	30	40	45
Public Buildings, Post Offices, etc	45	55	60
Court Rooms	30	35	45
Small Stores	40	50	60
Upper Floors Department Stores	40	50	55
Stores, General, Including Main Floor Dept. Stores	50	60	70
Hotel Dining Rooms	40	50	60
Restaurants and Cafeterias	50	60	70
Banking Rooms	50	55	60
Factories	65	77	90
Office Machine Rooms	60	70	80
Vehicles			
Railroad Coach	60[b]	70	80
Pullman Car	55[b]	65	75
Automobile	50	65	80
Vehicular Tunnel	75	85	95
Airplane	75	80	90

[a] These values are tentative. More detailed measurements by D. F. Seacord, Bell Telephone Laboratories (*Journal Acoustical Society of America*, Vol. 12, pp. 183–187, 1940) give average values and standard deviations of room noise in residences, offices, stores, factories, etc., in large American cities.
[b] For train standing in station, a level of about 45 db is the maximum which can ordinarily be tolerated.

DESIGN ROOM NOISE LEVEL

Measurements of sound levels in various types of rooms and locations have been observed by numerous investigators. However, close agreement upon these values has not been realized, and more detailed measurements are needed to accurately establish the normal sound levels in occupied spaces and enclosures subject to sound analysis and control. Typical sound levels, of a tentative nature, based upon earlier determinations, are listed in Table 2. The levels listed are weighted levels by the 40 db or 70 db network, depending upon the range of level existing. Levels taken upon the flat response network may be from 5 db to 20 db higher, as governed by the predominating frequencies which may influence the weighting level.

Table 3 lists sound levels based upon more recent surveys than Table 2, and upon the basis of the flat response network. The flat response network offers a more logical correlation of space sound level to fan sound level which, under present practice, is reported upon basis of the *flat response* reading.

The values listed were determined with the air conditioning or ventila-

TABLE 3. AVERAGE SOUND CONDITIONS IN VARIOUS TYPES OF ROOMS AND BUILDINGS[a]
Flat Response Network

TYPE OF ROOM OR BUILDING	DECIBELS
Broadcasting studios	20–30 (very quiet)
Residences, churches, libraries, apartments, auditoriums, executive offices, class rooms	40–55 (quiet)
Hospitals, court rooms, quiet offices, show rooms, small retail stores, tea rooms, hotel dining rooms, foyers, upper floors of department stores, recreation rooms	45–60 (moderately quiet)
Banking rooms, beauty salons, barber shops, general offices, restaurants, main floors of department stores, cocktail lounges, dairy bars, tap rooms, billiard halls	55–70 (average)
Gymnasiums, transportation waiting rooms, drug stores, grocery stores, cafeterias, super markets, recreation halls, post offices, swimming pools, locker rooms, garages, service stations, dance halls, laundries, dry cleaners, bowling alleys	65–80 (moderately noisy)
Warehouses, office machinery, field houses, hangars, skating rinks, loading platforms, packing plants, factories, machine shops, foundries, forge shops, round houses, steel mills	75–100 (noisy)

[a] Values based on recent surveys.

tion equipment out of operation, unless such equipment presented no acoustical addition to normal conditions. The windows and doors were closed to simulate the conditions of normal occupancy. In Table 2, minimum, representative and maximum levels are given for each type of space, classified as shown in the following paragraph. In Table 3, the minimum to maximum range is shown, with the same general classification.

Minimum sound level refers to spaces within well-constructed buildings, typified by double windows, carpeted floors, and acoustically-treated walls and ceilings. In such spaces heavy upholstered furniture is also usually used.

Representative sound level refers to spaces within average construction with average furnishings, and exposed to external sounds typical of the locality in which the space is usually found.

Maximum sound level refers to (1) any space within inexpensive construction where bare furnishings are used, and where noise is normally not an important factor, or (2) spaces in close proximity to very intense street traffic or industrial noise.

In general, if the sound level in the space resulting from the operation of the air conditioning equipment only, is equivalent to, or less than, the typical level (from Tables 2 or 3 or, better still, determined by actual site measurement) the installation will prove satisfactory. If the space level and the equipment level are equal and heard together, the resultant level will be 3 db higher than either space or equipment level alone. However, to minimize possible annoyance due to introduction of single or distinctive frequency components from the equipment, it is desirable to design for an equipment sound level of at least 5 db below the typical space level.

NOISE GENERATED BY FANS

Noise generated by fan wheels may be divided into two classifications, rotational noise and vortex noise. The rotational noise may be described

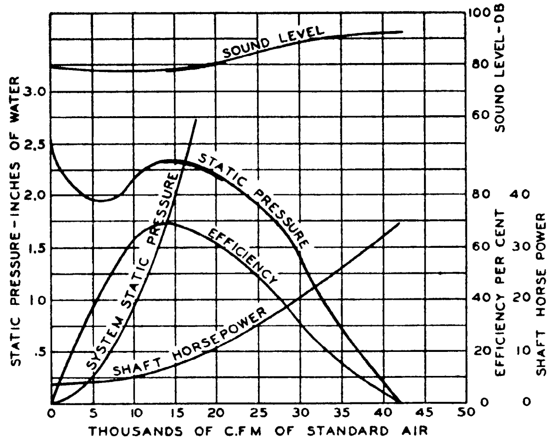

FIG. 1. SOUND LEVEL CHARACTERISTICS OF TYPICAL CENTRIFUGAL, MULTI-BLADE VENTILATING FAN

as that due to the thrust and torque applied to the air. Vortex noise is that due to the shedding of vortices from the blade, and is dependent on the angle of attack, velocity, air turbulence, and blade shape. Vortex noise is due to pressure variations on the blade as a result of variations of air circulation. Given the noise level at the outlet or inlet of one type of fan construction under specific conditions of size, tip speed, and total pressure, the noise levels at other values of tip speed, total pressure, and size may be approximated by the relationships:

1. For constant size and point of rating, the noise level of a fan will increase with increasing speed.

$$db \text{ (change)} = 50 \log_{10}\left(\frac{RPM_2}{RPM_1}\right) \tag{3}$$

2. For constant pressure and tip speed, the noise level of a given type of fan will increase with increasing fan size.

$$db \text{ (change)} - 20 \log_{10}\left(\frac{\text{Size}_2}{\text{Size}_1}\right) \tag{4}$$

Fan size refers to wheel diameter, housing height or some dimension that is directly proportional to linear units. Fan sizes based on arbitrary systems or systems of preferred numbers, have no significance.

The noise of a given fan is not constant at constant speed if the air delivery changes due to change of resistance. In general, a backward curved blade fan is lowest in noise at or near the point of maximum efficiency; a forward curved blade fan at or between the point of maximum efficiency and shut-off; an axial flow fan at or between the point of maximum efficiency and free delivery. The noise level of a double width fan may be taken as 3 db higher than for a similar single width fan operating under the same conditions of speed and pressure.

The sound level characteristic curve of a typical ventilating fan of the centrifugal multi-blade type is shown in Fig. 1. The accented portion of the curves denotes the range of minimum sound emission. The selection and application of the fan should be made within such good application range where quietness of operation is of major consideration. The various types of fans available possess individual sound level characteristics throughout their range of possible operation. Recourse to standard

Table 3. Average Sound Conditions in Various Types of Rooms and Buildings[a]

Flat Response Network

Type of Room or Building	Decibels
Broadcasting studios	20–30 (very quiet)
Residences, churches, libraries. apartments, auditoriums, executive offices, class rooms	40–55 (quiet)
Hospitals, court rooms, quiet offices, show rooms, small retail stores, tea rooms, hotel dining rooms, foyers, upper floors of department stores, recreation rooms	45–60 (moderately quiet)
Banking rooms, beauty salons, barber shops, general offices, restaurants, main floors of department stores, cocktail lounges, dairy bars, tap rooms, billiard halls	55–70 (average)
Gymnasiums, transportation waiting rooms, drug stores, grocery stores, cafeterias, super markets, recreation halls, post offices, swimming pools, locker rooms, garages, service stations, dance halls, laundries, dry cleaners, bowling alleys	65–80 (moderately noisy)
Warehouses, office machinery, field houses, hangars, skating rinks, loading platforms, packing plants, factories, machine shops, foundries, forge shops, round houses, steel mills	75–100 (noisy)

[a] Values based on recent surveys.

tion equipment out of operation, unless such equipment presented no acoustical addition to normal conditions. The windows and doors were closed to simulate the conditions of normal occupancy. In Table 2, minimum, representative and maximum levels are given for each type of space, classified as shown in the following paragraph. In Table 3, the minimum to maximum range is shown, with the same general classification.

Minimum sound level refers to spaces within well-constructed buildings, typified by double windows, carpeted floors, and acoustically-treated walls and ceilings. In such spaces heavy upholstered furniture is also usually used.

Representative sound level refers to spaces within average construction with average furnishings, and exposed to external sounds typical of the locality in which the space is usually found.

Maximum sound level refers to (1) any space within inexpensive construction where bare furnishings are used, and where noise is normally not an important factor, or (2) spaces in close proximity to very intense street traffic or industrial noise.

In general, if the sound level in the space resulting from the operation of the air conditioning equipment only, is equivalent to, or less than, the typical level (from Tables 2 or 3 or, better still, determined by actual site measurement) the installation will prove satisfactory. If the space level and the equipment level are equal and heard together, the resultant level will be 3 db higher than either space or equipment level alone. However, to minimize possible annoyance due to introduction of single or distinctive frequency components from the equipment, it is desirable to design for an equipment sound level of at least 5 db below the typical space level.

NOISE GENERATED BY FANS

Noise generated by fan wheels may be divided into two classifications, rotational noise and vortex noise. The rotational noise may be described

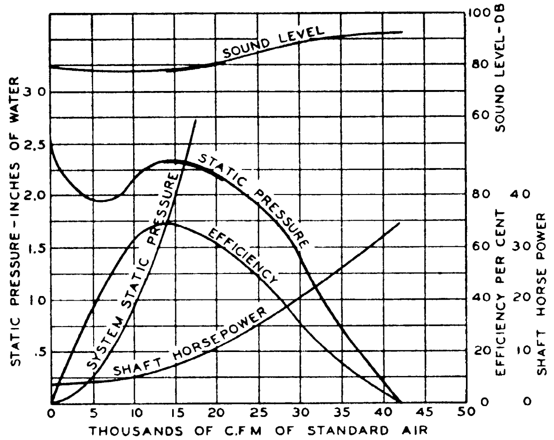

FIG. 1. SOUND LEVEL CHARACTERISTICS OF TYPICAL CENTRIFUGAL, MULTI-BLADE VENTILATING FAN

as that due to the thrust and torque applied to the air. Vortex noise is that due to the shedding of vortices from the blade, and is dependent on the angle of attack, velocity, air turbulence, and blade shape. Vortex noise is due to pressure variations on the blade as a result of variations of air circulation. Given the noise level at the outlet or inlet of one type of fan construction under specific conditions of size, tip speed, and total pressure, the noise levels at other values of tip speed, total pressure, and size may be approximated by the relationships:

1. For constant size and point of rating, the noise level of a fan will increase with increasing speed.

$$db\ (\text{change}) = 50 \log_{10}\left(\frac{RPM_2}{RPM_1}\right) \tag{3}$$

2. For constant pressure and tip speed, the noise level of a given type of fan will increase with increasing fan size.

$$db\ (\text{change}) = 20 \log_{10}\left(\frac{\text{Size}_2}{\text{Size}_1}\right) \tag{4}$$

Fan size refers to wheel diameter, housing height or some dimension that is directly proportional to linear units. Fan sizes based on arbitrary systems or systems of preferred numbers, have no significance.

The noise of a given fan is not constant at constant speed if the air delivery changes due to change of resistance. In general, a backward curved blade fan is lowest in noise at or near the point of maximum efficiency; a forward curved blade fan at or between the point of maximum efficiency and shut-off; an axial flow fan at or between the point of maximum efficiency and free delivery. The noise level of a double width fan may be taken as 3 db higher than for a similar single width fan operating under the same conditions of speed and pressure.

The sound level characteristic curve of a typical ventilating fan of the centrifugal multi-blade type is shown in Fig. 1. The accented portion of the curves denotes the range of minimum sound emission. The selection and application of the fan should be made within such good application range where quietness of operation is of major consideration. The various types of fans available possess individual sound level characteristics throughout their range of possible operation. Recourse to standard

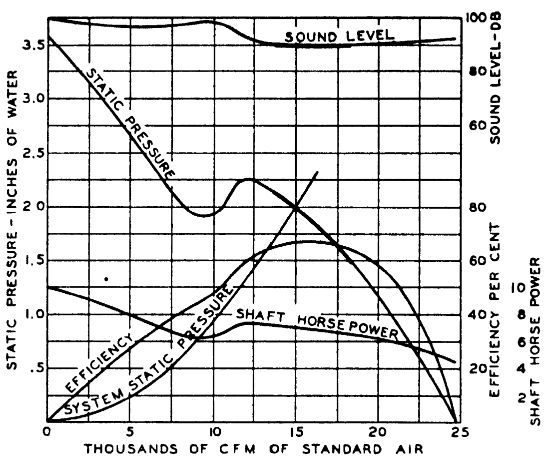

Fig. 2. Sound Level Characteristics of Typical Vaneaxial Fan

test rating information should be made to arrive at the sound emission of a particular type.

In general, that size of fan which is so selected as to operate at or near peak static efficiency, will also provide the lowest sound level attainable with the particular type and design of fan.

The characteristic trend of the sound level curve of a typical vaneaxial fan, adaptable to moderate pressure ventilation requirements, is shown in Fig. 2. The sound level of a particular fan is primarily governed by the operating speed required to produce a desired delivery against the system static pressure. Fig. 3 illustrates the variation of fan sound level in decibels with operating speed, the fan operating in connection with a conventional fixed system. The influence of high static pressures is evident in increased operating speed and higher sound level.

The range of sound levels to be experienced in fan application is widespread due to volumetric and pressure requirements which extend over a broad field. Fig. 4 illustrates the general scope of sound levels of centrifugal ventilating fans over a wide range of volumetric capacities and static

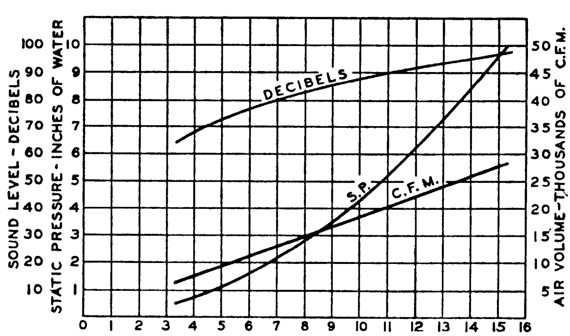

Fig. 3. Relation of Sound Level to Operating Speed of a Centrifugal Ventilating Fan

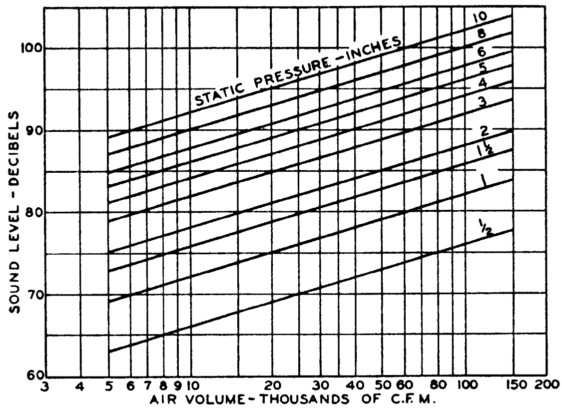

FIG. 4. TYPICAL SOUND LEVELS AT POINT OF MINIMUM SOUND EMISSION FOR CENTRIFUGAL VENTILATING FAN

pressures. The sound levels are based upon a general average of the sound emission that can be anticipated from the several types of fans adaptable to ventilating and air conditioning duties. The sound levels are typical of the centrifugal fans as a class, and specific types may exhibit sensible departure from the charted values. Exact application should be based upon applicable test data derived from the particular equipment under consideration.

NATURAL ATTENUATION OF DUCT SYSTEM

Straight Sheet Metal Ducts. The attenuation of sound in straight sheet metal ducts is a function of the length. shape, and size of the duct.[4] Attenuation values are given in Table 4. In general this attenuation is so negligible, except for long runs, that it may be disregarded for all practical purposes.

Elbows and Transformations. Due to reflective interference, attenuation will take place at elbows and transformations. The magnitude of the attenuation will depend on the size and abruptness of the elbow or transformation as shown in Table 5.

When the area of a duct increases abruptly, an attenuation of noise level takes place in the duct. In duct design practice the total area of the branch ducts is greater than the supply duct. Similarly with outlets, the area of the outlet, plus the area of the duct after the outlet is greater than the duct area before the outlet. Therefore in an outlet run, attenuation occurs in the duct as it passes each outlet. Table 6 gives the db reduction for various ratios of total branch duct and outlet area to supply duct area.

Grilles to Room. The large abrupt change in area between the grilles and the surfaces within a room results in an appreciable noise attenuation.

TABLE 4. ATTENUATION IN STRAIGHT SHEET METAL DUCT RUNS

Duct	Size, In.	Attenuation per Ft, db
Small	6 x 6	0.10
Medium	24 x 24	0.05
Large	72 x 72	0.01

TABLE 5. ATTENUATION OF ELBOWS[a]

Elbow	Size In.[b]	Attenuation per Elbow, db
Very small	2 wide	3
Small	3 to 15	2
Medium	15 to 36	1.5
Large	36 plus	1

[a] The attenuation in vaned elbows should be considered the same as in elbows having the same dimensions as the radius of curvature of the vanes. If the vanes are lined for the purpose of damping any vibrations in them, one third may be added to the attenuation values listed.

[b] These attenuation values are based on elbows having a center line radius 1.5 to 2 times the diameter or width of the duct. The attenuation will be greater if the ratio is less than 1.5 and less when the ratio is greater than 2.

This attenuation is a function of the total grille area (supply and return) and the total sound absorption of the room in *sabins*. (The sound absorption of a room in *sabins* is the summation of the products of each surface of the room measured in square feet multiplied by its corresponding absorption coefficient. The *sabin* is a unit of sound absorption equivalent to the absorption of one square foot of a totally sound-absorbent surface). The attenuation is given in Equation 5 as:

$$db\left(\begin{array}{c}\text{Attenuation between}\\ \text{grilles and room}\end{array}\right) = 10\log_{10}\frac{\text{Total Room Absorption in Sabins}}{\text{Total Grille Area}} \tag{5}$$

Values in Table 7 approximate the attenuation for various rates of air change, and general types of room surfaces.

DUCT SOUND ABSORBERS

The difference between the required sound attenuation and the natural attenuation must be supplied by the proper sound treatment of the ducts.

Selection of the Absorptive Material

When a sound wave impinges on the surface of a porous material, a vibrating motion is set up within the small pores of the material by the alternating sound waves. As the ratio of the cross-sectional area of the pores to their interior surface is small, the resistance to the movement of air in the pores is large. This viscous resistance within the pores of the material, converts a portion of the sound energy into heat. The decimal fraction representing the absorbed portion of the incident sound wave is called the absorption coefficient. Considerable absorption may also result, particularly in the low frequency range, from the flexural vibrations of the duct. In the selection and application of the absorptive material, the following points should be considered:

TABLE 6. ATTENUATION AT DUCT BRANCHES OR OUTLETS

Ratio $\frac{\text{Branch Duct + Outlet Area}}{\text{Supply Duct Area}}$ or $\frac{\text{Sum of Branch Areas}}{\text{Supply Duct Area}}$	Attenuation per Transformation, db
1.00	0.0
1.20	0.8
1.35	1.3
1.50	1.8
1.75	2.5
2.00	3.0

1. For the absorption of the low frequencies below 500 cycles per second the material should be at least 1 to 2 in. thick. Thin materials, particularly when mounted on hard solid surfaces, will absorb the high frequencies and reflect the low.

2. In order to provide as much low frequency noise absorption as possible by means of flexural vibration, it is desirable to fasten the absorptive panels discontinuously. This result may be attained to some extent by spot cementing, but better results are obtained when it is possible to fasten the absorptive panels to furring strips, leaving an air space behind. However, the exact resonance characteristics of the panels, and thus their absorption, are so unpredictable that flexural vibration cannot be relied upon for a specific value of attenuation.

Requirements for a good sound absorption material are: (1) high absorption at low frequencies;[5] (2) adequate strength to avoid breakage; (3) fire resistance and compliance with national and local code requirements; (4) low moisture absorption; (5) freedom from attack by bacteria

TABLE 7. APPROXIMATE ATTENUATION BETWEEN GRILLES AND ROOM

Outlet Velocity fpm	Air Change Min.	Live Room[b] $a^a = 0.05$ db	Medium Room[c] $a = 0.15$ db	Dead Room[d] $a = 0.25$ db
500	5	11	16	18
	10	14	19	21
	15	16	21	23
	20	17	22	24
750	5	13	18	20
	10	16	21	23
	15	18	23	25
	20	19	24	26
1000	5	14	19	21
	10	17	22	24
	15	19	24	26
	20	20	25	28
1250	5	15	20	22
	10	18	23	25
	15	20	25	27
	20	21	26	28

[a] Average absorption coefficient for the room.
[b] Live room-average absorption coefficient 0.05. Bare wood or concrete floor—hard plaster walls and ceiling—minimum of furniture.
[c] Medium room-average absorption coefficient 0.15. Carpeted floor, upholstered furniture, hard plaster walls and ceiling or bare room with acoustically treated ceiling.
[d] Dead room-average absorption coefficient 0.25. Heavy carpeted floor. Walls and ciling acoustically treated. Upholstered furniture.

and algae; (6) low surface coefficient of friction; (7) particles should not fray off at the higher design velocities; and (8) freedom from odor when either dry or wet.

With every application, the use of sound absorptive material should be considered in the dual function of insulation and sound absorption. It has been shown theoretically[6] that the reduction (in decibels per linear foot) of sound transmitted through a duct lined with sound absorbing material, is related in a rather complicated manner to the size and shape of the duct, to the frequency of the sound, and to the sound absorbing characteristics of the lining. Experimental evidence likewise indicates that there is no simple formula involving the variables which will apply accurately to all cases. However, it may be stated generally that the attenuation in decibels at a given frequency is directly proportional to the length of lined duct. It decreases as the cross-sectional area increases, and increases as the aspect ratio is increased.

The noise reduction varies to a considerable extent with the frequency of the sound. In calculating noise reduction, consideration should be given both to the comparative efficiency of the duct lining material at different frequencies, and to the frequency distribution of the noise to be quieted. In the case of fan noise, it is recommended that calculations be based upon the predominant frequency component in the fan sound level spectrum. Normally, most of the sound energy is in the region of this frequency, which generally corresponds to the blade frequency and is equal to rpm $\times$ no. of blades $\div$ 60.

Where the noise reduction is calculated upon the basis of the fundamental frequency component, the treatment indicated as required should be ample for the harmonics which are more easily absorbed than the fundamental. In quieting noise due to air turbulence and eddy currents where high frequencies predominate, the frequency 1024 should be used.

Since ventilating system noise contains many frequencies, an exception should be noted to the previous statement that attenuation in decibels is directly proportional to length of duct. Most sound absorbent materials are more efficient at high frequencies than at low frequencies. In consequence, the attenuation in the first five or ten feet of lined duct will be greater because the high frequencies are being absorbed. Thereafter, since low frequencies will be predominant, the overall noise attenuation per foot will gradually be less.

Duct Lining

By far the most commonly used method of obtaining sound absorption in ventilating systems is to line the duct with absorbing material. It is usually more convenient to line all four sides of the duct, but a lining on one side over a longer length of the duct will, in general, give the same effect for the same area of applied acoustical material. Subject to certain restrictions, the attenuation of a fully lined duct to single-frequency sounds may be expressed by the approximate Equation 6:[7]

$$R = 12.6L \frac{P}{A} a^{1.4} \tag{6}$$

where

R = attenuation, decibels.

L = length of lined duct, feet.

P = perimeter of duct, inches.

A = cross-sectional area of duct, square inches.

a = absorption coefficient of lining.

This formula was empirically developed for a set of duct sizes ranging from 9 x 9 in. to 18 x 18 in., for cross-sectional dimension ratios of 1:1 to 2:1, for frequencies between 256 and 2048 cycles, and for absorption coefficients between 0.20 and 0.80. The duct lining material used was 1 in. rock wool sheet. In Table 8 are listed the absorption coefficients of a material of this type in one-half and one inch thickness.

It is also possible to calculate the absorption by a very complicated mathematical theory.[8,9] Such calculations are in substantial agreement with Equation 6. This equation may be in error when applied to other

TABLE 8. ATTENUATION DATA FOR TYPICAL 1 IN. AND ½ IN. THICK DUCT LINING BOARD

FREQUENCY cycles per second	1-INCH THICKNESS			½-INCH THICKNESS		
	Absorption Coefficient a	$a^{1.4}$	Attenuation db	Absorption Coefficient a	$a^{1.4}$	Attenuation db
128	0.29	0.17	$2.1\ L\frac{P}{A}$	0.13	0.06	$0.8\ L\frac{P}{A}$
256	0.51	0.39	$4.9\ L\frac{P}{A}$	0.25	0.15	$1.9\ L\frac{P}{A}$
512	0.70	0.60	$7.6\ L\frac{P}{A}$	0.40	0.28	$3.5\ L\frac{P}{A}$
1024	0.80	0.73	$9.2\ L\frac{P}{A}$	0.72	0.63	$7.9\ L\frac{P}{A}$
2048	0.79	0.72	$9.1\ L\frac{P}{A}$	0.78	0.71	$8.9\ L\frac{P}{A}$

types of duct lining and to duct sizes and shapes greater than those specified. An empirically-derived chart[10] representing the average experimental data on a number of different types of materials, is shown in Fig. 5. Since individual materials vary, the curves of Fig. 5 are given only as representing the best available averages for duct sizes of cross-sections from 6 x 6 in. to 48 x 48 in. The dotted lines are plotted from Equation 6 and show that the slope is materially different from the average values.

Rectangular Cells (Plate or Cell Absorbers)

If the length of duct from the main duct to the grille is shorter than the length of lining indicated by Equation 6, the duct may be subdivided into smaller ducts as shown in Fig. 6, or it can also be even more subdivided by an egg-crate construction. In such a construction in which all the subdivided ducts are the same size, sound will be equally absorbed down each channel. It is, therefore, only necessary to calculate the sound attenuation of an individual channel. For this, Equation 6 is adequate.

When the number of splitter plates or cell partitions is large, the percentage free area of the gross duct size may be materially reduced. This leads to a further sound attenuation. Values of the attenuation possible, due to this cause, are given in Table 9.

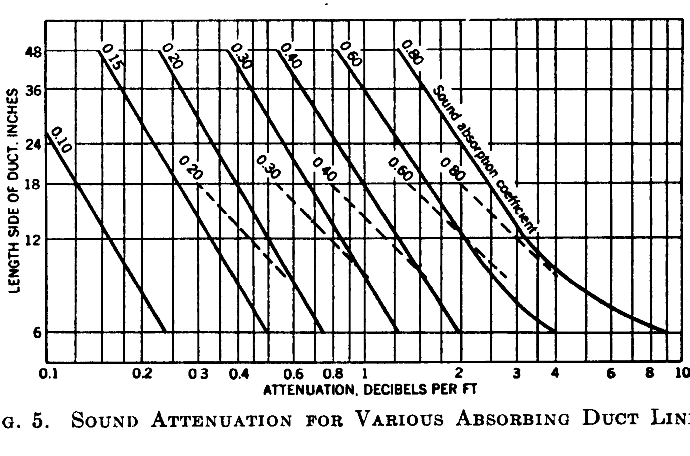

FIG. 5. SOUND ATTENUATION FOR VARIOUS ABSORBING DUCT LINERS

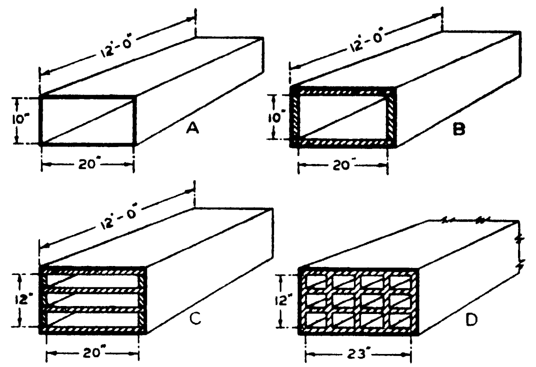

FIG. 6. ACOUSTIC TREATMENT OF DUCTS

A. Unlined metal duct.
B. Absorption lined duct (Case 1).
C. Splitter plate type absorber (Case 2). (Channels 20 in. x 3.33 in. inside).
D. Cell type absorber. (Cells 5 in. x 3.33 in. inside).
1 in. thick absorption material in all cases.

Sample Calculations for Duct Treatment

Example 1: An air conditioning installation is to be installed in a small theater. Determine the necessary sound treatment for the air distribution system to provide a satisfactory noise level in the theater utilizing these conditions:

Fan tip speed 4000 fpm, total pressure 1.25 in.	77 db
Acceptable room noise level (Table 2)	40 db
Required attenuation	37 db

Solution: Natural attenuation of supply duct.

Sheet metal duct 50 ft long 48 in. x 36 in. (Table 4) 50 x 0.01	0.5 db
Elbows, two size 48 in. x 36 in. (Table 5) 2 x 1	2.0 db
Attenuation grilles to theater air change 10 min (Table 7) outlet velocity 1000 fpm	22.0 db
Total natural attenuation	24.5 db

Difference between required and natural attenuation, 37 minus 24.5, is 12.5 db. This attenuation must be supplied by sound treatment in the duct, either in the form of duct wall lining or rectangular cells of the plate or cell absorber arrangement.

A similar analysis of the return duct system shows that 15 db attenuation is to be furnished by absorptive material. An inspection of the installation shows that the lining of the plenum on the suction side of the fan would prove the most economical, where it would secure the dual function of heat insulation and sound absorption.

Example 2: A 10 x 20 in. duct is connected to a private office space in a quiet location. Determine the length of lining necessary to attenuate average fan noise satisfactorily, using a lining material of a type to which Equation 6 applies, and having an absorption coefficient of 0.40 at 256 cycles. Assume that the duct is only 12 ft long as shown in Fig. 6, and that a 30 db reduction is required in this length.

Solution:

Case 1. (No splitters, duct lining only). From Equation 6,

$$R = 12.6 \times 12 \times \frac{60}{200} \times 0.40^{1.4} = 13 \text{ db.}$$

TABLE 9. END REFLECTION OF PLATE OR CELL ABSORBERS

Percentage free area of absorber	50	40	30	25	20
Attenuation db	1	2	4	5	6

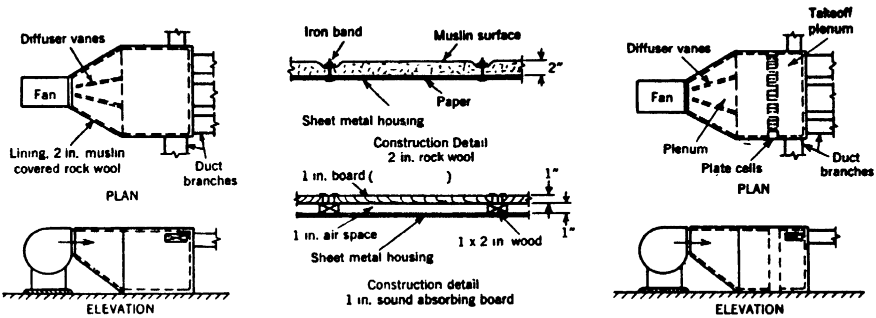

Fig. 7. Absorption Plenums With and Without Sound Cells

Case 2. (Two 1 in. splitter plates, 3 channels each 20 in. × $\frac{10}{3}$ in.). From Equation 6,

$$R = 12.6 \times 12 \times \frac{46.7}{66.7} \times 0.40^{1.4} = 29 \text{ db.}$$

Additional attenuation may be obtained by using additional splitter plates or use of egg-crate arrangement of absorbing material and application of Equation 6.

Plenum Absorption

In systems where individual ducts are directed to a number of rooms, and sound treatment is required in every duct, a sound absorption plenum on the fan discharge as shown in Fig. 7 will often prove the most economical arrangement. The absorption in the plenum may be approximated by Equation 7.

$$db \text{ (Attenuation)} = 10 \log_{10} \frac{\text{Plenum Absorption in Sabins}}{\text{Area Fan Discharge}} \qquad (7)$$

The area of the plenum should be at least ten times as great as the fan discharge area. The plenum should be lined with 2 in. of muslin covered rock wool blanket, or 1 in. sound absorbing board preferably nailed to wood strips on the inside of the plenum. With such a lining the plenum is particularly effective in reducing low frequency fan noise. The absorption of the plenum in sabins is the sum of the products of each interior area of the plenum measured in square feet multiplied by its corresponding absorption coefficient.

Outlet Sound Absorbers

Outlet sound absorbers are rectangular or plate cells installed directly behind an outlet or they may be the lining of a pan or plaque outlet. They are particularly effective in the elimination of high frequency whistles which are generated by air flow in the ducts. They are also employed in large systems with long runs where only a few outlets near the fan require treatment. Frequently outlet cells are the only means of correcting existing noisy installations, as the duct sections directly behind the outlets may be the only sections accessible for treatment. (See Fig. 8).

AIR SUPPLY OPENING NOISES

When air is introduced into a room through a grille or register at a constant velocity, sound energy is being introduced into the enclosure at

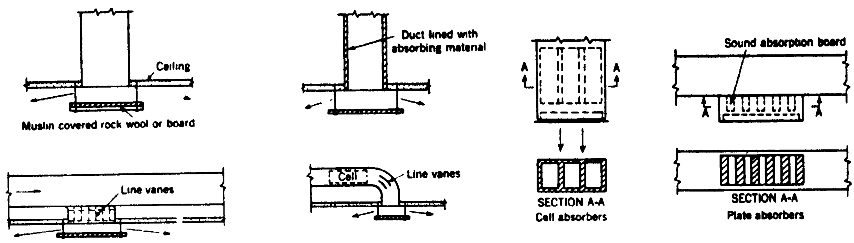

Fig. 8. Outlet Cells for Pan Outlets or Grilles

a constant rate.[11] Due to partial reflection at the boundaries of the enclosure, the intensity of sound at any point in the space builds up to some maximum value. In a large room at a point remote from the source of sound (the supply opening) the intensity can be shown to be substantially proportional to the rate at which sound energy is generated, and inversely proportional to the number of sound absorption units (sabins) in the room. It would thus appear that doubling the sound absorption of the room would halve the intensity and result in a noise level decrease of 3 db.

Grille noise is similar in character to fan vortex noise. Knowing the noise level at the face of a grille for a given grille blade setting, the noise will vary as given in Equation 8 where V is the velocity of the air through the grille

$$db \text{ (change)} = 50 \log_{10} \left(\frac{V_2}{V} \right) \tag{8}$$

For a change in blade setting Equation 9 applies, and in this case the total pressure is measured directly behind the face of the grille. For a typical air conditioning grille the noise level at the grille face may be approximately 48 db with a total pressure behind the grille of 0.1 in.

$$db \text{ (change)} = 25 \log_{10} \left[\frac{\text{(Total Pressure)}_2}{\text{(Total Pressure)}_1} \right] \tag{9}$$

The resultant room noise level can be approximated by Equation 10.

$$\text{Room Level} = \begin{bmatrix} \text{Noise Level at} \\ \text{Face of Grille} \end{bmatrix} - 10 \log_{10} \frac{\text{Total Room Absorption in Sabins}}{\text{Total Grille Area}} \tag{10}$$

Grille Selection

In practice the allowable total sound and the required air flow are usually known, and it is desired to determine the maximum allowable velocity. In comparing sound ratings of various grilles several factors must be known if the information is to be properly applied:

1. The threshold intensity on which the decibel ratings are based.
2. The distance from the grille at which data were taken.
3. If stated as sound level versus velocity for a given grille, the *core area* (not nominal area) must be known.
4. The sound absorbing characteristics of the test room.
5. Whether or not corrected for test room sound level; if not, the room level (without grille noise) must be known.
6. Methods used for recording data. (Characteristics of sound meter).

Since total sound and air flow are both functions of velocity and area, the solution of the problem implies a trial and error method. It has been

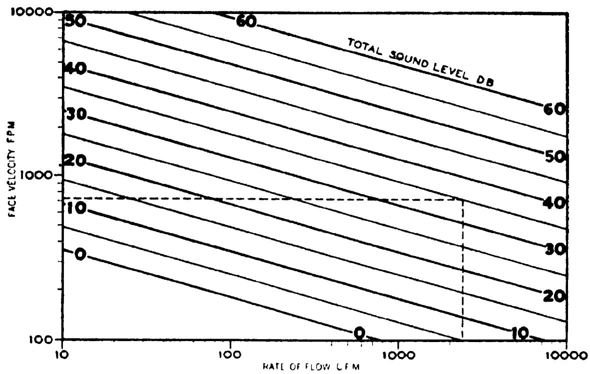

FIG. 9. AIR FLOW AND SOUND LEVEL CHART

found possible to present these data with sufficient practical accuracy as a family of uniform curves, as illustrated in Fig. 9, which are based on these assumptions:

1. Threshold intensity = 10^{-16} watts per square centimeter.[1]
2. Microphone location 5 ft from lower edge of supply opening on a line downward at 45 deg, and in a plane bisecting the supply opening perpendicularly.
3. Where data are given as sound level versus velocity, the rating is per square foot of core area
4. The room is assumed to have 100 sabins absorption.
5. Plotted data are sound levels of *supply openings only*, correction having been made for test room level.
6. Data taken with a direct reading sound-level meter with frequency weighing network intended to approximate the response of the human ear.

If the published ratings are in terms of decibels per square foot, correction must be made for area to secure the total sound level of supply openings of more or less than one square foot area from Equation 11.

$$\text{Decibel Addition} = 10 \log_{10} A \qquad (11)$$

where

A = core, square feet.

With Fig. 9 it is possible to find directly the velocity in feet per minute which will give a predetermined total sound at a predetermined rate of flow expressed in cubic feet per minute. The values used are arbitrarily chosen for the purpose of discussion, and do not necessarily represent data referring to any particular design of air supply opening. A correction chart is shown in Fig. 10 for a room having a sound absorption other than 100 sabins.

Example 3: Determine the core area (see Chapter 30) of an air supply grille which will maintain a noise level of not more than 40 db in a room having 100 sabins of sound absorption, if an air volume of 2400 cfm is required to maintain the proper air conditioning.

Solution: Assuming a grille noise rating of at least 5 db below the noise level of the room, Fig. 9 shows that the limiting grille velocity for a total sound level of 35 db is about 725 fpm, and the core area becomes fixed at 2400 ÷ 725 or 3.31 sq ft.

If the room absorption had been greater, the previously selected velocity of 725 fpm would be safe, since the sound level reduces. If the room absorption had been 200 sabins, a correction of plus 1.3 should be made by reference to Fig. 10, and the permissible velocity becomes that corresponding to a total sound level of 36.3, or approximately 800 fpm.

If the room had been highly reflective with an absorption of less than 100, the correction would be much more important. For instance, for a room of 35 sabins, a correction of minus 3 db should be made, and the maximum velocity corresponding to the 32 db total sound level would be approximately 600 fpm.

Where more than one supply opening must be considered, the problem is more complicated. If a similar supply opening is added in a far corner of a highly absorbent room, the change in noise level at the 5 ft station at the first supply opening is small; however, if the room is small, or highly reverberant or both, the intensity at the 5 ft station may be almost doubled and the noise level increased nearly 3 db thereby. The simplest method of handling this problem is to treat the room as though all the air were being supplied by one supply opening. Thus, if two outlets, each supplying 1000 cfm are used, the value 2000 cfm should be used with Fig. 9. Although this method may place an unwarranted limit on velocity when used in a large room, it is seldom that such a room has a noise level low enough to justify a more complicated, though more exact procedure.

In general, return grilles are selected for velocities about half the supply velocity, and when this is done, they may be neglected in sound computations. However, if supply and return grilles are the same size, resulting in the same face velocity, they must be treated as two supply openings. That is, if 1000 cfm are supplied and exhausted through grilles of the same area, 2000 cfm must be used in the solution with Fig. 9

CROSS TRANSMISSION BETWEEN ROOMS

Ducts serving more than one room permit cross talk between the rooms and should be lined with acoustical material. Where the rooms are close together and the ducts short, the ducts should be sub-divided to provide ample acoustical treatment. Lagging material similar in character to acoustical board, when placed on the outside of ducts, serves to prevent noise, originating outside the ducts, being carried inside the ducts and into the air stream.

A case, where outside lagging is desirable, occurs when ducts originate at the fan in the equipment room and pass through this room on the way to the room being conditioned or ventilated. Unless the ducts are lined, some of the mechanical noise from air in the equipment room may be transmitted through the wall of the duct into the air stream, and thereby carried into the room. In such cases, that portion of the duct which is exposed to the sounds in the equipment room should be lagged with material, such as cork, pipe covering or other sound damping material, to prevent the sound from entering the duct at this point. Numerical data are not available to permit a simple and practical calculating procedure to determine thickness of covering which should be used for this purpose.

Laboratory measurements have shown that the loss through a sheet of No. 22 gage metal is 24 db. When a sheet of rock wool insulation 1 in. thick and weighing 1.4 lb per square foot is added to this, the insulation value is increased to 29 db. In general, however, adding a layer of insulation or pipe covering does not materially increase the sound insulation value unless the material is dense, or unless it is surfaced with another sound impervious layer such as metal or board. Standard reference books should be consulted for sound insulating properties of various materials. Inside lining material, used in the case previously mentioned, would serve as an absorber of the sound transmitted through the duct walls, and thus act as a

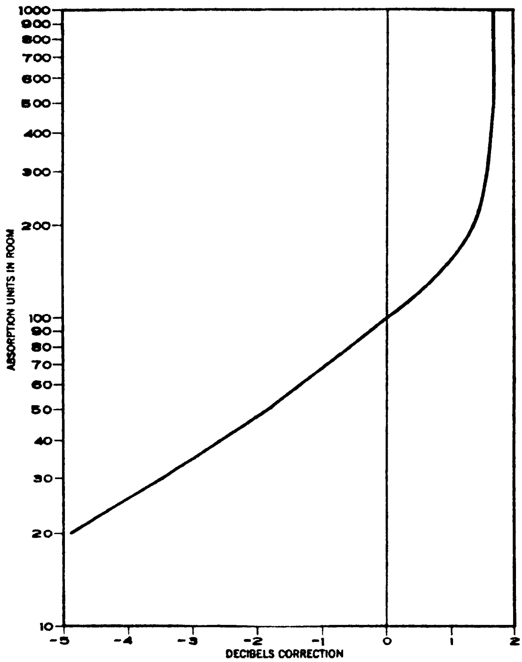

FIG. 10. ROOM ABSORPTION CORRECTION CHART

means of preventing the transfer of noise into the air stream. Inside lining may also be used in ducts to absorb noise which reaches the air stream from equipment such as fans, sprays and coils; noise due to eddying currents set up by elbows, dampers and similar obstructions; and noise transmitted from room to room where there is a common duct system.

CONTROLLING VIBRATION FROM MACHINE MOUNTINGS

It is impossible to select equipment which will operate without producing some mechanical noise and, since the equipment must be mounted in a building, it is probable that a part of this noise will be transmitted to the building to such a degree as to make noisy conditions in the rooms which are to be air conditioned.

Much of this noise may be transmitted by the duct if it is rigidly connected to the fan outlet. It is common practice to make the connection between the fan and the duct with a canvas sleeve which effectively restricts noise at this point. Noise may also enter the building through the mounting of the motor and the fan. Flexible mountings should be provided in all installations, but these mountings must be carefully designed so that they will actually reduce the energy transmitted between the machinery and the supporting floor. If a flexible material is used, it is desirable to investigate the installation so that it is not short-circuited by through bolts which are improperly insulated, and by electrical conduit which is not properly broken and is attached both to the equipment and to the building. The flexible mounting, if improperly engineered, may actually increase the energy transmitted between the equipment and the supporting floor.

In the proper isolation of vibration, which is usually in the lower range of frequencies and does not include the *airborne* vibrations known as sound, there is one basic formula which is important in the solution of the problem. It is the formula of transmissibility as governed by the equation:

$$T = \frac{1}{\left(\frac{f}{f_n}\right)^2 - 1} \tag{12}$$

where T = transmissibility of the support.
f = frequency of the vibratory force.
f_n = natural frequency of the machine unit on its support (damping = 0).

Equation 12 shows that the transmissibility approaches unity for disturbing frequencies considerably lower than the natural frequency of the mounting. As the disturbing frequency is increased, the transmissibility is also increased until at the resonant frequency, where $f = f_n$ the transmissibility becomes infinite. This is not true in practice because all materials have some internal damping effect. However, operating at or very close to the resonant frequency is always serious as forces and stresses may be multiplied 10 to 100 times. As the disturbing frequency becomes greater than the natural frequency, the transmissibility becomes a smaller quantity, and at the value of $f/f_n = \sqrt{2}$ it again has the value of unity. Beyond this point true isolation is first accomplished. At a ratio of 3 to 1 for f to f_n the isolation is effective enough for practical application, and experience and economical design have shown that a ratio of 5 to 1 is good. For high speeds, higher ratios for f to f_n are easily attained and give better results for effective vibration control, but for the lower speeds as experienced with compressor work the higher ratios become uneconomical.

For a given installation, the speed of the compressor is fixed by the specifications; therefore the value of f is fixed. That leaves only f_n to be determined, and that is accomplished by the choice of mounting material and design for the support of the machine. It is well to keep in mind that when trying to isolate vibration, no attempt should be made to isolate the driving and driven piece of equipment separately. The two should be mounted on a rigid frame, and then the entire assembly isolated according to the rules presented in this chapter.

The value of f_n can be controlled by the flexibility of the machine support, and when the deflection of the machine support is proportional to the load applied (such as with springs or nearly so with rubber in shear) the value of f_n can be determined by Equation 13:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{d}} \tag{13}$$

where g = gravitational constant.
d = static deflection of supporting material.
f = frequency of the vibratory force.
f_n = natural frequency of the machine unit on its support (damping = 0).

By the use of Equation 13 a set of curves may be plotted as shown in Fig. 11. The first line AB, plotted as the critical frequencies for the various static deflections, is a curve showing the worst possible conditions or resonant conditions.

Plotting another curve CD, which is $\sqrt{2}$ times curve AB, shows the area $MCDN$ in which the resilient material or mounting does more harm

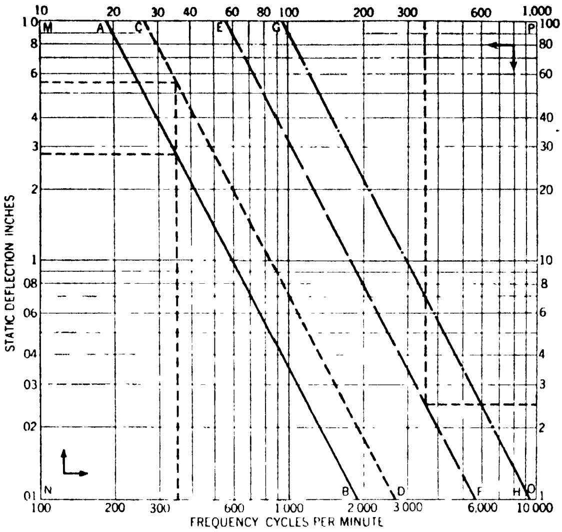

Fig. 11. Static Deflection for Various Frequencies

than good. Plotting curves *EF* (3 times curve *AB*) and *GH* (5 times curve *AB*) shows area *EGHF* which represents efficient and economical isolation. Area *GPOH* is excellent isolation, but for all except the highest speeds, becomes rather uneconomical because of the large deflections required.

Example 4: An electric motor driven compressor unit is to be isolated. The compressor is partially balanced and operates at a speed of 360 rpm. The speed of the motor is 1160 rpm, and is belt connected to the compressor. Total weight of the compressor and motor is 4500 lb.

Solution: The minimum disturbing frequency to be isolated is 360 cycles per minute. Assume that the desired ratio of forced to natural frequency is 3 as a minimum, and that 5 is desired. The desired natural frequency of the mounting is 360 ÷ 5 = 72 cycles per minute.

From Fig. 11 a deflection of 7 in. is required to attain a natural frequency of 72 cycles per minute. This value may be obtained from critical curve *AB* for 72 cycles, or from curve *GH* (5 times critical) for 360 cycles. For the minimum ratio of 3 the deflection would be 2.5 in.

The next step is to determine the total weight to be supported by the springs. For low speed partially balanced compressors, it has been found necessary to add a foundation weighing 2 to 3 times the weight of the motor and compressor, in order to maintain the machine movement below 0.03 in.

Compressor and motor	4,500 lb
Concrete foundation	9,000 lb
Total	13,500 lb

Practical application dictates the number of springs to be used, which is based on the design of the machine foundation and the supporting floor structure. However, it is desirable to design for at least 8 springs and one or two spares for cases of unknown weights. As many as 50 springs have been used on one installation. The distribution of the springs must be balanced against the masses to be supported, otherwise the foundation design and supporting structure determine the location of the springs.

The choice of the material used in the design of the resilient mounting is also important. For the slow-speed type compressor, a common speed found in practice is 360 rpm. For speeds below this, isolation should not be attempted except under careful supervision. Referring to Fig. 11, it is found that for 360 rpm the static deflection required for a ratio of f/f_n of 3 to 1 (line *EF*) is 2.5 in., and for a ratio of 5 to 1 (line *GH*) it is 7 in. For these values of deflection the only choice of material is the coil spring. This is also true for speeds up to about 700 rpm. In consideration of the transverse spring constant (so as to maintain good ratios among the various degrees of freedom) experience has shown that the spring should be designed with a *working height* equal to 1.0 to 1.5 times the outside diameter. A long spring of small outside diameter has very low transverse rigidity, and therefore requires some additional means of preventing side drift of the unit, and on very sensitive applications this may tend to destroy the isolation efficiency. For speeds of 700 to 1200 rpm the required deflections range from 0.22 in. to 1.75 in. For these conditions rubber in shear serves as a rather satisfactory material if protected from oil. For speeds higher than 1200 rpm cork specially made for vibration damping can be applied with good results. These limitations are by no means absolute, because certain liberties may be taken without impairing the result if all possible degrees of freedom have been taken into account in the design of the installation.

When a machine unit is properly isolated it will have a definite amount of movement which is determined by the ratio of the unbalanced forces to the total mass of the machine. If this resultant machine movement is too great for the necessary connections or the satisfaction of the customer, it can be reduced only in two ways without destroying the quality of the isolation; first, adding mass or dead weight to the machine (such as concrete) common in the application of low speed, partially balanced machinery; second, accurately balancing (both statically and dynamically) all moving parts so as to eliminate the vibration at the source. This latter method is the best engineering practice and is the modern trend. However, even with well balanced machinery, installed in the vicinity of quiet offices, it is usually necessary to isolate properly the equipment to prevent the transmission of vibration likely to cause complaints.

Where limitation of machine movement is desired during the starting and stopping periods, the application of friction or hydraulic damping will serve without seriously interfering with the efficiency of the isolation.

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CHAPTER 41

ELECTRIC HEATING

Resistors, Heating Elements, Electric Heaters, Unit Heaters, Central Heating and Air Conditioning, Electric Boilers, Electric Hot Water Heating, Heating Domestic Water Supply, Calculating Capacities, Induction and Dielectric Heating, Power Problems

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ELECTRIC heating deals with the conversion of electrical energy into heat and the distribution and practical use of the heat so produced. In certain regions, where the cost of electricity is favorable, electric heating is used extensively. Its use is also frequently dictated by special conditions.

Definitions of the terms *Electric Resistor*, *Electric Heating Element*, *Electric Heater* and other terms applying to heating practice, will be found in Chapter 1.

RESISTORS AND HEATING ELEMENTS

Commercial electric heating elements usually have solid resistors such as metal alloys or non-metallic compounds containing carbon. In some types of electric boilers, water forms the resistor which is heated by passing an alternating electrical current through it.

In one type of heating element, the resistors are exposed coils of nickel-chromium wire or ribbon, or non-metallic rods, mounted on insulators. This type is used extensively for operation at high temperatures for radiant heat, or at low temperatures for convection and fan circulation heating.

Some elements have metallic resistors embedded in a refractory insulating material, encased in a protective sheath of metal. Fins or extended surfaces add heat-dissipating area. Elements are made in many forms, such as strips, rings, plates and tubes. Strip elements are used for clamping to surfaces requiring heat by conduction, in some types of convection air heaters, and in low temperature radiant heaters. Ring and plate elements are used in electric ranges, waffle irons, and many small air heaters. Tubular elements may be immersed in liquids, cast into metal, and, when formed into coils, used in electric ranges and air heaters.

Cloth fabrics, woven from flexible resistor wires and asbestos thread, are used for many low temperature purposes such as heating pads, aviators' clothing and radiant panel heating installations.

Special incandescent lamps are used as heating elements in certain applications where radiant heat is desired. These use carbon or tungsten filaments as resistors, and are designed to produce maximum energy in the infra-red portion of the spectrum.

ELECTRIC HEATERS

Conduction electric heaters, which deliver most of their heat by actual contact with the object to be heated, are used in such applications as aviators' clothing, hot pads, soil heaters, and water heaters. Conduction heaters are useful in conserving and localizing heat delivery at definite points. They are not suitable for general air heating.

Radiant electric heaters, which deliver most of their heat by radiation, have heating elements with reflectors to concentrate the heat rays in the desired directions. They are not satisfactory for general air heating, as radiant heat rays do not warm the air through which they pass. They must first be absorbed by walls, furniture, or other solid objects which then give up the heat to the air. For a discussion of electrically heated panels as applied to radiant heating, see Chapter 23.

Gravity convection electric heaters, designed to induce thermal air circulation, deliver heat largely by convection, and should be located and used in much the same manner as steam and hot water radiators or convectors. They generally have heating elements of large area, with moderate surface temperature, and are enclosed to give a stack effect to draw cold air from the floor line. The flexibility possible with electric heating elements should discourage the use of secondary mediums for heat transfer. Water

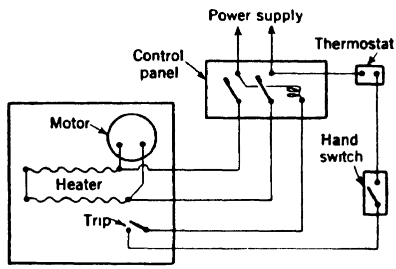

FIG. 1. WIRING DIAGRAM FOR UNIT HEATER

and steam add nothing to the efficiency of an electric heater and entail expensive construction and maintenance.

Induction and dielectric heaters are described in a later section.

UNIT HEATERS

Electric unit heaters include a built-in fan unit which circulates room air over heating elements. They are adapted to the same uses as other types of unit heaters, if conditions are favorable to electric heating. They are very adaptable for heating of small offices, locker rooms, etc., in otherwise unheated buildings. In small unattended equipment rooms, thermostatically controlled electric unit heaters are frequently used to maintain a temperature above freezing.

The best location for electric unit heaters depends upon local conditions. Various designs and arrangements are available, as with steam unit heaters (see Chapter 24).

The arrangement of the wiring circuits is very important. In principle, they are all the same and include as essential elements, a magnetic control contactor, a thermostat, and a master hand switch. All heaters should be designed with a safety thermal trip wired in series with the magnetic contactor, and with the hand switch and thermostat. A typical wiring diagram for single phase power supply is shown in Fig. 1. A main disconnect switch should be provided. For three-phase power supply, a 3-pole contactor should be used with the heater arranged for 3-phase connection. On large sizes, separate over-load protection for the motor should be provided.

If the line voltage is more than 120 to ground, it is advisable to supply the thermostatic control circuit through a transformer.

CENTRAL HEATING AND AIR CONDITIONING

Electric heating elements can be used for the prime source of heat in a central fan heating system or in the heating phase of an air conditioning system. They can be used in the same manner as steam heating units for tempering, preheating or reheating the air at the main supply fan location, and as booster heaters at the delivery terminals of the duct system. In the humidification phase of air conditioning, electric heating elements can be used to provide moisture by the evaporation of water.

In coordinating the input of heat energy and the volume of air circulation, a basic difference between electric heating and steam heating enters into the problem. Steam is approximately a constant-temperature

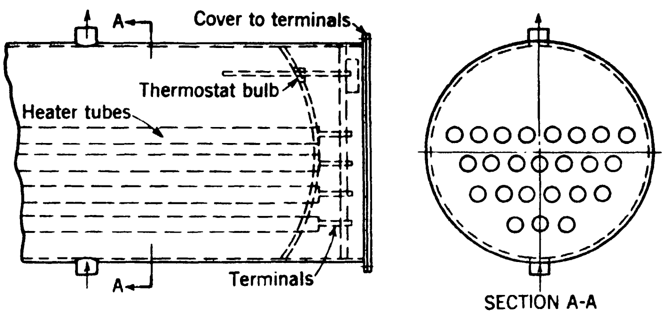

FIG. 2. RESISTANCE TYPE BOILER FOR STEAM OR HOT WATER

source of heat for any given pressure, and a change in air volume flowing over steam coils does not greatly affect the temperatures of the coil surface. The amount of steam condensed (heat input) varies in proportion to the air volume, but the surface temperature of the steam coils remains about the same. Electric heat is quite different, having a constant input of energy. If the volume of air flow over electric heating elements is changed, and no change is made in the electrical power input, there will be a corresponding change in the temperature of the air delivered. This occurs because the electrical energy input remains constant, and the surface temperature of the heating elements will vary as is necessary to force the air to accept all the heat. With electric heat the total heat is constant unless some compensating action is performed by controls. Automatic variation of the electrical heat input synchronized properly with the air flow, can be successfully accomplished by various special methods of control. By-pass dampers as used with steam units will not control electric heat.

Electric heaters are useful in balancing the heat distribution in central fan systems. Even in those instances where steam is the principal heat source, the temperature of individual rooms can be controlled locally by separate electric booster heaters. These heaters can be installed in branch ducts or behind the air outlet grilles in each room. With this arrangement, the central heating unit distributes air at an average temperature, controlled from a thermostat centrally located, such as in the main return duct. The electric booster heaters may be controlled by thermostats mounted in each individual room to permit the occupant to maintain any desired temperature independent of the rest of the building.

ELECTRIC BOILERS

Steam or hot water generating boilers using electrical energy are entirely automatic, and are well adapted to intermittent operation. Small electric boilers usually have heating elements of the enclosed metal resistor type immersed in the water. Boilers of this construction may be used either with direct or alternating current since the heat is delivered to the water by contact with the hot surfaces. To lessen the likelihood of burning out of heating elements, they should be of substantial construction, with a low heat density per unit of surface area, and provision should be made for

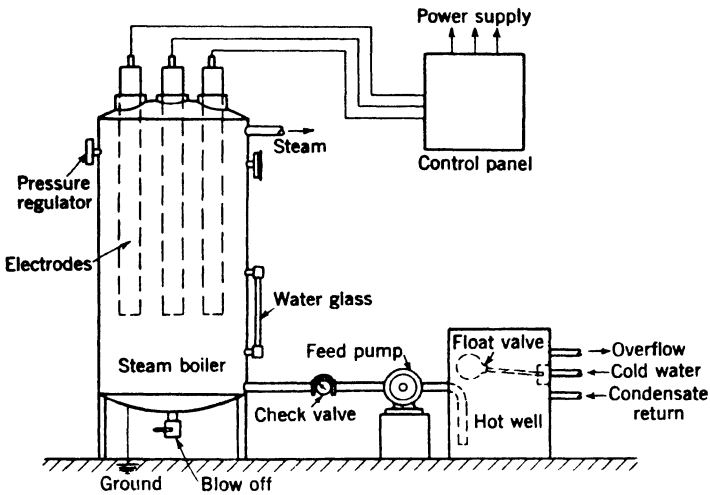

FIG. 3. DIAGRAMMATIC ARRANGEMENT OF AN ELECTRODE BOILER

cleaning off deposits of scale which restrict the heat flow. A typical resistance type of steam or hot water boiler is shown in Fig. 2.

Large electric boilers are usually of the type employing water as the resistor, using immersed electrodes. With this type only alternating current can be used, as direct current would cause electrolytic deterioration. Such a type of electrode boiler is shown in Fig. 3.

Electric steam boilers are useful in industrial plants which require limited amounts of steam for local processes, and for sterilizers, jacketed vessels and pressing machines which need a ready supply of steam. It sometimes is economical to shut down the main plant fuel-burning boilers when the heating season ends, and to supply steam for summer needs with small electric steam boilers located close to the operation.

ELECTRIC HOT WATER HEATING

Electric water heating, using an electric boiler in place of a fuel-burning boiler, like electric steam heating, is generally confined to auxiliary or other limited applications. The use of insulated water-storage tanks, in which to store heat generated by electricity during off-peak hours at extremely low rates, is a development which has some special applications.

In this system of heating, the primary storage tank is simply a large, well-insulated, pressure type steel tank, equipped with electric heating elements, automatic time switches, and automatic limit controls for temperature and pressure. The heating system installed in the building may be of any standard design. A system of this kind requires very careful design to avoid excessive overall radiation losses during periods of low heat demand. It is also important to provide for sudden changes in heat demand. A typical water heating boiler is illustrated in Fig. 2.

HEATING DOMESTIC WATER BY ELECTRICITY

Electric water heaters of the automatic storage type for domestic hot water supply are simple and reliable. In many sections of the country low electric rates have been established by the electric utilities to secure this load. In many localities, electric rate schedules divide the current

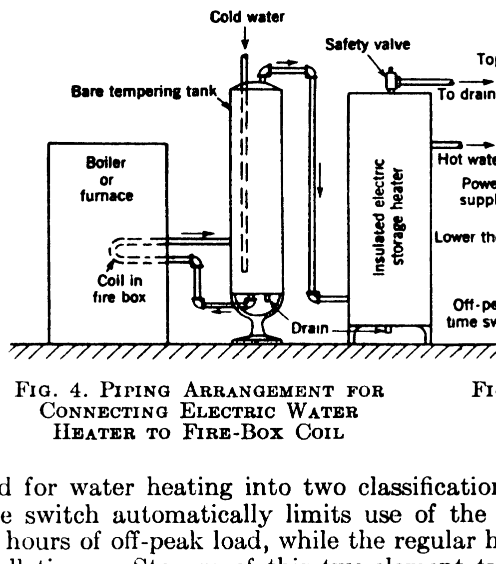

FIG. 4. PIPING ARRANGEMENT FOR CONNECTING ELECTRIC WATER HEATER TO FIRE-BOX COIL

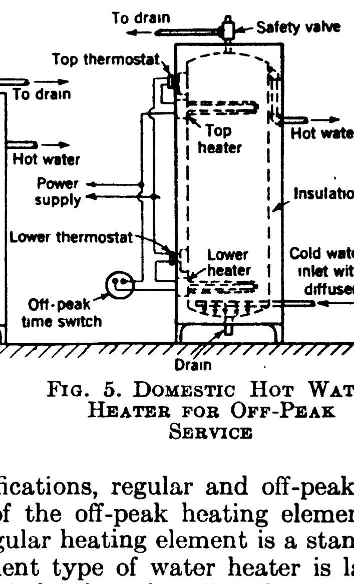

FIG. 5. DOMESTIC HOT WATER HEATER FOR OFF-PEAK SERVICE

used for water heating into two classifications, regular and off-peak. A time switch automatically limits use of the off-peak heating element to the hours of off-peak load, while the regular heating element is a stand-by at all times. Storage of this two-element type of water heater is larger than average to help carry over the periods when the off-peak element is timed out. Some utilities now offer a schedule which, beyond a stipulated minimum, lowers the rate for all electric service if an electric water heater is installed.

Competition with other fuels, especially gas, seems to be the major controlling factor in the use of electricity. The first cost of electric storage heaters is greater than for gas, owing to the need for larger tank storage due to off-peak service and slower recuperating capacity.

In residential work, to effect a saving in the cost of operation, it is sometimes desirable to use a furnace coil or indirect heater in connection with an electric water heater. In this case it is important to make the proper connections in order to benefit by any heat obtained from the furnace, and at the same time to prevent dangerous overheating. The proper piping connections are shown in Fig. 4, and in this case the electric heater will only furnish heat when insufficient heat is supplied from the furnace. This arrangement has a further advantage in the summertime in that the bare tank through which the cold water passes on its way to the electric heater serves as a tempering tank, absorbing heat from the basement air and requiring the use of less energy in the electric heater.

A typical domestic hot water heater as shown in Fig. 5 is arranged with upper and lower heating elements for the usual type of off-peak heating service. The lower heating element is under the control of the off-peak time switch. However, the upper heating element is usually connected to the line so that, in case the supply of hot water in the tank becomes exhausted, the top thermostat can turn on the top heater and heat a small supply of water. The top heater will not heat the water in the tank below its location, but when the off-peak period arrives the lower heater is turned on and the entire tank becomes heated.

CALCULATING CAPACITIES

In calculating electric heating capacity, one kilowatt is equal to 3413 Btu per hour or 14.2 sq ft equivalent direct steam radiation.

All of the energy applied to an electric resistor is transformed into heat. The output of an electric heater is a fixed constant, unaffected by the temperature of the surrounding air, and the total load on an electric heating system is the total wattage of the connected electric heaters.

ELECTRIC HEATING BY INDUCTION AND DIELECTRIC MEANS

These methods differ radically from resistance heating. They have many important industrial uses, and open up a whole new field of special application where extreme speed or control of heat location are vital.

Metals and other electrical conductors can be heated by induction. The work is placed in an alternating magnetic field within, or adjacent to, a coil, and heat is produced in the body of the piece by eddy currents. While induction heating has certain limitations, it has great advantages in certain applications such as melting metals, forging, brazing, heat treating and particularly for localized heating and zonal hardening of metals. It is possible to apply localized heat so rapidly that conduction cannot draw the heat away before it has time to accomplish the desired purpose at a particular spot. Surfaces and local areas can be hardened without distortion or scale formation.

Commercial 60-cycle alternating current may be used in special cases, such as induction heating of large pressure vessels hut special higher frequency generating equipment is generally required. It should be carefully selected for the particular kind of work to be done. Motor-generators with frequencies in the vicinity of 250 cycles per second, are used for many melting furnaces. Motor-generators having frequencies between 2,000 and 10,000 cycles per second, are generally used for heat treating and hardening sizeable parts. For heating or brazing thin sections or small parts, electronic tube oscillators, spark discharge oscillators, or mercury arcs are used to produce frequencies ranging up to 500,000 cycles per second. Work coils used with high frequency induction heating are generally copper tubes through which cooling water is circulated. These must be specially designed for each application.

Non-conductors of electricity can be heated internally by dielectric means by placing the materials in a high frequency electrostatic field between electrode plates. This process is distinctly different from the induction heating process. High voltages and very high frequencies, often up to 50 million cycles, are needed to produce the desired rate of heating. The main field for dielectric heating is with materials which are poor thermal conductors. Food can be sterilized, plywoods bonded, plastics heated, granular or crystalline material dehydrated, deep-pile fabrics dried, and countless other products heated quickly and uniformly. Dielectric heating is well suited to many continuous production processes, as the materials can pass through the heating field quickly and without the necessity of contact with the electrode surfaces.

POWER PROBLEMS

The cost of electric energy varies because of several factors. Distribution costs differ for large and small users. The fact that electricity cannot be economically stored, but must be used as fast as generated, makes it impossible to operate electric plants at uniform loads; hence, even the time of use may affect the cost of electricity. Special low rates are sometimes available during certain prescribed hours of use.

Since cost of production and distribution depends not only upon the quantity of energy used but also upon the maximum rate of use, electric energy is often sold on a demand rate basis. In some cases, the demand charge is based upon the rated connected load; in other cases, upon the maximum demand indicated by a demand meter.

Homes are almost universally supplied with lighting current of 115 volts, which can only be used economically for small heaters. Usually the service lines will not permit more than plug-in devices. The *National Board of Fire Underwriters* permits approved heaters of 1320 watts or less to be plugged into approved baseboard receptacles, but such heaters cannot be served on a circuit supplying much other load without overloading the circuits. There is an increasing trend toward supplying homes with three wire 115–230 volt service. Where homes have such service, larger heaters can be installed. For industrial purposes, heaters should be designed to use polyphase power, which is usually supplied at 208, 220, 440 or 550 volts. All polyphase heaters should be balanced between phases. In ordering electric heaters, proper voltage must be specified, as the heat produced will vary as the square of any variation in voltage.

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CHAPTER 42

CORROSION AND WATER FORMED DEPOSITS, CAUSES AND PREVENTION

Definitions, Classification and Characteristics of Water, Causes and Prevention of Scales and Sludges, Causes and Prevention of Slimes, Under-Water Corrosion, Atmospheric Corrosion, Buried Pipe Lines, Handling Water Treating Chemicals, Legal Regulations

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THE surfaces of heating and ventilating equipment that are in intimate contact with water sometimes are affected by the chemical characteristics of contacting waters to such an extent that prohibitive amounts of insoluble materials are formed or corrosion ensues at an insufferable rate. To avoid or to correct such troubles, it is desirable that heating and ventilating engineers have a general appreciation of industrial water chemistry. The principal purpose of this chapter is to provide those criteria by which the average engineer may judge whether a problem is one that will yield to rather simple remedies, or will require the skill of an experienced water technologist.

DEFINITIONS

The following definitions for water-formed deposits, corrosion, and closely allied terms have been proposed:

Water-Formed Deposits. A water-formed deposit[1] is any accumulation of insoluble material derived from water or formed by the reaction of water upon surfaces in contact with water.

Deposits formed from or by water in all of its phases may be further classified as scale, sludge, corrosion products, or biological deposits.

Scale. Scale[1] is a deposit formed from solution directly in place upon a confining surface. It is a deposit which will retain its physical shape when mechanical means are used to remove it from the surface on which it is deposited. Scale, which may or may not adhere to the underlying surface, is usually crystalline and dense, frequently laminated, and occasionally columnar in structure.

Sludge. Sludge[1] is a water-formed sedimentary deposit. It usually does not cohere sufficiently to retain its physical shape when mechanical means are used to remove it from the surface upon which it deposits. Sludge is not always found at the place where it is formed. It may be hard and adherent, and baked to the surface on which it has been deposited.

Biological Deposits. Biological deposits[1] are water-formed deposits of biological organisms or the products of their life processes. Biological deposits may be microscopic in nature, such as slimes, or macroscopic, such as barnacles or mussels. Slimes are usually composed of deposits of a gelatinous or filamentous nature.

Corrosion. Corrosion[2] is destruction of a metal by chemical or electrochemical reaction with its environment. In the corrosion process, the reaction products formed may be soluble or insoluble in the contacting environment. Insoluble corrosion products may deposit at or near the attacked area, or be carried along and deposited at a considerable distance from the attacked area.

Corrosivity. Corrosivity[3] is the capacity of an environment to bring about destruction of a metal by the process of corrosion. Corrosivity is a property of the environment, but it has no significance until the metal in question is specified.

CLASSIFICATION AND CHARACTERISTICS OF WATER

For industrial use there is no accepted conventional classification of water. Rather, each industry usually develops a body of ideas applicable to its own water problems.[4] For heating and ventilating engineers, it is perhaps most convenient to distinguish between mineralized waters and condensates.

Mineralized Waters

All the waters found in streams, wells, lakes, and the ocean are mineralized. The same is true of all municipal supplies even though they may have been *treated*. For a given area, ground waters are likely to be more highly mineralized than are surface waters. Conversely, surface waters

TABLE 1. MINERAL ANALYSES TYPIFYING COMPOSITION OF WATERS AVAILABLE AND USED INDUSTRIALLY IN THE USA

Substance	Unit	Location or Area[a,b]								
		(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
Silica	SiO_2	2	6	12	37	10	9	22	14	..
Iron	Fe	0	0	0	1	0	0	0	2	..
Calcium	Ca	6	5	36	62	92	96	3	155	400
Magnesium	Mg	1	2	8	18	34	27	2	46	1,300
Sodium	Na	2	6	7	44	8	183	215	78	11,000
Potassium	K	1	1	1		1	18	10	3	400
Bicarbonate	HCO_3	14	13	119	202	339	334	549	210	150
Sulphate	SO_4	10	2	22	135	84	121	11	389	2,700
Chloride	Cl	2	10	13	13	10	280	22	117	19,000
Nitrate	NO_3	1	.	0	2	13	0	1	3	..
Dissolved Solids	...	31	66	165	426	434	983	564	948	35,000
Carbonate Hardness	$CaCO_3$	12	11	98	165	287	274	8	172	125
Non-Carbonate Hardness	$CaSO_4$	5	7	18	40	58	54	0	295	5,900

[a] All values are parts per million of the unit cited to nearest whole number (see Reference 5).
[b] Numbers indicate location or area as follows:
(1) Catskill supply—New York City
(2) Swamp Water (Colored) Black Creek, Middleburg, Florida
(3) Niagara River (Filtered) Niagara Falls, New York
(4) Missouri River (Untreated) Average
(5) Well Waters—Public Supply—Dayton, Ohio—30–60 ft.
(6) Well Water—Maywood, Illinois—2090 ft.
(7) Well Water—Smithfield, Va.—330 ft.
(8) Well Water—Rosewell, N. Mexico
(9) Ocean Water—Average

are more likely to be contaminated with municipal sewage and trade wastes. Virtually all mineralized waters also contain biological organisms.

Mineralogical Characteristics. The character and amount of extraneous inorganic materials—including deleterious gases—dissolved and suspended therein, describe the mineralogical characteristics of any water. Revealing such information is the function of a mineralogical chemical analysis.

The analyses in Table 1 disclose the composition of the public water supplies used by about 45 percent of the total population of the cities, in the United States, having more than 20,000 inhabitants.[5]

All values recorded are in terms of parts per million.* This is the approved standard terminology for reporting the results of mineral analyses.[6] Values reported in the other terms commonly used may be converted into the standard form by using the factors listed in Table 2.

*Parts per million are hereinafter abbreviated ppm. A part per million signifies a unit weight of material per million unit weights of the solution.

TABLE 2. CONVERSION FACTORS FOR WATER ANALYSES

To Convert	Into	Multiply by
Grains per U. S. gallon	ppm	17
Grains per Imperial gallon	ppm	14¼
Grams per liter	ppm	1000
Mg per liter	ppm	1

Data for dissolved gases or pH values have been omitted in Table 1 because even waters of the same mineral contents may vary widely in these respects. Unpolluted natural waters usually have pH values within the range 5 to 8, depending upon their *free* CO_2 contents. Polluted waters, which include those derived from wells or swamps in marshy ground, may have pH values well below 5.

Biological Characteristics. The slime-forming organisms are mostly lower plants, grouped by botanists into the Phylum Thallophyta. This group is distinguished by the absence of leaves, roots, or stems from the mosses, ferns, and seed plants which comprise the three other phyla of the plant kingdom.

The Thallophyta (see Table 3) are divided into algae, which can synthesize chlorophyll for the production of sugar, and the fungi which lack chlorophyll, and must therefore secure already synthesized carbohydrates.

All Thallophyta are of universal distribution and many of them are slime forming. Of the five divisions of algae, only three (the green, the blue-green, and the diatoms) are found in fresh water. Of the five divisions of fungi, all may occur in fresh water, the principal slime formers being indicated in Table 3.

The methods of analysis commonly used in the sanitary examination of a water have, as their principal object, to identify and count pathogens.

Most slime-forming organisms are not pathogens. When a water is subjected to a biochemical analysis for the purpose of evaluating its slime producing characteristics, tests, widely different from sanitary bacteriological tests, must be made. Tests upon the water itself are seldom satisfactory, and true indications of the sliming characteristics of a water can only be determined by the analysis of deposits on surfaces having a temperature close to the temperature of the final design equipment. The

TABLE 3. PRINCIPAL SLIME FORMERS

Phyla	Rough Division of Phyla
Algae	Single celled, sometimes forming slimy sheets. Many celled in either sheets or fronds.
Fungi	*Bacteria* (Schizomycetes) frequently forming slimy surface coatings. *Slime Molds* (Myxomycetes) forming slimy sheets as one stage of their life history. *Sac fungi* (Ascomycetes) of which one division, the yeasts, occasionally form slimy aggregates. The alga-like fungi (*Phycomycetes*) and the stalked fungi (Basidomycetes) rarely form slimes but their filaments may hold together the slimes of other organisms.

Table 4. Plant Water Supplied Examination

WATER SUPPLY—	200 feet deep well—average water temperature 53 F—water is producing a brown stain in plumbing fixtures.
SAMPLE—	The sample was scraped from the surface of the shell and tube condenser of #2 Freon Compressor on the meat chilling room. The sterile sample bottle was filled one-half with deposit and the balance with circulating water. No preservative was added—pH at time of collection was 7.4.
ANALYSIS REQUIRED— MACROSCOPIC ☑ BACTERIOLOGICAL ☑	MICROSCOPIC ☑ ORGANIC CONTENT ☑
PROBLEM—	A 25-ton Freon—12 Compressor has head pressure about 10 lb higher than during initial operation, without change in water temperature. Deposits have been observed on heat exchanger surfaces. It is desired to know the nature of these deposits and if they are the cause of this increased head pressure.
MACROSCOPIC EXAMINATION—	Heavy brown flocculant material settles rapidly in clear water. pH—7.3 Odor—woody, mouldy.
MICROSCOPIC EXAMINATION—	Inorganic Material—small amount white crystals. Amorphous Material—small amount—brown. Iron Bacteria—profuse growth of crenothrix—(Photo usually included).
CULTURAL EXAMINATION—	Sabouraud's Agar 1. Aerobic gram positive spore-forming rod with mucoid sheaths. (Photo usually included). 2. Short gram negative coccibacilli (Photo usually included).
TOTAL COUNT	100,000 organism/cc.
ORGANIC CONTENT (by WEIGHT, DRY BASIS)—60%	
DISCUSSION—	The presence of common slime-forming organisms in the deposit combined with high organic content indicates that the deposit is bacterial in origin. Heat transfer reductions would be caused by such a deposit. These deposits, combined with crenothrix, can cause corrosion of both ferrous and non-ferrous metals.
RECOMMENDATIONS—	It is recommended that the water be treated at the suction side of the deep well pump with chlorine in quantities sufficient to maintain a free chlorine residual of 1.0 ppm at the discharge of the shell and tube cooler. This treatment can be scheduled on an intermittent basis.

results of such a test are commonly reported in the manner illustrated in Table 4.

Condensates

All condensates result from the chilling of water vapor. Such chilling may result from natural causes, thus producing *dews*, *sweats*, *rain*, *and snow*, or from artificial causes, as in steam condensing equipment, producing condensate or return water.

In the heating and ventilating field, the biological characteristics of condensates are likely to be of concern *only* where the condensate is used as cooling water in recirculating systems. The deleterious gas contents of condensates, however, very often create serious corrosion troubles.

The data in Table 5 typify the chemical composition of the atmosphere in rural and metroplitan areas, and of stack gases when various types of common fuels are used. The curves in Fig. 1 disclose the solubility of the major deleterious gases present in such atmospheres in otherwise pure water, when the partial pressure of the gas is one pound per square inch absolute.

The most common deleterious gases entrained by steam are oxygen and carbon dioxide. In rare instances, hydrogen sulphide, sulphur dioxide, or ammonia are present.

In most steam condensing equipment,[7,8] the non-condensable gases entrained with steam accumulate so that the amount present in the vapor space is several hundred times higher than in the incoming steam and

TABLE 5. DATA TYPIFYING THE DELETERIOUS GAS CONTENT OF DIFFERENT ATMOSPHERES

Name of Gas	Chemical Formula	Air				Flue Gases					
		Rural		Metro-politan		Bitum. Coal		Fuel Oils		Natural Gas	
		% by Volume	Partial Pressure psia	% by Volume	Partial Pressure psia	% by Volume	Partial Pressure psia	% by Volume	Partial Pressure psia	% by Volume	Partial Pressure psia
Oxygen	O_2	21	3.143	21	3.143	2	0.299	7	1.048	10	1.497
Carbon Dioxide	CO_2	0.03	0.004	0.06	0.009	15	2.245	13	1.946	10	1.497
Sulphur Dioxide	SO_2	None	None	0.003	0.004	0.07	0 010	0.03	0.004	0.0001	0.0015

the amount dissolved in the condensate may therefore approach, or even exceed for short periods, the amount entrained by the steam.

CAUSES AND PREVENTION OF SCALES AND SLUDGES

Scales may be formed on surfaces of equipment in contact with water, and sludges in the body of the water, by the separation from the water of dissolved or suspended solids. According to the nature of a particular piece of equipment and the method of its operation, such separation may be promoted by one or more than one, of several factors:

a. The concentration of solids may be increased by the evaporation of water.

b. The dissolved solids may be rendered less soluble in the water by changes in temperature.

c. Conditions may favor the decomposition of unstable compounds with the formation of less soluble compounds.

Figs. 2 and 3 show that the solubilities of both calcium carbonate and calcium sulphate decrease with the rising temperature within a moderate range of temperatures. Surfaces transferring heat into water, such as condensers and coolers, are more susceptible to scale and sludge formation than are the cold parts of the same system using the same water.

The most common of the unstable soluble salts are the bicarbonates of calcium, magnesium, and occasionally iron. Under conditions favoring the removal of carbon dioxide, as when the water is strongly aerated or when it is boiled, the bicarbonates are readily converted to the relatively insoluble carbonates (or, in the case of iron in the presence of oxygen, ferric hydroxide or oxide may be formed). Conversely, carbonates are readily converted to the more soluble bicarbonate by the addition of carbon dioxide or other acidic materials. This explains the increase in the apparent solubility of calcium carbonate at decreasing pH values (increasing concentration of hydrogen ion) shown in Fig. 2, the carbonate really going into solution largely as bicarbonate.

It is sometimes desired to evaluate the tendency in a particular water toward the separation of calcium carbonate, which may be desirable as a

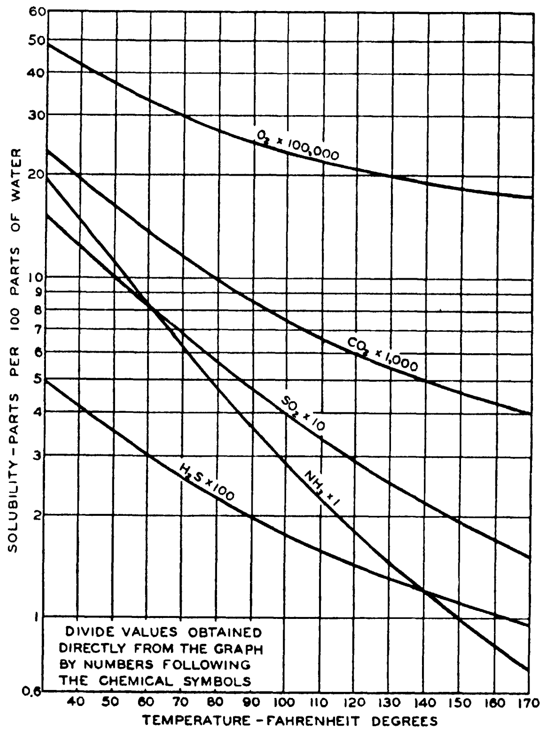


FIG. 1. SOLUBILITY OF GASES AT PARTIAL PRESSURE OF 1 PSI

means of establishing a corrosion-resistant film on metal surfaces, or in other circumstances may be undesirable because of the impedance of the calcium carbonate film to heat transfer. This tendency is indicated approximately by the Langelier Index,[9] which is obtained by subtracting the actual pH of a particular water from the pH at which it is estimated precipitation of calcium carbonate would just begin. This estimate may be made by the use of Fig. 4.

There are various expedients which may be employed for avoiding or mitigating difficulties due to scales:

a. The water may be treated before use to remove elements such as calcium, magnesium, and iron, which form relatively insoluble compounds. In the various softening processes this removal of these elements is accompanied by the addition of other elements, particularly sodium, the compounds of which are relatively soluble.

b. The water may be treated within the equipment to promote the separation of dissolved solids as sludges, rather than as scale which is, in most cases, more objectionable.

c. The increase in total solids, due to the evaporation of water, may be controlled by the displacement, continuously or intermittently, of some of the used water by fresh supply.

d. Substances, such as the polyphosphates, having the property of inhibiting the precipitation of calcium carbonate from solutions supersaturated with it, may be added.

e. The pH of the water may be lowered (hydrogen ion concentration raised) to reduce the tendency for precipitation of carbonate. This is permissible only to such an extent as will not cause a serious increase in rate of corrosion.

The choice of the best expedient must be made for each type of equipment, and will be affected by local considerations.

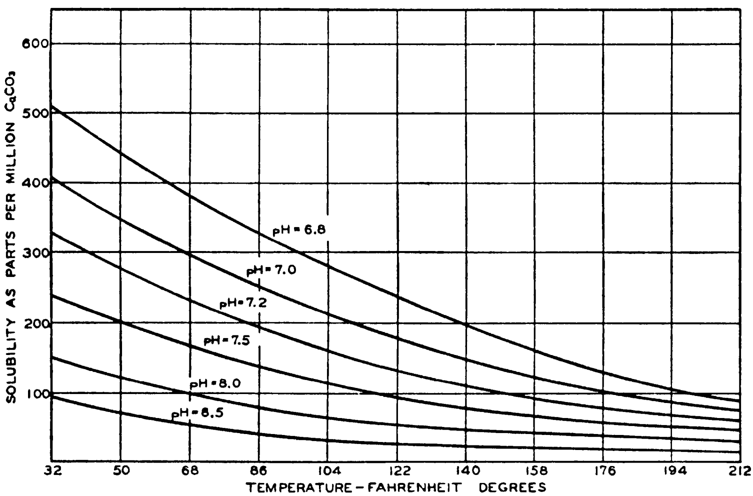


FIG. 2. SOLUBILITY OF CALCIUM CARBONATE IN DISTILLED WATER CONTAINING CARBON DIOXIDE

(pH Values at Approximately 73 F)

Fig. 2 Adapted from (1) Ind. & Eng. Chem., 20 (1928) 1197—by Baylis. (2) J.A.C.S. 50 (1929) 2086—Frear & Johnson.

Once-Through Equipment and Closed Recirculating Systems

Where abundant supplies of water are available at low cost, the cooling water may pass through the equipment once, undergoing a slight rise in temperature. Little difficulty from scale should be experienced in this case unless the carbonate hardness is more than 200 ppm, or the water has been treated to induce incipient calcium carbonate precipitation. Closed recirculating systems in which the water is cooled indirectly, as in radiators, and returned to the equipment, should usually be subject to little trouble with scale. However, in both once-through and closed recirculating systems, slimes may cause trouble.

If there is some tendency for scaling, it may usually be prevented by the addition of small amounts, about 5 ppm or less, of polyphosphate.[10] Alternately, a minor lowering of pH by the addition of carbon dioxide or sulphuric acid may be effective if permissible from corrosion standpoint.

Open Recirculating Systems

Where water from condensers and similar equipment is passed through a spray pond or cooling tower and then returned to the equipment, there is an increase in the concentration of solids because of the evaporation of some of the water into the cooling air, and, moreover, the aeration removes carbon dioxide. Both factors promote the tendency to deposit scale. If the conditions are particularly adverse, it may be necessary to subject the water to a softening treatment before use, this being the more feasible because of the reduced water requirement in such a recirculating system. When this is not practicable, or when the tendency to scale formation is only moderate, a considerable improvement may be effected by the

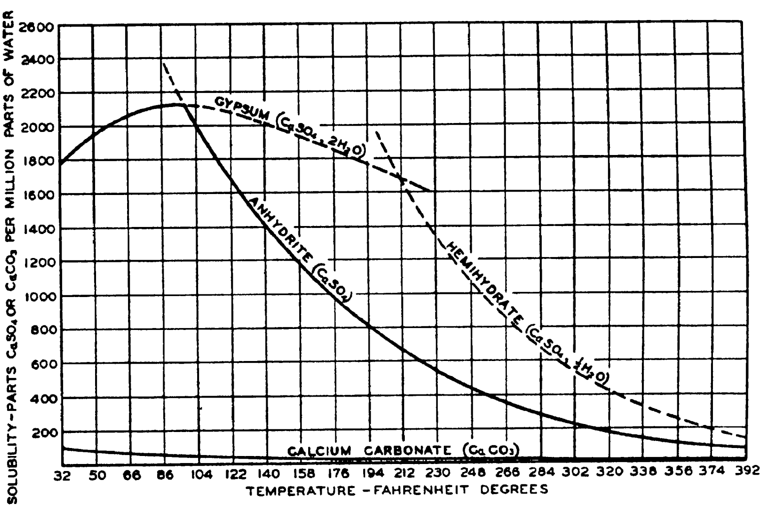


Fig. 3. Solubility of Calcium Sulfate and of Calcium Carbonate for Comparison

($CaCO_3$ in Equilibrium with Normal CO_2 Content of the Atmosphere)

Fig. 3 Adapted from Bull. No. 15, Univ. of Mich. "Formation and Properties of Boiler Scales" by P. E Partridge.

addition to the water of organic compounds such as gelatine, glucosates, dextrine, and tannin which tend to prevent precipitated material from forming adherent scales. In systems of this kind, the loss of liquid as spray from the cooling towers or spray ponds may limit adequately the final concentration of solids in the cooling water. If not, provision must be made for sufficient purging of used water.

Heating Systems

In hot water heating systems or in steam heating boilers where all condensate is returned, troubles from scaling should not be severe. If necessary, sodium phosphate or sodium carbonate may be added to the water to prevent the formation of adherent calcium sulphate scale.

Boilers and High Temperature Equipment

Where temperature exceeds 250 F, complete softening of the water is the only practical method for minimizing sludge formation. This is usually accomplished by artificial or natural zeolites (called also ion-exchange materials) or by hot-process precipitation softeners.

In boilers operating at pressures above 100 psi virtually all the calcium, magnesium, silica, iron, and manganese salts entering with the feed water are potential scale or sludge formers.

In low pressure boilers (100–250 psig), the formation of adherent calcium sulphate (anhydrite) scales is most to be feared. Such deposits form on the *hottest* evaporative surfaces. It is a material of low heat conductivity. Even a layer of egg shell thickness may so impede the rate of heat transfer as to bring about over-heating of the metal.

The ortho-phosphates of sodium are most frequently used to prevent sulphate scales. The concentration of phosphate required is such as to cause the precipitation of calcium phosphate as sludge, thus keeping the boiling water under-saturated with respect to calcium sulphate. To a lesser extent, sodium carbonate (called also soda ash and sal soda) is also used. Most of the effective boiler compounds contain either phosphates or soda ash, or both. Certain organic materials and colloids are sometimes found to minimize scale formation. Where chemicals are introduced directly into the boiler in amounts adequate to prevent scale, sludge is formed in amounts proportionate to the calcium and magnesium salts entering with the feed water. To prevent troublesome accumulation of this sludge, as well as soluble salts, as evaporation occurs, some *blowdown* of boiler water is necessary.

CAUSES AND PREVENTION OF SLIMES

A water containing slime-producing organisms will produce prohibitive amounts of slime *only* when the conditions of use are such as to propagate their life processes. Whenever sufficient food material from normal water or from airborne dust combines with optimum temperature conditions, such as exist on cooling surfaces and air washers, serious quantities of slime will be produced.

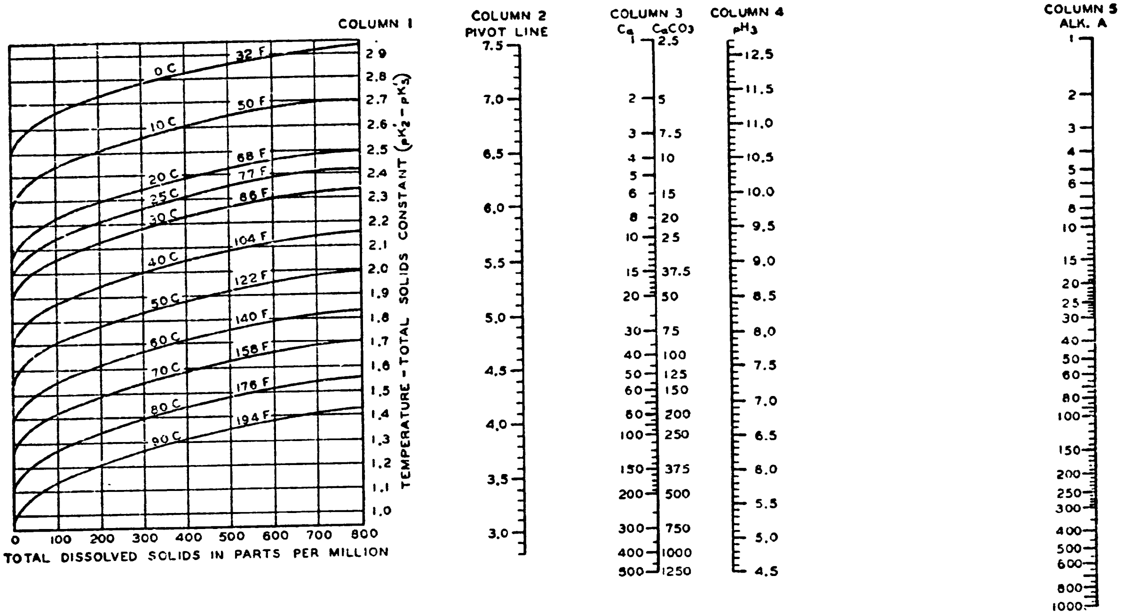
Some natural well waters do not contain sufficient foods to support luxuriant slime growths. Algae, which require light for carrying on their life processes, are likely to cause difficulty in cooling towers and other areas where sunlight is abundant. The ordinary slime-forming bacteria are capable of using a wide variety of nitrogenous and cellulose material as food sources. These bacteria thrive best under dark conditions such as exist in condensers and other heat transfer surfaces. Other organisms capable of causing similar difficulties use such a wide variety of food material as algae,[11] iron compounds,[12] and inorganic sulphates.[13]

At present, the use of toxic chemicals and irradiation are the two general means employed in slime control. The value of ultra-violet light, used so broadly in the beverage industry, is somewhat in dispute.

Anti-fouling paints have been developed and are fairly satisfactory for the prevention of the growth of macro-organisms such as *barnacles* and *mussels*, but these paints must be renewed at frequent intervals, and are not applicable to inaccessible areas such as the inside of pipe lines and cooling towers. Satisfactory *anti-sliming* paints have *not* been found.

Names and other pertinent data relating to some of the more common chemicals used in slime control are shown in Table 6.

Chlorine is the only chemical to which is attributed the ability to destroy slime-forming organisms. The others are presumed to poison marine organisms, most of which *recover* when the chemical is not used regularly.



(Data required for determining pH saturation.

(1) Total alkalinity, as parts per million of $CaCO_3$.

(2) Calcium in parts per million.

(3) Total dissolved solids, in parts per million.

(4) Temperature, in degrees centigrade, at which pH saturation is desired.

FIG. 4. GRAPH AND NOMOGRAM FOR DETERMINATION OF pH SATURATION BY LANGELIER'S FORMULA

(Applicable within pH Range 7.0—9.5)

Based on article in Oct. 1936 issue of American Water Works Association Journal and later corrections for Tables 2 and 4 of 'article. Prepared for Charles P. Hoover of the Columbus, Ohio, Water Softening and Purification Plant by M. L. Riehl.

Instructions for Using Chart:

1. Knowing temperature and total dissolved solids, find temperature and total solids constant on Col. 1.

2. Align this constant with given value of calcium on Col. 3 of Chart, then locate point on Col. 2 of Chart (Pivot Line).

3. Align this point on Pivot Line with given alkalinity on Col. 5; read pH saturation on Col. 4.

Saturation index is pH actual minus pH saturation.

e.g.—pH actual,	pH saturation.	Saturation index.
7.6	8.1	−0.5 (corrosive)
8.4	7 8	+0.6 (scale forming)

While chlorine is the most generally used chemical, the use of others may occasionally prove to be more practicable. Choice of the chemical is conditioned largely by the design and operation of the system.

Open Recirculating Systems

In spray ponds and cooling towers of the open type, light-loving algae growths are likely to cause blocking of the distribution piping and troughs. These organisms are most troublesome in areas accessible to sunlight. Algae slimes are usually stringy in character.

In open recirculating systems, continuous use of small quantities of chlorine is generally most satisfactory. In once-through systems, where large quantities of water are used, intermittent treatment a few times each day will usually result in satisfactory slime removal and chemical economies.

Neither the phenols nor copper sulphate may be used for the removal of slime already formed. For this purpose, chlorine gas is used. After being cleaned, the other chemicals may be used to prevent the reestablishment of slime in the system. The removal of green algae from a cooling tower should never be used as an indication that the true slime-forming organisms on heat exchanger surfaces have been removed. The more resistant slime formers, which so materially reduce heat transfer efficiency, will often be unaffected by treatment which completely eliminates algae.

TABLE 6. COMMON CHEMICALS USED FOR SLIME CONTROL

CHEMICAL	TRADE NAME	PHYSICAL STATE[a]
Chlorine	Chlorine	Gas
Hypochlorites	Calcium Hypochlorites Sodium Hypochlorites	Crystalline
Chlorinated Phenols Sodium—	Chlorophenylphenate Tetrachlorophenate Pentachlorophenate	Briquettes Briquettes Briquettes
Potassium Permanganate	Permanganate of Potash	Crystalline
Copper Sulphate	Blue Vitriol	Crystalline

[a] As Shipped.

Closed Once-Through Systems

In equipment where light is excluded, slime formations are due to fungi. Usually, they predominate on the heat exchange surfaces. Bacteria form thick, soft slime. Yeast and molds form tough rubbery slimes. Chlorine and hypochlorite solutions, fed intermittently, are used to prevent such slimes.

UNDER-WATER CORROSION

When deleterious substances are present in water, the corrosivity of the solution is increased in proportion to the amount of deleterious substances present, the temperature, and usually the rate of flow of the solution over the metal surfaces. There are other relevant factors, but their influence in general is subordinate to those mentioned. Dissolved oxygen, acid gases, and chloride salts are the corrosion accelerators most frequently encountered.

Neutral and slightly alkaline waters saturated with air, corrode iron at a rate about triple that for the same water free of air. Hot water containing oxygen will corrode iron at a rate three to four times that for the same water when cold.

Corrosion of iron decreases as the pH of water solutions increases, and practically ceases at a pH of 11. If the metal contains film forming agents, such as chromium, nickel, and silicon, or if the water contains inhibitors such as silicates and chromates, corrosion may in some instances be minimized.

Cold Water Services

Where water from municipal supplies is used industrially in a closed system with little or no increase in temperature, it is seldom necessary or feasible to treat the water to reduce its corrosivity. When it is mandatory, the addition of caustic soda to *maintain* a pH over 11, *plus* the addition of sufficient sodium sulphite to maintain a residual of over 100 ppm (as Na_2SO_3), usually suffices to prevent serious troubles. However, in some cases the cost may be prohibitive.

When the use of sodium sulphite or a comparable chemical for oxygen removal is prohibited, as in potable waters, the addition of small amounts of lime to maintain a Langelier Index (see Fig. 4) of 0.5 or more may prove helpful.

In systems exposed to the atmosphere, as for example air washers or storage tanks, both laboratory[14] and field tests[15] have shown that the addition of alkalies to maintain a pH greater than 8.5, plus the addition of other chemicals that produce protective films on the metal surface, will measurably decrease corrosion. Sodium dichromate, sodium silicate, and tri-sodium orthophosphate have been shown to be effective film formers in the order mentioned.

Caustic soda is usually used to raise the pH value, and sodium dichromate is most often employed as a film former in industrial waters. In old systems, not previously inhibited, about 500 ppm of sodium dichromate are usually maintained at the start. After two or three months, and in new systems, a residual of about 300 ppm of dichromate usually proves effective. When insufficient dichromate is employed, pitting is sometimes accelerated. Aeration does not impair the efficiency of dichromates, but does deplete the caustic soda concentration.

In large industrial systems, the use of vacuum deaeration has been shown to be effective.[16] In small systems, the equipment required can seldom be justified economically.

Soft water, as for example the effluent from zeolite softeners, is likely to be several times more corrosive to iron than hard waters. In small installations, the use of copper or brass pipe usually is a practical expedient. Cement lined pipe and tanks suitably resist attack.

Where the water contains slime-forming organisms, especially those bacteria that thrive on iron, chlorination of the water is imperative to inhibit tuberculation and subsequent pitting.

Bitumastic paints, applied at *regular* intervals upon *well cleaned* surfaces, will measurably prolong the life of equipment handling cold waters.

Hot Water Services

As a usual thing, corrosion does not create important troubles when temperatures are maintained below 140 F.

In closed systems where little fresh water is introduced, such as in a hot water space heating system, corrosion is usually negligible because the oxygen released in heating the water is purged through the vents.

Where large amounts of fresh water are constantly entering and are being heated, the use of mechanical deaeration is the most universally satisfactory expedient to employ. Where the use of such equipment cannot be justified economically, anti-corrosive chemicals, and the use of corrosion resistant metals, are the more practical expedients to be used.

Treating Chemicals. Alkalies, such as lime and caustic soda, silicates of soda (water glass), the poly-phosphates of soda, sodium sulphite, and sodium dichromate are usually used. Organic compounds, such as the glucosates, dextrines, and tannins are sometimes used, but their value is still a controversial matter. When any chemical is used, so many relevant factors are involved that it is always advisable to seek adequate technical counsel in inaugurating the treatment. Very often, where such precaution is not taken, new troubles are created that are more aggravating than the original difficulty.[17]

Silicate of soda is used to protect iron, lead, and brass water pipe.[18] For most waters, a solution of $Na_2O:3SiO_2$ is recommended. Sodium silicate, equivalent to about 10 ppm added silica, should be fed to the water for the first month after which it may be reduced to give 5 or 6 ppm added silica. Where careful control of the silicate feed is exercised, the water is not injured for domestic use by this treatment. The rate of corrosion of iron pipe has been reduced by 70 percent, and dezincification of brass pipe practically stopped, by this simple treatment. The amount required and the effect are not the same in all waters.

Pipe Materials. Brasses with 60 to 67 percent copper are dezincified in some corrosive waters, and in certain localities are not much more serviceable than galvanized iron or steel pipe. The zinc in brass pipes is leached out locally, leaving a plug of porous copper. The weakening of such pipe is especially noticeable under the threads. Dezincification is retarded by the use of silicate of soda (8 ppm added silica).[19]

In salt or fresh water, there is no material difference in rate of pitting of wrought iron, steel, low metalloid steels, or copper bearing steels. This is contrary to the relative performance of these metals in atmosphere.

Refrigerating Systems

Corrosion in refrigerating systems is confined to surfaces in contact with brines or those in contact with the refrigerant.

Brines. Refrigerating brines usually are comprised of sodium chloride, calcium chloride, or calcium and magnesium chlorides. The corrosivity of dilute brines is higher than their more concentrated solutions. The corrosivity of sodium brines, other conditions being fixed, is about 1.5 times greater than brines of the alkaline earth metals.

Brines are excellent electrolytes. Contact of dissimilar metals of wide potential differences, when in contact with brines, results in rapid corrosion by galvanic action.

The leakage of air, acid refrigerants, or both, accelerates the corrosivity of brines. Ammonia precipitates calcium and magnesium salts, thus clogging the system at restricted points.

The addition of caustic soda and sodium dichromate to brine solutions to inhibit corrosion of iron, is a more or less general practice. Sodium silicate and sodium phosphate are also used at times, but tests indicate they are not as effective as is sodium dichromate. It has been suggested[20] that 125 lb of sodium bichromate per 1000 cubic feet of calcium chloride brine, and 200 lb per 1000 cubic feet of sodium chloride brine, be added to inhibit brines; that when salt or calcium chloride is added to "strengthen" brine, sodium dichromate also be added in the amounts shown in Table 7.

Refrigerants. The common refrigerants, except those of the hydrocarbon type, will attack the common metals and alloys if *moisture is present.* Even a very small amount of water may cause severe corrosion

TABLE 7. QUANTITIES OF SODIUM DICHROMATE TO BE ADDED TO MAINTAIN INITIAL CONCENTRATION

Specific Gravity of Brine to be Strengthened	Lb Sodium Dichromate per 100 lb $CaCl_2$ Added
1.16	0.695
1.18	0.621
1.20	0.556
1.22	0.502
1.24	0.455
	Lb of Sodium Dichromate per 100 lb NaCl Added
1.12	1.79
1.14	1.47
1.16	1.32
1.175	1.18

with certain refrigerants. The amount required need only be sufficient to produce a water film on the metal surface.

With the halogenated hydrocarbons, complete elimination of water is much to be desired. Where ammonia is used, copper and its alloys, aluminum and zinc, are attacked especially at elevated temperatures. When sulphur dioxide is used, more than 50 ppm (0.005 percent) of water will cause appreciable corrosion of virtually all the common materials.

Minimizing Condensate Corrosiveness

There are four expedients that may be utilized to minimize corrosion in steam condensate systems: (1) treatment of the boiler feedwater so as to eliminate deleterious gases entrained with the steam, (2) design of the condensing equipment to minimize dissolution in the condensate of the deleterious gases entrained with the steam, (3) chemical treatment of the condensate, (4) use of resistant metals.

Boiler Feedwater Treatment. Elimination of oxygen from boiler feedwater and, therefore, from the steam developed, can be accomplished either mechanically or chemically. In some steam generating stations, both expedients are employed.

Tests[21] have indicated that in small low-pressure heating boilers, where the boiler input contains less than about 50 ppm of carbonate hardness, the CO_2 in the steam can be controlled by adding calcium hydroxide to the boiler. In Fig. 5 are shown the equilibria conditions proposed for boilers operating at pressures up to about 5 psi gage. This expedient may *not* be used in higher pressure boilers, because of the possibilities of scale and sludge formations. In the latter, the only method used to date for treating the feedwater consists of removing the alkaline earth salts, *i.e.*, softening, and subsequent acidulation followed by deaeration at temperatures near the atmospheric boiling point of water.[22]

Design of Condensing Equipment. In the design of water heaters and comparable types of condensing equipment,[23] it is possible to shift the accumulation of non-condensible gases to a location away from the condensate level and, subsequently, vent these gases to the atmosphere.

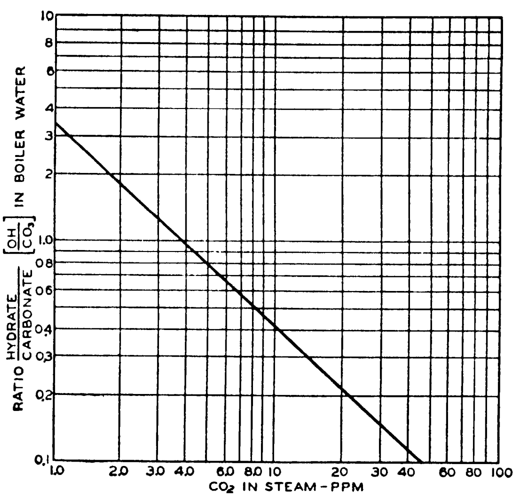


FIG. 5. RELATION OF HYDRATE/CARBONATE CONTENT IN *Hard* BOILER WATER AND CO_2 IN STEAM AT ABOUT 5 PSI OPERATING PRESSURE

(*All analytical values are ppm by weight*)

Venting an amount of steam equal to about one-half percent of the total steam entering the condenser is the optimum vent rate.

Venting is of little practical value when the CO_2 content of the incoming steam is below about 5 ppm. When the steam contains more than 5 ppm, venting provides a means of producing a condensate containing a minimum of about 3 ppm. However, even as little as 3 ppm of dissolved CO_2 can produce active corrosion if large amounts of condensate are flowing.

Chemical Treatment of Condensate. Condensates containing comparatively large amounts of oil, are practically non-corrosive, due to the protective film provided by the oil. When oil is intentionally added to condensate,[24] inadequate quantities may accelerate rather than decelerate corrosion on those surfaces not covered by the oil. Sodium silicate added to CO_2-bearing condensate has been shown to decrease, but not entirely prevent, corrosive action. It is not definite whether the protection afforded by silicate solutions is due to the establishment of a protective film on the metal surface or to neutralization of the CO_2 by the alkali in the silicate solution.

It has been postulated that ammonia,[25] cyclohexylamine,[26] ethylene diamine, and morpholine[27] will retard corrosion of condensate lines. Tests with benzylamine have also been reported.[28] Where copper and its alloys are involved, the use of alkaline inhibitors is believed inadvisable. The use of small amounts of sodium hexametaphosphate has been suggested too, but tests[29] indicate that this salt accelerates rather than decelerates, the rate of attack of steel by condensate containing CO_2 and oxygen. Whether chemical treatment of steam or condensate is feasible, must be determined not only upon the basis of the acuteness of corrosion troubles, but also upon the uses to which the steam or condensate is put.

Use of Resistant Metals. For economic reasons, the metals known to resist corrosion can seldom be used exclusively for condensate lines in any

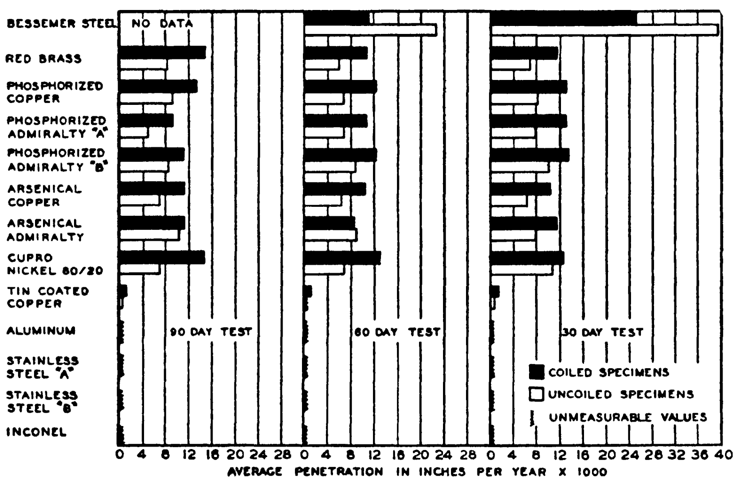


FIG. 6. COMPARATIVE CORROSION RESISTIVITY OF 10 MATERIALS EXPOSED TO CONDENSATE

sizable enterprise. Nevertheless, there may be instances where the use of a limited amount of the more costly, but resistant, materials can be justified. The data in Fig. 6 are the results of tests[30] designed to reflect the corrosion resistance of the more commonly used metals to attack by condensate containing oxygen and CO_2.

In contemplating the use of a resistant metal, as a section of a condensate line, it should be remembered that, if other conditions are *right,* corrosive attack will merely be transferred down stream in the system. Galvanic corrosion resulting from the contact of dissimilar metals in a condensate line seldom occurs. No paint or similar protective coating has thus far proven satisfactory. Tests of cement lined and vitreous lined pipe have shown the linings to be readily dissolved by hot condensates.

ATMOSPHERIC CORROSION

Most of the problems originated by atmospheric corrosion occur in connection with the fire-side of boilers and furnaces (including their flues and stacks), sewer vents, air ducts, coal and ash handling equipment. Usually such equipment is fabricated from common types of ferrous metals.

Generally little or no atmospheric corrosion occurs at temperatures higher than the boiling point of water, because at such temperatures little or no condensate is formed. If it does form at the higher temperatures, only negligible amounts of carbon dioxide and oxygen present in the atmosphere, will dissolve in the hot liquid, but sulphur gases may dissolve and cause rapid attack. Oxygen, sulphur dioxide, sulphur trioxide, and carbon dioxide are the deleterious gases most frequently accountable for corrosion in moist atmospheres.

Coal Storage and Handling Equipment

Virtually all coals contain sulphur in the form of pyrite, and some moisture. In storage, the pyrite is likely to be decomposed by oxidation. Moisture dissolves the products of decomposition forming sulphurous and sulphuric acid. The acid solutions vigorously attack the supporting metal.

Rubber linings have been developed for coal chutes and bins to effectively resist corrosion and the abrasive action of the coal, but they are expensive.[31] Concrete linings for steel bunkers have also been effectively employed.[32]

The use of high chromium steels is not always a sure cure, especially with coals treated with dust allaying agents high in chlorides.

Flues, Stacks, and Fire-side of Boilers

The surfaces of flues and boilers contacting the products of combustion, seldom experience corrosive attack when the equipment is in operation. Breechings, smoke hoods and canopies in contact with flue gas may, however, be subject to attack during the warming-up period of an appliance, or when the rate of operation is so low that the temperature of the flue gas is below the dew-point. It is common practice to use cast-iron or acid resistant vitreous enameled steel in flue gas connections to appliances, to prolong the life of these parts. The shut-down period, when condensation of moisture occurs on the metal surfaces, is usually the time when most damage is done.[33] In those sections of the stacks where flue gas temperature drops below the dew-point, corrosion is inevitable during operation.

It is clear that where long shut-down periods are anticipated, a practical method for mitigating corrosion is to clean the surface thoroughly and to provide adequate clean, dry air circulation to prevent condensation. (See also Care of Idle Heating Boilers, Chapter 15).

Protective coatings with organic binders are destroyed rather rapidly above 400 F because of the decomposition of the organic materials. The surfaces of metals, whose temperature does not exceed 400 F, may be protected by periodically applying paints such as those specified in the following paragraphs entitled Air Ducts.

Air Ducts

The most practical method for protecting air duct surfaces made of steel from atmospheric corrosion, is to apply protective paints. One of the most effective protective coatings is red lead paint.

Three coats of paint should be applied, of which the first two coats should be rust inhibitive paint such as red lead paint, with the second coat tinted to a light brown color with carbon black, and the finishing coat may be red lead paint tinted to a black or brown color, black paint made according to Federal Specification TT-P-61, red iron oxide paint conforming to Federal Specification TT-P-31, or white or light tinted paint made according to Federal Specification TT-P-40.

Another paint which has had some use for priming iron and steel is zinc chromate paint.

Under some conditions, a chlorinated rubber base paint made according to Federal Specification TT-P-91 may be used for the finishing coat, particularly where the presence of highly corrosive gases or contact with strong alkaline water would injure the standard paints. Rubber base paints should be used only for the finishing coat over regular priming and second coats.

BURIED PIPE LINES

Lines that are cold and in intimate contact with the earth are corroded from the same causes as in mineral waters, but pitting is usually more intense due to variations in concentration of salts and oxygen in solution, acidity, drainage, and presence of solid materials (such as cinder) in contact with metal pipe. Galvanic currents, induced by contact of certain dissolved constituents in the soil, often act over a large area, and accelerate corrosion where they leave the pipe line.

Certain bacteria that thrive in the absence of oxygen have the power to obtain hydrogen and dissociate sulphates in the soil, with a resultant production of hydrogen sulfide which attacks the iron to form iron sulphide.

Stray electric currents from electric power generating stations sometimes find their way into buried steel structures, and do damage in proportion to the current density where the current leaves the metal to enter the ground.

Pipe Materials

Under many conditions where steel would be corroded, the use of corrosion-resistant metals other than steel may be desirable, even if greater in first costs. Wholly austenitic stainless steels are very resistant to underground corrosion. In most environments copper, red brass, and copper-silicon alloys will resist corrosion and may, at times, be used to advantage. However,[34] soils with a high content of organic matter, or alkaline soils in which the ratio of chlorides and carbonates to sulfate is high, may be corrosive. Copper should not be embedded directly in cinders or in tidal marshes where it may be subjected to attack by sulfur compounds. Lead[35] corrodes chiefly in soils deficient in oxygen or containing cinders. Because lead is corroded to a considerable extent in most soils, lead coatings applied to steels are not adequate for underground use.

Galvanized iron pipe will resist corrosion for various periods of time, depending on the soil and how long the galvanized coating lasts. The zinc used for the galvanized coating, is on the electrochemical protective side of the iron, and the zinc is corroded and changed to zinc compounds before the iron is attacked. This accounts for the protection afforded by galvanized iron. Even if some protection is obtained, eventually the galvanized coatings are destroyed by chemical action and the corrosion of the steel begins.

Protective Coating

Protective coatings for buried pipe lines are in a class by themselves because of the unusual service conditions, and because it is not possible to maintain them by recoating when necessary. Buried steel pipe lines have been protected against corrosion with considerable success by the use of very thick bituminous coatings applied in molten condition. The best results are obtained by applying the bituminous coatings over a standard priming coat such as red lead or a bituminous paint, and for long service it has been found that after the bituminous coatings are applied, a wrapping of asbestos fabric saturated with bitumens will prevent movement and displacement of the bituminous coatings, and add greatly to the length of time satisfactory protection will be maintained.

Cathodic Protection

Protection is obtained by rendering the structure cathodic to the surrounding water or soil by means of a controlled difference of potential. This method, which has proved satisfactory and economical on a number of gas and oil pipe lines underground, has also been applied with some success to the protection of the inside of water storage tanks and other structures that are in contact continuously with water. Protective coatings that insulate a large portion of the metal surface will reduce very materially the total amount of protective current that must be impressed on bare anodic areas to arrest corrosion.

Because of differences in environmental conditions, it is necessary to determine or estimate the minimum current density required for each structure, and design the anode or anodes so that the necessary protection can be obtained most economically. In water having relatively high electrical conductivity such as in sea water, this is comparatively easy compared with fresh water. In the latter, the composition of the water is a major factor. It is therefore desirable to obtain an accurate estimate of the minimum current density required. The current is then controlled by the potential between the anode and the structure to be protected.

Rectifiers have generally proved to be the most practical means for supplying the necessary current for protection of surfaces in contact with neutral waters.[36]

HANDLING WATER TREATING CHEMICALS

Virtually all the chemicals used in water conditioning are injurious if taken internally in large doses. Many also cause severe skin irritation. Thus, they should be handled with caution.

Caustic soda, lime, and concentrated sulphuric acid will burn the flesh. In addition, if mixed with small amounts of water, sufficient heat may be generated so that spattering occurs or the container becomes too hot to handle.

The chlorophenol compounds, even in the low concentrations used in water conditioning, have been reported[37] to produce dermatitis. Chrom-itch is not uncommon among workers handling chromates. The amines are said to be absorbed through the skin.[38] Morpholine is said to cause kidney and lung trouble when so absorbed.

Chlorine gas irritates the skin, eyes, and mucous membranes. Concentrations as low as 0.004 percent by volume in air cause dangerous illness in 0.5 to 1 hour.

When relatively large amounts of the non-gaseous chemicals are to be handled, protective clothing, including goggles, should always be provided, and a shower head or its equivalent provided at or very near the point where the chemicals are mixed. *Chemicals should always be washed from the skin with large volumes of water.*

For the handling of chlorine and chlorinators, the *U. S. Public Health Service*[39] stipulates the following safety requirements:

1. Suitable gas masks and a small bottle of ammonia for testing for leaks should be kept at convenient points immediately outside the room or enclosure in which chlorine is being stored or is in use. Gas masks should be inspected at regular intervals and kept in serviceable condition. Note:—All-purpose masks offer adequate protection only when the concentration of acid gases does not exceed two percent—see Safe Practices Pamphlet #64 National Safety Council.

2. Chlorinating equipment and cylinders of chlorine should be housed preferably in separate buildings above the ground level.

3. The room or building housing chlorinators in service should be maintained at a temperature above 60 F, but never in excess of the normal summer temperature. The cylinders of chlorine should be shielded, where necessary, from excessive heat or cold. Direct heat should not be applied to cylinders of chlorine, nor should hot water be poured over them or come in contact with the cylinder valve.

4. Adequate ventilation should be provided for all enclosures in which chlorine is being fed or stored.

5. All joints of tubing connecting chlorine cylinder and chlorinators should be kept absolutely tight and inspected frequently to insure tightness. Tubing should slope upward from the cylinder.

LEGAL REGULATIONS

In a number of states, the water used for humidification, even in industrial plants, is required to meet drinking water standards insofar as bacteriological quality is concerned. A ruling of the *U. S. Department of Agriculture, Meat Inspection Division*, prohibits the use of chromate in water used for air washing when the air later contacts foodstuffs.[40]

There is an ever growing consciousness on the part of public health officials, of the necessity for regulations to *protect* potable water supplies. Attesting this is an ordinance[41] now in effect in Detroit, Michigan, which stipulates in part:

"No physical connection shall be maintained between lines carrying city water and pipes, pumps, or tanks supplied from any other source. Where dual supplies are necessary or desired, lines carrying city water must be protected against back flow of polluted water by an *atmospheric gap*. Secondary supplies and emergency sources shall include: surface waters from rivers, lakes, ponds, lagoons, and reservoirs; well waters both deep and shallow; any supply of water which has been stored, held, or reserved after being used for industrial purposes; cooling water, or water which has in any way been treated, processed, or has been subjected or exposed to any contamination of a bacteriological or chemical nature; and water from any other source than the city supply."

The *U. S. Public Health Service* stipulates:

"Salts of barium, hexavalent chromium, heavy metal glucosides, or other substances with deleterious physiological effects, shall not be allowed in the water supply system."

The same agency recommends that the concentration of the substances listed be held below the values cited in Table 8. The Board of Directors of the *American Water Works Association* has accepted these values as standard for *all* public water supplies in the United States.[42] While their action is not binding, prudence dictates that no form of treatment should be used that will result in raising the concentration of the substances listed above the value cited.

Since virtually all of the permissible chemicals used for scale, slime, and corrosion control have deleterious, physiological effects if taken internally

TABLE 8. RECOMMENDED MAXIMUM ALLOWABLE CONTENT IN WATER SUPPLY[a]

SUBSTANCE	MAX CONCENTRATION, ppm	SUBSTANCE	MAX CONCENTRATION, ppm
Copper	3.0	Arsenic	0.05
Iion & Manganese (Total)	0.3	Selenium	0.05
Magnesium	125.0	Phenols (Total)	0.001
Zinc	15.0	Poly-phosphate of Sodium	10.0
Lead	0.1	pH Value @ 25C	10.6
Fluorine	1.0		

[a] *U. S. Public Health Service.*

in *relatively large doses*, they should always be carefully *proportioned.* To insure this, the Detroit ordinance stipulates that the chemical feeding device must have the following major characteristics:

"1. There shall be a visible means of checking the quantity of material being applied by the feeding device.

2. A water metering device, sealed to prevent tampering, shall be installed to measure the flow of water being treated.

3. The device shall be constructed so that in the event of back-flow or vacuums, the maximum amount of material that may be possibly back-siphoned from the device or any of its attachments or parts shall not exceed one fluid ounce.

4. Should there be a failure of the water metering device or the water supply, the feeding device shall automatically cease operating."

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CHAPTER 43

OWNING AND OPERATING COSTS

Fixed Charges: Amortization, Interest, Taxes, Insurance, Rent; Maintenance Costs, Service Costs: Operating Refrigerating Equipment, Condenser Water, Heating

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THE purpose of this chapter is to discuss owning and operating costs of heating, ventilating, and air conditioning systems. The discussion will dwell particularly on air conditioning systems, as most owners or prospective purchasers will be interested in comparing the owning and operating costs with the possible investment return due to increased patronage, improved efficiency of employees, or improvement and maintenance of quality in a manufactured product.

Owning and Operating Costs may be grouped under three headings: (1) *Fixed Charges,* (2) *Maintenance Costs,* and (3) *Service Costs.*

FIXED CHARGES

Fixed Charges, which are the cost of owning the system, include: (1) Amortization, (2) Interest, (3) Taxes, (4) Insurance, and (5) Rent.

Amortization

Amortization cost will depend on: (1) the total first cost, and (2) the amortization period.

The *total first cost* of an installation is the actual dollar outlay or capital expenditure required to buy and install the air conditioning, or heating and ventilating system ready for operation. It can be divided into two parts: (*a*) the first cost of the air conditioning or heating and ventilating system itself, and (*b*) other first costs incurred because of the installation of the air conditioning or heating and ventilating system.

The *first cost* of air conditioning or heating and ventilating systems includes the following:

1. Heat producing equipment including boilers, burners, controls, etc.
2. Heat distributing equipment including direct radiation, piping, etc.
3. Air handling equipment including fans, air heaters, air conditioners, filters, controls, etc.
4. Air distribution system including ducts, outlets, grilles, etc.
5. Refrigerating equipment including piping, pumps, etc.
6. Water conservation devices including towers, evaporative condensers, etc.
7. Insulation of pipes, ducts and equipment.

The best procedure for establishing the first cost of any system is to first determine the heating and cooling load and then, after a thorough engineering study, select the type of system. The installed cost of a system will vary between wide limits depending upon the type of equipment selected, the design of the distribution system, equipment and labor costs in the locality, etc. An approximate cost, within ± 10 percent, may then be quickly determined for a selected design in a given locality by adding

TABLE I. COST DOLLARS PER TON FOR A COMPLETE SYSTEM, INCLUDING HEATING AND COOLING COILS, FAN, DUCTS, REFRIGERATION EQUIPMENT, TEMPERATURE CONTROLS, ETC.*

1	2	3	4	5	6	7	8	9	10	11	12	13
			Unit Costs		Total Cost[a]			Unit Costs[b]				
Refrigeration Tons	Floor Area sq ft	Air Distributed cu ft	Refrigeration Equipment per Ton	Ducts, Fans, Heating Coils, Etc. per cfm	Refrigeration Equipment	Air Handling Equipment	Total	Per Ton	Per Sq Ft Floor	Per cfm	Evaporative Condensers Per Ton[c]	Cooling Towers Per Ton[c]
25	8,333	12,500	250	1.05	6,250	13,100	19,350	775.00	2.32	1.55	190.00	
50	16,666	25,000	224	.98	11,200	24,500	35,700	730.00	2.15	1.43	155.00	
75	25,000	37,500	212	.95	15,900	35,700	51,600	690.00	2.07	1.38	120.00	
100	33,332	50,000	204	.92	20,400	46,000	66,400	664.00	1.99	1.33	90.00	52.00
150	50,000	75,000	195	.91	29,200	68,200	97,400	650.00	1.95	1.30	60.00	50.00
200	66,664	100.000	192	.90	38,200	90,000	128,200	641.00	1.93	1.28		48.00
250	83,333	125,000	187	.88	46,800	110,000	156,800	628.00	1.88	1.25		46.00
300	100,000	150,000	182	.86	54,600	129,000	183,600	612.00	1.84	1.23		44.00
400	133,328	200,000	165	.84	66,000	168,000	234,000	585.00	1.76	1.17		42.00
500	166,666	250,000	158	.82	79,000	205,000	284,000	568.00	1.71	1.14		40.00

* This table was compiled during latter part of 1950.

[a] Includes only final water, drain and electrical connections within the equipment room.

[b] Columns 9, 10, and 11 represent costs where well water or city water is available for condensing purposes. Where conservation of water is required, add column 12 or 13 to column 9 to obtain total cost per ton.

[c] Does not include cost of structural reinforcing if required to support the weight of cooling towers or evaporative condensers.

the estimated unit costs of the component parts of the system. A reasonably precise estimate of the cost of the components may be obtained from cost records of recent installations of a comparable design, or may be obtained from manufacturers or contractors.

TABLE 2. PROBABLE USEFUL LIFE OF EQUIPMENT[a]

	LIFE IN YEARS
1. HEAT PRODUCING EQUIPMENT	
(*a*) Boilers	20
(*b*) Stokers and burners	20
2. HEAT DISTRIBUTING EQUIPMENT	
(*a*) Piping—copper	same as bldg.
(*b*) Piping—iron	20
(*c*) Radiation—concealed	25
(*d*) Radiation—direct	25
(*e*) Valves and specialties	10
3. AIR HANDLING EQUIPMENT	
(*a*) Filters—automatic	20
(*b*) Heating and cooling coils	20
(*c*) Spray humidifiers and dehumidifiers	10
(*d*) Fans	15
(*e*) Air conditioning units	10
(*f*) Motors	20
(*g*) Electrical starting equipment	20
(*h*) Pneumatic control systems	15
(*i*) Electric control systems	15
4. AIR DISTRIBUTING EQUIPMENT	
(*a*) Ductwork	same as bldg.
(*b*) Outlets, grilles	20
(*c*) Duct insulation	15
5. REFRIGERATING EQUIPMENT	
(*a*) Centrifugal refrigerating machines	20
(*b*) Reciprocating refrigerating machines	20
(*c*) Motors and starters	20
(*d*) Piping—copper	20
(*e*) Piping—steel	20
(*f*) Pumps	20
6. WATER SAVING DEVICES	
(*a*) Evaporative condensers	15
(*b*) Cooling towers	15
(*c*) Wells	25

[a] Taken from *U. S. Bureau of Internal Revenue* Schedule of Probable Useful Life, revised 1942.

In Table 1 an attempt has been made to give approximate averages of the cost of air conditioning systems where the refrigeration requirement is one ton for every 333 sq ft of floor area, and where the air quantity to be distributed is 1½ cfm per sq ft of floor area. This table should be used with caution, as the floor area per ton of refrigeration and the cfm per square foot of floor area may vary between wide limits.

Other first costs, incurred because of the installation of the air conditioning or heating and ventilating system, include costs of electrical work, plumbing, miscellaneous piping, building alterations, cutting, patching, furring in of ducts or pipes, foundations, structural supports, remodeling or redecorating after installation, consulting engineer's fees, licenses, permits, etc. These vary so widely that no approximations are possible, and each case must be considered alone.

The length of the amortization period to be used depends upon: the type and remaining life of the building or space for which the system is to be used; the type of equipment to be employed as a part of the system; the character of the business; and the lease or ownership conditions.

TABLE 3. OWNING AND OPERATING COST

FIRST COST		ANNUAL SERVICE COST	
Cost of mechanical system		Electric Power Costs	
Other costs		Fans	
First Cost (FC)—Total		Pumps—Chilled water	
ANNUAL FIXED CHARGES		Pumps—Condenser water	
		Pumps—Well water	
Amortization—Depreciation period Y years		Cooling tower fans	
Interest rate $I\%$		Cooling tower pumps	
Amortization and Depreciation $\frac{FC}{Y} =$		Refrigeration machines	
Interest: $\frac{Y+1}{2Y} \times I =$		Miscellaneous or other	
Taxes		Gas	
Insurance		Coal	
Rent		Oil—for boilers or Deisel engines	
Annual Fixed Charges: (Total)		Steam	
ANNUAL MAINTENANCE COSTS		For Direct Heating	
Replacement or servicing of air filters		For Ventilation—preheaters	
Outside Maintenance service		For Ventilation—reheaters	
Water Treatment		For Turbine driven equipment.	
Lubricating oil and grease		For Engine driven equipment.	
Painting for corrosion protection or other purposes		Sewers	
Replacement of worn parts		Charges for discharging water into public drainage systems	
Refrigerant		Condenser water	
Wages of engineer or operator		Annual Service Costs—TOTAL	
Annual Maintenance Cost—TOTAL		SUMMARY	
		Annual Fixed Charges	
		Annual Service Costs	
		Annual Maintenance Costs	
		Annual Owning and Operating Costs—TOTAL	

Depreciation, due to deterioration and obsolescence, must also be considered in arriving at the amortization period. Deterioration and maintenance generally go hand in hand. If a long depreciation period is to be used, then the item for maintenance, repair and replacement of wearing parts must be greater than for a short depreciation period.

An approximation of the useful life of various items of equipment and parts of systems is shown in Table 2. It should be noted that if an appropriate maintenance item is not established, the rate of equipment deterioration may be increased substantially.

Interest

The interest chargeable may be based on the average money rate for the period in which the first cost of the equipment will be amortized.

Some accountants do not include interest in the annual fixed charges, as they consider it a negligible item.

The formulas for computing interest and amortization are given in Table 3. This table will also serve as a check list of the various components to be considered in determining owning and operating costs.

Taxes

The taxes chargeable will be the proportion of property tax caused by the increased valuation of the property due to air conditioning.

Insurance

The rate for insurance may vary considerably depending on the type of structure in which the equipment is located and upon other governing factors A rate of about $0.60 per $1,000 may be considered as being representative of normal installations.

TABLE 4. APPROXIMATE MAINTENANCE COST FOR LARGE AIR CONDITIONING INSTALLATIONS, USING HIGH QUALITY EQUIPMENT

	DOLLARS PER TON PER YEAR
Repairs for refrigeration machinery	0.96
Refrigerant	0.40
Oil and grease	0.11
Painting (Water boxes and dehumidifiers)	0.40
Filters, clean and re-oil 4 times per year	1.36
Controls, outside service	0.24
Cleaning air conditioners	1.19

Rent

If the equipment under consideration is to be located in rented or leased quarters, it may be necessary to include an item for space rental.

MAINTENANCE COSTS

Maintenance costs include replacement parts and the labor required for making repairs, replacing parts, cleaning, painting, etc. It should be noted that major overhauling or complete replacement may restore the capital value of certain items of equipment, and in such cases the costs incurred may not necessarily be charged as maintenance costs. Generally, routine labor requirements will be the function of an operating engineer or staff, and the responsibility of this group may extend beyond the equipment being discussed here; hence, it is important to include only an equitable share of the time of this group. Extraordinary repairs involving special machinery will usually be covered by contract with equipment service divisions, and should be accounted for on that basis.

Many of the items included in maintenance costs are highly variable and depend on the type and quality of the purchased equipment. For large air conditioning installations, using high quality equipment, some approximate costs per ton are given in Table 4.

SERVICE COSTS

Service costs include the costs for power, water, steam, coal, oil, etc., consumed to operate the system.

From the selected equipment and type of installation, it is possible to segregate the relatively constant power loads and the total brake horsepower. Annual power cost can then be figured from the following formula:

$$\text{annual power cost} = \frac{0.746(\text{bhp})HR}{\eta} \tag{1}$$

where

bhp = brake horsepower.
H = annual operating hours.
R = power rate, dollars per/kwhr.
η = motor efficiency (decimal).

In using Equation 1 it must be pointed out that the electric rate, R must reflect the proper combination of energy and demand rates. These vary widely between the utility companies, and sometimes the rate structure is such that it is largely the demand charge which determines the proper value of R to use in Equation 1.

Operating Refrigerating Equipment

In an air conditioning system the refrigerating equipment is usually the largest power consuming item to be considered. Also, the prediction of operating cost is more difficult because the power required for summer cooling is affected by many factors of a variable nature.

Table 5 gives the equivalent full load operating hours of refrigerating equipment used for summer cooling for the period of May 15th to October 15th. This table was calculated from the following equation:

$$H_e = m(b + cf) \tag{2}$$

where

H_e = equivalent full load operating hours of refrigeration equipment used for summer cooling during period May 15 to October 15.
m = total hours during period May 15 to October 15 that the establishment is open for business.
b = fraction of maximum load from internal heat under average operating conditions.
c = fraction of maximum load which is due to external sources at maximum design conditions.
f = ratio of the number of hours for a particular city, when the outside wet-bulb exceeds 65 F, during the period June 1 to October 1 to the total number of hours during that same period. Total hours are assumed as 8 hr per day period for barber shops, department stores, funeral parlors, offices, short hour restaurants, and specialty shops, and 12 hr per day period for drug stores, long hour restaurants, and theaters.

It should be pointed out that certain southern cities may have seasons longer than the 5-month period indicated in Table 5. If it is desired to consider a longer season of operation, the ratio of full load operating hours to hours open for business is smaller; in other words, the *refrigeration load factor* is lower. This is true because the extra increment of days added will be a relatively light load, since the table already includes the more severe part of the season.

The season electrical power cost for refrigerating equipment is then given by the following equation:

$$\text{season power costs} = \frac{0.746\ (\text{bhp}_t)\ TH_e R}{\eta} \tag{3}$$

where

bhp_t = brake horsepower per ton (see Fig. 1) for *average* load during period. (Due allowance should be made for poorer compressor efficiency at light load.)
T = tons of refrigeration at maximum design load.
H_e = equivalent full load refrigeration operating hours (Table 5).
R = power cost, dollars per kwhr, including demand and energy charges.
η = motor efficiency at average load (decimal).

In considering refrigeration power consumption, it should be noted that the use of weather records for a specific year may lead to large inaccuracies in estimating operating costs, since there may be wide variations from year to year, and therefore, average yearly weather records should be used rather than those for any individual year.

If the refrigeration compressor is steam turbine driven, the same general method can be followed, taking into account average water rate per brake horsepower-hour and the cost of steam.

Condenser Water

Condenser water cost estimates can also be based on equivalent full load operating hours of the refrigeration equipment. The varying temperature of the water at its source, as well as the temperature of the discarded water, must, however, be taken into account. In general, when water is purchased, control is provided to hold the leaving water temperature (or condensing temperature) constant; and in such case the entering water temperature becomes the major variable, and the gallons per minute per ton can readily be calculated for any water temperature rise.

The following equation for cost of condenser water is useful:

$$B = 0.060\ a\ T\ H_e\ C \tag{4}$$

where

B = cost of water for refrigeration during period, dollars.
a = average gallons per minute (ton).
T = tons of refrigeration at maximum design load.
H_e = equivalent full load refrigeration operating hours (Table 5).
C = water cost, dollars per 1000 gal.

The average gallons per minute per ton must take into account the variable water temperature. When well water is used as a source, and

Table 5. Equivalent Full Load Operating Hours of Refrigeration Equipment Used for Summer Cooling May 15 to Oct. 15[a]

Application	Hr Open for Business	Atlanta	Boston	Chicago	Detroit	Los Angeles	New Orleans	New York	Philadelphia	Oklahoma City	St. Louis	Washington D.C.
Barber Shops	1280	1010	650	720	720	680	1080	830	860	1020	890	940
Department Stores	940	840	560	610	610	580	890	700	720	840	750	780
Drug Stores	2100	1630	950	1060	1060	980	1790	1280	1330	1650	1420	1530
Funeral Parlors	600	440	300	330	330	310	470	370	380	440	400	410
Offices	940	870	560	620	620	580	900	710	740	880	770	810
Restaurant (Short Hour)	1290	970	535	620	620	570	1060	760	800	980	830	880
Restaurant (Long Hour)	2100	1510	820	930	930	850	1690	1170	1210	1530	1300	1400
Specialty Shops (5 & 10)	1090	800	530	590	590	560	860	670	690	810	720	750
Theaters—Continuous	1500	1010	700	750	750	720	1080	850	870	1020	910	950
Theaters—Neighborhood	900	640	420	450	450	430	650	500	520	650	550	580

[a] Modern Air Conditioning, Heating and Ventilating, by W. H. Carrier, R. E. Cherne and W. A. Grant (*Pitman Publishing Corp.* 1940, p. 73).

entering and leaving temperatures are considered constant, the average gallons per minute per ton obviously are equal to the design gallons per minute per ton. However, when the source is river or lake water, its maximum seasonal temperature will generally be reached at the same time that the refrigeration load factor is highest. The average gallons per minute per ton should be calculated from known or estimated water temperatures, because they vary through the season. Maximum water main temperatures are given in Chapter 34, but should always be verified locally. In lieu of this tedious work, the average gallon per minute per ton may be taken as 80 percent of design gallons per minute per ton with reasonable accuracy,

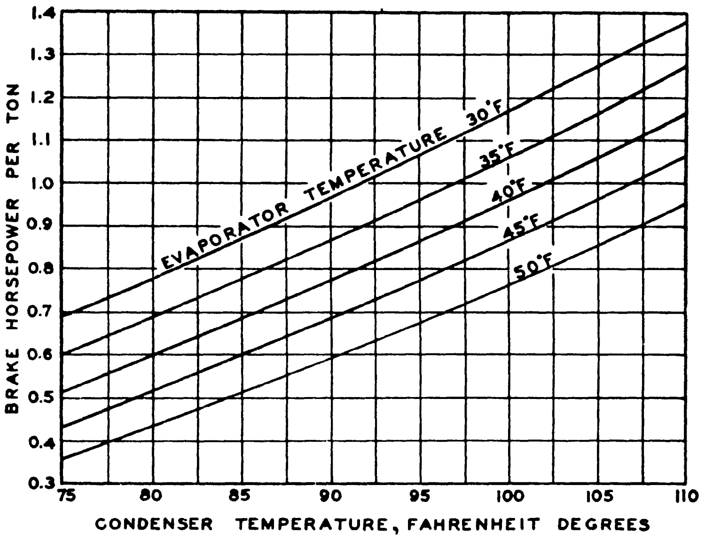

FIG. 1. TYPICAL BRAKE HORSE POWER REQUIREMENTS FOR REFRIGERATION[a]

[a] Values given are representative of "F-12" reciprocating machines of about 25 tons capacity in air conditioning applications. Requirements of smaller machines are usually higher, and for larger machines may be lower. Values shown are for liquid refrigerant at condenser temperature (no subcooling). Subcooling of the liquid may decrease these values approximately 0.3 percent to 0.5 percent for each Fahrenheit degree the liquid temperature is lowered.

for the condition of variable temperature of entering water obtained from rivers and lakes.

When cooling towers or evaporative condensers are used, the windage and evaporation losses are usually between 2 percent and 3 percent of the water circulated.

Heating

The method of estimating fuel consumption to balance the building heat loss is given in Chapter 17. It is important to include the fuel required to heat ventilation air as used in ventilating and air conditioning systems. In estimating fuel consumption for ventilation air, the tendency of the conventional control systems to use less than the estimated quantity of outside air in cold weather should be considered in its effect in lowering fuel consumption. In addition, the heat required to accomplish winter humidifying must not be neglected, when this feature is included in the equipment.

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CHAPTER 44

INDUSTRIAL AIR CONDITIONING

General Requirements for Manufacture, Processing and Preservation; Design Conditions and Application Data; Classification of Problems; Moisture Content and Regain; Conditioning and Drying; Chemical and Biochemical Reactions; Crystallization; Control for Machining, Polishing, and for Static Electricity Elimination; Laboratory Conditions; Calculations; Safeguarding Health and Maintaining Safety; Contaminant Control; Dilution Systems for Contaminants; Heat Storage; Radiation; Odors

INDUSTRIAL air conditioning is concerned with the design and application of equipment for obtaining proper conditions for (1) the manufacturing, processing, and preserving of material, equipment, and commodities; and (2) maintaining the health, safety and efficiency of workers. This chapter includes a general discussion of these conditions and also a comprehensive list of specific requirements for various types of products.

GENERAL REQUIREMENTS FOR MANUFACTURE, PROCESSING AND PRESERVATION

In order to apply air conditioning to industrial processes, the air conditioning engineer must have a thorough understanding of the processing problems involved. Individual processes and machines are changing rapidly, and air conditions must be revised constantly to meet the new conditions.

Table 1 lists the temperatures and relative humidities required for storage of certain commodities, and for manufacturing and processing of others. In some cases the temperatures and relative humidities listed in Table 1 have no direct influence upon the product itself, but do affect the efficiency of employees, and in turn affect workmanship, uniformity and the cost of production. Sometimes, a compromise between the known optimum condition for processing and that required for worker comfort is unavoidable.

Air conditioning for industrial processes is so extensive and involved that a detailed treatment is beyond the scope of this chapter. It is possible to cover only a few salient points of the general subject. In many industries the exact conditions to be maintained are determined and known only by the manufacturer who specifies them. In other industries there is a wide variance between manufacturers' requirements, depending on results desired, experience, and cost considerations.

CLASSIFICATION OF PROBLEMS

In general, any industrial air conditioning problem in processing may be classified under one or more of the following:

1. Control of regain.
2. Control of rate of chemical reactions.
3. Control of rate of biochemical reactions.
4. Control of rate of crystallization.
5. Control of temperature for close tolerance machining and grinding.

TABLE 1. TEMPERATURES AND HUMIDITIES APPLICABLE TO INDUSTRIAL AIR CONDITIONING*

PROCESS	TEMP. F	R.H. %
BAKERY		
Mixer (bread dough)	75–80	40–50
Fermenting	75–80	70–75
Proof box	92–96	80–85
Bread oven	375–450	
Cake oven	300–430	
Bread cooler (room or tunnel) Vacuum 28.6 in.	70–75	80–85
Cold room	40–45	
Make up room	75–80	65–70
Cake mixing	70–75	65
Cake mixing (sponge)	95–110	
Crackers and biscuits	60–65	50
Wrapping	60–65	60–65
Dried ingredients, storage	70	55–65
Fresh ingredients, storage	30–45	80–85
Flour storage	65–80	50–65
Shortening (depending on type), storage	45–70	55–65
Sugar, storage	80	35
Water, storage	32–35	
Wax paper, storage	70–80	40–50

METHODS OF MIXER COOLING

1. 35–40 F water circulated through mixer jacket.
2. 15–25 F brine.
3. Direct expansion, refrigerant circulated through mixer jacket.
4. Cracked ice added to dough in mixer.
5. Cold air introduced into mixer during mixing process.
6. Cooled agitators are used in mixers.

MIXER LOAD CALCULATIONS

Refrigeration is required to remove: excess ingredient, if any; heat generated by the beating and mixing of dough; excess heat in mixer body; heat of hydration of flour and water; and heat absorbed by mixer from atmosphere during mixing process.

Additional factors are the design and speed of mixer, consistency, kind and mass of dough.

Data Used in Calculations:

1 bbl. flour = 200 lb
Heat of hydration = 6.5 Btu per lb of flour
Specific heat of flour = 0.42 Btu per lb
Water is 65% of weight of flour
Flour is 65% of batch
Sponge is 60% of batch

Total motor output is converted to heat in the mixer.

In fermenting rooms recent practice is to use direct radiation for heating, atomizing sprays for humidifying, and gravity convection cooling surfaces for temperature reduction and dehumidifying thereby eliminating objectionable air currents.

Proof box and bread cooler conditions vary slightly for dark bread.

Cakes are sterilized by ultra-violet rays before wrapping.

PROCESS	TEMP. F	R.H. %
BANANAS		
Ripening	68	90 to 95
Storage	60	85 to 90
Shipping	54 to 56	85 to 90

Forced ripening is accomplished in 4 to 5 days while slow ripening is extended to 7 to 9 days with lower temperatures.

The green fruit with a pulp temperature of 56 F is placed in a ripening room at 68 F and 90 to 95 percent relative humidity until it begins to change color. The temperature is then lowered sharply to 60 F (or slightly less) dependent upon how quickly the fruit is to be moved.

Typical refrigeration load for ripening room per carlot, with an 85 F ambient temperature:

Insulation & electric load = 0.4 ton
Live load = 0.9 ton
Pull down load (from 68 F to 60 F) = 1.6 ton

Minimum carlot—20,000 lb which represents about 300 stems.

Since bananas give off minute quantities of ethylene and possibly other gases necessary for ripening, ventilation during the early stages of ripening is undesirable.

Cooling unit to be sized to provide a complete air change every 1½ to 2 min.

* Information in Table 1 is drawn from many sources. See bibliography at end of chapter.

TABLE 1. TEMPERATURES AND HUMIDITIES APPLICABLE TO INDUSTRIAL AIR CONDITIONING—(Continued)

BANANAS (Continued)

Provide a heating system with a rated capacity to warm the fruit at a rate of not less than 2 deg per hr.
Specific heat—0.9 Btu per (lb)(F deg).

Approximate rate of evolution of heat by bananas when stored at temperatures indicated:

Bananas at 54 F = 3,300 Btu (per ton) (24 hr).
green at 68 F = 8,360 Btu (per ton) (24 hr).
turning at 68 F = 9,240 Btu (per ton) (24 hr).
ripening at 68 F = 8,360 Btu (per ton) (24 hr).

ROOM	TEMP. F	R.H. %
BREWING		
Storage: Hops	30 to 32	55 to 62
Grain	80 max.	60 max.
Liquid yeast	32 to 34	
Lager	32 to 34	75 min.*
Ale	40 to 46	75 min.*
Fermenting Cellar. Lager	40 to 45	75 min.*
Ale	55	75 min.*
Racking Cellar	32 to 35	75
* If wooden tankage is used, otherwise humidity controlled to prevent condensation on walls and ceiling.		

Wort cooled to 47 F for lager, 54 to 59 F for ale, by evaporative cooling or use of double pipe or plate type coolers with counter flow of cooling medium.

Fermentation produces 250 Btu per lb of sugar fermented. Lager fermented five days at 55 to 60 F, ale at 68 to 75 F, then cooled to storage temperature.

Bottled beer pasteurized by heating to 140 F in twenty minutes, maintaining temperature for eighteen minutes, then cooling to 80 F in twenty minutes. Canned beer requires one-third less time.

Cold water, brine, direct expansion ammonia or propylene-glycol and water solutions may be used as the cooling medium.

PROCESS	TEMP. F	R.H. %
CANDY (CHOCOLATE)		
Candy centers for coating	80 to 85	40 to 50
Hand dipping room	60 to 65	50 to 55
Enrober room	75 to 80	55 to 60
Enrobing		
Loading End	80	50
Enrober	90	13
Stringing	70	40 to 50
Tunnel	40 to 45	
Packing	65	55
Panned specialty room	70 to 75	45
General candy storage	65 to 70	40 to 50
Tempering room = Ventilation only		

Tunnel discharge room requires a dew point lower than the temperature of the product leaving the tunnel.

During the summer months the product is usually held in the tempering room for 24 hours prior to shipping.

Recovery of sugar fly in coating kettle rooms is accomplished with the use of cyclone type dust collecting devices. Supply air to coating kettles is maintained at 85 F dry-bulb and 61 F wet-bulb temperature.

Bacteriological control is employed.

Load calculations for hand dipping rooms are based on 100 lb of 90 F chocolate per (worker) (hr).

Specific heat of chocolate = 0.30 to 0.56 Btu per (lb)(F deg).
Latent heat of fusion = 34 to 40 Btu per (lb)(F deg).
Freezing point =
Sweet milk chocolate = 86 F.
Dark chocolate = 90 to 92 F.

PROCESS	TEMP. F	R.H. %
CANDY (HARD)		
Manufacturing	75 to 80	30 to 40
Mixing and cooling	75 to 80	40 to 45
Tunnel	55	
Packing	65 to 75	40 to 45
Storage	65 to 75	45 to 50
Tempering = (Ventilation only)		

TABLE 1. TEMPERATURES AND HUMIDITIES APPLICABLE TO INDUSTRIAL AIR CONDITIONING—(Continued)

CANDY (HARD) (Continued)

Hot rooms used in the drying of jellies and gums are maintained at 120 to 150 F. A purge system using 100 percent outside air, bypassing the heating coil, is incorporated to produce rapid cooling of both the product and the room.

Cold rooms for cooling marshmallows and cast creams are maintained at 75 to 80 F with a relative humidity of 45 to 50 percent. Uniform air distribution is essential.

Standard starch drying equipment is employed.

Filtration of air is required.

Process	Temp. F	R.H. %
CHEWING GUM		
Manufacturing	77	33
Rolling	68	63
Striping	72	53
Breaking	74	47
Wrapping	74	58

Process	Temp. F	R.H.
CERAMICS		
Refractory	110 to 150	50 to 90
Molding room	80	60 to 70
Clay storage	60 to 80	35 to 65
Decalcomania production	75 to 80	48
Decorating room	75 to 80	48

Relative humidity has no effect on the manufacture of the products.

Temperature and humidity must be controlled in the decorating shop in the whiteware plants, and the decalcomania production room.

Dust control is essential, and the dust count must be held down to four million particles per cubic foot due to the danger of silicosis.

Process	Temp. F	R.H. %
CEREAL		
Packaging	75–80	45–50

Process	Temp. F	R.H. %
CITRUS FRUIT		
Lemons (storage)*	58	84 to 88
Grapefruit (storage)*	60	86 to 88
* Prior to transcontinental shipment		

Careful consideration must be given to air volumes, air temperatures, humidity, ventilation and distribution.

In regions where end rot is prevalent, it may be necessary to store grapefruit at temperatures of 32 to 34 F for a period not exceeding six weeks.

Lemon and grapefruit storage requires an air conditioning system to (1) maintain a constant temperature, (2) maintain a high relative humidity, (3) ventilate to maintain 0.1% CO_2 content, (4) obtain uniform air distribution, and (5) provide air washing or air filtration.

Load calculations include (1) transmission losses, (2) internal load-fruit cooling and respiration, cooling fruit boxes, electric load, and people, and (3) outside air load—72 cfm per carlot.

Specific heat—Btu per (lb) (F deg):

Lemons 0.94, Grapefruit 0.87, Boxes 0.40

HEAT OF EVOLUTION IN BTU PER (TON) (24 HR)

Temp.	Lemons	Grapefruit
32	580	460
40	810	1070
60	2070	2770
80	6200	4180

TABLE 1. TEMPERATURES AND HUMIDITIES APPLICABLE TO INDUSTRIAL AIR CONDITIONING—(Continued)

CITRUS FRUIT (Continued)

Approximately 10 percent of the cooling load is considered as latent heat load.

In a conventional system the air required is one cfm per storage box or 650 cfm per carload, resulting in a small temperature rise in the supply air making possible the required high humidities.

In a combination system the air required is one cfm per sq ft of floor area or 300 cfm per carlot with the addition of auxiliary humidifying nozzles to maintain the required high humidities.

Installation of metal ducts is required. Ductwork made of insulation board is sometimes preferred.

Process	Temp. F	R.H. %
DISTILLING		
Storage:		
Grain	60	35 to 40
Liquid Yeast	32 to 34	
General Manufacturing	60 to 75	45 to 60
Aging	65 to 72	50 to 60

Mashing done at 150 to 155 F, then cooled to 64 to 68 F.

Yeast culture fermented at 170 F, then cooled to 70 to 72 F.

Yeast propagated at 150 to 154 F.

Mash heated to 165 F and then cooled to 80 F, then pitched with yeast and fermented at maximum temperature of 85 F.

Cooling for various distilling processes normally accomplished by use of river or well water depending on temperatures and availability.

Low humidity and dust control important where grains are ground.

Viscous filters preferred as mold spores and bacteria are trapped in the viscous film, preventing propagation.

Process	Temp. F	R.H. %
ELECTRICAL PRODUCTS		
Electronics and X-Ray:		
Coil & transformer winding	72	15
Tube assembly	68	40
Electrical Instruments:		
Manufacture and Laboratory	70	50 to 55
Thermostat assembly & calibration	76	50 to 55
Humidistat assembly & calibration	76	50 to 55
Small Mechanisms:		
Close tolerance assembly	72	40 to 45
Meter assembly and test	74 to 76	60 to 63
Switchgear:		
Fuse and cutout assembly	73	50
Capacitor winding	73	50
Paper storage	73	50
Conductor wrapping with yarn	75	65 to 70
Lightning arrestor assembly	68	20 to 40
Thermal circuit breakers assembly and test	76	30 to 60
Water wheel generators:		
Thrust runner lapping	70	30 to 50
Rectifiers:		
Processing selenium and copper oxide plates	74	30 to 40
Dust control is essential in these processes		

Process	Temp. F	R.H. %
FLOOR COVERING		
Linoleum:		
Mechanical oxidizing of linseed oil	90 to 100	20 to 28
Printing	80	30 to 50
Stoving process	160 to 250	60 gr*
* grains per pound abs. hum.		

Some operations are stabilized against possibility of mold growth.

Precise control of temperature and humidity is required for mechanical oxidizing of linseed oil. The rate of flow and the temperature of cooling water in the jacket surrounding the tank must be controlled.

Air filtration is recommended for the stoving process.

Table 1. Temperatures and Humidities Applicable to Industrial Air Conditioning—(Continued)

Process	Temp.* F	
FOUNDRIES		
Core making	60 to 70	
Mold making:		
Bench work	60 to 70	
Floor work	55 to 65	
Pouring	40	
Shakeout	40 to 50	
Cleaning Room	55 to 65	
* Winter design temp.		

The charging room is usually unheated.

In core making fume exhaust hoods are required for oven and for cooling areas adjacent to ovens.

In mold making provide hoods at transfer points with wet collector dust removal system. Use 600 to 800 cfm per hood.

Pouring rooms require two-speed powered roof ventilators. Design for minimum of two cfm per sq ft floor area at low speed. Shielding is required to control radiation from hot surfaces. Proper introduction of air will minimize preheat requirements.

In shakeout room provide hoods with wet collector dust removal system. Exhaust 400 to 500 cfm per sq ft grate area. Roof ventilators are generally not effective.

In cleaning room provide hoods for grinders and cleaning equipment with dry cyclones or bag type collectors.

Winter ventilation (preheated) is required to the extent of replacing exhausted air. Summer ventilation s usually supplemented by use of pedestal fans.

Spot coolers are sometimes used in larger installations.

Process	Temp. F	R.H. %
FUR		
Drying	110	
Shock treatment	18 to 20	
Storage	40 to 50	55 to 65

Shock treatment, for eradication of any insect infestations, requires the lowering of the temperature to 18-20 F for 3 to 4 days, then raising it to 60-70 F for 2 days, then lowering it once again to 18-20 degrees for 2 days and raising it to the storage temperature.

Furs remain pliable, oxidation is reduced and color and luster are preserved when stored at 40 to 50 F.

Mold growth is prevalent with humidities above 80 percent, while hair splitting is common where humidity is lower than 55 percent.

Process	Temp. F	R.H. %
LEATHER		
Drying:		
Vegetable tanned	70	75
Chrome tanned	120	45
Storage	50-60	40-60

After leather is moistened in preparation for rolling and stretching, it is placed in an atmosphere held at room temperature with a relative humidity of 95 percent.

Leather is usually stored in warehouses without temperature and humidity control. However, it is necessary to keep humidity below 85 percent to avoid mildew.

Air filtration is recommended for toggle machines.

Process	Temp. F	R.H. %
LENSES (OPTICAL)		
Fusing	75	45
Grinding	80	80

The air must be free of dust and temperature held constant.

To acquire desired cleanliness of air a combination of impingement and electrostatic filters are used Dust collectors are required for grinding operations.

TABLE 1. TEMPERATURES AND HUMIDITIES APPLICABLE TO INDUSTRIAL AIR CONDITIONING—(Continued)

Rooms	Temp. F	R.H. %
LIBRARIES AND MUSEUMS		
Museum	70–80	40–50
Book storage	70–80	40–50

Spray type dehumidifiers used to eliminate SO_2. Water treatment is essential to maintain between 8.5 and 9.0 pH.

Reheat is usually needed for refrigeration cycle due to the low sensible heat load.

In extremely cold weather a lower relative humidity is required to prevent condensation on walls.

Positive high and low limit relative humidity controls are used. Do not locate water or steam piping where leakage can cause damage.

Check Figures for Cooling Estimates:

	Low	Medium	High
Square feet floor area per person	40	60	80
Watts per sq ft of floor area		1	2
Room sensible heat, Btu per (hr)(sq ft)	20	35	45
Grand total heat, Btu per (hr)(sq ft)	30	51	75
Sensible heat factor	0.73	0.83	0.90
Cooling load per person, tons	0.12	0.23	0.40
cfm per sq ft of floor area	0.92	1.60	2.10

MALTING (BREWING)

Steeped 24 to 72 hr in 45 to 65 F water.

Germinated six days at 55 to 75 F.

Kilned at temperatures of 120 to 175 F.

MALTING (DISTILLING)

Germinated twenty days at 63 F.

Germination produces total heat of 16,000 Btu/bushel at varying rate depending upon grain and process.

Process	Temp. F	R.H. %
MATCHES		
Manufacture	72 to 74	50
Drying	70 to 75	40
Storage	60 to 62	50

Water evaporated is 18 to 20 lb per million matches simultaneously with the setting of the glue.

The match machine will turn out about 750,000 matches per hour.

Process	Temp. F	R.H. %
MUSHROOMS		
Sweating-out period	120 to 140	
Spawn added	60 to 75	nearly sat.
Growing period	48 to 60	moderate
Storage	32 to 35	80 to 85

As spawn starts to grow, it is necessary to abruptly cool the mushroom house by 15 deg in a 12 hr period (approx.). Usually, this is the controlling factor in selection of refrigeration equipment, unless there is portable equipment available for such a purpose.

Ductwork is usually of wood, due to the deterioration of ferrous metals during sweating-out period.

Air filtration is essential in spawn rooms. Viscous filters are preferred, as mold spores are trapped in the viscous film. Odorless oil should be used.

Heat of emission is 4 Btu per (hr)(sq ft of growing surface).

Table 1. Temperatures and Humidities Applicable to Industrial Air Conditioning—(Continued)

Process	Temp. F	R.H. %
PAINT APPLICATION		
Lacquers:		
Air drying	70–90	60
Baking	180–300	
Oil Paints		
Air drying	60–90	60
Paint Spraying	75	

Spray booths to have 100 fpm face velocity. Make-up air must be preheated. Ovens must have air removed to keep fumes below explosive concentration. Equipment must be explosion-proof throughout

Room or Process	Temp. F	R.H. %
PHARMACEUTICAL		
Powder storage (prior to mfg.)	70 to 80	30 to 35
Manufactured powder storage and packing areas	75 to 80	15 to 35
Milling room	80	35
Tablet compressing	70 to 80	40
Tablet coating room	80	35
Effervescent tablets and powders	90	15
Hypodermic tablets	75 to 80	30
Colloids	70	30 to 50
Cough syrups	80	40
Glandular products	78 to 80	5 to 10
Ampule manufacturing	80	35
Gelatin capsules	78	40 to 50
Capsule storage	75	35 to 40
Micro-analysis	80	50
Biological manufacturing	80	35
Liver extracts	70 to 80	20 to 30
Serums	74 to 78	50
Animal rooms	75 to 80	40
Small animal rooms	75 to 78	47 to 48

Gelatin capsules require varying relative humidities, depending upon size of capsule. Moisture content should not exceed 0.25 gr per cu ft. Various kinds of gelatin require different temperatures.

Penicillin incubation process requires holding temperature within ½ deg F, with temperatures and humidity rigidly controlled during all manufacturing phases.

Ampule filling requires a 20 percent relative humidity when especially fine powders are used.

Uncoated tablet manufacturing requires accurate control of temperature and relative humidity, since low relative humidity causes formation of a hard outer layer, and high relative humidity retards drying at the proper rate.

Liver extracts require a low relative humidity after they are dried. Temperatures higher than 80 F will cause the extracts to deteriorate.

Tablet coating requires the control of the temperatures of all ingredients and the temperature of air introduced to coating pans.

Sterile conditions are essential in many pharmaceutical processes. Provide suitable air exhaust to remove surplus material from tablet compressing machine.

Air filtration is generally required, with positive air filtration in some areas.

Animal Rooms in Pharmaceutical Laboratories

In the following tabulation each of the items mentioned is equivalent in metabolism to one man:

Quantity	Animal	Weight Each
672	White Mice	21 g
110	Rats	200 g
73	Rats	400 g
70	Guinea Pigs	410 g
21	Rabbits	2.6 kg
16	Cats	3.0 kg
16	Monkeys	3.0 kg
5	Dogs	14.0 kg

Dogs are generally the worst offenders as far as odor generation is concerned.

Decontamination of exhaust air is recommended in populated areas.

For good air quality conditions the following space per animal and total air circulation (all outdoor or decontaminated in recirculation) should be provided:

Animal	cu ft	cfm
Mice	3	0.5
Rats	4	0.75
Guinea Pigs	6	1.0
Rabbits	10	1.9
Hamsters	8	1.5
Cats	35	10.0
Dogs	150	28.0

TABLE 1. TEMPERATURES AND HUMIDITIES APPLICABLE TO INDUSTRIAL AIR CONDITIONING—(Continued)

PROCESS	TEMP. F	R.H. %
PHOTO STUDIO		
Dressing Room	74	30–40
Drying Room	75–80	50
Printing Room	70	70
Setting Room	72	65
Developing Room	70–75	60
Storage Room	60	45

Heat liberated during printing, enlarging and drying processes is removed through an independent exhaust system, which also serves the lamp houses and dryer hoods.

Dust control is essential, and absolute filtration is required in some areas.

PROCESS	TEMP. F	R.H. %
PHOTO MATERIAL		
Drying	20–125	40–80
Cutting & Packing	65–75	40–60
Film Base & Paper Storage	70–75	40–65
Coated Paper & Film Storage	70–75	40–65
Safety Base Film Storage	60–80	45–50
Nitrate Film Storage	40–50	40–50

Spray water must have algae inhibitor. Positive dust control must be maintained and absolute filtration is essential.

In nitrate film area take proper precautions against fires.

Recirculated air used for film drying is passed through activated carbon filters.

Relative humidity for film storage should never exceed 60 percent with a minimum of 25 percent.

PROCESS	TEMP. F	R.H. %
PLASTICS		
Manufacturing areas:		
Thermo setting moulding compounds	80	25 to 30
Cellophane wrapping	75 to 80	45 to 65

Absolute filtration is required in some areas.

Collection and removal of dust and fumes is essential.

PROCESS	TEMP. F	R.H. %
PLYWOOD		
Hot Pressing (resin)	90	60
Cold Pressing	90	15 to 25
POULTRY RAISING		
Incubator	70–75	30–60
Brooder:		
1st week	70–75	60
2nd week	50–60	60
Battery room:		
Starting	70	70–75
Growing	50–60	50–60
Egg storage	45–60	70–80

Maximum ventilation is required for laying quarters during the summer months, while an attempt is made to maintain a temperature 10 deg above or below outdoor temperature during the winter, to minimize condensation on the exterior walls. This applies to houses not having forced ventilation systems.

PROCESS	TEMP. F	R.H. %
PRECISION MACHINING		
Spectrographic analysis	75–80	45–50
Gear matching & special assembly room	75–80	35–40
Gasket storage	100	50
Cement & glue storage	65	40
Precision parts machining	75	45–50

TABLE 1. TEMPERATURES AND HUMIDITIES APPLICABLE TO INDUSTRIAL AIR CONDITIONING—(Continued)

PROCESS	TEMP. F	R.H. %
PRECISION MACHINING (Continued)		
Accurate gaging & inspection	75	45–50
Precision gage manufacture & adjustment	68–75	45–50
Precision gear drive assembly	72	42–50
Watch main spring calibration	76	45
Gage room	78	50
Precision parts—honing	75–80	35–45

For general manufacturing and assembly areas no attempt is made to control conditions during hot weather. An ample supply of outside air and air motion are relied upon to provide personal comfort.

Low relative humidity is usually maintained to prevent corrosion.

Temperature control within a narrow range is more important than temperature maintained.

Dust control is required wherever polishing operations occur.

Air distribution is important to maintain constant conditions throughout the area.

PROCESS	TEMP. F	R.H. %
PRINTING		
Pressroom:		
Multicolor offset lithography	75–80	46–48
Other sheet fed printing	75–80	45–50
Newspaper & other web printing	75–80	50–55
Stock room:		
Multicolor offset lithography	73–80	[a] 5–8% above pressroom
Other paper storage	70–80	b
Binding, cutting, drying, folding and gluing	70–80	45–50
Roll storage	73–80	50
[a] 5 to 8% above pressroom [b] same as pressroom		

Lithography requires constant humidity control of entire pressroom with paper conditioned 5 to 8 percent higher relative humidity at start. All printing requires conditioned paper otherwise it will not lie flat, with 40 to 45 percent R.H. low limit to eliminate static electricity; and 60 percent R.H. high limit to prevent swelling of the rolls and slow ink drying. Temperature is not critical, but extremes should be avoided due to softening of the rolls at high temperature and improper ink distribution at low temperatures.

Hoods must be provided for gas dryers, with solvent recovery recommended for all except job shops. Exhaust system with dust collectors incorporated is required for type and plate cleaning areas. Check use of gasoline and other solvents.

Summer—winter central systems are recommended for all except job shops where unitary equipment with heating coils and humidifiers may be used. Normal air cleaning is adequate. Air distribution must prevent drafts on paper in storage or process. Gas flame dryers impose unusual loads.

Air from press and storage rooms should not be recirculated through office areas.

PROCESS	TEMP. F	R.H. %
REFRIGERATION EQUIPMENT		
Valve Manufacturing	75	40
Compressor Assembly	70 to 76	30 to 45
Refrigerator Assembly	75	30 to 50
Testing	65 to 82	47

PROCESS	TEMP. F	R.H. %
RUBBER DIPPED GOODS		
Manufacture	90	.
Cementing	80	25–30
Dipping surgical articles	75–90	25–30
Storage prior to manufacture	60–75	40–50
Laboratory (*ASTM* Standard)	73.4	50

Solvents used in manufacturing processes are usually explosive and toxic, requiring positive ventilation.

Volume manufacturers usually install a solvent recovery system.

TABLE 1. TEMPERATURES AND HUMIDITIES APPLICABLE TO INDUSTRIAL AIR CONDITIONING—(Continued)

PROCESS	TEMP. F	R.H. %
TEXTILES		
Cotton: Opening	70–75	55–60
Picking	75	55–60
Carding, Winter	75–80	55
Carding, Summer	83	90
Carding, Summer or	85	80
Carding, Summer or	87	70
Drawing	80	60
Roving	80	60
Ring Spinning		
Conventional	80–85	70
Long Draft	80–85	55
Frame Spinning	80–85	55–60
Spooling & Warping	78–80	65
Weaving	78–80	70–85
Cloth Room	75	65–70
Combing	75	50–65
Linen: Carding	75–80	60
Spinning	75–80	60
Weaving	80	80
Woolens: Pickers	80–85	60
Carding	80–85	65–70
Spinning	80–85	50–60
Dressing	75–80	60
Weaving		
Light goods	80–85	60
Overcoating (32 oz.)	80–85	60–65
Drawing	75	50–60
Worsteds: Carding	80–85	65
Combing	80–85	65–70
Gilling	80–85	65–70
Top Storage	70–85	75–80
Drawing	80–85	65
Cap Spinning	80–85	50–55
Spooling & Winding	75	65–70
Weaving	80	55–70
Finishing	75	60
Silk: Preparatory	80	60–65
Weaving	80	60–70
Dressing	80	60–65
Spinning	80	65–70
Throwing	80	60
Rayon: Spinning	80–90	50–60
Throwing	80	55–60
Weaving		
Regenerated	80	50–65
Acetate	80	55–60
Spun rayon	80	80
Picking	75–80	50–60
Carding, roving, drawing	80–90	50–60
Knitting		
Viscose or cuprammonium	80–85	65
Acetate	80–85	60
Laboratory (*ASTM*)	70	65
Rayon synthetic fiber processing:		
Viscose		
Preparatory	80	60
Weaving	80	60
Celanese		
Preparatory	80	70
Weaving	80	70–75
Nylon		
Preparatory	80	50–60
Weaving	80	50–60

Cotton: Relative humidity maintained in ring spinning depends on staple, twist and whether leather aprons are used for conveying long draft stock. Aprons readily absorb moisture causing cotton to stick when relative humidity is above 55 percent.

With conventional 3 or 4 roll spinning, relative humidity may be as high as 70 percent dependent upon draft, twist and staple.

Relative humidity carried in cotton weaving depends upon construction of the cloth. When automatic machines are to be tended and the warps are heavily sized, it may be as high as 90 percent.

Woolen: In woolen spinning, the relative humidity maintained for mules is generally 55 percent, with conditions over frame spinning at 55 to 60 percent. Both types of spinning depend on the class of stock spun, also the regain in the roving.

TABLE 1. TEMPERATURES AND HUMIDITIES APPLICABLE TO INDUSTRIAL AIR CONDITIONING—(Concluded)

TEXTILES (Continued)

Worsteds: Top storage temperature depends on whether cellar long period conditioning at low temperature or quick conditioning at high temperature is used. Weaving relative humidity depends upon staple, quality and construction.

Filtration of air is essential.

Rayon manufacture: The steeping room, where sheets of raw material are dipped in caustic soda then broken into a fine matted crumb, is held at 70 F and 55 percent relative humidity. Relative humidity is held down to prevent condensation on cold pipes and jackets.

In churn room, where sodium cellulose is converted into cellulose xanthate, temperatures of 75 to 80 F are maintained while humidity control is not important. Room temperatures are held below 85 F during the summer months.

The crumb is dumped into aging tanks located in a room held at 73 F with no humidity control.

For spinning operations it is desirable to limit the temperature to 90 F with a minimum relative humidity of 70 percent for the summer, while 70 percent and 75 F are desirable for the winter months.

The storage room, where the material is held for later processing design conditions, are 85 F and 100 percent relative humidity.

The material is washed, desulfurized, bleached and then washed again. It is then placed in a drier where controlled conditions of 100 deg and 65 percent relative humidity are required to bring the rayon back to the proper regain.

In the coning room, where winding machines place the rayon yarn on cones, temperature is held at a maximum of 80 F. Relative humidity is held at 55 percent.

PROCESS	TEMP. F	R.H. %
TOBACCO		
Cigar and cigarette making	70 to 75	55 to 65
Softening	90	85 to 88
Stemming and stripping	75 to 85	70 to 75
Packing and shipping	74 to 76	65
Filler tobacco casing and conditioning	75	75
Filler tobacco storage and preparation	78	70
Wrapper tobacco storage and conditioning	75	75

In preparation for stripping, the tobacco undergoes a softening operation, whereby it is automatically heated, moistened and then cooled.

Control of moisture regain, and of chemical and biological reactions is required in tobacco processing.

Exact temperature and humidity conditions maintained are usually trade secrets as they affect the finished product.

6. Control of dew-point for protection of highly polished surfaces.
7. Control of humidity for static electricity elimination.
8. Control of conditions for material test laboratories.

Moisture Content and Regain

In the manufacture or processing of hygroscopic materials such as textiles, paper, wood, leather, tobacco and foodstuffs, the temperature and relative humidity of the air have a marked influence upon the rate of production and upon the weight, strength, appearance, and general quality of the product. The moisture content of materials having a vegetable or animal origin, and to a lesser extent minerals in certain forms, comes to equilibrium with the moisture of the surrounding air. This moisture content is known as *regain*. Standards of *regain* are fixed in the trade, and are the fundamental basis for the control of certain physical qualities of the material during manufacture.

Manufacturing economy requires that the moisture content be maintained at a level favorable to rapid and satisfactory manipulation, and to a minimum loss of material through breakage. A uniform condition is desirable in order that high speed machinery may be adjusted permanently for the desired production with a minimum loss from delays, wastage of raw material, and defective product. *Moisture content* refers to free moisture (as in a sponge) and to hygroscopic moisture (which varies with atmospheric conditions). It is usually expressed as a percentage of the total weight of material. *Regain* is more specific and refers only to hygroscopic moisture. It is expressed as a percentage of the *bone-dry* weight of material. For example, if a sample of cloth weighing 100.0 g is dried to a bone-dry weight of 93.0 g, the loss in weight, or 7.0 g, represents the weight of moisture originally contained. This expressed as a percentage of the total weight (100.0 g) gives the moisture content of 7 percent. The regain, which is expressed as a percentage of the bone-dry weight, is $(7.0/93.0) \times 100 = 7.5$ percent.

The use of the term *regain* does not imply that the material as a whole has been completely dried out and has re-absorbed moisture.

A basis for calculating the regain of textiles is obtained by drying, under standard conditions, a sample from the lot, and the dry weight thus obtained is used in the calculations to determine the regain.

Table 2 shows the regain or hygroscopic moisture content of several organic and inorganic materials when in equilibrium at a dry-bulb temperature of 75 F and various relative humidities. The effect of temperature as compared to the relative humidity is comparatively unimportant, although sudden changes in temperature cause a slight change in regain even when the relative humidity remains stationary. Changes in temperature do, however, affect the rate of absorption or drying, although this property generally varies with the nature of the material, its thickness and density.

When hygroscopic materials absorb moisture from the surrounding air, they deliver to the air sensible heat equivalent to the latent heat released by the moisture to the material. This amount of heat should be included in the load estimate.

Conditioning and Drying

In general, the materials may be exposed to desirable humidities for treatment coincidentally with the manufacture or processing of the materials, or they may be treated separately in special enclosures. This latter treatment may be classified as conditioning or drying. The usual purpose of conditioning or drying is to establish a desired condition of moisture content and to regulate the physical properties of the material. When the final moisture content is lower than the initial one, the term *drying* is applied (See Chapter 46). If the final moisture content is to be higher, the process is termed *conditioning.* In the case of some textile products and tobacco, for example, drying and conditioning may be combined in one process for the dual purpose of removing undesirable moisture, and accurately regulating the final moisture content. Frequently, conditioning or drying is made a continuous process in which the material is conveyed through an elongated compartment by suitable means, and subjected to various controlled atmospheric conditions.

Control of Rate of Chemical Reactions

A typical example of control of the rate of chemical reactions occurs in the manufacture of rayon. The pulp sheets are conditioned, cut to size, and passed through a mercerizing process. It is essential that, during this process, close control of both temperature and relative humidity should be maintained. The temperature controls the rate of reaction directly, while the relative humidity maintains a constant rate of evaporation from the

TABLE 2. REGAIN OF HYGROSCOPIC MATERIALS

Moisture Content Expressed in Percent of Dry Weight of the Substance at Various Relative Humidities—Temperature, 75 F

Classification	Material	Description	Relative Humidity—Per Cent									Authority
			10	20	30	40	50	60	70	80	90	
Natural Textile Fibers	Cotton	Sea island—roving	2.5	3.7	4.6	5.5	6.6	7.9	9.5	11.5	14.1	Hartshorne
	Cotton	American—cloth	2.6	3.7	4.4	5.2	5.9	6.8	8.1	10.0	14.3	Schloesing
	Cotton	Absorbent	4.8	9.0	12.5	15.7	18.5	20.8	22.8	24.3	25.8	Fuwa
	Wool	Australian merino—skein	4.7	7.0	8.9	10.8	12.8	14.9	17.2	19.9	23.4	Hartshorne
	Silk	Raw chevennes—skein	3.2	5.5	6.9	8.0	8.9	10.2	11.9	14.3	18.8	Schloesing
	Linen	Table cloth	1.9	2.9	3.6	4.3	5.1	6.1	7.0	8.4	10.2	Atkinson
	Linen	Dry spun—yarn	3.6	5.4	6.5	7.3	8.1	8.9	9.8	11.2	13.8	Sommer
	Jute	Average of several grades	3.1	5.2	6.9	8.5	10.2	12.2	14.4	17.1	20.2	Storch
	Hemp	Manila and sisal—rope	2.7	4.7	6.0	7.2	8.5	9.9	11.6	13.6	15.7	Fuwa
Rayons	Viscose Nitrocellulose Cupramonium	Average skein	4.0	5.7	6.8	7.9	9.2	10.8	12.4	14.2	16.0	Robertson
	Cellulose Acetate	Fiber	0.8	1.1	1.4	1.9	2.4	3.0	3.6	4.3	5.3	Robertson
Paper	M. F. Newsprint	Wood pulp—24% ash	2.1	3.2	4.0	4.7	5.3	6.1	7.2	8.7	10.6	U. S. B. of S.
	H. M. F. Writing	Wood pulp—3% ash	3.0	4.2	5.2	6.2	7.2	8.3	9.9	11.9	14.2	U. S. B. of S.
	White Bond	Rag—1% ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2	U. S. B. of S.
	Com. Ledger	75% rag—1% ash	3.2	4.2	5.0	5.6	6.2	6.9	8.1	10.3	13.9	U. S. B. of S.
	Kraft Wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9	U. S. B. of S.
Misc. Organic Materials	Leather	Sole oak—tanned	5.0	8.5	11.2	13.6	16.0	18.3	20.6	24.0	29.2	Phelps
	Catgut	Racquet strings	4.6	7.2	8.6	10.2	12.0	14.3	17.3	19.8	21.7	Fuwa
	Glue	Hide	3.4	4.8	5.8	6.6	7.6	9.0	10.7	11.8	12.5	Fuwa
	Rubber	Solid tires	0.11	0.21	0.32	0.44	0.54	0.66	0.76	0.88	0.99	Fuwa
	Wood	Timber (average)	3.0	4.4	5.9	7.6	9.3	11.3	14.0	17.5	22.0	Forest P. Lab.
	Soap	White	1.9	3.8	5.7	7.6	10.0	12.9	16.1	19.8	23.8	Fuwa
	Tobacco	Cigarette	5.4	8.6	11.0	13.3	16.0	19.5	25.0	33.5	50.0	Ford
Foodstuffs	White Bread		0.5	1.7	3.1	4.5	6.2	8.5	11.1	14.5	19.0	Atkinson
	Crackers		2.1	2.8	3.3	3.9	5.0	6.5	8.3	10.9	14.9	Atkinson
	Macaroni		5.1	7.4	8.8	10.2	11.7	13.7	16.2	19.0	22.1	Atkinson
	Flour		2.6	4.1	5.3	6.5	8.0	9.9	12.4	15.4	19.1	Bailey
	Starch		2.2	3.8	5.2	6.4	7.4	8.3	9.2	10.6	12.7	Atkinson
	Gelatin		0.7	1.6	2.8	3.8	4.9	6.1	7.6	9.3	11.4	Atkinson
Misc. Inorganic Materials	Asbestos Fiber	Finely divided	0.16	0.24	0.26	0.32	0.41	0.51	0.62	0.73	0.84	Fuwa
	Silica Gel		5.7	9.8	12.7	15.2	17.2	18.8	20.2	21.5	22.6	Fuwa
	Domestic Coke		0.20	0.40	0.61	0.81	1.03	1.24	1.46	1.67	1.89	Selvig
	Activated Charcoal	Steam activated	7.1	14.3	22.8	26.2	28.3	29.2	30.0	31.1	32.7	Fuwa
	Sulfuric Acid	H_2SO_4	33.0	41.0	47.5	52.5	57.0	61.5	67.0	73.5	82.5	Mason

surface of the solution, and maintains a solution of known strength throughout the mercerizing period.

Another well-known example in this class is the *drying* of varnish which is an oxidizing process dependent upon temperature. High relative humidities have a retarding effect on the rate of oxidization at the surface, and allow the internal gases to escape freely as the chemical oxidizers *cure* the varnish from within. This produces a surface free from bubbles and a homogeneous film throughout. Desirable temperatures for *drying* varnish vary with the type. A relative humidity of 65 percent is beneficial for obtaining the best processing results.

Control of Rate of Biochemical Reactions

In the field of biochemical control, industrial air conditioning has been applied to many different and well-known products. All problems involving fermentation are classed under this heading. As biochemistry is a subdivision of chemistry, subject to the same laws, the rate of reaction may be controlled by temperature. An example of this is the dough room of the modern bakery. Yeast develops best at a temperature of 80 F. A relative humidity of 70 percent is maintained to hold the surface of the dough open to allow the carbon dioxide gases formed by the fermentation to pass through and produce a loaf of bread, when baked, of even, fine texture without large voids.

The curing of fruits, such as bananas and lemons, also comes under this classification. Bananas require a cycle of temperatures and relative humidities for ripening. The starches in the pulp of the fruit must be changed and the skin cured and colored, after which the fruit is cooled to maintain as low a rate of metabolism as possible. Ideal storage conditions range between 56 and 60 F, with about 85 percent relative humidity, and ventilation at the rate of three or four air changes per hour.

The curing of lemons is an entirely different problem. Bananas are cured for a quick market, while lemons are held for a future market. The process, therefore, varies in the temperature used. Temperatures from 54 to 59 F have been found to be best suited for this process. A high relative humidity of 84 to 88 percent is necessary to hold shrinkage to a minimum and, at the same time, develop the rind so it will be sufficiently tough to permit handling.

Tobacco from the field to the finished cigar, cigarette, plug or pipe tobacco, offers another interesting example of what may be done by industrial air conditioning in the control of color, texture and flavor. In the processing of tobacco, control of moisture regain, and of chemical and biochemical reactions, is involved, and only through close atmospheric control can the best quality of leaf be developed.

Control of Rate of Crystallization

The rate of cooling of a saturated solution determines the size of the crystals formed. Both dry- and wet-bulb temperatures are of importance, as the one controls the rate of cooling, while the other, through evaporation, changes the density of the solution.

In the coating pans for pills, gum, and nuts, a heavy sugar solution is added to the tumbling mass. As the water evaporates, each separate piece is covered with crystals of sugar. A smooth, opaque coating is only accomplished by blowing into the kettle the proper amount of air at the right dry- and wet-bulb temperatures. If the cooling and drying are too slow, the coating will be rough and semi-translucent, and the appearance unsightly; if too fast, the coating will chip through to the interior. Only by balancing temperature, relative humidity, and volume of air to the sugar solution, can the proper rate be obtained and a perfect coating assured.

Control of Temperature for Close Machining Tolerances

Where tolerances must be held within 2 or 3 ten-thousands of an inch, as in the manufacture of precision instruments, tools, and high quality lenses, temperature variations may cause expansion and contraction of material to an extent that will seriously affect the quality of the work. This type of work usually requires close temperature control to assure accuracy and uniformity of the product.

Usually the temperature level with respect to the product is not as important as controlling the temperature within close limits. For this reason, conditions are usually selected within the comfort range.

Control of Dew-Point for Protection of Polished Surfaces

In the manufacture of certain metal articles, the presence of finger prints, tarnish, or *etching* can not be tolerated in the finished article. If these articles are manufactured under conditions of effective temperatures that will cause the hands to perspire, an unsatisfactory product will result. The salt and acid contained in body perspiration, when deposited on the highly polished article, can show corrosion and rust within a few hours if examined under a microscope.

It is therefore important to maintain temperatures and relative humidities (dew-point) low enough to prevent sweating of the hands. In addition, the manufacture of polished surfaces usually requires a better-than-average job of air filtering to avoid abrasion of the surfaces.

Control of Humidity for Reduction of Static Electricity

The presence of static electricity is often detrimental to the satisfactory and economical processing of many light materials, such as textile fibers, paper, etc. It is also extremely dangerous where explosive atmospheres or materials are present. Fortunately, this hazard is minimized by increasing the relative humidity to at least 55 percent, if the material being processed is not damaged thereby.

It must be borne in mind that for successful elimination, the air that actually comes in contact with the material in the machine must be at a relative humidity of 55 percent or more. As some machines consume a great deal of power, which is converted directly into heat, the temperature in the machine may be considerably higher than the temperature adjacent to the machine where the relative humidity is normally measured. In such cases, the relative humidity in the machine will be appreciably lower than that elsewhere in the room, and it may therefore be necessary to maintain a room relative humidity of 65 percent, or even more, to maintain the desired humidity.

Control of Conditions for Material Test Laboratories

Laboratories having controlled conditions of temperature and humidity, are becoming more common, not only for the purposes of scientific research, but also for routine testing and for quality production control. A control of temperature and humidity within fairly close plus or minus limits is usually required. Laboratories designed for scientific research may require control of conditions over a wide range, whereas the routine testing laboratory or quality control laboratory will usually be designed to maintain the *A.S.T.M.* Standard Conditions of 50 percent relative humidity and 23 C (73.4 F) temperature.

CALCULATIONS

The methods for determining the heating and cooling loads for the various industrial processes are similar to those outlined in Chapters 11 and 12. Some factors affecting heating or cooling requirements are given in Table 1. Because of the large number of motors and heat producing units usually found in an industrial application, it is particularly important that operating allowances, for the latent and sensible heat loads, be definitely ascertained and used in the calculations to determine the total design load.

GENERAL REQUIREMENTS FOR HEALTH, SAFETY AND EFFICIENCY

Control of Atmospheric Contaminants

Safeguarding the health and maintaining the safety and efficiency of workers, require control of dusts, fumes, smokes, mists, fogs, vapors, and gases, and control of the effective temperature, which includes temperature, humidity and motion of air about the worker.

General ventilation may be relied upon in some cases to control air contaminants. Chapter 10 gives information on natural ventilation. If mechanical ventilation is to be used, Chapter 32, Fans; Chapter 30, Air Distribution; Chapter 31, Air Duct Design; Chapter 33, Air Cleaning; and Chapter 34, Spray Apparatus, furnish information on a broad range of industrial design conditions.

Specialists in the field of industrial hygiene should be consulted in case of doubt concerning the presence of airborne industrial hazards to health. Chapter 8, Air Contaminants; Chapter 6, Physiological Principles; and Chapter 7, Air Conditioning in the Prevention and Treatment of Disease, will be of help in determining the atmospheric conditions which should be maintained around the worker. Local codes, ordinances, or state labor laws must likewise be observed, particularly for ventilation requirements for hazardous trades. Comfortable conditions are desirable because they are likely to increase the efficiency of workers. For purposes of analysis, both sensible and latent heat should be included as contaminants of industrial atmospheres. A recent small scale survey has indicated that more than half of the air conditioning and ventilation installations in a typical industrial plant were concerned with removal of either sensible or latent heat as a source of air contamination.

Contaminant Control Systems

In general, systems for control of atmospheric contamination in industrial plants will be of three types:

1. *Local exhaust systems* will be indicated where the contamination originates at concentrated areas and is characterized by low or imperceptible air motion, or where the contaminant is a dust, mist or fume requiring a capture velocity exceeding 25 fpm. Design of this type system is discussed in Chapter 45, and will not be further treated in this chapter.
2. A system employing the *dilution* method will usually be indicated where the contamination originates at scattered points dispersed generally throughout the area.
3. Combination of local *exhaust* and *dilution* methods is often economical, since well designed exhaust hoods or openings, removing from the space that portion of the contamination load which is susceptible to such treatment, will often reduce greatly the air volumes required for dilution purposes. The choice of the type of system should be made on the basis of economic comparisons.

Design of Dilution Systems

The first step in the design of a system employing the dilution method is to determine as exactly as possible the nature and extent of the contaminating load. This will often be difficult, and may require construction of pilot production models. Often, however, the required data will be available from production records, showing the weight or volume rate of loss of the contaminating agent to the atmosphere, or it may be estimated from parallel operations in other plants, or by applying experienced engineering judgment. However obtained, the determination of the nature and magnitude of the contaminating load is an indispensable step in the proper design of the corrective system. Designs based on number of air changes per hour, or other rule-of-thumb methods, are hopelessly inadequate, and lead either to unsuccessful operation or to excessive and unnecessarily high cost of installation.

1. *Gases and Vapors.* Once having established the nature and magnitude of the contamination load, it is rarely necessary to completely remove contaminating agents from the atmosphere. For cases involving diffusible vapor or gas contaminants, maximum allowable concentrations (MAC) of commonly encountered gases and vapors have been established, and these data are tabulated in Chapter 8. From these data, and the previously established rate of addition of the contaminant to the space, the volume of air required to dilute the addition to a tolerable level can be calculated by the equation:

$$Q = \frac{V \times 10^6}{(\mathrm{MAC}) - (\mathrm{SAC})} \tag{1}$$

where

Q = quantity of air circulated, cubic feet per minute.
V = rate of generation of contaminant, cubic feet per minute.
MAC = maximum allowable concentration, ppm by volume.
SAC = concentration in supply air, ppm by volume.

The rate of generation of the contaminating vapor will often be available as a *weight* or *volume* of liquid evaporated into the space per unit time. These may be converted to the units of Equation 1 by applying the principle that a pound-mol of a gas or vapor will occupy approximately 359 cu ft at standard pressure and temperature. Thus,

$$\text{cfm (vapor)} = \frac{W}{M_w} \times 359 \times \frac{t + 460}{492} \tag{2}$$

where

W = rate of generation of contaminant, pounds of liquid solvent per minute.
M_w = molecular weight.
t = air temperature, Fahrenheit.

A special case occurs where local concentrations of solvent vapors *at the breathing zone,* resulting from concentrated sources of contamination, are intolerably higher than the average design concentration when using dilution methods. Data are available for calculations, but involve many assumptions regarding boundary conditions, such as convection area and random air movement in the vicinity.

2. *Dusts and Fumes.* Maximum allowable concentration of various dusts, fumes and mists are also tabulated in Chapter 8. However, the dilution method as a means of treating particulate contaminating agents should be used with care, since the allowable air movement in spaces will ordinarily be lower than the capture velocity required for such particles. Exhausting at the source (see Chapter 45) will generally be the recommended treatment for these particulate contaminants.

3. *Sensible Heat.* Excessive sensible heat contamination is subject to treatment, similar to that for vapors, by the dilution method, the difference being that the rate of generation of the contaminant must be expressed in units of energy rather than volume or weight, and that the effect will be expressed as excessive temperature. In this case, the circulated air required will be:

$$Q = \frac{H}{(t_l - t_o) \times d \times c} \tag{3}$$

where

Q = quantity of air circulated, cubic feet per minute.
H = rate of generation of heat, Btu per minute.
t_l = allowable temperature in the space, Fahrenheit.
t_o = temperature of supply air, Fahrenheit.
d = density of air in pounds per cubic foot.
c = specific heat of air.

In some cases, such as ventilation systems not employing refrigeration, unusually large and uneconomical air quantities may be required when the desired or tolerable temperature t_l approaches too closely the temperature t_o of the dilution air. This condition may sometimes be corrected by a combination of treatment by dilution and central exhaust, large sources of heat load being eliminated by exhaust through hoods at the source.

Heat Storage in Structure

A special condition is sometimes encountered in large masonry structures when, due to the heat storage capacity and time lag of the structure, a prolonged period of hot weather may cause storage of such large quantities of heat in the structure that they continue to be a source of heat load after the outside weather has moderated.

In general, the solution to ventilation problems by the simple *dilution method* is limited to cases where a practical and economical equilibrium may be established between the maximum rate of generation of the contaminant distribution and cost of the air required for its removal.

Control of Radiant Heat

One of the most difficult problems of ventilation engineers is found in the so-called *hot industries* (steel mills, paper mills, foundries, etc.) where radiant heat from high temperature surfaces is a most important factor, the magnitude of which is not always appreciated. Since ventilation cannot remove radiant heat, it is worthwhile to consider using low emissivity materials and shielding to minimize conversion to sensible heat due to interaction with surfaces within sight of the radiant source.

Odors

There is little information available either on the rate of generation of common odors, or on the maximum concentration which would be tolerated by a majority of persons. Thus a quantitative specification covering systems for locker rooms and toilet rooms, or other applications where odors constitute the contaminant, is not now possible. Pending research in this important field, it is suggested that the data in Table 1, Chapter 6, showing the outside air supply required per person, at various socio-economic levels, be used.

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CHAPTER 45

INDUSTRIAL EXHAUST SYSTEMS

Classification of Systems, Hood Design Principles, Capture Velocity and Hood Suction, Duct System Design, Resistance of System, Efficiency of System, Air Flow Producing Equipment, Protection Against Corrosion and Abrasion

IN many industrial plants some type of exhaust system designed to collect and remove dusts, fumes, mists, vapors, and gases is essential in order to promote efficiency, economy, and safety of operation. Definitions of these various contaminants are included in Chapter 8, Air Contaminants.

The theory of air flow in an exhaust system in the following paragraphs *A* to *D* will be found in a publication[1] of the *American Foundrymen's Association.*

A. When the air flow producing equipment of an exhaust system is properly operated, it will produce a negative pressure (below atmospheric) in the exhaust side of the system sufficient to overcome all resistance, and to sustain the desired air velocity; and, further, will overcome all resistances on the discharge or positive pressure side of the system so that the air drawn through exhaust inlets will be discharged against atmospheric pressure.

B. An exhaust system is entirely dependent on a *sufficient volume* of air flowing into the exhaust inlets to catch the matter to be exhausted before such matter has an opportunity to diffuse into the general atmosphere of the work place or room.

C. The velocity of the air flowing into an exhaust inlet is usually of secondary importance, and becomes an essential factor only when a certain velocity is required to overcome some force action on the matter to be caught. The velocity within an exhaust system is only important to the extent that it shall be sufficient to convey the entrained matter and prevent it from settling or dropping out in the piping system. Velocity in terms of velocity pressure is most essential in designing a system, because it is the basis upon which all calculations are made. In testing and checking a system the velocity, as determined from the velocity pressure reading obtained by means of a Pitot tube, is the only true indication of the exact air flow in a pipe or system.

D. The total pressure within an exhaust system is only of importance in determining the power required to operate the system. Total pressure tests do not indicate the proper functioning of an exhaust system as related to the volume and velocity of the air flowing into an exhaust inlet.

General design information is included in this chapter which is intended to relate primarily to industrial exhaust systems.

CLASSIFICATION OF SYSTEMS

In general, there are two basic layouts of exhaust systems, the central and the multiple unit system. In the central system a fan is located near the center of operations with a piping system radiating to the various machines to be served. In the multiple unit system, which is sometimes employed where the machines to be served are widely scattered, or where the operations are apt to be independent or intermittent, small individual exhaust fans are located at the center of the machine groups or at each machine. The unit arrangement has the advantage of flexibility.

Exhaust systems may be classified with respect to the nature of the material to be handled by them as (a) those handling dusts and certain fumes and mists of large particle size; and (b) those handling vapors, gases and certain fumes and mists of small particle size. Design details differ in systems serving dust producing operations and those exhausting the more vapor-like matter, even though the same basic theories govern both classes.

Dust or gas may be captured by enclosure or by open hoods with positive inward air movement. With some classes of machinery it is not feasible to hood the machines closely, and in these cases open hoods over or adjacent to the machines are provided to collect as much as possible of the dust and fumes. Examples of these classes include such machines or operations as pickling tanks, melting furnaces, arc welding, and monument finishing operations.

Open hoods should be placed as close to the source of dust or fumes as possible, with due regard to the movements of the operator. In no instance should the operator be located between the source of dispersion and the exhaust hood or enclosure. When the hood must be placed at some distance above the machine, it should be large enough to cover a large area, as dispersion (considering dust) is usually quite rapid.

Some consideration should be given to the natural movement of the contaminant. In many cases there are convection currents and other atmospheric disturbances in the work room. These disturbances diminish the tendency of dust and fumes to settle from the room air.

In some classes of operation, the main objective is to prevent the escape of dust into the surrounding atmosphere, and the removal of some dust from the machine or enclosure may be merely incidental. The dust-creating apparatus is enclosed within a housing which is made as tight as practicable, and sufficient suction is applied to the enclosure to maintain an inward air flow through all cracks and openings, thus preventing escape of the dust. While the exhaust system is required to handle only the air which enters through the crevices and openings in the enclosure, in many installations leakages are very high, and great care is required to reduce such leakages to a minimum.

Certain dust and fume producing operations are best carried on by isolating the process in a separate compartment or room, and then applying general ventilation to this space.

HOOD DESIGN PRINCIPLES[2, 3, 4, 5, 6, 7]

The first and most important steps in the design of a local exhaust system are to determine the number and shape of hoods or enclosures, and the size of the branch connections. No general rules, however, can be given since hood and duct designs are determined by the characteristics of the operations to which they are applied. When a tentative decision regarding the set-up has been made, it is next necessary to obtain the suction and air velocities required to effect control. At this point, the designer must rely upon the prevailing practice, and on such physical data relating to hoods, duct systems, and collectors as are available.

In general, the most important requirements[8] of an efficient local exhaust system are:

1. Hoods, ducts, fans, motors and collectors should be of adequate size and type.
2. The air velocities should be sufficient to control and convey the materials collected.
3. The hoods and ducts should be placed so as not to interfere with the operation of a machine or any working part.
4. The system should do the required work with a minimum power consumption.
5. When flammable contaminants are conveyed, the piping should be provided with an automatic damper in passing through a fire-wall (Refer to *Pamphlet* No. 91, *National Board of Fire Underwriters*).[9]

6. Ducts and all metal parts should be grounded to reduce the danger of dust explosions by static electricity. Motor and starting equipment should conform to Article 500—Hazardous Locations—of the National Electrical Code.[10]

7. The exhaust system should be readily accessible for inspection and maintenance.

CAPTURE VELOCITY AND HOOD SUCTION

The removal of dust or contaminant by means of an exhaust hood requires a movement of air, at the point of origin, sufficient to carry it into the exhaust system. The air velocity necessary to accomplish this depends upon the physical properties of the material to be controlled and the direction and speed with which it is dispersed. If the dust to be removed is already in motion, as is the case with high-speed grinding wheels, the hood must be installed in the path of the particles so that a minimum air volume may be used effectively. It is always desirable to design and locate a hood so that the volume of air necessary to produce results is as small as possible. This will reduce the size of equipment, the power required by the system, and also the heating load requirements in winter.

TABLE 1. MINIMUM AIR VELOCITIES REQUIRED AT POINT OF ORIGIN TO CAPTURE CONTAMINANT EFFECTIVELY

Condition of Generation of Contaminant	Minimum Capture Velocity, FPM	Process
Released without noticeable movement	50–100	Evaporation of vapors, exhaust from pickling, washing, degreasing, plating, welding, etc.
Released with low velocity	100–200	Paint spraying in booth; inspection, sorting, weighing, packaging, low speed conveyor transfer points, rotating mixtures, barrel filling.
Active generation	200–500	Foundry shakeout, high speed conveyor transfer points, crushers, screens.
Released with great force	500–2000	Grinding, tumbling mills, abrasive cleaning.

Capture Velocities

Data for the selection of *capture velocities* of many operations are not available, but it is safe to assume that for most dusty operations velocities should not be less than 200 fpm at the point of origin. Recommended minimum capture velocities for various processes are given in Table 1.

The method for determining, approximately, the quantity of air that must be exhausted to produce these capture velocities at the point of origin, is given in Equation 1:

$$Q = V(10X^2 + A) \tag{1}$$

where

Q = quantity of air exhausted, cubic feet per minute.
V = air velocity in feet per minute at X distance in feet from the hood and on the centerline of the hood.
X = distance in feet, along the hood centerline, from the face of the hood to the point where the air velocity is V feet per minute.
A = area in square feet of the hood opening.

TABLE 2. BRANCH PIPE SIZE FOR WOODWORKING MACHINE HOODS

Based on a Pipe Velocity of 4000 fpm.

Type of Machine	Size, In.		No. of Branches	Minimum Diameter, In.		
	Min.	Max.		Bottom Branch	Top Branch	Others
Self feed table saw			2	5	4	
Other single saws		18	1		4	
	18		1		5	
Saws with Dado Head			1		5	
Band saws		2	2	4	4	
	2	3	2	5	4	
	3	6	2	5	5	
Disc sanders		18	1	4		
	18	28	1	5		
	26	32	2	4	4	
	32	38	2	5	4	
	38	48	3	5	4	4
Triple drum sanders		30	1	7		
	30	36	1	8		
	36	42	1	9		
	42	48	1	10		
Single drum sanders:		350	1	4[a]		
(area in sq in.)	350	700		5		
	700	1400		6		
	1400	2800		7		
Horizontal belt sanders		9	2	5	4	
	9	14	2	6	4	
Vertical belt sanders		6	1	4		
	6	9	1	5		
	9	14	1	6		
Jointers		8	1	4		
	8	20	1	5		
Single planers		20	1	5		
	20	26	1	6		
	26	36	1	7		
Tenoner			2	5	5	

[a]Not over 10 in. diameter.

No set rule can be given regarding the shape of a hood for a particular operation, but it is well to remember that its essential function is to create an adequate velocity distribution. The fact that the zone of greatest effectiveness does not extend laterally from the edges of the opening, may frequently be utilized in estimating the size of hood required. Where complete enclosure of a dusty operation is contemplated, it is desirable to leave enough free space to equal the area of the connecting duct. Hoods for grinding, polishing, and buffing should fit closely, but, at the same time should provide an easy means for changing the wheels. It is advisable to design these hoods with a removable hopper at the base to capture the heavy dust and articles dropped by the operator. Such provisions are of assistance in keeping the ducts clear. The air quantity required to capture dust which is thrown or projected in a direction away from the hood at considerable velocity, may often be reduced by effective baffling or partial enclosure of an operation. This procedure is strongly urged where dusts are directed beyond the zone of influence of the hood.

Air Flow from Static Readings

State codes for local exhaust systems at certain operations list minimum static suction requirements which may range from $1\frac{1}{2}$ in. to 5 in. water column. Frequently, in grinding, buffing, and polishing operations, a large part of the wheel must be exposed, and the dust-laden air within the hood is thrown outward by the centrifugal action of the wheel, thus counteracting useful inward draft.

The static suction at the throat of a hood is frequently used in practice as a measure of the effectiveness of control. Where the hood coefficient is known, the volume of air flow through any hood may be determined from the equation:

$$Q = 4005\, fA\sqrt{h_s} \tag{2}$$

where

Q = quantity of air exhausted, cubic feet per minute.
A = area of connecting duct, square feet.
h_s = static suction measured at approximately 3 diameters from throat of hood, inches of water.
f = orifice or restriction coefficient which varies from 0.6 to 0.95, depending on the shape of the hood.[5]

An average value of f is 0.8, although for a well-shaped opening a value of 0.85 to 0.9 may be used. The factor f is determined from the equation:

$$f = \sqrt{\frac{h_v}{h_s}} \tag{3}$$

where h_v is the velocity head in the connecting duct.

The *static suction* is not a good measure of the effectiveness of a hood, unless the area of the opening and the location of the operation, with respect to the hood, are known. This is clearly indicated by Equation 4 which shows that the velocity at any point along the axis varies approximately inversely as the square of the distance. This formula, coupled with Equation 2, should serve to indicate the velocity conditions to be expected when operations are conducted externally to the hood opening.

Axial Velocity Formula for Hoods

When the normal flow of air into a hood is unobstructed, Equation 4 may be used to determine the air velocity at any point along the axis:[11]

$$V = \frac{0.1\, Q}{x^2 + 0.1\, A} \tag{4}$$

where

V = velocity at point, feet per minute.
Q = quantity of air exhausted, cubic feet per minute.
x = distance along axis, feet.
A = area of opening, square feet.

Design Based on Total Air Flow

Where the foregoing factors are not known, the usual method of designing an exhaust system is to base the air flow through the system on rates of flow (through each hood) which have been found by expeience to provide adequate control. For woodworking systems the sizes of branch connections in common use are given in Table 2.

Similar data for grinding and buffing wheels are given in Table 3.

Velocity Contours

It is possible, by use of a specially constructed Pitot tube,[12] to map contours of equal velocity in any axial plane located in the field of influence. It has been found that the positions of these contours for any hood can be

TABLE 3. BRANCH PIPE SIZES FOR GRINDING AND BUFFING HOODS

Based on a Pipe Velocity of 4500 fpm.

Type of Wheel	Wheel Size Diameter, In.		Maximum		Branch Pipe Minimum Diameter, In.
	Min.	Max.	Width In.	Area Sq In.	
Grinding	..	9	1	30	3
	9	18	3	175	4
	18	24	4	300	5
	24	30	5	500	6
	30	36	6	700	7
Disc Grinding		20	..	300	4
	20	30		...	5
Buffing, Polishing and Scratch Brushing	.	8	2	50	3½
	8	16	3	150	4
	16	24	4	300	5
	24	30	6	600	6

expressed as percentages of the velocity at the hood opening, and are purely functions of the shape of the hood.[13]

Further, the velocity contours are identical for similar hood shapes when the hoods are reduced to the same basis of comparison. These facts are applicable to all hood problems so that, when the velocity contour distribution is known, the air flow required can be determined. Fig. 1 shows the contour distribution in two axial planes perpendicular to the sides of a rectangular hood having a side ratio of one-half. The distribution shown is idential for all openings with a similar side ratio, provided the mapping is as shown in the figure. The contours are expressed as percentages of the velocity at the opening.

Low Velocity Systems

On multiple installations of the same operation, it is often possible to institute a great saving in power cost by designing an exhaust system using low velocities in the main ducts. Such a system, for use in grinding and shaping porcelain, has been described.[14] In these operations, the separate machines are grouped around a central plenum chamber, and exhausted by means of a low pressure fan connected to the plenum. In one such case, a power saving of over 90 percent was obtained. A similar design technique[15] has been described for use in ventilating plating tanks.

Canopy Hoods

Canopy hoods are being replaced by other types of hoods, such as slotted hoods at tank operations. Where canopy hoods are used, they should extend 6 in. laterally from the tank for every 12 in. elevation and, wherever possible, they should have side and rear aprons so as to prevent short-

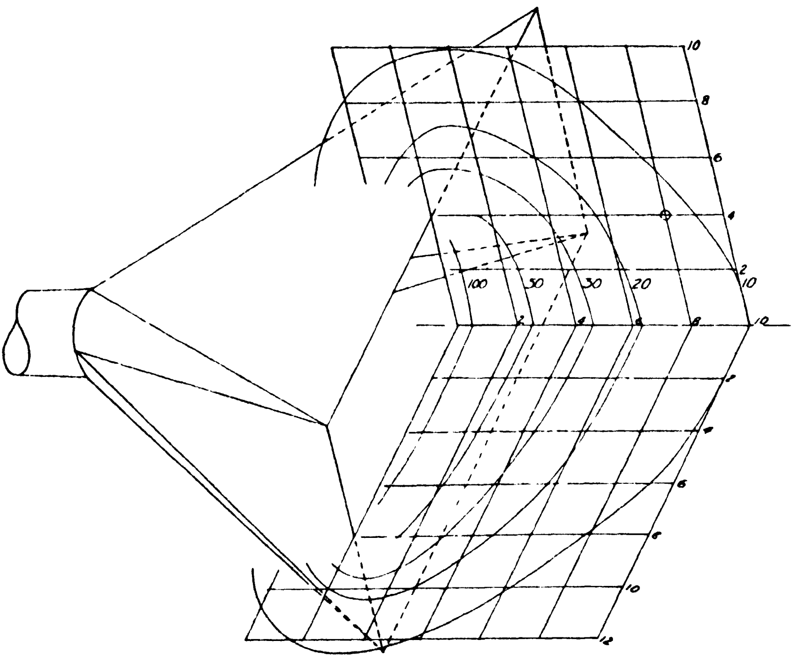

FIG. 1. VELOCITY CONTOURS FOR RECTANGULAR OPENING WITH A SIDE RATIO OF ONE-HALF. CONTOURS ARE EXPRESSED AS PERCENTAGES OF THE VELOCITY AT THE OPENING

circuiting of air from spaces not directly over the vats or tanks. In most cases, hoods of this type take advantage of the natural tendency of the vapors to rise, and air velocities may be kept low. Cross drafts from open doors or windows disturb the rise of the vapors, and therefore consideration must be given them. The air velocities required also depend upon the character of the vapors given off. The recommended minimum capture velocity is 100 fpm.

The quantity of air which must be exhausted to obtain any given capture velocity is expressed by the following equation:

$$Q = 1.4\,PDV \tag{5}$$

where

Q = quantity of air exhausted by hood, cubic feet per minute.
P = perimeter of the tank, feet.
D = distance between tank and hood opening, feet.
V = capture velocity, feet per minute.

Lateral Exhaust Systems

Lateral exhaust, as developed for chromium plating,[16] is preferred to canopy type hoods. The method operates by drawing air and fumes laterally across the top of vats or tanks into slotted ducts located at the top and extending fully along one or more sides of the tanks. The slot width is usually based on a slot velocity of 2000 fpm, but should not be less than 1 in. wide. The hood should not be required to draw the air laterally for a distance of more than 24 in., and the level of the solution should be kept 6 to 8 in. below the top of the tank. If width of tank is over 24 in., a double lateral exhaust should be used with slots on both sides.

It has also been determined that a similar control may be used for tanks wider than 3 ft when the same velocity (2000 fpm) is maintained through a slot which is increased $\frac{1}{4}$ in. for every foot of width greater than 3 ft. When these slots must be extended more than 6 ft in length, some method of spreading the flow is necessary to provide even air flow distribution through the entire slot length. This can be accomplished by tapering the slot, which incidentally will add to the resistance of the system. A more economical approach is to place properly spaced vanes in the side ducts, or to branch the side ducts.[17]

Spray Booths

In the design of an efficient spray booth, it is essential to maintain an even distribution of air flow through the opening and about the object being sprayed. While in many instances spraying operations can be performed mechanically in wholly enclosed booths, the volatile solvent vapors produced by spraying operations may reach injurious or explosive concentrations. At all times, the concentrations of these vapors, and particularly those containing benzol, should be kept well below 100 parts per million in the breathing zone of the worker. Vapors from many spraying operations are dangerous to the health of the worker, and care should be taken to minimize exposure to them.

It is recommended in the design of spray booths that the exhaust duct be located at the end of the booth opposite the opening. In front of this duct should be placed baffle plates which will cause a uniform air velocity distribution across the frontal area. The air volume should be sufficient to maintain a velocity of not less than 150 fpm over the open area of the booth.

Spray booths may be of either the dry or wet type. The latter is the more modern design, provided with a water-wash section for the removal of the solid over-spray contaminants, and for the absorption of water-soluble thinners or solvents.

The most modern innovation is the electrostatic spraying and detearing unit. Objects to be sprayed are passed through a high tension electrostatic field, which not only produces a more evenly sprayed surface, but materially reduces excessive over-spray. The detearing unit removes teardrops of sprayed material from the edges or ends of air-drying sprayed objects as these objects pass through a high tension electrostatic field.

Hoods for Chemical Laboratories

Hoods used in chemical laboratories are generally provided with sliding windows which permit positive control of the fumes and vapors evolved by the apparatus. Their design should offer easy access for the installation of chemical equipment, and should be well lighted. Air velocities should not exceed 100 fpm when the window is fully open.

Kitchen Hoods

The length and width of kitchen hoods should be such as to extend beyond the extreme projection of the ranges, broilers, etc., over which they are installed. The minimum projection or overlap should be 12 in. Where space conditions permit, range hoods should be about 2 ft high in order to provide a reservoir to confine momentary bursts of smoke and steam until the exhaust system can evacuate the hood. Range hoods should be located as low as possible to increase their effectiveness.

In general, the amount of air to be exhausted from restaurant range hoods is at the rate of 100 cfm per square foot of face area. In some cases, where the application is principally frying, and where it is not practicable to install a hood 2 ft high, it is recommended that the face velocity be increased from 100 to 150 fpm, depending on peak load conditions in the kitchen. Exhaust connections to range hoods should always be made at the top and back of hoods, and should be spaced preferably not more than 6 ft apart, and be rectangular in shape with the long side parallel to the back of the hood. Exhaust openings into range hoods should be designed to maintain a velocity of 1500 to 1800 fpm.

An approved fire damper with fusible link should be (and is required by code in many states) installed in the main exhaust duct or branch adjacent to the range hood. Should there be more than one hood connected to a common duct, then the branch duct to each hood should be provided with a fire damper. Access doors should be provided at the fire damper for purpose of inspection, cleaning, or for renewal of fusible link. All exhaust piping to range hoods, commonly called grease ducts, should be provided with tight fitting cleanout doors of adequate size to permit easy removal of grease. Some engineers use filters to advantage in hoods which are subject to grease conditions.

Hoods over steam tables should be of construction similar to range hoods. It is good practice to design such hoods with a face velocity of 60 to 70 fpm. Hoods over dishwashing machines are usually relatively small, and generally 1500 to 2000 cfm per hood are allowed, which is equivalent to a velocity of approximately 100 fpm per square foot of face area. Range hoods in diet kitchens are constructed the same as restaurant range hoods, but with less exhaust air per square foot of face area, depending upon the nature of the food cooked.

Hoods are not often used in private residences, unless they are quite large and the consideration of expense is not important. For such residences the hoods should be designed on the same basis as diet kitchens. Most all residence kitchens can be effectively and economically ventilated by the installation of a built-in kitchen ventilator, which should be located in an outside wall and in close proximity to the kitchen range. It has been found that the capacity of the built-in kitchen ventilator should be at least 350 cfm regardless of the size of kitchen. This can be justified on the

Table 4. Approximate Conveying Velocities

Material Conveyed	Design Velocity FPM
Vapors, gases, fumes, very fine dust	2,000
Fine dry dusts	3,000
Average industrial dusts	3,500
Coarse particles	3,500–4,500
Large particles, heavy loads, moist materials	4,500 and over

basis that the smaller the kitchen the more concentrated the heat will be, thus requiring a more rapid rate of air change. Standard size built-in kitchen ventilators are generally available in three sizes, namely, 350, 500 and 800 cfm. The proper size to use will depend on design conditions and available wall space.

DUCT SYSTEM DESIGN

In designing a duct system, it is necessary to recognize a few fundamental principles (see also Chapter 31). Knowing the quantity of air required, the size of the duct may be computed from Equation 6:

$$A = \frac{Q}{V} \tag{6}$$

where

A = cross-section area of duct, square feet.
Q = air quantity to be exhausted by the duct, cubic feet per minute.
V = velocity of air, feet per minute.

Air Velocities in Ducts

Where it is necessary to transport the particulate material collected in an exhaust system, minimum carrying velocities must be maintained in the ducts preceding the collector. It has been found that good results are obtained when design air velocities in horizontal runs are not less than 2000 fpm, or not greater than 5000 fpm. When the dust being carried is organic and other than wood flour, or similar material, a velocity of 2500 fpm is adequate. Approximate required conveying velocities are given in Table 4.

For duct systems wherein the air has no dust or solid load, a lower velocity is desirable, which may range from 1500 to 2500 fpm. In view of the fact that the horsepower required by a system depends directly on the resistance, and the resistance is a function of the velocity, economical design requires velocities of this magnitude.

The equal friction method is generally used for designing a duct system, as this insures equal resistance to air flow in all branches throughout the system (see Chapter 31). Long main ducts do not generally provide the most economical layout. Where it is necessary to ventilate a large number of machines, or machines which are widely separated, it is desirable to locate the fan at approximately the center of the system. With this arrangement it is possible to choose a fan which will deliver the required air quantity against a lower resistance pressure, and this will generally result in a horsepower saving.

When a system carrying dust is designed with an oversize main duct to allow for future extension, the air velocity may be found to be too low to carry the dust, and serious plugging may occur. In this case it is desirable to install an orifice in the end of the pipe to allow for the lower air quantity.

Construction

The interior of all ducts should be smooth and free from obstructions at joints, and soldered air-tight. Other sealing mediums are permissible where soldering is impracticable.

Ducts should be constructed of galvanized sheet metal, except when the presence of corrosive fumes or gases, temperatures above 400 F, or other factors would make galvanized material impracticable. For the usual ex-

TABLE 5. GAGES OF METALS FOR EXHAUST SYSTEMS[a]

Diameter of Round Pipe or Greatest Dimension of Rectangular Pipe, Inches	Thickness of Duct Material U. S. Gage Number	
	For Highly Abrasive Matter	For Other Matter
Up to 8 inclusive	20	22
Over 8 to 18 inclusive	18	20
Over 18 to 30 inclusive	16	18
Over 30	14	16

[a]Fundamentals of Design, Construction, Operation and Maintenance of Exhaust Systems (*American Foundrymen's Association*, p. 53).

haust systems, the metal thicknesses shown in Table 5 are recommended. Elbows and angles should be a minimum of two gages heavier than straight lengths of equal diameter. Hoods should be a minimum of two gages heavier than straight sections of a connecting branch.

Longitudinal joints of ducts should be lapped and riveted or spot-welded on 3-in. centers maximum. Girth joints or ducts should be made with lap in direction of air flow, with 1 in. lap for duct diameters through 19 in., and $1\frac{1}{4}$ in. lap for diameters over 19 in. Elbows and angles should have an inside or throat radius of two pipe diameters whenever possible. Large radii are recommended for heavy concentrations of highly abrasive dusts. Elbows 6 in. or less in diameter should be constructed of at least 5 sections and, if over 6 in. in diameter, of 7 sections, with angles pieced proportionally. Hoods should be free of sharp edges or burrs, and reinforced to provide necessary stiffness. Transitions in mains and sub-mains should be tapered with a taper 5 in. long for each 1 in. change in diameter whenever possible. All branches should enter the main at the large end of the transition at an angle not to exceed 45 deg, or preferably 30 deg. Branches should be connected only to the top or sides of mains, with no two branches entering diametrically opposite to each other. Dead end caps should be provided within 6 in. from last branch of all mains and sub-mains. Cleanouts should be provided every 10 ft and near each elbow, angle, or duct junction in horizontal sections. Ducts should be supported sufficiently to place no loads on connected equipment, and to carry weight of a system plugged with material. The maximum distance between supports should be 12 ft for 8 in. or smaller ducts, and 20 ft for larger ducts. Six inches minimum clearance should be provided between ducts and the ceiling, wall or floor. Blast gates for adjustment of the system should be placed near the connection of a branch to the main, and means of locking gates after the adjustments have been made should be included. Rectangular ducts should be used only when clearances prevent the use of round construction. Rectangular ducts should be as nearly square as possible. The weight of metal and the lap, and other construction details, should be the equal of round duct construction having a diameter equal to the longest side. All pipes passing through roofs should be equipped with collars so arranged as to prevent water from leaking into the building.

The main trunks and branch pipes should be as short and straight as possible.

Cleanout openings having suitable covers should be placed in the main and branch pipes so that every part of the system can be easily reached in case the system clogs. Either a large cleanout door should be placed in the main suction pipe near the fan inlet, or a detachable section of pipe, held in place by lug bands, may be provided.

Every pipe should be kept open and unobstructed throughout its entire length, and no fixed screen should be placed in it, although the use of a trap at the junction of the hood and branch pipe is permissible, provided it is not allowed to fill up completely. The passing of pipes through fire-walls should be avoided wherever possible, and floor sweep connections should be so arranged that foreign material cannot be easily introduced into them.

At the point of entrance of a branch pipe into the main duct, there should be an increase in the latter equal to their sum. Some state codes specify that the combined area be increased by 25 percent. While this is not always good practice, and is frequently done at the expense of a reduced air velocity, it is often done where future expansion of the exhaust system is contemplated.

Duct Resistance

The resistance to flow in round galvanized duct, riveted and soldered at the joints, may be obtained from Figs. 1 or 2, Chapter 31. The pressure drop through elbows depends upon the radius of the bend. For elbows

TABLE 6. LOSS THROUGH 90-DEG ELBOWS

RATIO OF ELBOW CENTER LINE RADIUS TO PIPE DIAMETER OR DEPTH	APPROXIMATE LOSS IN PERCENT OF VELOCITY HEAD
1	80
1½	31
2	22
2½	19

whose centerline radii vary from 100 to 250 percent of pipe diameter, the loss may be estimated from Table 6.

RESISTANCE OF SYSTEM

The resistance of the exhaust system is composed of three factors: (1) loss through the hoods, (2) collector drop, and (3) friction drop in the duct system.

The loss through the hoods is usually assumed to be equal to one-half the suction at the hoods. Where possible the resistance of the particular collector to be used should be obtained from the manufacturer.

Friction drop in the pipes must be computed for each section where there is a change in area or in velocity. The velocities should be found in each section of pipe, starting with the branch most remote from the fan. The friction drop for these sections can be determined by reference to Table 6 in this chapter, and Figs. 1 or 2, Chapter 31. Total friction loss in the piping system is the friction drop in the most remote branch, plus the drop in the various sections of the main, plus the drop in the discharge pipe.

EFFICIENCY OF EXHAUST SYSTEMS

The efficiency of an exhaust system depends upon its effectiveness in reducing the concentration of dusts, fumes, vapors, and gases below the safe or threshold limits.[18]

Too much emphasis cannot be placed on the necessity of testing exhaust systems frequently by determining the concentration of atmospheric contamination at the worker's breathing level.[19] Commonly accepted values of threshold limits for usual atmospheric contaminants will be found in Tables 3, 4 and 5, Chapter 8.

AIR FLOW PRODUCING EQUIPMENT

In any type of exhaust system, some form of air flow producing equipment is required to create the pressure necessary to cause the air to flow through the system to the discharge stack. The principal types of air moving equipment are centrifugal exhaust fans, disc or propeller fans, axial flow fans and venturi ejectors.

TABLE 7. CORROSION RESISTING MATERIALS FOR EXHAUST SYSTEMS[a]

MATERIAL	ACID[b]							
	ACETIC	CHROMIC	HYDRO-CHLORIC	HYDRO-FLUORIC	NITRIC	PHOS-PHORIC	SUL-PHUROUS	SUL-PHURIC
METALS	Dil. / Conc.	Dil. / Conc.	Dil. / Conc.	Dil. / Conc.	Dil. / Conc.	Dil. / Conc.	Dil. / Conc.	Dil. / Conc.
Aluminum ..	Good	Fair	Poor	No Data	Poor / Good	Poor	Poor	Poor
Magnesium and Alloys	No Data	Good / Poor	No Data	Poor / Good	No Data	No Data	No Data	No Data
Lead and Lead-Coated . .	Poor	Good	Poor	Poor	Poor	Poor	Good	Good / Poor
Moly Alloy (60 Ni–20Mo–20 Fe)	Good	No Data	Fair	No Data	Poor	Poor	No Data	Good
Monel Metal ...	Fair	Poor	Fair / Poor	Good[d]	Fair / Poor	Fair	Fair	Good[e] / Poor
Bronze . .	Poor						Good	
Silicon Iron .	Fair / Good	No Data	Fair	Poor	Good	Good	No Data	Good
Stainless Steel[c] (18 Cr-8 Ni)	Good	Good	Poor	No Data	Good	Poor	Good	Poor / Good[e]
Enameled Steel	No Data	No Data	Good	Poor	Good	Poor	No Data	Good
MISCELLANEOUS								
Asbestos Comp.			Good except against strong acids and alkalies					
Wood	Some woods are decomposed or softened faster than others.							
Rubber. . ..					Poor			Poor
Plastics . .	In general plastics resist weak acids and are decomposed by concentrated acid.							

[a] Standard Practice Sheet No. 115 (Division of Industrial Hygiene, New York State Labor Department).
[b] Acid mists in air are more corrosive than as liquid in storage tank. Galvanized iron not resistant to acid.
[c] Stainless steel of (24 Cr—10 Ni) fairly resistant at low temperature for HCl and H_3PO_4.
[d] Under most conditions.
[e] At room temperatures.

PROTECTION AGAINST CORROSION AND ABRASION

Manufacturers generally provide special fans for the handling of various industrial wastes. When corrosive or abrasive materials are conveyed, the fan blades and interior of the fan housing should be protected from wear. This may be accomplished by placing the collector on the suction side of the fan. Excellent protection against many corrosive acids may be obtained by lining the interior surfaces of the ducts and fans, including wheels, with rubber.

The removal of gases and fumes in many chemical plants requires that metals used in the construction of the exhaust system be resistant to chemical corrosion. A list of the materials which may be used to resist the action of certain fumes is given in Table 7. Hoods and ducts, when short, may frequently be constructed of wood and be quite effective. Rubberized paints are available and may be applied as protective coatings in handling such gases as chlorine and hydrochloric acid.

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CHAPTER 46

INDUSTRIAL DRYING SYSTEMS

Drying Terminology, Mechanism of Drying, Internal and External Conditions, Periods of Drying, Approximate Equations for Estimating Drying Time, Equilibrium Moisture Content, Applications of Hygrometry to Drying, Dryer Calculations, Drying Methods and Equipment; Radiant, Conduction and Convection Drying; Solution of Drying Problem

THE term drying, in a broad sense, encompasses the removal of water, and occasionally other liquids, from gases, liquids, or solids. However, the common usage of the word confines the meaning principally to the removal of water or solvent from solids by thermal means. Dehumidification is the term that is commonly assigned to the drying of gases. This is usually accomplished by condensation or adsorption by various drying agents, and is treated in Chapter 37. Distillation, and more particularly fractional distillation, is associated with the drying of liquids.

It is usually more economical to employ, whenever possible, mechanical means of separating as much water as is practicable from the solid materials before undertaking drying or dehydration steps. These mechanical methods such as filtration, screening, pressing, centrifuging, or settling usually require much less power, and frequently less capital outlay, thereby making the operation cheaper in terms of cost per pound of water removed.

DRYING TERMINOLOGY[1]

The generally accepted definitions of terms used in drying technology follow:

Bound moisture refers to liquid (held by a solid) which exerts a vapor pressure less than that of the pure liquid at the same temperature. Liquid may become *bound* by retention in small capillaries, by solution in cell or fiber walls, by homogeneous solution throughout the solid, and by chemical or physical adsorption on solid surfaces. Bound moisture can be removed from a solid only under specific conditions of humidity in the external surroundings.

Capillary flow refers to the flow of liquid through the interstices and over the surface of a solid. It is caused by liquid-solid molecular attraction.

Commercial dry basis expresses the moisture content of a product as pounds of water per pound of solid as it leaves the dryer, *i.e.*, per pound of commercially dry solid.

The constant-rate period is that drying period during which the rate of water removal per unit of drying surface is constant.

The critical moisture content is that obtaining when the constant-rate period ends and the falling-rate period begins.

Dry basis indicates the moisture content of a wet solid as pounds of water per pound of bone-dry solid. The advantage of using this basis is that the absolute amount of moisture loss is obtained simply by subtracting the moisture contents before and after drying. (See definition of *Wet Basis.*)

Dryer efficiency is that fraction of the total heat, supplied by fuel, used to evaporate water. *Overall efficiency* is sometimes used to distinguish overall system efficiency from the *efficiency of the drying space* or *evaporative efficiency*.

Equilibrium moisture content is that to which a given material can be dried under specific conditions of air temperature and humidity.

Evaporative efficiency compares the amount of evaporation actually obtained in a dryer with that which would obtain by saturation of the air.

The falling-rate period is that drying period during which the instantaneous drying rate continually decreases.

Fiber saturation point is the moisture content of cellular materials (wood, etc.) at which the cell walls are completely saturated while the cavities are liquid-free. It may be defined as the equilibrium moisture content at the humidity of the surrounding atmosphere approaches saturation.

Free moisture content is that liquid content which is removable at a given temperature and humidity. Free moisture may include both bound and unbound moisture.

The funicular state is that condition in drying a porous body when capillary suction causes air to be sucked into the pores.

Humidity denotes the amount of water vapor actually present in a gas, and is generally expressed as weight of vapor per unit weight of any gas.

A hygroscopic material is one that may contain bound moisture.

Initial moisture distribution refers to the moisture distribution throughout a solid when drying begins.

Internal diffusion. Diffusion is a single-phase phenomenon; internal diffusion must therefore occur as solid through solid, liquid through liquid, or gas through gas. Internal diffusion occurs when the moving phase obeys the fundamental laws of diffusion.

The moisture content of a solid is usually expressed as moisture quantity per unit weight or volume of the dry or wet solid. A weight (dry or wet) basis is preferred.

Moisture gradient refers to the internal distribution of water in a solid at a given moment in the drying process, the nature of which depends on the characteristics of the solid involved.

A non-hygroscopic material is one that can contain no bound moisture.

Pendular state is that state of a liquid in a porous solid when a continuous film of liquid no longer exists around and between discrete particles and, therefore, flow by capillarity cannot occur. This state succeeds the funicular state.

Unaccomplished moisture change refers to the ratio of the free moisture present at any time to that initially present.

Unbound moisture in a hygroscopic material is that moisture in excess of the equilibrium moisture content corresponding to saturation humidity. All water in a non-hygroscopic material is unbound water.

Wet basis expresses the moisture in a material as a percentage of the weight of the *wet* solid. This basis is less satisfactory than the dry-weight basis on which the percentage change of moisture is constant for all moisture contents. Fig. 1 shows the relationship between the *dry-* and *wet-weight* bases, and indicates that when the *wet-weight* basis is used to express moisture content, a 2 or 3 percent change at high moisture content (above 70 percent) actually represents a 15 to 20 percent change in evaporative load. An evaporative increase of this amount might well increase the load above the capacity of a dryer.

MECHANISM OF DRYING[1]

When a solid dries, two fundamental processes are involved: (1) the transfer of heat to evaporate the liquid, and (2) the transfer of mass as vapor and internal liquid. These two processes occur simultaneously, and the factors governing the rate of each process determine the rate of drying.

In any commercial drying problem, a principal objective is to supply the required heat in the most efficient manner. Consequently, heat transfer may occur by convection, conduction, or radiation, or by any combination of these mechanisms. The various types of industrial dryers may be shown to differ fundamentally with respect to the method used for transferring heat to the solid. In general, heat must flow first to the outer surface of the solid and then into the interior. An important exception is drying with high frequency electrical currents where heat is *generated within the solid*, producing a higher temperature at the interior than at the surface, and consequently, causing heat to flow from inside the solid to the outer surfaces.

Mass transfer in drying occurs as liquid or vapor flow, or both, within the solid, and as vapor flow from the external wet surfaces. The nature of liquid concentration gradients in solids during drying depends on the mechanism of internal liquid flow, and this mechanism, in turn, depends to a large extent upon the physical and chemical characteristics of the solid being dried.

Internal vs. External Conditions

A study of how a solid dries may be based on the *internal* mechanism of liquid flow, or on the effect of the *external conditions* of temperature, humidity, air flow, state of subdivision, etc., on the drying rate of the solid. The former procedure involves a fundamental study of the liquid flow conditions within a solid during drying. The latter procedure, although less fundamental, is more generally used because the effects are easier to establish and the results have greater immediate application in dryer design and operation.

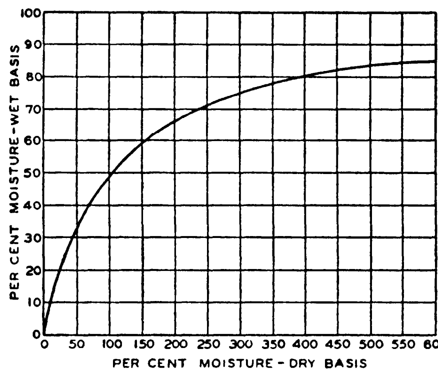

FIG. 1. RELATION BETWEEN WET-WEIGHT AND DRY-WEIGHT BASES

Internal Mechanism of Liquid Flow. Internal liquid flow may occur by several mechanisms, depending on the structure of the solid. Several mechanisms of flow are as follows:

1. *Diffusion* in continuous, homogeneous solids.
2. *Capillary flow* in granular and porous solids.
3. Flow caused by *shrinkage* and *pressure* gradients.
4. Flow caused by a *vaporization-condensation* sequence.
5. Flow caused by *gravity*
6. Flow caused by an electrical potential, *electro-osmosis*
7. Flow caused by temperature gradients, *thermal diffusion*

Although more than one of these mechanisms of flow may be effective at one time, only one predominates as a rule at a given time in a solid during drying. However, a different mechanism may predominate at a different time in the cycle. The mechanism of moisture flow is usually established experimentally from a study of moisture gradients.

External Variables. The principal external variables involved in any drying problem are: *temperature*, *humidity*, *air flow*, *state of subdivision* of the solid, *agitation* of the solid, *method of supporting the solid*, and the *contact* between hot surfaces and wet solid. All these variables do not necessarily occur simultaneously in one problem.

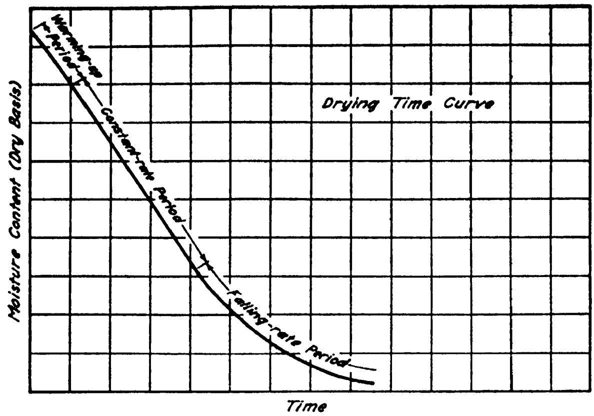

FIG. 2. MOISTURE CONTENT W VS. DRYING TIME θ[2]

Periods of Drying[1]

A typical drying time curve for a wet solid is shown in Fig. 2. This curve is a plot of the moisture content at any time in a solid undergoing drying. It is the usual method of presenting experimental drying data. Although Fig. 2 shows that the moisture content is subject to a continuous variation with time, a more precise illustration of the nature of this variation can be obtained by differentiating the curve and plotting the drying rate (pounds of water per hour per pound of dry material) against the moisture content (pounds of water per pound of dry material) as shown in Fig. 3, or plotting the rate of drying against time as shown in Fig. 4. These *rate curves* show that the drying process is not a smooth, continuous one in which a single mechanism controls throughout. The rate curve in Fig. 4 has the advantage of showing how long each drying period predominates.

Section AB on each curve represents a *constant-rate period*. In Fig. 2, it is shown by a straight line of constant slope $dW/d\theta$, which becomes a horizontal line on the rate curves in Figs. 3 and 4.

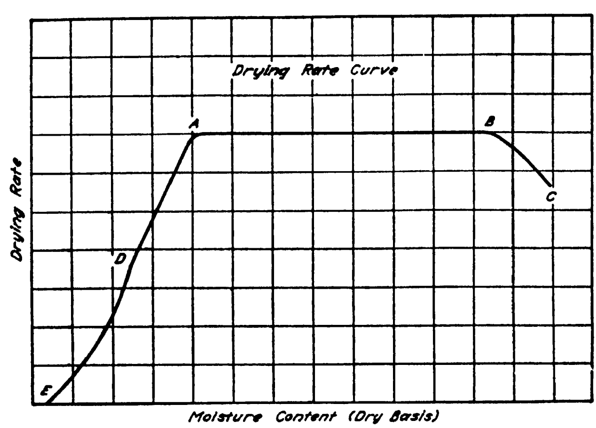

FIG. 3. RATE OF DRYING $\frac{dW}{d\theta}$ VS. MOISTURE CONTENT W[2]

The curved portion of Fig. 2 is termed the falling-rate period, and, as shown in Figs. 3 and 4, it is typified by a continuously changing rate. Point A, where the constant rate ends and the drying rate begins to decrease, is termed the *critical moisture content.*

The portion of the curves designated by CB represents a warming-up period, and it may, or may not, be a significant item depending on the total time involved.

Constant-Rate Period. Drying during the constant-rate period is equivalent to evaporation from a free-water surface on the surface of the solid. The rate of drying in this period is determined by the rate of diffusion of water vapor through an air film at the wet surface of the solid. A constant rate of evaporation on the surface of the solid maintains the surface at a constant temperature, which, in the absence of other heat effects, is very nearly the wet-bulb temperature of the air. If heat flows to the surface of evaporation by radiation and conduction, or both, in addition

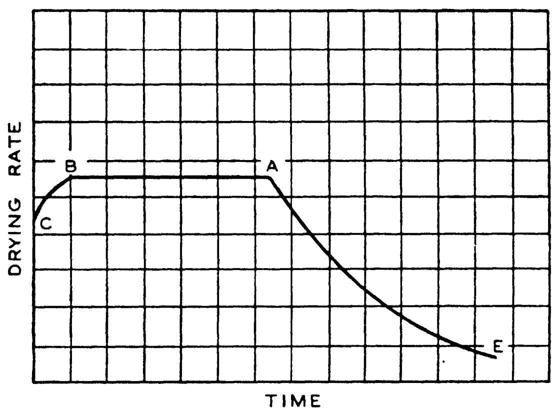

FIG. 4. RATE OF DRYING, $\frac{dW}{d\theta}$ VS. TIME θ

to convection, the surface temperature will be constant at some value between the air temperature and the wet-bulb temperature. This higher temperature in turn produces a higher constant rate of evaporation.

In those dryers in which heat is transferred to a wet solid by conduction through hot surfaces, and heat transfer by convection is not a factor, the wet surfaces approach the boiling point temperature rather than a wet-bulb temperature.

When all the heat for evaporation in the constant-rate period is supplied by a hot gas, a dynamic equilibrium is established between the rate of heat transfer to the material and the rate of vapor removal from the surface. This equilibrium between heat and mass transfer rates can be expressed as follows:

$$\frac{dw}{d\theta} = \frac{h_t A \Delta t}{\lambda} = k_g A \Delta p \tag{1}$$

where

$\frac{dw}{d\theta}$ = drying rate, pounds of water per hour.

h_t = total heat transfer coefficient, Btu per (hour) (square foot) (Fahrenheit degree).
A = area of heat transfer and evaporation, square feet.
λ = latent heat of evaporation at t_s, Btu per pound.
k_g = mass transfer coefficient, pounds per (hour) (square foot) (atmosphere).
Δt = $(t_a - t_s)$ = temperature difference between air and surface of evaporation, Fahrenheit degrees.
t_a = air temperature, Fahrenheit.
t_s = temperature of surface of evaporation, Fahrenheit.
Δp = $(p_s - p_a)$ = vapor pressure difference, atmospheres.
p_s = vapor pressure of water at t_s, atmospheres.
p_a = partial pressure of water vapor in air, atmospheres

When $h_t = h_c$, the coefficient of heat transfer by convection only, then t_s under equilibrium conditions becomes t_w, the wet-bulb temperature of the air, and p_s is the vapor pressure at this temperature. If heat is also supplied by radiation, then h_t is the sum $(h_c + h_r)$ where h_r is the radiation coefficient and h_c is the convection coefficient, and t_s becomes higher than the wet-bulb temperature. A similar result occurs when heat reaches the surface of evaporation by convection and conduction. When the surface is at the wet-bulb temperature, the value of Δp in millimeters of mercury, is almost exactly one-half the wet-bulb depression $(t_a - t_w)$, in *Centigrade* degrees.

Effect of Air Velocity. The principal effect of air velocity is on h_c and k_g, since the rate of transfer of heat and mass in the constant-rate period depends mainly on the rate of diffusion of heat and vapor through the air film at the surface of the solid, and air velocity is the chief factor affecting the thickness of this film. The influence of air velocity may be expressed by the following relationship:[3]

$$h_c = 0.0128\ G^{0.8} \qquad (2)$$

where

h_c = convection heat transfer coefficient, Btu per (hour) (square foot) (Fahrenheit degree).
G = mass velocity of dry air, pounds per (hour) (square foot).

For estimating the constant rate in drying from plane surfaces with air flow parallel to the surface of evaporation and with no radiation or conduction effects, the following heat transfer expression can be used:

$$\frac{dw}{d\theta} = \frac{0.0128 G^{0.8} A}{\lambda} (t_a - t_w) \qquad (3)$$

where

t_w = wet-bulb temperature of the drying air, Fahrenheit degrees

Heat transfer coefficients, rather than mass transfer coefficients, should be used to estimate drying rates, because heat transfer coefficients are generally more reliable, and, unless the temperature of the drying surface is measured, it must be calculated from heat transfer considerations before mass transfer coefficients can be applied for drying-rate predictions. The assumption that the surface of drying is at the wet-bulb temperature of the air, introduces a more serious error in the computation of mass transfer than of heat transfer.

Determination of True Surface Temperature. Frequently, radiation and conduction are of sufficient magnitude to cause the temperature of evaporation to exceed the wet-bulb temperature of the air. When this occurs, it is necessary to estimate the true surface temperature in order to calculate the constant rate. This may be done by means of a heat balance equating the total heat transferred by convection, conduction, and radiation to the latent heat of evaporation.

Constant-Rate Period in Through-Circulation Drying. The equation for estimating the rate of evaporation when air flows *across* a free water surface must be modified for the case of air-flow *through* a permeable bed of solids. The constant rate in through-circulation drying depends on the air rate, air temperature, air humidity, size of the particles making up the permeable bed, and physical characteristics of these particles.[4]

The following general expression for the constant rate in through-circulation drying for the system water and air, was developed[5] from experiments on the rate of evaporation of water from the surface of wet spheres and cylindrical particles with through-circulation of air:

$$\frac{dW}{d\theta} = \frac{0.42aG^{0.59}(\Delta H)_m}{D_p^{0.41}} = \frac{0.37c_s aG^{0.59}\Delta t_m}{\lambda \rho_s D_p^{0.41}} \tag{4}$$

where

$\frac{dW}{d\theta}$ = constant rate, pounds of water per (hour) (pound of dry stock).
a = drying area, square feet per cubic foot of bed volume.
G = superficial mass velocity, pounds of dry air per (hour) (square foot).
ΔH_m = logarithmic mean of inlet and outlet humidity driving force across the air film adjacent to the particle through which the water vapor diffuses, pounds per pound (the surface humidity is taken as the humidity corresponding to the wet-bulb temperature of the drying air).
ρ_s = bulk density of dry granular bed, pounds per cubic foot.
D_p = average diameter of particle, feet.
Δt_m = logarithmic mean difference between temperature entering and leaving the bed and the wet-bulb temperature, Fahrenheit degrees.
c_s = humid heat, Btu per (pound of dry air) (Fahrenheit degree).
λ = latent heat of evaporation, Btu per pound.

Equation 4 applies when the Reynolds number D_pG/μ is greater than 300, where μ is the viscosity of the air stream. For values less than 300, a modification of Equation 4 has been presented.[6]

Evaporation from Liquid Drops. For the important problem of spray drying, evaporation rates of liquid drops must be estimated. Below a value of Reynolds number (D_pG/μ) of 10 for spherical particles, the heat transfer coefficient across the gas film surrounding the drop is given by

$$h = \frac{2k_f}{D_p} \tag{5}$$

where

h = film heat transfer coefficient, Btu per (hour) (square foot) (Fahrenheit degree).
k_f = thermal conductivity of gas film, Btu per (hour) (square foot) (Fahrenheit degree per foot).

Equation 5 is applicable when the Reynolds number for liquid drops is less than 10. Drop diameters are almost always less than 500 microns, and usually in the range of 20 to 150 microns.

The rate of evaporation of drops may be expressed in terms of heat transfer or mass transfer. In terms of heat transfer, the evaporation rate is given by the equation:

$$\frac{dw}{d\theta} = \frac{2\pi k_f D_p}{\lambda}(t_a - t_s) \tag{6}$$

where

$\frac{dw}{d\theta}$ = evaporation rate, pounds per hour.

A similar expression based on mass transfer is

$$\frac{dw}{d\theta} = \frac{2\pi M d_v D_p}{RT}(p_s - p_a) \tag{7}$$

where

M = molecular weight of the diffusing vapor.
d_v = diffusivity of the vapor, square feet per hour.
T = absolute temperature of the gas, Fahrenheit degrees.
R = gas constant, (cubic feet) (atmosphere) per (Fahrenheit degree).
p_s = vapor pressure at the particle surface corresponding to the liquid temperature, atmospheres.
p_a = vapor pressure of liquid in the drying medium, atmospheres.

Both Equations 6 and 7 are based on the assumption that Equation 5 applies. If Equation 6 is integrated for a constant drop diameter (*i.e.*, if it is assumed that the solid being dried in the liquid drop creates a structure which becomes rigid at a fixed D_p) and evaporation proceeds as from a pure liquid drop, an expression for the time of evaporation is obtained as follows:

$$\theta = \frac{W\lambda\rho_s D_p^2}{12k_f(t_a - t_s)} \tag{8}$$

where

θ = time, hours.
W = water content of the drop as it enters the drying chamber, pounds per pound.
ρ_s = density of dry particle, pounds per cubic foot.

The temperature difference between drop and gas ($t_a - t_s$) is essentially constant for a single drop evaporating in a large mass of gas. However, in spray dryers this is not true, and an overall average temperature difference must be used in Equation 8 in this case.

When the drop diameter varies as evaporation proceeds, the expression for the time of evaporation becomes

$$\theta = \frac{\rho_L\lambda[(D_{p1})^2 - (D_{p2})^2]}{8k_f(t_a - t_s)} \tag{9}$$

where

ρ_L = density of the evaporating liquid, pounds per cubic foot.
D_{p1} = drop diameter at the start of evaporation, feet.
D_{p2} = drop diameter of dry particle, feet.

Equation 9 assumes that the drop density is essentially that of the liquid.

Drying at Air Temperatures above the Boiling Point of the Liquid. When the temperature of the drying air is maintained above the boiling point of the liquid being evaporated, or when superheated vapors are used for drying, the usual equations for mass transfer, expressing rate of evaporation

as a function of the vapor pressure difference, lose significance, since large errors are introduced in the expression for vapor pressure driving force due to its apparently small value. Such cases can be treated conveniently on a basis of heat transfer, since a temperature difference must always exist in order that drying may proceed. At drying temperature above 260 F, recirculation has no retarding effect on the drying process.

Constant-Rate Period When Heat Transfer Depends on Conduction and Radiation. In indirect drying, where heat transfer and drying do not depend on the flow of heated gases, the drying rate depends either on heat conduction through retaining walls to wet material in contact with such surfaces, or on radiation, or both. This applies to drum dryers, agitated pan dryers, indirect continuous sheeting dryers, steam tube rotary dryers, vacuum rotary and vacuum tray dryers, and infra-red dryers.

A principal difference between indirect drying and direct drying is that, with the former, the material is usually at a higher temperature than the surrounding air, so that heat is actually transferred to the air instead of from the air.

Generally, the individual heat transfer coefficients for indirect dryers are difficult to determine or estimate, and therefore, an overall coefficient, as defined by Equation 10, is generally used:

$$q = UA\ (t_h - t_s) \tag{10}$$

where

q = rate of heat transfer, Btu per hour.
U = overall heat transfer coefficient based on the temperature difference between the heating medium and the product, Btu per (hour) (square foot) (Fahrenheit degree).
t_h = temperature of the heating medium, Fahrenheit degrees.
t_s = temperature of the solid, Fahrenheit degrees.

The overall coefficient is a function of dryer type. Thus, in agitated pan dryers, U depends on the degree of agitation, temperature of the surface, physical properties of the wet material, etc., and will sometimes vary throughout a drying cycle as the physical properties of the solid vary with a changing moisture content.

As long as U and the temperature difference in Equation 10 remain constant, a constant drying rate will be maintained. However, as drying proceeds the material temperature will begin to increase after some critical moisture content is reached, and, as in the case of direct dryers, a falling-rate period is encountered. U is frequently defined, for the entire drying period, on the basis of an overall mean temperature difference. Therefore,

$$q = UA(\Delta t)_m \tag{11}$$

The Falling-Rate Period. In the discussion of the periods of drying, it was shown that the drying process is discontinuous, consisting of a period of a constant rate of evaporation and a period in which the rate continuously decreases. (See Figs. 3 and 4). This latter period is usually designated as the falling-rate period. It begins when the constant-rate period ends at the critical moisture content. If the critical moisture content is less than the required final moisture content, the constant-rate period will constitute the whole of the drying process. On the other hand, if the initial moisture content is less than the critical moisture content, as in the case of some slow-drying materials, such as soap and wood, then no constant rate will appear, and the whole of the drying process will be in the falling-rate period. This period, in the most general case, can be divided into two zones which may be termed (1) the zone of unsaturated surface drying, and (2) the zone where internal liquid flow controls.

The zone of unsaturated surface drying follows immediately after the critical point and results from a progressively decreasing wetted surface. With the surface no longer completely wetted, dry portions of the solid protrude into the air film, so that the rate of evaporation per unit of total surface is reduced. The effective wetted surface in this zone is frequently a linear function of the water content, so that the curve representing rate of drying *vs.* water content of the solid is straight in this region, as shown by line AD in Fig. 3. The mechanism of drying is essentially the same as during the constant-rate period.

The zone where internal liquid flow is in control, is usually the second zone of the falling-rate period. In this phase the rate of internal liquid movement controls the drying rate, and in drying to low moisture contents, this period may be the principal factor determining the drying time.

Studies of internal moisture flow have indicated the possibility of several controlling mechanisms, the more significant ones having been postulated previously as diffusion, capillarity and pressure gradients due to shrinkage. Of these mechanisms, internal moisture movement by diffusion has been treated extensively, while capillary flow and flow caused by shrinkage and pressure gradients, have received only preliminary consideration.

When diffusion does control in the falling-rate period, it obeys the same fundamental laws of diffusion as those applying to the diffusion of heat. On this basis, the integrated diffusion equation for the falling-rate period (for the case where the surface is dry or at its equilibrium moisture content and the solid has a uniform initial moisture distribution) expresses the average moisture content as a function of time as follows:

$$\frac{W - W_e}{W_o - W_e} = \frac{8}{\pi^2}\left[e^{-\theta d(\pi/2L)^2} + \frac{1}{9}e^{-9\theta d(\pi/2L)^2} + \frac{1}{25}e^{-25\theta d(\pi/2L)^2} + \cdots\right] \tag{12}$$

where

W, W_o, W_e = the moisture contents, on a dry basis, at any time θ; at $\theta = 0$, the start of the diffusional flow period; and in equilibrium with the external conditions, respectively, pounds of water per pound of dry solid.

d = the liquid diffusivity, square feet per hour.

L = one-half the thickness of the solid layer through which the liquid is diffusing, feet.

In Equation 12 it is assumed that evaporation is occurring from two opposite faces of the solid. When evaporation occurs from only one surface, substitute the total thickness of the solid layer for L in Equation 12.

Equation 12 is based on the assumption that d is constant. However, this is rarely true, and d has been shown to vary with moisture content, temperature and humidity.[7]

When the time becomes large, a limiting form of Equation 12 is obtained as follows:

$$\frac{W - W_e}{W_o - W_e} = \frac{8}{\pi^2}e^{-\theta d(\pi/2L)^2} \tag{13}$$

From Equation 13 an expression for the rate of drying may be derived to give

$$\frac{dW}{d\theta} = -\frac{\pi^2 \mathrm{d}}{4L^2}(W - W_e) \tag{14}$$

where $dW/d\theta$ = drying rate, pounds per (hour) (pound dry material).

Equation 14 states that the rate of drying, when internal diffusion controls for long times, is directly proportional to the free moisture content $(W - W_e)$, the liquid diffusivity d, and that the time of drying varies as the square of the material thickness. However, Equation 14 holds only when $(W - W_e)/(W_o - W_e) < 0.6$. When this ratio exceeds 0.6, the curve of drying rate *vs.* moisture content is concave upward.

Equations 12, 13, and 14 hold only for a slab-shaped solid, the length of which is large compared with its thickness.

The falling rate frequently can be expressed with fair accuracy over the required range of moisture content by an equation similar to Equation 14:

$$\left(\frac{dW}{d\theta}\right)_f = -K(W - W_e) \tag{15}$$

where K is a function of the constant rate as follows:

$$K = \frac{(dW/d\theta)_c}{(W_c - W_e)} \tag{16}$$

where

$(dW/d\theta)_c$ = the constant drying rate, pounds per (hour) (pound dry material).
W_c = the critical moisture content, pounds per pound dry material.

Substituting in Equation 16 the proper expression for $(dW/d\theta_c)$ the value of K becomes

$$K = \frac{h_t(t_a - t_s)}{\rho_s L\lambda(W_c - W_e)} \tag{17}$$

and hence, the falling rate for this case is given by

$$\left(\frac{dW}{d\theta}\right)_f = -\frac{h_t(t_a - t_s)(W - W_e)}{\rho_s L\lambda(W_c - W_e)} \tag{18}$$

For materials obeying Equation 18, the drying time varies directly as the thickness. When the surface temperature in the constant-rate period is at the wet-bulb temperature, t_w can be substituted for t_s and $0.0128\ G^{0.8}$ can be substituted for h_t in Equations 17 and 18.

The drying time for each case of the falling-rate period may be obtained by integration of Equations 14 and 18, respectively, to give:

1. Diffusion law

$$\theta_f = \frac{4L^2}{\mathrm{d}\pi^2}\log_e\left(\frac{W_c - W_e}{W - W_e}\right) \tag{19}$$

2. Proportional-to-thickness law

$$\theta_f = \frac{\rho_s L\lambda(W_c - W_e)}{h_t(t_a - t_s)}\log_e\left(\frac{W_c - W_e}{W - W_e}\right) \tag{20}$$

Table 1 gives an approximate classification of materials which are most likely to obey Equations 19 and 20.

Equations 18 and 20 hold for cross-circulation drying. When through-circulation drying is involved, the appropriate constant-rate expression given by Equation 4 must be used to determine K in Equation 16. Thus, for through-circulation drying in the falling-rate period when Equation 15 holds, the rate is given by

$$\left(\frac{dW}{d\theta}\right)_f = -\frac{0.37c_s aG^{0.59}(\Delta t)_m}{\rho_s \lambda D_p{}^{0.41}(W_c - W_e)}(W - (W_e) \tag{21}$$

where the symbols have been defined for Equations 4, 12, and 16.

Critical Moisture Content. In order to use the above equations for estimating the drying time in the falling-rate period, it is necessary to know values of the critical moisture content. Such values are usually difficult to obtain without making actual drying tests which, in themselves, would give the required drying time and thereby obviate the necessity of the calculations.

Table 1. Approximate Classification of Materials Most Likely to Obey Equations 19 and 20

Materials obeying Equation 19	Materials Obeying Equation 20
1. Singe-phase solid systems such as soap, gelatin, glue. 2. Wood and similar solids below the fiber saturation point. 3. Last stages of drying starches, textiles, paper, clay, hydrophilic solids, and other materials when bound water is being removed.	1. Coarse granular solids, such as sand, paint pigments, minerals, etc. 2. Materials in which moisture flow occurs at concentrations above the equilibrium moisture content at atmospheric saturation, or above the fiber saturation point.

It appears that the constant-rate period ends when the moisture content at the surface reaches some specific value. If the rate of drying is great, the moisture gradients within the solid will be steep and the average moisture content considerably greater than that at the surface. It is for this reason that the critical moisture content (average through the material) increases with increase in rate of drying, and with an increase in thickness of the layer being dried.

Approximate Equations for Estimating Drying Time

An estimate of the overall drying time for a given drying problem usually involves an estimate of the time required for the constant-rate period, plus an estimate of the time for the falling-rate period. An approximate equation for the overall drying time applicable to the cross-circulation drying of materials of the type listed in Table 1 as obeying Equation 20, may be written as follows:

$$\theta_t = \theta_c + \theta_f = \frac{(W_o - W_c)\lambda L\rho_s}{h_t(t_a - t_s)} + \frac{\rho_s L\lambda(W_c - W_e)}{h_t(t_a - t_s)}\log_e\frac{W_c - W_e}{W - W_e}$$

$$= B\left[\frac{W_o - W_c}{W_c - W_e} + \log_e\frac{W_c - W_e}{W - W_e}\right] \tag{22}$$

where

$$B = \frac{\rho_s L\lambda(W_c - W_e)}{h_t(t_a - t_s)} = \frac{1}{K}.$$

θ_t = total drying time, hours.
θ_c = drying time for constant-rate period, hours.
θ_f = drying time for falling-rate period, hours.
W_o = initial moisture content, pounds per pound of dry solid.
W_c = critical moisture content, pounds per pound of dry solid.
W_e = equilibrium moisture content, pounds per pound of dry solid.
W = moisture content at time θ_t, pounds per pound of material.
h_t = total overall heat transfer coefficient Btu per (hour) (square foot) (Fahrenheit degree).
t_a = air temperature, Fahrenheit degrees.
t_s = temperature of surface of material, Fahrenheit degrees.
L = depth of material in tray, feet.
λ = latent heat of evaporation at t_s, Btu per pound.
ρ_s = density of dry solid, pounds per cubic foot.

Equation 22 will apply to those materials satisfying Equation 20 when drying to very low moisture content is not involved.

For through-circulation drying, an expression similar to Equation 22 is obtained. Thus, the total drying time for through-circulation drying is given by

$$\theta_t = B'\left[\frac{W_o - W_e}{W_c - W_e} + \log_e \frac{W_c - W_e}{W - W_e}\right] \tag{23}$$

where $$B' = \frac{2.7\rho_s\lambda D_p^{0.41}(W_c - W_e)}{c_s aG^{0.59}(\Delta t)_m}$$

The drying times estimated from Equations 22 and 23 apply only to cross-circulation drying and through-circulation drying, respectively. Drying times for other methods, such as rotary drying or drum drying, must be estimated by other methods.

Equilibrium Moisture Content

In the drying of solids it is important to distinguish between hygroscopic and non-hygroscopic materials. A hygroscopic material is one which retains a definite percentage of moisture under definite conditions of air humidity. This bound moisture is in a state of equilibrium with the water vapor in the surrounding air, and a decrease in the water vapor content will decrease the amount of equilibrium bound water. Water so retained by a solid in equilibrium with the humidity of the surrounding air, is designated as the *equilibrium moisture content.* Such moisture may be held as adsorbed surface films or condensed in fine capillary structures at reduced vapor pressure.

The equilibrium moisture content varies with the temperature and humidity of the surrounding air. Consequently, any correlation of equilibrium moisture content should take these two factors into account. However, at low temperatures, *e.g.*, 60 to 120 F, a plot of equilibrium moisture content *vs.* percent relative humidity, expressed as 100 (p/p_s), is essentially independent of temperature. Such a plot usually results in a curve of double curvature with a point of inflection (see Fig. 5).

The equilibrium moisture content at a given relative humidity is not independent of temperature for all temperature ranges. As the temperature increases at a given relative humidity, the equilibrium moisture content tends to decrease. A limiting condition exists when temperatures above the boiling point of the adsorbed liquid are encountered. In such cases, relative humidity loses its significance with regard to equilibrium moisture content, and complete dryness of most hygroscopic materials is possible, even when a large amount of vapor exists in the atmosphere. This makes possible drying by means of superheated vapors.

In the special case of the dehydration of hydrated inorganic salts, such as copper sulfate, sodium sulfate, and barium chloride, temperature and humidity are very important in obtaining the desired degree of dehydration. Thus, in the drying of wet salt crystals to obtain a product with the maximum number of molecules of hydrate water, it is necessary to dry under closely controlled conditions of air temperature and humidity. Generally, the temperature is low and the humidity is high.

The equilibrium moisture content of a hygroscopic material may be determined in a number of ways. The requirement for any method is a

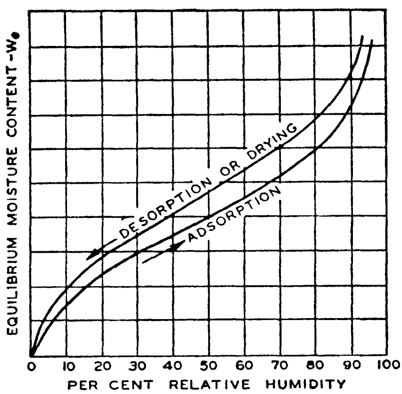

FIG. 5. TYPICAL EQUILIBRIUM MOISTURE CONTENT CURVES[1]

source of constant humidity and constant temperature air into which the sample may be placed. The determination may be made under either static or dynamic conditions, the latter being preferred if the data are to be used for drying calculations.

Probably the simplest static procedure is to place a series of samples in ordinary laboratory desiccators over sulfuric acid solutions of known concentration, which thereby produce atmospheres of known relative humidity. The sample in each desiccator is weighed periodically until a constant weight is obtained. The moisture content at this final weight represents the equilibrium moisture content for the particular relative humidity involved. The value of equilibrium moisture content so obtained will depend on whether it is reached by losing moisture, as in drying, or by gaining it, *i.e.*, whether the sample is at a moisture content higher or lower than the equilibrium value. The equilibrium moisture content reached by losing moisture, *i.e.*, by drying, is generally higher than that reached when moisture is adsorbed, as shown in Fig. 5.

The equilibrium moisture content of a solid has particular significance in drying because it represents a limiting final moisture content for specific conditions of air humidity and temperature. Drying costs can be unnecessarily high if a material is dried to a moisture content less than that which it normally possesses in equilibrium with atmospheric air. For example, if a dryer dries a material to 1 percent final moisture, and on standing under normal atmospheric humidities it regains moisture to 5 percent, the material is considered to be overdried, so that probably the dryer would be capable of a considerably higher capacity and efficiency with a 5 percent final moisture content.

Applications of Hygrometry to Drying

Drying of a solid by hot air or hot gases may be divided into two processes: (1) transfer of heat to evaporate the water, and (2) removal of the

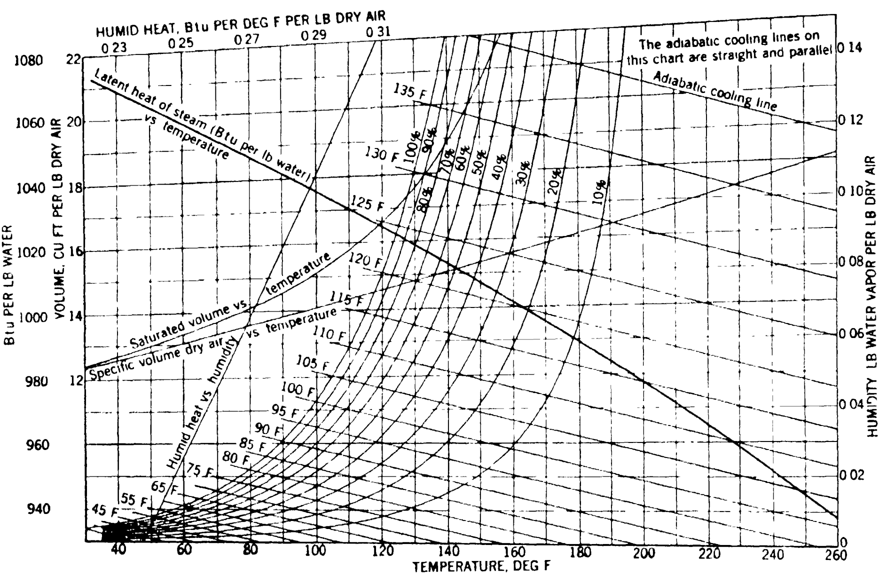

FIG. 6. PSYCHROMETRIC CHART

vapor by the air or gas stream. Likewise, two processes are involved in the *design* and *operation* of direct dryers: (1) the estimation of the drying rate or drying time, and the effect of the external variables on the drying rate; and (2) the calculation of the heat and air quantities required. The first estimates concerning drying time have been considered in the first part of this chapter. The second calculations are based on the use of the psychrometric chart, Fig. 6.

In drying, the humidity chart finds its greatest utility in analyzing the operation of existing dryers, in making design calculations, and in checking calculations of air quantities. It is equally useful in interpreting the humidity-temperature relations within the dryer. The adiabatic cooling lines on the humidity chart indicate the relation between the temperature and the humidity which are present in air passing through an adiabatic dryer, *i.e.*, one in which all of the sensible heat given up by the air in cooling is used to evaporate water from the wet stock. Referring to the section of the humidity chart shown in Fig. 7, where AB is one adiabatic saturation line, it follows that air entering an adiabatic dryer at temperature t_1

and a humidity H_1 will cool, following this cooling line toward point A. Air leaving with a humidity H_2 will consequently have cooled to t_2, the wet-bulb temperature of the air throughout the dryer being t_w. When heat is lost to the surroundings, the operation is somewhat lower than t_2, so that the actual humidity-temperature relation is represented by the line Bb, having less slope than the adiabatic saturation line. The ratio $(t_1 - t_2)/(t_1 - t_3)$ then gives a measure of the evaporative efficiency of the dryer. For the case of dryers containing steam coils maintained at a constant temperature, the humidity-temperature relation is obviously represented by the vertical line Bc, assuming the initial and final humidities to be H_1 and H_2 as before. The heat supplied within the dryer itself is usually less, but may be greater, than the total heat requirements of the dryer. If less, the cooling is indicated by some such line as Bd, and if greater, by a line such as Be having a positive slope.

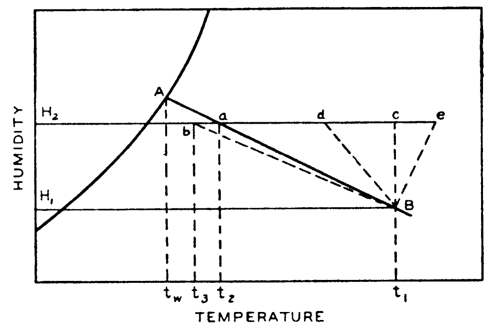

FIG. 7. HUMIDITY-TEMPERATURE RELATIONS IN DRYERS[1]

DRYER CALCULATIONS

As shown in the foregoing part of this chapter calculations for drying during the constant-rate period are different from those applying to the falling-rate period, and in contrast are subject to relatively simple mathematical analysis.

The constant rate of drying by convection is directly proportional to the temperature difference between air and wet solid, and also proportional to the 0.8 power of the air velocity as shown by Equation 3. Usually the wet surface is assumed to attain the wet-bulb temperature of the air passing over it, and evaporation takes place at a constant rate under equilibrium conditions. This is a conservative assumption, however, and when conduction and radiation effects occur, the constant rate may be increased by 30 to 60 percent over that for pure convection.

Fig. 8 permits a ready estimate of the constant drying rate for various air temperatures and humidities. The chart is based on the difference between the dry-bulb and wet-bulb temperatures of the entering stream of air, and on an air velocity of 300 fpm. It may be assumed satisfactory for tray drying of any material in the constant-rate drying period. It does not apply to rotary or through-circulation drying.

A curve for correcting the air velocity in any given problem is incorporated in Fig. 8. This curve is based on the variation of drying rate with the 0.8 power of the velocity, as given by Equation 3.

The use of Fig. 8 in practical drying problems is as follows: Since the drying conditions of temperature and relative humidity are fixed, the corresponding absolute drying rate is read from Fig. 8. This value is then multiplied by the correction factor corresponding to the air velocity employed. The rate so obtained, however, does not include any effects of radiation or of conduction through unwetted surfaces. These effects tend to increase the rate of evaporation so that the chart is conservative.[3]

It has been demonstrated empirically for certain materials that the rate of drying during the falling-rate period is approximately proportional to the free water content of the material. Actual calculations of drying time

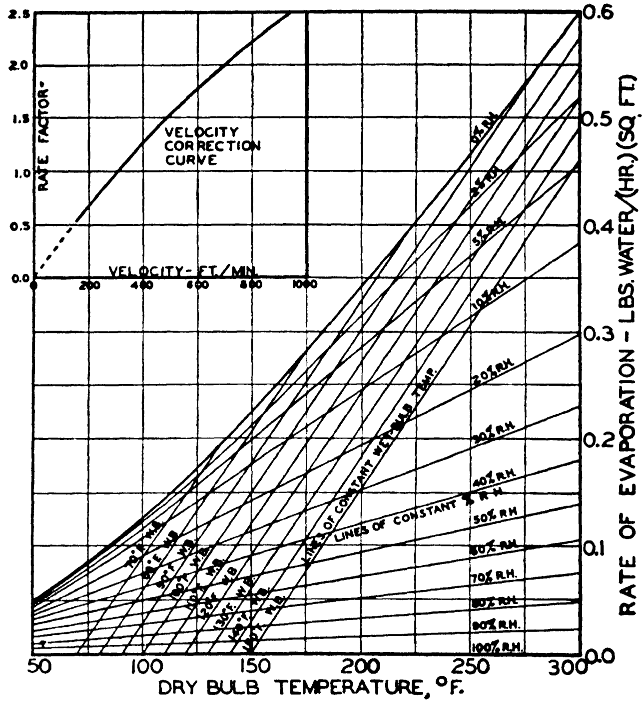

FIG. 8. RATE OF EVAPORATION CHART[3]

during the falling-rate period for this case require only a knowledge of the critical moisture content and the constant rate. For other cases of the falling-rate period, calculations are not feasible. Consequently, it is best to determine drying times for design purposes by means of pilot tests. However, when tests are not feasible, drying times may often be estimated approximately from Equations 22 and 23.

The following nomenclature will be used in the discussion of design calculations:

H = humidity ratio of air, pounds of water vapor per pound of dry air.
N_a = pounds of dry air supplied to the dryer per unit of time.
S = pounds of stock dried per unit of time in a continuous dryer.
S' = pounds of stock charged per batch to a discontinuous dryer.
θ = time, hours.
Q = total heat supplied to the drver, Btu

t = air temperature, Fahrenheit degrees.
t' = stock temperature, Fahrenheit degrees.
t'' = average stock temperature over short time interval, in a batch dryer, Fahrenheit degrees.
t_w = wet-bulb temperature, Fahrenheit degrees.
s_1 = specific heat of the stock, Btu per pound.
Q_{rc} = total radiation and conduction losses, Btu per hour.
W = pounds of water per pound of dry stock.
λ = heat of evaporation of water, Btu per pound.
c_s = humid heat of air, *i.e.*, heat necessary to raise 1 lb of dry air + H lb of steam 1 F deg.

Subscript (1) designates conditions at the point where the material in question (air or stock) enters, and (2) where it leaves the dryer.

Air dryers may be divided into two classes, batch and continuous.

In any continuously operating dryer, the relation between moisture content of the stock and quantity of air required for the drying operation is given by the equation

$$N_a\ (H_2 - H_1) = S(W_1 - W_2) \tag{24}$$

where H_2 is constant.

In discontinuous dryers, the drying operation is given by the equation

$$N_a(H_2 - H_1) = S' \frac{dW}{d\theta} \tag{25}$$

where H_2 is a variable during a portion of the cycle.

In the continuous dryer, the heat consumption per unit time is

$$\frac{Q}{\theta} = N_a c_{s1}(t_2 - t_1) + N_a(\lambda_2 + t_2 - t'_2)(H_2 - H_1) + S(t'_2 - t'_1)(s_1 + W_1) + Q_{rc} \tag{26}$$

Equation 26 assumes continuity of operation. For charge or batch operations, the total time of the drying cycle may be broken up into a number of periods, sufficiently short so that over each period average values of t, t' and H may be employed, provided the third term of the right hand member of the equation is modified to read:

$$S'(t''_2 - t''_1)\ (s_1 - W_1)$$

and in the second term t'_2 be replaced by

$$\frac{t'_1 + t''_2}{2}$$

Theoretically, these periods should be very short and the equation integrated. Practically, the error introduced by using a small number of long periods and employing average values of the variables over each, is not serious. The evaluation of Equation 25 may be approximated in a similar manner.

The first term of the right hand member of Equation 26 represents heat lost as sensible heat in the effluent air. In many drying operations this becomes excessive. Each pound of air supplied should remove the maximum amount of moisture. This is best accomplished by bringing the air into contact with the stock with sufficient intimacy so that the air leaving the dryer is saturated, or nearly so. Counter-current, as against parallel, flow of air and stock gives rise to optimum operating conditions, resulting in a minimum quantity of air required (N_a), and a corresponding minimum loss, as sensible heat, in the exit air. Similarly, continuous operation is superior to intermittent operation.

Despite the fact that the sensible heat loss increases with the rise in temperature of the air, the percentage of heat lost from this source decreases if the increase in moisture carrying capacity of the air (due to high temperature) is actually utilized. To secure maximum thermal efficiency in drying, a high drying temperature and high saturation of the outlet air are imperative.

The second term of the right member of Equation 26 represents the latent heat of evaporation of the water plus the heat to raise this water to the temperature of evaporation. The third term of the equation represents heat to raise the temperature of the stock plus the water which remains unevaporated in the stock.

The changes taking place in the air during the drying process can be illustrated on the skeleton psychrometric chart, Fig. 9. The case illus-

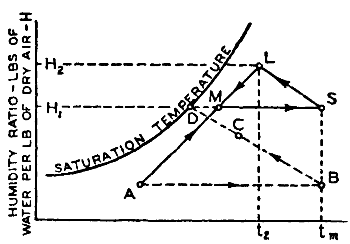

FIG. 9. CHANGES IN AIR DURING DRYING PROCESS

trated is typical of tunnel and rotary dryers where heat is applied to the air at one point only. After the first adjustment stage, during which both the material and the dryer reach the working temperature, the only heat losses from the dryer are those of radiation and conduction from the housing, and these are practically negligible for an insulated dryer. Hence, the drying process can be considered to be adiabatic.

If 100 percent outside air is used, the air can be considered to enter at point A, Fig. 9 (the prevailing outside air condition), and be heated to point B (the maximum permissible temperature t_m or the temperature determined by previous test). As the air evaporates moisture, it cools along the constant wet-bulb line BD to point C. The difference between the moisture content of air at B and at C represents the moisture *pick up* of the air. The maximum possible *pick up* from B to D is never achieved in practical dryers, the actual *pick up* being anywhere from 10 to 75 percent of the maximum.

In order to conserve heat and to control the wet-bulb temperature at which the drying takes place, recirculation is used. The process is shown on Fig. 9. The outside air at A is mixed with recirculated air until the moisture level is raised to the desired point. The mixture is represented at point M, the heaters heat the mixture to the desired dry-bulb temperature t_m at point S. The moisture is picked up from S to L. Point L is the condition at which air is exhausted.

Actual dryer operation is somewhat more complicated because even if radiation and conduction losses are neglected, the wet-bulb temperature of the air remains constant only as long as surface evaporation of water is taking place. When sub-surface evaporation is occurring, some heat from the air is used to heat the material and hence, there is a drop in wet-bulb temperature. Fig. 10 illustrates the drying process in a tunnel dryer in which the air is flowing parallel to the product.[2]

In design calculations using Equation 26, the following steps outline the procedure:

1. The unit drying rate, pounds of water per hour, is determined from the experimental drying time curve and the amount of product to be dried per hour. The drying time may also be approximated from previous experience.

2. The experimental data or experience also determine the drying condition, *i.e.*, point L Fig. 9. This fixes H_2. Where experimental data are lacking L may be approximated from Regain Tables (see chapter Industrial Air Conditioning) since the relationship between the vapor pressure in the product and in the air at equilibrium for the desired final moisture content must prevail in the dryer. The air temperature must be not greater than the maximum permissible product temperature.

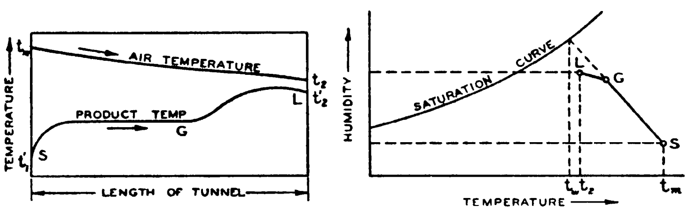

Fig. 10. Temperature and Moisture Conditions in a Tunnel Dryer
Parallel Flow Air and Product

3. The rate of air circulation N_a must be determined and also the supply air condition S. In the design of some dryers, such as rotary or tunnel types, it is customary to determine S first and then to calculate the air rate N_a. In other types of dryers, such as tray dryers or through circulation dryers, where a fixed air velocity is maintained, N_a is calculated first and then point S is found. Because of the many variables involved, it is generally not possible to select S except on the basis of past experience or on the basis of experimental drying tests.

4. The prevailing outside air conditions establish point A and hence, the line A L. The percent of recirculated air can then be calculated.

5. The physical arrangement of the dryer must then be selected to handle the desired quantity of product, and at the same time circulate the calculated air quantity at the desired velocity.

6. Equation 26 can then be used to calculate the heat requirements.

DRYING METHODS AND EQUIPMENT

Drying systems are sometimes classified according to the method of heat transfer that is employed, since the entire problem of drying resolves itself into individual problems of heat transfer and the thermodynamics of air and water vapor. The methods of heat transfer are radiation, conduction and convection. Many types of dryers have been built on these principles for different purposes.

Drying systems can also be classified, according to the method of product handling, as batch operation, semi-continuous and continuous.

Radiant Drying

Sun drying, the oldest form known to man, is still practiced where the material is amenable to such treatment, where the necessary time can be allowed, and where there is little danger of rain or atmospheric pollution.

In artificial systems, radiating surfaces, heated by steam, electricity or other means, afford a good method of heat distribution and control. Radiant heating sets up convection currents, and in low-temperature dryers only about one-third to one-half of the total heat for evaporation is actually supplied to the material by radiation. At high temperatures the radiation output increases rapidly, according to the *fourth-power law.* The total radiation may be computed by the equations and tables given in Chapter 5. In general, fins and irregular surfaces do not increase radiation, hence, the area to be used in calculations is the area of a smooth-surface envelope enclosing the radiating elements.

A certain amount of air circulation is required through a radiant dryer in order to carry off the vapor.

Radiant heat from infra-red lamps has been accepted by certain industries as practicable for their specific problems. An example of successful application is found in the drying of lacquers.

Lacquers and similar surface films can be very effectively dried by radiation. Special electric lamp units have been developed which give off a high percentage of infra-red and similar heat rays. For continuous manufacturing processes these units are mounted in tunnels through which conveyors pass. For local applications, as for example paint drying in automobile repair shops, they may be mounted on portable racks. Objects of relatively large surface area in proportion to their weight, and fabricated materials having a rather high heat absorption, may be satisfactorily heated by such a source.

For drying, baking, pre-heating and de-hydrating, where a low temperature infra-red heat source is desired, or where use of glass-enclosed radiant lamps is objectionable for safety reasons, electric heating units employing low temperature metal sheathed resistors, are available.

Conduction Drying

Drying rolls or drums, Fig. 11,[8] flat surfaces, open kettles and immersion heaters are examples of the direct-contact method. Intimate contact of the material with the heating surface is important, and in some cases agitation is desirable to increase the uniformity of heating or to prevent overheating.

Greatest resistance to heat transfer occurs on the air side of the material being dried. The rate of heat transfer from the surface of the heated material to the air, and hence the rate of drying, may be increased by (a) forced convection or air circulation, and (b) vacuum operation to lower the boiling point of the liquid being evaporated.

A rather interesting method of conduction drying was put into practical use during the war for the drying of blood plasma, and has since been expanded to other fields such as the preservation of bacteria and other micro-organisms. This has come to be known as freeze drying or drying by sublimation. The material to be dried is first frozen and then placed in a high vacuum chamber connected to extremely low temperature condensers. The water is removed by vaporizing from the solid directly to the gas without ever becoming liquid.

Convection Drying (Direct Dryers)

A limited amount of convection drying takes place in almost any dryer such as those described in the preceding paragraphs. However, to be

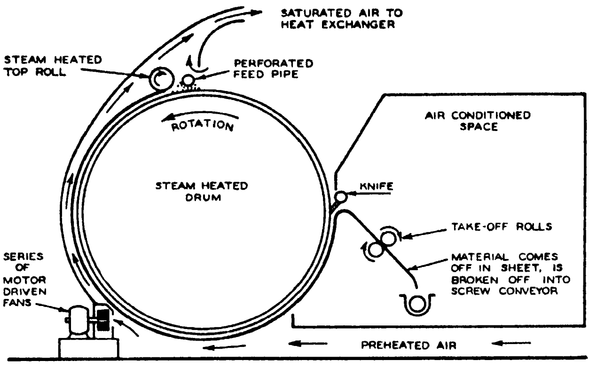

FIG. 11. DRUM DRYER[8]

classified as a convection dryer the principal source of heat is the heated air or other gases circulated in the dryer. There are a number of mechanical means of accomplishing this circulation of air or gases, each of which has some particular virtue. Brief descriptions of some important types of convection dryers follow:

Rotary Dryers. These dryers are cylindrical drums which cascade the material being dried through the air stream. (See Fig. 12). The driers may be heated directly or indirectly, and the air circulation may be parallel or counter-flow. A variation is the rotating louver type dryer, which introduces the air beneath the flights thus securing very intimate contact.

Cabinet and Compartment Dryers. These are generally considered batch dryers wherein each charge is dried to completion before removal.[8] A wide range includes types from the heated loft with only natural convection, and usually poor and non-uniform drying, to the self-contained units with forced draft and properly designed baffles which give positive results. It is also possible to evacuate some of the systems for low temperature drying of delicate or hygroscopic materials. These dryers are usually loaded with material spread in trays to increase the exposed surface. The trays are loaded directly into the dryer or may be stacked on trucks which are wheeled in. (See Fig. 13).

Tunnel Dryers. Tunnel dryers are a modification of the compartment dryer, and as a rule are continuous or semi-continuous in operation. Heated air or combustion gas is usually circulated by means of fans, although a few natural draft units are still in use. The material is handled on trays or racks on trucks, and moves through the dryer either intermittently or continuously. The air flow may be parallel, counter-flow or a combination of the two, obtained by center exhaust. Further, the air flow may be across the surface of the trays or up or down through the bed, or in any

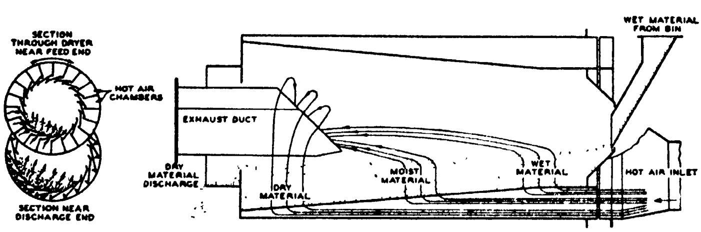

FIG. 12. CROSS SECTION AND LONGITUDINAL SECTION THROUGH CIRCULATION DRYER[8]

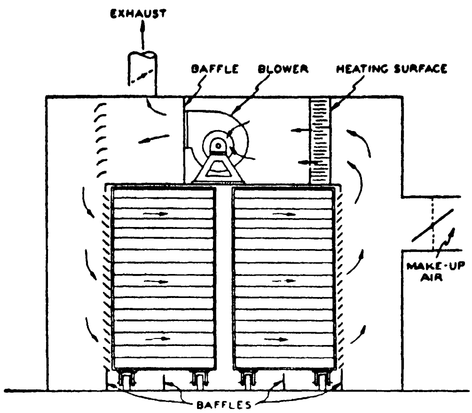

FIG. 13. COMPARTMENT DRYER, SHOWING TRUCKS, WITH AIR CIRCULATION[8]

combination of directions. By reheating the air in this type of dryer or recirculating it, a high degree of saturation is achieved before exhausting the air. This reduces the waste of sensible heat.

A variation of this type dryer is the strictly continuous type having one or more mesh belts which travel through the dryer carrying the product, such as Fig. 14. Innumerable combinations of temperature, humidity and air direction and velocity are possible. The labor requirement is low on such a dryer, as it can be loaded and unloaded mechanically. There is the disadvantage of hot air leaks at the entrance and exit, although these can be minimized by means of baffles or inclined ends where the material enters and leaves from the bottom.

Spray Dryers. In recent years the spray dryer has become important for the drying of liquids in many fields, especially in the food industry.

The liquid is atomized by means of pressure nozzles, air jets or centrifugal bowls into the air stream of a tower or chamber. Inlet air temperatures may run from 250–300 F up as high as 1200 F. Drying times are very short because of the minute particle size. Particles as small as 5 to 10 microns are formed in spray dryers. The dry powder is separated from the air by cyclone separators which are sometimes followed by cloth bags or scrubbing towers.

Because of the high inlet temperatures and the relatively large volume of air required, the efficiency of the spray dryer is not too good and, consequently, is seldom used for dilute solutions (less than 30 percent solids).

Fig. 15 shows a typical arrangement for a spray drying system.[8]

A common and important feature of all spray processing is the direct conversion of the spray liquid to a granular product suitable for packaging

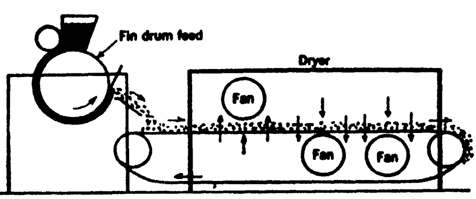

FIG. 14. SECTION OF CONTINUOUS DRYER, BLOW-THROUGH TYPE

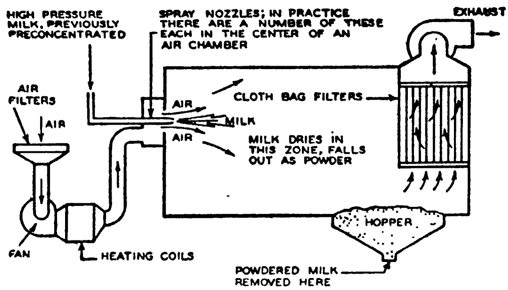

FIG. 15. SPRAY DRYER OF THE PRESSURE-SPRAY ROTARY TYPE[8]

without grinding or other intermediate handling. Another aspect is the unusually high rate of drying attained. In a well designed system 15 to 30 seconds is a fair time for the passage of the sprayed particle through the drying zone; the particle temperature need not rise materially above the wet-bulb temperature of the drying air. This makes the process particularly adaptable to the drying of heat-sensitive material, some of its most important applications being the drying of milk, eggs, potato flour, soap and blood.[9]

SOLUTION OF TYPICAL DRYING PROBLEM

Since there are so many types of dryers which may be used, and so many special conditions surrounding each particular problem, it is usually recommended that those having experience with the dryer to be used be consulted. The following example, however, will serve as a guide for typical dryer calculations.[10]

Example 1: Assume 900 lb per hour of ceramic powder is to be produced. The powder has a specific heat of 0.22 and density of 98 lb per cu ft, wet. Initial moisture content is 19 percent on a wet basis; final moisture content is to be one-half of one percent on a wet basis.

A continuous belt dryer is a logical choice, and previous experience indicates that rubber belts will withstand temperatures up to 200 F, which is also about the highest desirable product temperature. Experience also indicates that a drying time of 45 min is possible at about 160 F dry-bulb and 100 F wet-bulb.

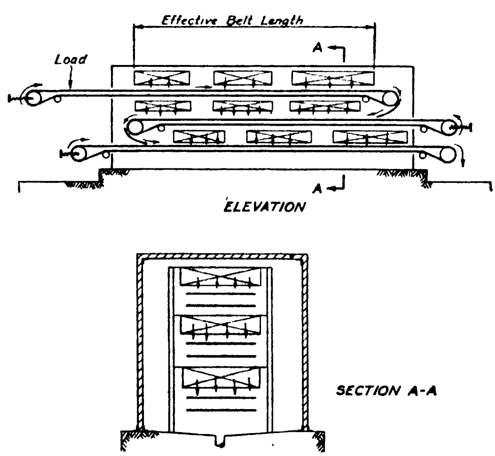

FIG. 16. CONTINUOUS BELT DRYER FOR CERAMIC POWDER[10]

Step 1: Let x = pounds moisture at final condition.

Then, $$\frac{x}{900} = 0.005$$

or, $$x = 4.5 \text{ lb moisture}$$

and therefore the solid will amount to 895.5 lb.

Likewise the weight of the initial moisture x can be found

from $$\frac{x}{895.5 + x} = 0.19$$

or, $$x = 210 \text{ lb.}$$

The weight of moisture to be removed is 205.5 lb per hour, and wet material entering dryer is 1105.5 lb per hr.

Step 2: Previous tests indicate that a ½ in. layer of powder gives satisfactory results, and that a desirable air velocity is 50 fpm applied at a right angle to the belt. Based on 45 min (¾ hr) drying time, the dryer holding capacity will have to be $1105.5 \times 0.75 = 830$ lb of wet material or $\frac{830}{98} = 8.45$ cu ft of material.

Assuming a 48 in. wide belt with an effective 42 in. width for the product, each foot of belt length carries 0.146 cu ft or 14.3 lb of wet material. Hence, the linear feet of belt must be $\frac{830}{14.3} = 58$ ft and the total area of exposed product is 232 sq ft. Based on 50 fpm velocity directed at a right angle to the belt the total air circulation will be $232 \times 50 = 11{,}600$ cfm.

For space economy and in order to expose periodically fresh layers of powder to air, a multiple vertical belt arrangement (Fig. 16) with belts traveling in opposite directions, is selected. Three belts each 19.4 ft long will be required. Fig. 16 illustrates the physical arrangement of the dryer.[10] The housing will be about 25 ft long, 8 ft wide and 7 ft high.

Step 3: At the drying conditions of 160 F dry-bulb and 100 F wet-bulb, the air moisture content (from Fig. 6) is 0.028 lb per pound of air. Make-up air will be assumed at 80 F dry-bulb and 72 F dew-point (summer weather), or a humidity of 0.0168 lb per pound of air. The *pick-up* is therefore 0.0280 − 0.0168 or 0.0112 lb per pound of air. Then $\frac{205.5}{0.0112} = 18400$ lb of air per hour or 307 lb per min.

At the elevated temperature, the total air quantity of 11,600 cfm represents 675 lb of air per min. Hence,

Make-up air = 46 percent

Recirculated air = 54 percent

Step 4: Although the drying condition and drying rate should preferably be determined from experience or test results, the drying conditions can sometimes be estimated if the regain characteristics of the product or a similar product are known. In this case the regain characteristics of clay could be used as a guide. Inspection of regain data for typical clays indicates that at about one-half of one percent the vapor pressure of the moisture in the product is about 0.7 in. Hg less than the vapor pressure of free moisture at the product temperature. An assumption is made that the product temperature approximates the air wet-bulb temperature. If an assumption is made regarding the percent recirculated air the desired vapor pressure in the dryer can be estimated. For example assume a use of 46 percent outside air or 307 lb per min. Then the moisture *pick-up* $= \frac{205.5}{60 \times 307} = 0.0112$ lb per pound of air, with a consequent total of $0.0168 + 0.0112 = 0.0280$ lb of moisture per pound of air in the leaving outside air, which has about 1.25 in. Hg vapor pressure. This is assumed to be the vapor pressure of the moisture in the product, and thus the vapor pressure of free water at product temperature can be $1.25 + 0.7 = 1.95$ in. Hg. The temperature corresponding to 1.95 in. Hg is 100 F, and thus the air wet-bulb can be estimated to be 100 F. At 100 F wet-bulb temperature and 0.028 lb moisture per pound of air, the dry-bulb temperature is 160 F. Obviously, the assumed percentage of recirculated air affects the results, and therefore it is important that it be based on experience. About 50 percent recirculation is reasonable for the type dryer considered in this example.

Step 5: The *pick-up* of moisture per pound for the total air circulated is $\frac{205.5}{60 \times 675}$ = 0.0051 lb. 0.0280 − 0.0051 = 0.0229 lb moisture per lb of air for the supply air.

Assuming an existing wet-bulb of 100 F, the supply air dry-bulb will be 182 F. The mixture of recirculated air at 160 F dry-bulb and 100 F wet-bulb, with outside air at 80 F dry-bulb and 72 F dew-point, will be at approximately 120 F dry-bulb and 89 F wet-bulb.

Step 6: The heat required may be determined from Equation 26 by substitution of the following values: N_a = 307 × 60 = 18420 lb of air per hr; S = 900 lb; $c_s = 24 + 0.45\left(\frac{0.028 + 0.0229}{2}\right) = 0.251$; t_1 = 80 F; t_2 = 160 F; t_2' = 100 F; λ = 1100 (approx.); W = 0.005 lb; s_1 = 0.22.

$$Q = 18420\,(0.251)\,(160-80) + 18420\,(1100 + 160 - 100)\,(0.028 - 0.0168) + 900\,(100 - 80)\,(0.22 + 0.005) + Q_{rc}$$
$$= 609{,}000 \text{ Btu per hr} + Q_{rc}$$

The heat input requirement is therefore 609,000 Btu per hr plus radiation and convection losses (Q_{rc}) which may be computed from the known construction of the dryer surfaces and the heat transfer coefficients.

Summer conditions were used in *Example 1* in order to obtain the maximum heat requirement which would be the case, except under the unusual condition where radiation and conduction losses are a large percentage of the total. In winter it is usually possible to take advantage of drier makeup air, and either speed up the process or operate at a lower dry-bulb temperature.

Controls for the system selected for *Example 1* would consist of a thermostat in the main return air duct controlling the heat input to maintain constant dry-bulb temperature. A wet-bulb controller in the return circulating duct would maintain constant desired wet-bulb temperature by simultaneous positioning of three sets of dampers in the makeup air, the exhaust air and the recirculated air ducts.

LETTER SYMBOLS USED IN CHAPTER 46

A = area of heat transfer and evaporation, square feet.
a = drying area, square feet per cubic foot of bed volume
B = a constant (for use in Equation 22).
B' = a constant (for use in Equation 23).
c_s = humid heat, Btu per (pound of dry air) (Fahrenheit degree).
d = diffusivity of the liquid or vapor, square feet per hour.
D_p = average diameter of particle, feet.
D_{p1} = drop diameter at start of evaporation, feet.
D_{p2} = drop diameter of dry particle, feet.
e = Naperian base of logarithms = 2.718.
G = mass velocity of dry air, pounds per (hour) (square foot).
ΔH_m = logarithmic mean of inlet and outlet humidity driving force across the air film adjacent to the particle through which the water vapor diffuses, pound per pound. (The surface humidity is taken as the humidity corresponding to the wet-bulb temperature of the drying air).
H_1 = humidity ratio of entering air, pounds of water vapor per pound of dry air.
H_2 = humidity ratio of leaving air, pounds of water vapor per pound of dry air.
h = film heat transfer coefficient, Btu per (hour) (square foot) (Fahrenheit degree).
h_c = coefficient of heat transfer by convection, Btu per (hour) (square foot) (Fahrenheit degree).
h_r = coefficient of heat transfer by radiation, Btu per (hour) (square foot) (Fahrenheit degree).
h_t = total gas film heat transfer coefficient, Btu per (hour) (square foot) (Fahrenheit degree).
K = a constant (a function of the constant drying rate).

k_f = gas film thermal conductivity, Btu per (hour) (square foot) (Fahrenheit degree per foot).
k_g = mass transfer coefficient, pounds per (hour) (square foot) (atmosphere).
L = material thickness, feet.
M = molecular weight of the diffusing vapor.
N_a = dry air supplied to the dryer, pounds per hour.
Δp = $p_s - p_a$ = vapor pressure difference, atmospheres.
p_s = vapor pressure of water at t_s, atmospheres.
= vapor pressure at the particle surface corresponding to the liquid temperature, atmospheres.
p_2 = partial pressure of water vapor in air, atmospheres.
p_a = vapor pressure of liquid in the drying medium, atmospheres.
Q_{rc} = radiation and conduction loss, Btu per hour.
Q = total heat supplied to dryer, Btu.
q = rate of heat transfer, Btu per hour.
R = gas constant (cubic feet) (atmospheres) per (Fahrenheit degree).
S = weight of stock dried in a continuous dryer, pounds per hour.
S' = weight of stock charged in a discontinuous dryer, pounds per batch.
s = specific heat of stock, Btu per pound.
T = absolute temperature of the gas, Fahrenheit degrees.
t_a = air or gas temperature, Fahrenheit degrees.
t_h = temperature of the heating medium, Fahrenheit degrees.
t_s = temperature of particle, solid or surface of evaporation, Fahrenheit degrees.
t_w = wet-bulb temperature of drying air, Fahrenheit degrees.
t_1 = entering air temperature, Fahrenheit degrees.
t_2 = leaving air temperature, Fahrenheit degrees.
t_1' = entering stock temperature, Fahrenheit degrees.
t_2' = leaving stock temperature, Fahrenheit degrees.
t'' = average stock temperature over short interval of time, in batch dryer (t_1'' = entering, t_2'' = leaving) Fahrenheit degrees.
Δt = $(t_a - t_s)$ = temperature difference between air and surface of evaporation, Fahrenheit degrees.
Δt_m = logarithmic mean between temperature entering and leaving the bed, and the wet-bulb temperature, Fahrenheit degrees.
U = overall heat transfer coefficient, Btu per (hour) (square foot) (Fahrenheit degrees temperature difference between heating medium and product).
W = moisture content on dry basis at any time θ, pounds of water per pound
W_c = critical moisture content, pounds water per pound dry material.
W_d = water content, dry basis, of the drop as it enters the drying chamber, pounds per pound of dry solid.
W_e = moisture content at equilibrium with external conditions, pounds per pound dry material.
W_o = initial moisture or moisture content at start of diffusional period, pounds per pound.
w = pounds of water.
$\frac{dW}{d\theta}$ = drying rate, pounds of water per (hour) (pound dry material).
$\left(\frac{dW}{d\theta}\right)_c$ = constant drying rate, pounds per (hour) (pound dry material).
$\left(\frac{dW}{d\theta}\right)_f$ = falling rate, pounds water per (hour) (pound of dry stock).
$\frac{dw}{d\theta}$ = drying rate or rate of evaporation, pounds of water per hour (Eq. 1).
θ = time, hours.
θ_c = drying time for constant rate period, hours.
θ_f = drying time during falling rate period, hours.
θ_t = total drying time, hours.
λ = latent heat of evaporation at t_s, Btu per pound.
μ = viscosity of the air stream, pounds per (hour) (square foot).
ρ_L = density of evaporating liquid, pounds per cubic foot.
ρ_s = bulk density of dry granular bed, density of dry particle, pounds per cubic foot.

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CHAPTER 47

TRANSPORTATION AIR CONDITIONING

Railway Passenger Car Air Conditioning; Streetcar and Trolley Coach Heating and Ventilating; Passenger Bus Air Conditioning; Automobile Air Conditioning; Aircraft Air Conditioning; Ship Air Conditioning, Heating and Ventilating, Refrigeration Systems, Air Condition Space Treatment, Systems and Controls

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THE principles of air conditioning applying to stores, restaurants, hospitals, theaters, and homes are applicable to railway passenger cars, passenger buses, automobiles, streetcars, trolley coaches, airplanes and ships. However, equipment used for mobile applications differs from that used for stationary purposes in that it must meet additional requirements. Equipment must be compact, accessible for quick inspection and servicing, light-weight and unaffected by vibration and impact. Freedom from vibration which could be transmitted to supporting vehicle and thus to passengers, is essential.

RAILWAY PASSENGER CAR AIR CONDITIONING

The railway passenger car represents a very difficult air conditioning problem. Space is strictly limited so that all equipment and ducts must be reduced to minimum size. Electric power supply and water supply also are limited. All equipment must withstand severe vibration and shock, and must be very reliable since servicing points are frequently far apart.

During the heating season it is necessary to heat conventional cars with steam from the locomotive at pressures that may vary from 250 psig to only 5 or 10 psig on the last car in long trains. Passengers in window seats must sit only a few inches from cold outside walls and windows, and must also be very close to standing radiation installed along sides of cars. Sudden changes in load may be caused by changes in sun, wind, or train movement. Even in coldest weather, outside doors must be opened frequently.

During the cooling season, the problem is further complicated by a high concentrated internal load due to the passengers. Also, air distribution problems are increased by low ceilings and short air throws.

Heating

The heating of passenger cars is accomplished by using a split system consisting of an overhead air circulating system with heating and cooling coils, and standing radiation (floor heat) along car sides. The floor heaters, which usually consist of finned tubing, may be made more efficient by using covers designed to increase gravity air circulation, and to direct the warm air from finned heating surface along cold outside walls and car windows. In some new cars, wall convector panels are used and extend the full length of the car, with air intakes along the floor and outlets at window sill height and at window head height in dead-light panels. The heated panel protects passengers from cold outside walls, and the chimney effect of the panel duct increases air flow and improves heating surface efficiency.

Steam may be used directly in finned tubing, or steam may be used to heat a liquid (usually a mixture of water and diethylene glycol) which is circulated through the finned tubing by means of circulators. If steam is used directly, it is difficult to distribute the heat uniformly along the length of the car. In the case of steam, it is customary to divide long finned tubes into separate sections which are fed independently. This is not true zoning since it is not based on principles for zoning. (See Chapter 29.)

The finned tubing at the floor must have sufficient capacity to offset effects of cold walls and windows during normal operation, and to heat the entire car to a minimum temperature of 60 F during standby when the overhead system is not operating. The maximum capacity required (determined by standby requirements) varies with car construction and design temperatures, but is approximately 90,000 Btu per hour. This requires a heating capacity in finned tube of approximately 650 Btu per linear foot.

The overhead air heating coil must have sufficient capacity to heat the outside air brought into the car for ventilation, and to supply approximately 20 percent of the internal heat loss of the car so as to permit supply of floor heat at all times at an output that will not be objectionable to passengers sitting near it. The usual capacity of the overhead heating coil is approximately 100,000 Btu based on 2400 cfm of circulated air, with 600 cfm of this being outside air for ventilation. All Btu figures are approximations of actual heating requirements, and do not include heat losses in the trainline (or leakage) or losses in the undercar piping. The heat losses in the undercar piping can become a major portion of the train heating boiler load on cars having many undercar loops and steam regulating devices.

Refrigeration

For cooling and dehumidification during summer, refrigeration may be obtained from ice bunkers, steam jet systems, or mechanical compressors (driven directly from car axle by electric motors or by gas engines). Refrigeration required varies with load conditions, but 7½ tons per car is one capacity frequently used. Evaporative type condensers are sometimes used in combination with the usual air condenser on either steam jet or mechanical refrigeration.

When an electric motor (approx. 10 hp) is used to drive the air conditioning compressor, the electric power source is a problem. If the power source is an axle-driven generator, there is an appreciable increase in the drag on locomotive, and the power available for refrigeration, when train is stopped at a station, is limited to storage batteries. One solution to this problem is to use a d-c generator driven by a gas engine. Another is to use a Diesel-driven alternator in a special head-end car to furnish power to the entire train. Recently there have been installations in which a Diesel-driven alternator is mounted on an individual passenger car to supply the power requirements of the car. The attractiveness of this type of installation can be increased by utilizing spare alternator capacity in winter for electric heating. If this capacity is supplemented by exhaust heat from the Diesel engine, there is sufficient capacity to heat the entire car and to provide hot water for wash rooms when outside temperatures are above approximately 30 F. This feature is important on trains using Diesel locomotives, since it eliminates the need for firing the steam heating boiler in the locomotive during a portion of the year.

Air Distribution and Cleaning

Railway cars present critical problems in air distribution because air space per passenger is small (60 to 190 cu ft), and the sun load is great.

An average passenger car contains approximately 5000 cu ft of air, and may seat as many as 80 passengers. The occupants are continually liberating heat, carbon dioxide, moisture, odors, and some organic matter from their breath, skin and clothing. The heat and moisture can be removed by cooling and dehumidification, but other constituents can be successfully handled only by proper ventilation and air cleansing. In an average car, from 2000 to 2500 cfm are circulated by the air conditioning unit. Some of this air may be recirculated, but a portion of it should always be brought in from outside. The amount of outside air desirable depends upon the type of car, number of passengers, air temperature, humidity, odors, and whether or not the occupants are smoking, and will vary from 15 to 90 percent of the total air circulated.

Careful attention must be exercised in specifying the rate of outside air taken in so as to fit the type of service adequately, and yet not supply more ventilation than is necessary. Conditioning of this outside air is a major factor in determining size of both summer and winter equipment.

For normal conditions, 10 cfm of outside air per passenger are provided. When smoking is permitted, at least 15 cfm should be admitted. In some dining cars, and deluxe sleeping cars, outside air rates as high as 20 to 30 cfm per occupant are used. A ceiling duct lengthwise along the center of the car is usually used to distribute the air to the interior by fans or blowers. A perforated ceiling supplied from an overhead duct, or delivery grilles and plaques designed to give considerable entrainment and mixing, are used to deliver air to the car space.

Smoking rooms present a special problem. The cloud of smoke that usually hangs near the ceiling can be broken up by directing incoming air along the ceiling at a velocity somewhat higher than that used for the rest of car. The air is exhausted through the washroom or lavatory. For compartments, provision is made in the door or partition for removal of used air. Lower berths are provided with a low velocity air outlet.

Recirculating air grilles are usually of straight flow types. Outside air intakes are usually located in the vestibule, on the side of the car, or on the roof, depending upon location of the cooling coils. On many recently air-conditioned cars, there are no dampers or shutters at the outside air intakes; the percentage of the outside air is controlled by adjusting flow through the recirculating grille.

A considerable number of coach cars are now being equipped with return air ducts fitted in the structure of baggage racks. Part of the air circulated is returned to the blower unit through these ducts, and part through the car body. This arrangement reduces quantity and velocity of air returning through the car body, and removes smoke fumes at the source. This, and any other design features aimed at taking recirculated air at the floor and adjacent to both end doors (rather than drawing all recirculated air to one end of the car) also reduces infiltration of cold air in ankle height strata, when doors are opened during the heating season.

All air circulated by the blower is filtered before passing over the cooling and heating coils. In some cars outside air and recirculated air are filtered separately before mixing, while in others air from the two sources is mixed before passing through a common filter. Filters in use are combinations of metal, wool, cloth, spun glass, hemp, paper, hair and wire screen. Most filters have a viscous coating of oil for greater cleaning efficiency. Some types may be cleaned, re-treated, and re-used, while other types are discarded when dirty. Applications are also being made of electric precipitation for air cleaning. In this system the coarser particles are removed from the air by mechanical separation; finer materials, by electrostatic action.

Activated carbon units sometimes are used in addition to the regular filters for adsorbing odors and other impurities, thus reducing the amount of outside air necessary for ventilation.

Temperature and Humidity Control

Controls in a passenger car should be as automatic as possible. The regular train crew cannot be relied on to make adjustments for the comfort of passengers. For this reason the latest systems of temperature control have only an off-on switch to be operated by the train crew. When the system is in operation, heating or cooling is provided automatically as required.

When heating, it is important that floor-heat finned tubing be controlled at stable temperatures. Wide fluctuation in its temperature is highly objectionable because of location close to the passengers. Stable operation may be secured by controlling the floor heat on the basis of outside conditions (see Chapter 29 on zoning), and using an overhead air circulating system to maintain final car temperatures.

Because of window condensation and other problems, usually no attempt is made to raise relative humidity in a railroad car in winter time.

When cooling, the steam jet refrigeration system is controlled in an on-off manner. Some means are ordinarily provided for operating mechanical compressor systems at partial capacity. In this case split evaporators are used, so that evaporator surface and compressor capacity can be reduced together under light load conditions.

Under very light cooling loads, the relative humidity in a car tends to increase because of long off periods of refrigeration equipment. This can be prevented by starting refrigeration equipment on low capacity at an established outdoor air temperature, operating it continuously, and using a heating coil in the overhead system to re-heat sufficiently to maintain desired car temperatures. Under higher load conditions the heating coil becomes inoperative, and compressor and evaporator capacity are increased as needed.

Room type sleeper cars introduce a further problem of providing individual adjustment of room temperatures. Sometimes this individual control is secured by adjusting air volume, but such adjustment is unsatisfactory for overall comfort, and may affect the air supply to other rooms. Another method is to use the heaters at the floor to control the room temperature, but this tends to cause unstable and improper floor heat temperatures which may be objectionable to the passengers. A simpler and basically more satisfactory system is to use a small booster heater in individual overhead air supply ducts to each room under manual control of the occupant. In this case, the floor heat and basic overhead systems are automatically adjusted for varying load conditions just as in a simple coach type car. A fixed amount of heat regulated by the occupant can be added by the room booster heater to maintain desired individual room temperature. Temperature lag is less when room boosters are used instead of gravity floor heating control. Any adjustment of the booster will give the occupant immediate change in space conditions. The use of floor heat surfaces for room control may also cause low or excessive surface temperatures close to the passenger, with resulting discomfort.

STREETCAR AND TROLLEY COACH HEATING AND VENTILATING

Streetcars and trolley coaches present a special problem in the maintenance of satisfactory comfort conditions because of the frequent opening of the doors and highly fluctuating passenger load. Space limitations for ducts, and a desire to keep outlet grilles well above the floor to facilitate car cleaning, add further problems to the distribution system. Maintaining comfort conditions at the driver's station cannot be overlooked, since his term of occupancy is considerably longer than that of any passenger, and because he is usually dressed more lightly than passengers. A separate source of heat is usually provided for the operator, and is under his control.

Heating and Ventilating

Recently-built streetcars and trolley coaches obtain heat from air blown over the main accelerating resistors and track switch resistors, to heat the passenger space. In the modern streamlined streetcars, designated P.C.C., approximately 2400 cfm are drawn from the car and blown over these resistors to dissipate their heat. In trolley coaches, an amount of 800 cfm is customary. The heated air is then delivered to the passenger space or diverted to the outside by means of dampers, as required. If available heat from this source is insufficient, auxiliary electric heaters in the supply ducts may be cut in. The air distribution system is 100 percent recirculating when the maximum heating requirement is being met. At conditions other than maximum heating demand, part of the air drawn from the car is exhausted to the atmosphere. Outside air enters the car, under these conditions, by infiltration at all cracks and through doors when opened at stops.

The most recently-built streetcars have added ventilating fans in the roof structure to introduce outside air through ceiling diffusing grilles. By governing the outside air volume introduced through these roof fans in coordination with the heated air distributing system, it is possible to maintain a slight pressurization of the passenger space and avoid inrush of air when the doors are opened to load passengers. The roof fans also provide an effective means of maintaining lower inside temperature during summer operation. Ventilation tests on P.C.C. streetcars indicate that with 90 F outside temperature and above, 12,000 cfm are required to provide sufficient air change to keep inside temperature within a few degrees of outside air temperature, and to provide enough air movement over passengers for comfort. The best results have been obtained by operating the ventilating fans and keeping the windows closed. New cars provided with adequate ventilating capacity have been built with fixed sash.

Control

Temperature control, consisting of equipment especially designed to withstand the vibration present on transportation equipment, is used. Automatically operated dampers are used to control flow of heated air to the passenger space, or to direct heated air to the atmosphere. An automatically operated rheostat or multi-point switch is used to vary the speed of the ventilating fans. Recent control system applications employ one thermostat to operate both heating dampers and ventilating fans in a modulating or graduated manner, with a compensating thermostat in the heat supply duct to correct for wide fluctuations in temperature of air leaving the resistors. Ventilating fans are usually stopped or operated at lowest speed during the heating cycle, and then their speed is gradually increased as the car temperature rises above the heating-cycle control point.

PASSENGER BUS AIR CONDITIONING

The passenger bus designed for urban transportation operation presents a greater problem to the designer of heating systems than does the inter-urban bus. More frequent stops, and rapidly changing passenger load create this problem on urban vehicles. Provision of heat for the driver independent of the passenger heating system, is a further problem. The inter-urban bus, however, is usually a deluxe vehicle and may require a comfort cooling system. Space and weight limitations and vibration must be considered.

Heating

Recent designs of bus heating systems obtain improved air distribution. Heat in the engine coolant liquid is used to warm air by means of suitable finned coils and this heated air is distributed throughout the passenger space by ducts and outlets directed toward the floor. Some designs include finned surface near the floor in an application similar to that in railway passenger cars. Forced air circulation over this finned floor heating surface has been provided to increase its effectiveness. Oil burning booster heaters have been applied to many Diesel-powered buses to raise the temperature of the engine coolant for maximum engine operating efficiency, and to provide sufficient heat for the passenger space.

Ventilation

Air for ventilation is usually brought into a bus at the front, and distributed throughout the length of the passenger space by a duct or ducts near the ceiling. Except for a few designs employing 100 percent outside air for heating, no heating of the ventilating air has been provided. One recently designed distribution system for an inter-urban bus provides for a fixed minimum of outside air, and is arranged to increase the percentage of outside air to 100 percent when the heating or cooling load diminishes. The distribution ducts and diversion damper arrangement of this system make available two supply ducts and one return duct for heating and for cooling, with a changeover to all three ducts to supply air during the intermediate ventilating cycle. This system permits utilization of atmospheric cooling and ventilation to the greatest degree when it can be most economically employed in the interval between the heating and cooling demand.

Conventional throw-away type filters or renewable filters are used in intake air ducts for many vehicles. Electrostatic filters have been successfully used in some installations. The need for elimination of dirt is great, but the problem is complicated by space limitations and limited power.

Refrigeration

Summer conditioning systems for inter-urban vehicles range in cooling capacity from 36,000 to 48,000 Btu per hour. Mechanical compression systems using refrigerants are used, and are powered by water-cooled gasoline engines of approximately 14 hp.

Complete systems add from 800 to 1300 lb to weight of the coach. Sometimes an auxiliary generator driven by the refrigeration system engine is used and serves to help charge the bus battery, thereby offsetting power drain imposed by the ventilating blower. Belted reciprocating compressors and direct-driven V-type and rotary compressors are used, with engine speeds up to about 1800 rpm. Air-cooled condensers for this service require about 5000 cfm of outdoor air, and this is provided by either centrifugal or propeller type fans belted or direct-driven by the air conditioning engine. Preventing noise and vibration from affecting passengers is of vital importance. Installations must be made for quick daily engine servicing. In all cases fuel is obtained from the main bus tanks, and in some, the main engine cooling system cools the air conditioning engine.

Control

Automatic temperature control is receiving more attention in the design of new vehicles. Some municipalities and states have enacted laws requiring that buses operated on their streets and roads be so equipped. The simplest control systems for heating of urban buses consist of a single thermostat to start and stop the blower of the heater unit. Improved heating systems employ a thermostat to control liquid flow to the heater cores by means of modulating valves, in combination with a means of stopping the heating blower when no heat is required. Heating and ventilating control is accomplished by controlling the volume of outside air, over and above the minimum required in accordance with the temperature in the passenger space, by means of automatic modulating dampers in the outside air intake, or by varying the speed of the ventilating air blowers.

In a large proportion of inter-city buses equipped with mechanical refrigeration, a single thermostat is used to start and stop the cooling operation. This may be accomplished by automatically starting an engine-driven compressor on the cooling demand or by engaging a clutch to drive the compressor. Modulated or graduated control of engine-driven compressors may be accomplished by automatic regulation of the engine throttle controlled from a thermostat in the passenger space. Complete control systems are available to coordinate operation of the heating, ventilating and cooling equipment from a single thermostat, with automatic changeover from heating to ventilating to cooling.

AUTOMOBILE SUMMER AIR CONDITIONING

Recently summer cooling has been applied to automobiles. The average present-day automobile with little insulation, large, single glazed window areas, and high infiltration and exfiltration losses, requires about 15,000 Btu per hour of cooling capacity. One system utilizes a reciprocating compressor belted from the main engine fan shaft, thus operating at varying speeds up to 3000 rpm. The resulting refrigeration capacity varies from about 6000 Btuh at idling speed, to 24,000 Btuh at maximum car speed.

A dry air condenser is placed in front of the engine radiator, and the liquid and suction refrigerant lines run back under the car floor to the evaporator which is located in back of the rear seat. Conditioned air is delivered into the car just above the shelf near the back of the rear seat. A return grille is provided under the rear seat, and the recirculated air is filtered. Outdoor air is provided by infiltration. Power for the air circulating blowers is obtained from the car storage battery. Equipment of this nature increases the car weight approximately 200 lb.

AIRCRAFT AIR CONDITIONING

In the space of a few years, heating, cooling and ventilating of airplanes has progressed from comparatively simple systems to highly complex multi-purpose designs. The attendant control problem has become correspondingly complex. On older, non-pressurized planes, the heating system consisted either of a steam boiler and radiator or a single stage or double stage heat exchanger. On both types, the cabin temperature was adjusted by positioning the face and bypass dampers. While these were sometimes moved by an automatic modulating control, in the majority of cases they were positioned by one of the ship's crew, with results which, while not satisfactory, were passable. As these planes cruised at less than 200 mph and normally operated at low altitudes, changes in outside air temperatures were generally gradual enough so that manual readjustment of controls could maintain reasonably comfortable cabin conditions. Nearly all of these heating systems were marginal in respect to heat available, and the main problem was lack of heat, rather than inadequate control.

Non-Pressurized Cabins

With the advent of the combustion type heater, and use of larger and faster planes, use of manual controls became impracticable. The combustion type heaters reach full rating in less than a minute after being turned on, and as they are rated at 100,000 Btu per hr and up, and since several heaters are generally used, it would take full time of one crew member to keep cabin temperature regulated. As ships of this type are not pressurized, the heating system is still comparatively simple.

In one type, two 100,000 Btu heaters are placed in parallel positions and the ram air from an external scoop is passed through the heaters and discharged through a series of distributing outlets located in the cabin ceiling. The cabin air is discharged through grilles located in the bottom walls of the cabin. An auxiliary nose heater is used by the crew to obtain additional heat for the cockpit or for windshield defrosting.

The cabin is maintained at the desired temperature by means of an automatic control which operates both heaters simultaneously. This control consists of two duct thermostats, one being mounted in the air inlet duct between the air scoop and the heaters so that it is affected by outside ambient temperatures, and the other being mounted in the heater outlet duct so that it is affected by the discharge air temperatures. A thermostat in the cabin is so located that a continuous stream of cabin air passes through it. This type of control has been found to respond to a 1 deg temperature change in less than a second. Its theory of operation follows.

As the outside temperature starts to drop, the outside air duct thermostat decreases in resistance, unbalancing an electronic bridge. This unbalance is amplified by vacuum tubes and causes a power tube to close a relay, turning on the combustion heaters. The resulting increase in temperature is sensed by the warm air duct thermostat which increases in resistance, thus re-balancing the bridge and causing the relay to open. If there were no loss by radiation or convection from the aircraft cabin, these two duct thermostats would be sufficient for adequate control. However, the cabin thermostat is given approximately 30 times more influence than the duct thermostats and so acts as the master controller, and the duct thermostats prevent over-heating or under-heating and keep the discharge air from alternating between extreme cold and extreme heat.

In a slightly more elaborate system, two combustion heaters supply a plenum chamber which is maintained at a constant temperature. Air from the plenum chamber is then mixed with outside air to maintain desired cabin temperature. All of the warm air is discharged into the cabin through the walls. The discharge grilles are located on the floor under seats, and a modulating type controller varies proportions of heated and outside air necessary to maintain desired cabin temperature. The same type of control system as previously described is used, except that an amplifier operates a two-phase motor capable of positioning control dampers instead of operating a relay which would merely open and close the fuel valve. An auxiliary duct, running from the plenum chamber, is used by the pilot as a source of windshield defrosting air.

Pressurized Cabins

With the advent of the new high speed pressurized transport planes, and the addition of cabin cooling in addition to heating, the control problem becomes more complex. On all of these airplanes, the heat of compression from cabin supercharger must be controlled, the air cycle or expansion turbines must be turned on and also, the heat exchanger or combustion heaters, which are used in the system when cooling is required for additional heat, must be automatically controlled.

Assuming that one of these airplanes is operating in an extremely cold climate, the sequence of operation would be as follows:

The automatic controller for the supercharged-air inter-cooler would be in full closed position, so that none of the heat of compression would be removed, and the air would by-pass the expansion turbine and its compressor and the secondary after-cooler. An additional automatic controller would be operating the combustion heater and supplying the additional heat necessary to maintain the desired cabin temperature. If a heat exchanger were used as a supplemental source of heat, a modulating control operating a damper on this exchanger would run towards full heat position.

As the airplane enters a warm climate and heat requirements drop, the combustion heater would cease operation or the heat exchanger would go to full cold position, and the modulating control on the supercharger compressor would move towards the cold position. When the outside ambient temperature rises so high that cooling is desired, the cabin supercharger intercooler would be opened wide. If further cooling were required, the air would to into an air cycle turbine, which is modulated to deliver the required amount of cold air to maintain a comfortable cabin temperature.

Pressure in the cabin is maintained by providing a controlled, constant rate of air flow into the cabin sufficient to maintain ventilation, and adjusting the cabin-pressure relief valve setting, by means of a cabin pressure selector, to maintain the desired cabin pressure. Limits on maximum inside to outside pressure may be of the order of 4 or 5 psi, and safety controls should be provided to prevent exceeding this limit. There is a maximum rate at which the cabin can change to a newly selected value, this rate being in some cases also adjustable.

The requirements for controls of this nature are extremely rigid. It is commonplace for ships of this type to experience changes in outside ambient temperatures of as much as 100 deg in a space of 5 min. For this reason, speed of sensing a change and rapidity of response in the control system is essential if satisfactory control is to be accomplished. The older type thermostats cannot be used in airplanes, due to mass of the thermostat and to vibration experienced on all airplanes. All modern controls use some type of bridge system with temperature sensitive resistors as sensing elements. In some types of controls, the bridge system feeds a sensitive balanced relay, which in turn runs a modulating motor or controls an on-off power relay. A recent sensitive and quickly responding type uses an electronic amplifier, which in turn controls a two-phase motor, or, through relays, controls a d-c motor, or merely closes and opens a power relay for on-off applications.

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In addition to extreme speed and accuracy which are required of all aircraft temperature controls, they must be able to operate under great extremes of temperature, pressure and humidity, and also withstand continuous extreme vibration. Heaters should have, in addition to control from thermostats, suitable limit controls to prevent over-heating due to failure of air supply, or any other cause. Also, there should be safety devices to shut off fuel in case of flame failure.

On the latest *high speed jet airplanes*, the temperature control problem is still more severe than on the latest transports; as in addition to the accuracy required, control response must be phenomenally fast. For example, on some, the air going to the cabin from the jet engine compressor can change the temperature at the rate of 150 deg per second. This, coupled with the fact that on smaller size pursuit ships air is changed in the cabin as much as four times per minute, makes the instantaneous sensing of change and an extremely rapid control movement essential. Also, in airplanes operating at Mach numbers in excess of 0.7, the control must react to the large adiabatic temperature rises encountered. Some of these problems are so new that controls still have not been developed to meet all of the desired conditions. However, present studies being made by control manufacturers should result in developments of such controls in the near future.

SHIP AIR CONDITIONING

In air conditioning a ship, the designer is faced with all problems that would normally arise on shore installation plus additional factors. Mechanical ventilation is an absolute necessity for the comfort of passengers and ships' personnel, and for utility and preservation of cargo and stores. Ships are constructed with water-tight bulkheads dividing the vessel into several compartments. This complicates the running of duct work and results in a multiplicity of both supply and exhaust fans. Temperature and humidity requirements of various spaces aboard ship vary widely. Passenger staterooms and public spaces must have year-round air conditioning with is also being applied more and more to the quarters of the officers and crew. Boiler rooms, galleys, laundries, etc., must have ventilation, and some must have tempered air. Cargo spaces are quite likely to need dehumidification in addition to ventilation.

Inasmuch as ventilation is such a necessary factor aboard a ship, the majority of recently built vessels have been utilizing the air distribution system for heating purposes.

A ship is a self-contained structure quite likely to be away from repair ports for long periods. Adequate spare parts, therefore, form an integral part of equipment furnished. The same type of equipment should be used in as many places as possible throughout the ship, in order to reduce the number of spare parts to be carried.

Preliminary system design is simplified by uniformity of conditions which apply to most ships. Among such conditions are: (1) the necessity for the vessel to supply its own power, (2) the availability of an unlimited supply of low cost steam at suitable pressure, (3) limitations of available space and permissible weight.

The problem of heat transfer and insulation must be given careful consideration. The thermal conductivity of ship building material, such as steel, copper and brass, is many times the value of building material used ashore. The length of ducts between heat sources and fans necessitates extra duct insulation. Hull insulation must be of high quality, with attention given to fireproofness, low density, low thermal conductivity, ruggedness, vermin resistance, and ease of application. Board types are most common.

Duct insulation must have the same characteristics as hull insulation. Semi-rigid and rigid board are most common. Corrugated asbestos is not used because in the presence of moisture, it tends to disintegrate. There is a growing use of natural cork on chilled air ducts because of great difficulty in applying an adequate vapor seal due to space limitations.

SHIP HEATING AND VENTILATING

It is usually most economical in weight and space to use steam duct heaters for heating spaces served by mechanical supply systems. Spaces which do not have mechanical supply, or do not require ventilation in cold weather, are heated by steam convectors or, where the load is large, by unit heaters. Ventilation air, except that supplied to auxiliary and main machinery spaces, is usually preheated to temperatures of 50 to 70 F. No recirculation is used in ventilation and heating systems, but a manual reduction (25 to 50 percent) of air quantity is made during the heating cycle. All heaters are automatically controlled, and preheaters are designed and installed to minimize possibility of freezing of condensate. Preheaters are frequently located close to the outside air intake in order to conserve insulation and, for the same reason, zone reheaters are located as close as possible to each zone. Where a reheater serves only one space, it is commonly located in the space.

Table 1. Design Conditions for Ships

Area	Outside Design Temperatures			Inside Conditions		
	Heating	Ventilating	Cooling	DB & R.H.		Effective Temp.
	F	F	F	F	%	F
Living Quarters	0	95	95(DB) 80(WB)	80	50	73–74
Public Spaces	0	95	95(DB) 80(WB)	80	55	$73\frac{1}{2}$–$74\frac{1}{2}$
Naval Vessels	+10	88(DB) 80(WB)	88(DB) 80(WB)	85	50	75–78

Combinations or variations of duct type and convection or radiant heaters are used, depending upon basic design requirements, such as weight and space limitations, and the economic justification of the cost of the type selected.

Because of many different requirements of various spaces aboard ship, some design temperatures and humidities are given in Table 1. The resulting quantities of air should be checked against typical heating and ventilation practices for ships.*

Machinery Spaces

The prime purpose of machinery space ventilation is to maintain a habitable temperature for the operating personnel. It is more practicable to use *spot cooling* of personnel at working areas than to attempt to obtain uniform ambient temperature. The permissible temperature rise (above

* See Ventilation and Heating of Maritime Commission Ships, by J. W. Markert (*Heating and Ventilating*, Feb. 1943, p. 333).

outside) at working stations is usually 15 deg, while the overall temperature rise is usually between 30 and 50 deg.

These spaces must be exhausted adequately, preferably by mechanical means. Every attempt should be made to remove air at or close to the heat sources. The capacity of the mechanical exhaust systems should be greater than the supply, taking into consideration the expansion of the supply air, to insure an indraft through access openings to the space.

Heat is not required for machinery spaces, except for those fitted with electrically operated equipment, which may remain inactive during periods while in port when heating to about 50 F will be required.

Living Spaces

The minimum quantity of ventilation air required for any sleeping or office space, including hospital space, should be that which will limit the temperature rise over the outside air to not more than 10 deg (a rise of 7 deg is more satisfactory), or a minimum of 30 cfm per person, whichever is the greater.

In spaces fitted for eating, recreation, or manual work, the rise may be taken at 10 deg with not less than 20 cfm per person. The same requirement applies to mechanical exhaust, although natural exhaust may be used where only a short run of duct exists.

Heat should be furnished to maintain the following temperatures:

Staterooms, Berthing, Messing and Office Spaces	70 F
Working Spaces and Shops	60 F
Hospital Spaces	75 to 78 F

Toilets, Washrooms, Showers and Baths

Spaces for these purposes should be fitted with mechanical exhaust ventilation for odor and steam removal. Usually, the surrounding living or working spaces are exhausted through them. Air requirements on merchant vessels are commonly estimated on the basis of a complete air change in 4 min. Where the available air is limited by outside air requirements of air conditioning systems, a lesser quantity (one change in 6 min) is used for private bathrooms. Heat is obtained by use of convector radiators which should maintain a temperature of 70 F.

Galleys, Bakeries, Laundries, and Food Handling Spaces

The problem of the ventilation of spaces fitted for cooking and food preparation is primarily one of heat and smoke or fume removal. Complete mechanical exhaust is always provided, and the mechanical supply is made equivalent to at least 50 percent of the exhaust. Sufficient natural supply to provide an indraft is required. The exhaust quantities should be predicated on restricing the ambient temperature rise to 15 deg above the outside summer design air conditions. The resulting quantity will change the air in these spaces in about 23 sec to 1 min. All of the exhaust should be arranged to remove air from the space through hoods fitted over the heat producing equipment. The tempered mechanical supply should blow air directly on the personnel, but away from the equipment, to minimize interference with the flow of exhaust air to the hoods.

The problem of ventilating laundries is somewhat similar to that for galleys. The exhaust should be about 20 percent greater than supply to insure air indraft. Supply air is generally heated to a temperature of from

45 to 60 F. Ventilation should be sufficient to change the air in the spaces in 1 to 4 min.

Storerooms and Cargo Spaces

The ventilation provided for these spaces depends on the type of vessel, location of space, and nature of cargo.

Ventilation is required for all closed spaces. Even if ventilation is not necessary to preserve the stores or cargo, it is required to prevent the accumulation of toxic or combustible gases and odors. One air change in 15 to 30 min is common practice, except where inflammable liquids or proximity to hot spaces requires additional ventilation. Mechanical supply and natural exhaust are usually used. However, where inflammable gases may exist, natural supply and mechanical exhaust are provided.

Many ships are fitted with dehumidification facilities for eliminating damage to the dry cargo by preventing condensation and dampness. The dehumidification load consists of moisture removed from the ventilation air passed through the dehumidifier, plus the moisture on, or given off by, the cargo, packaging, dunnage, battens, and other materials in the ship's holds. The most severe outside condition requires a moisture removal of 90 grains (140–50) per pound of dry air, with 88 F cooling water. The largest cargo ships are provided with equipment for removing about 250 lb of water per hour.

The dehumidification systems generally utilize silica gel or lithium chloride with inhibitor. (See Chapter 37). In most cases central drying equipment is provided. On large passenger ships consideration is given to the use of two dehumidifying units, because the cargo-carrying spaces are usually concentrated at the extreme ends of the vessel. A simple duct system distributes the dry air to the hold supply ventilation system. These ventilation systems use outside air when weather conditions are favorable. Recirculation and dehumidification are used only when necessary, *i.e.*, when the weather dew-point approaches or exceeds the temperature in the hold. Two control stations are generally provided, one in machinery space and one in chart or wheelhouse. These stations are arranged so that either may change the cargo conditioning system controls from use of outside air to use of conditioned air.

SHIP REFRIGERATION

The kind of refrigeration equipment chosen for a particular vessel depends on the same factors which would govern selection on land.

Small tonnage systems use reciprocating or radial (Freon) compressors. Ships requiring in excess of approximately 125 tons of refrigeration generally use centrifugal compressors. Steam jet refrigeration has proven satisfactory on several large foreign liners and is being seriously considered in this country for similar applications. The two essential prerequisites of the steam jet (low-cost steam and condenser cooling water) are available.

The type of equipment used for the refrigerated cargo space will determine the equipment to be used for air conditioning. This consideration can reduce materially the overall cost, because a stand-by unit is always provided for cargo refrigeration.

AIR CONDITIONING SPACE TREATMENT FOR SHIPS

The application of air conditioning to new American passenger ships is well established. All passenger staterooms, except steerage and third class, are usually air conditioned. This includes staterooms for ship's personnel and offices within conditioned passenger areas. Third class staterooms are air conditioned on some ships, depending on the particular trade. Theaters, lounges, smoking rooms, beauty shops, barber shops and similar closed public spaces are usually air conditioned.

All messrooms, recreation rooms, officers' offices, crew's inboard rooms, and those having fixed portlights are usually air conditioned on passenger vessels. The treatment depends on the requirements of the operator and the proposed itinerary of the vessel.

SHIP SYSTEMS AND CONTROLS

The types of comfort conditioning systems used to date generally have followed conventional lines, except for those serving staterooms, offices, and similar small spaces. Large public spaces are fitted with individual systems which supply dehumidified and cooled air during the cooling cycle, and warm air during the heating cycle. In many cases these rooms are fitted with large glass windows and doors, and require direct radiation to offset the downdraft which would occur in cold weather. Finned-tube radiation running the full length of the glass area is commonly used for this purpose. Introduction of warm air at the sill, in lieu of direct radiation, is also used.

Systems serving most public spaces are designed to provide all outside air as long as the refrigeration load is less than the capacity of the cooling equipment. Many central systems are simplified by using 100 percent outside air all-year-round. The important problem in ship air conditioning concerns the treatment of the small spaces such as passenger staterooms, offices, and crew quarters. Low headroom, congested quarters, double berths, and unsymmetric arrangements make each space a problem in air distribution and treatment.

The simplest system used for small spaces consists of a central filter bank, supply fan, preheater, and cooling and dehumidifying coil. The preheater steam valve and cooling coil water valve are controlled in sequence by a duct thermostat, in the fan discharge, set to maintain a constant outlet air temperature. Zone reheaters are provided to take care of variations in heating loads. The reheater steam valve is controlled by a sub-master thermostat at the reheater outlet. Control of room temperature is obtained by operating manual dampers in the air supply to the space. A recirculation exhaust fan is frequently provided, and operates in conjunction with automatic dampers to utilize the maximum quantity of outside air consistent with capacity of the cooling coils.

One system utilizes the same central supply equipment and recirculation exhaust system as the one just described, except that zone reheaters are replaced by individual space hot water reheaters. Each reheater is provided with a control valve, controlled by room thermostat. Generally, a forced circulation single pipe hot water system is used. Fig. 1 shows a diagrammatic arrangement of this system. It should be noted that reheat is used to eliminate overcooling of the individual spaces during mild cooling conditions. This system is generally used for staterooms and small spaces devoted to first and second class passengers. The average total air per person is about 60 cfm, and average outside air per person is about 18 cfm.

A third system, used to a limited extent, is similar to the system just described, except that each room is provided with an induction unit (floor type where possible) which reheats the primary air supply. Control of

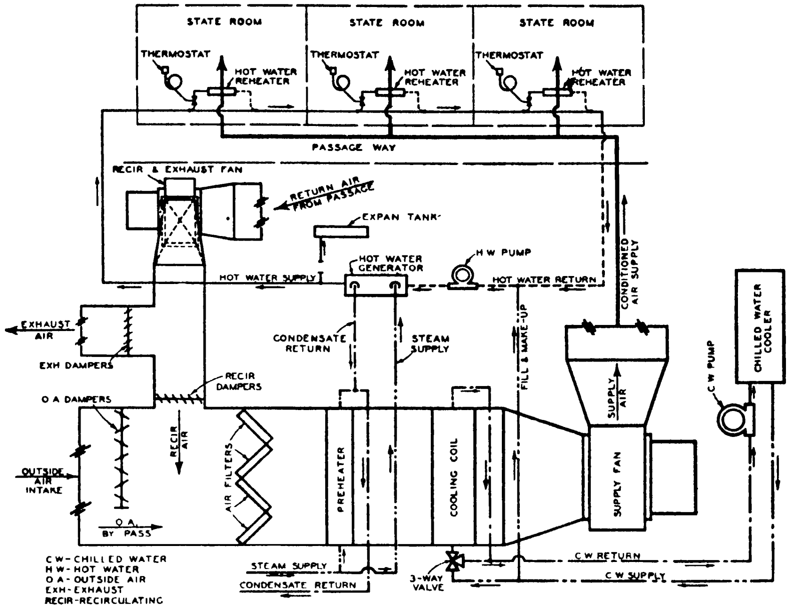

FIG. 1. ARRANGEMENT OF AIR CONDITIONING SYSTEM WITH CENTRAL SUPPLY EQUIPMENT AND INDIVIDUAL HOT WATER ROOM REHEATERS

heating coil in the induction unit is the same as noted for the system described in the previous paragraph. The average primary air supply is about 40 cfm per person. Recirculation is not always used. The amount of induced air varies with unit design.

A fourth less common arrangement is similar, but makes use of hot water in the induction unit in winter and cold water in the induction unit in summer. Control of the valve on the unit is obtained by use of a summer-winter type thermostat. The unit is provided with drip pan and drain piping to remove condensation. No recirculation is used.

On cargo ships, tankers and vessels not carrying passengers or not having air conditioning, heating of crews' spaces and officers' staterooms is usually obtained through a central system having filters, preheaters and reheaters. A minimum temperature of air leaving the preheater is controlled by a duct thermostat. The reheater is controlled by a sub-master discharge-duct thermostat, reset by outside master control. Relationship between sub-master and master control (wherein discharge temperature is raised as outdoor temperature drops), is set according to a schedule based on the ship's itinerary.

Duct work for all systems described is designed for conventional velocity. However, if power is available, and suitably strong duct construction and adequate sound absorbing facilities are provided, high velocity systems may be used.

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CHAPTER 48

WATER SERVICES

Sizing Cold Water Supply Piping, Procedure for Sizing Cold Water Systems, Cooling Water Piping, Estimating Heating Load and Storage Capacity, Methods of Heating Water, Computing Heat Transfer Surface, Hot Water Supply Piping, Control of Service Water Temperature, Safety Devices, Solar Water Heaters, Domestic Hot Water by Heat Pump

PROPER design of the water distributing system in a building is necessary in order that the various fixtures may function properly. The amount of either hot or cold water used in any building is variable, depending on the type of structure, usage, occupancy, and time of day. It is necessary to provide piping, water heating, and storage facilities of sufficient capacity to meet the peak demand without wasteful excess in either piping or equivalent cost.

SIZING COLD WATER SUPPLY PIPING

One of the important items that must be determined before any part of the water-piping system can be sized, is the probable rate of flow in any particular reach of piping. The rate of flow in the service line, risers, and main branches, however, will rarely be equal to the sum of the rates of flow of all connected fixtures. In fact, the probability that every fixture in a large group will be in use at the same time is so remote that it would be very poor engineering practice to design the piping to take care of such simultaneous flow.

The demand load in building water supply systems cannot be determined exactly and is not readily standardized. The two main problems to be considered are: (1) the satisfactory supply of water for a given fixture, and (2) the number of fixtures which may be assumed to be in use at the same time.

The minimum flow that will be satisfactory to the consumer depends greatly on the consumer, his standard of living, his professional needs, size of family, garden requirements, and similar factors. Depending on these factors, the per capita water consumption for domestic use usually varies between 20 and 80 gal per day. Experience indicates that the type of dwelling also has considerable influence on the water consumption.

In *apartment houses* the per capita daily water consumption is generally higher than in single-family houses. This is due to the use of a central metering system which is not conducive to the saving of water, and to the long hot water lines which cause high heat losses and an increase in the wasting of the cooled water. In designing water supply systems for apartment houses, a daily per capita water consumption of 50 gal may be considered a safe design figure.

Although a considerable number of *housing projects* have been developed throughout the United States, conclusive water consumption data have not yet been gathered. Nevertheless, it seems that the daily per capita water consumption in housing projects falls in between the consumption in apartment houses and that in single dwellings at the same geographical location.

Table 1. Proper Flow and Pressure Required During Flow for Different Fixtures

Fixture	Flow Pressure[a]	Flow gpm
Ordinary basin faucet	8	3.0
Self-closing basin faucet	12	2.5
Sink faucet—⅜ in.	10	4.5
Sink faucet—½ in.	5	4.5
Bathtub faucet	5	6.0
Laundry tub cock—½ in.	5	5.0
Shower	12	5.0
Ball-cock for closet	15	3.0
Flush valve for closet	10–20	15–40[b]
Flush valve for urinal	15	15.0
Garden hose, 50 ft, and sill cock	30	5.0

[a] Flow pressure is the pressure in the pipe at the entrance to the particular fixture considered.
[b] Wide range due to variation in design and type of flush-valve closets.

In general, a daily per capita water consumption of 40 gal can be used as a safe design figure for *housing projects.*

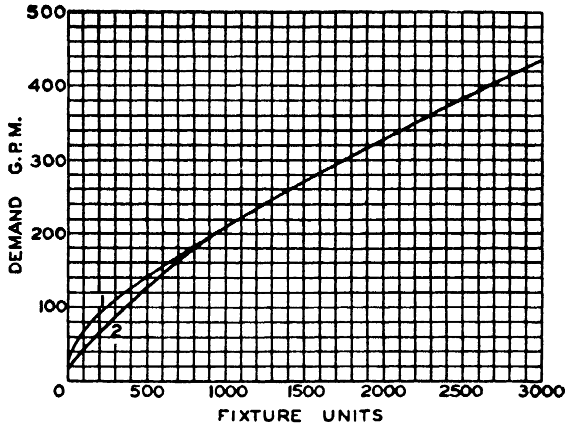
Table 1 gives the rate of flow desirable for many common types of fixtures, and the average pressure necessary to give this rate of flow The pressure necessarily varies with fixture design; with some, a much greater pressure is necessary to give the same rate of flow than with others. In general, the lower the quality of the faucet the greater will be the pressure required.

In estimating the load, the rate of flow is frequently computed in *fixture units.* One *fixture unit* is equivalent to 7.5 gal per min. Table 2 gives the demand weights in terms of fixture units for different plumbing fixtures under several conditions of service, and Fig. 1 gives the estimated demand in gallons per minute corresponding to any total number of fixture units. Fig. 2 shows an enlargement of Fig. 1 for a range up to 250 fixture units.

Table 2. Demand Weights of Fixtures in Fixture Units[a]

Fixture or Group[b]	Occupancy	Type of Supply Control	Weight in Fixture Units[c]
Water closet	Public	Flush valve	10
Water closet	Public	Flush tank	5
Pedestal urinal	Public	Flush valve	10
Stall or wall urinal	Public	Flush valve	5
Stall or wall urinal	Public	Flush tank	3
Lavatory	Public	Faucet	2
Bathtub	Public	Faucet	4
Shower head	Public	Mixing valve	4
Service sink	Office, etc	Faucet	3
Kitchen sink	Hotel or restaurant	Faucet	4
Water closet	Private	Flush valve	6
Water closet	Private	Flush tank	3
Lavatory	Private	Faucet	1
Bathtub	Private	Faucet	2
Shower head	Private	Mixing valve	2
Bathroom group	Private	Flush valve for closet	8
Bathroom group	Private	Flush tank for closet	6
Separate shower	Private	Mixing valve	2
Kitchen sink	Private	Faucet	2
Laundry trays (1-3)	Private	Faucet	3
Combination fixture	Private	Faucet	3

[a] For supply outlets likely to impose continuous demands, estimate continuous supply separately and add to total demand for fixtures.
[b] For fixtures not listed, weights may be assumed by comparing the fixture to a listed one using water in similar quantities and at similar rates.
[c] The given weights are for total demand. For fixtures with both hot and cold water supplies, the weights for maximum separate demands may be taken as ¾ the listed demand for the supply.



No. 1 for system predominantly for flush valves.
No. 2 for system predominantly for flush tanks.

FIG. 1. ESTIMATE CURVES FOR DEMAND LOAD

The estimated demand load for fixtures used intermittently on any supply pipe will be obtained by multiplying the number of each kind of fixture supplied through that pipe by its weight from Table 2, adding the products, and then referring to the appropriate curve of Figs. 1 or 2 to find the demand corresponding to the total fixture units. In using this method it should be noted that the demand for fixture or supply outlets other than those listed in the table of fixture units is not yet included in the estimate. The demands for outlets (such as hose connections, air conditioning apparatus, etc.) which are likely to impose continuous demand during times of heavy use of the weighted fixtures, should be estimated separately and added to the demand for fixtures used intermittently, in order to estimate the total demand.

So far, the information presented possible the determination of the *design rate of flow* in any particular section of piping. The next general step is to determine the *size* of piping.

As water flows through a pipe, the pressure continually decreases along the pipe, due to loss of energy from friction. The problem is then one of ascertaining the minimum pressure in the street main, and the minimum

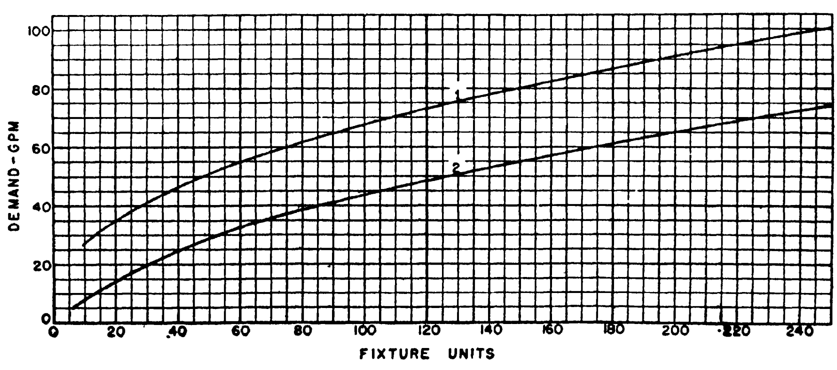


FIG. 2. SECTION OF FIG. 1 ON ENLARGED SCALE

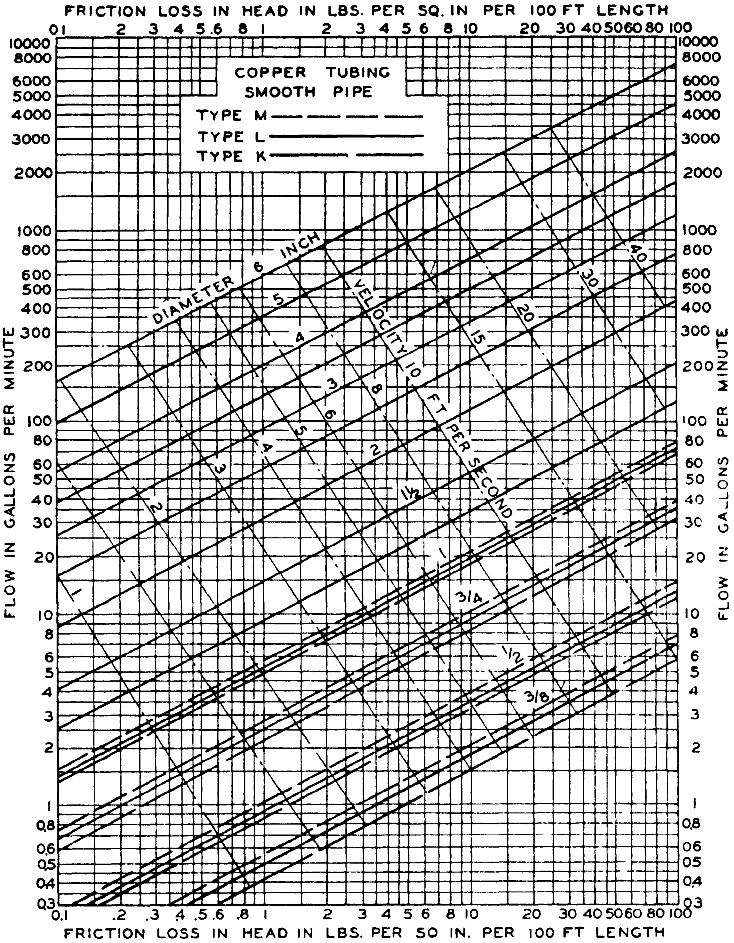


FIG. 3. FLOW CHART FOR COPPER TUBING

pressure required for the operation of the topmost fixture. (A pressure of 15 psi is ample for flush valves, but reference should be made to the manufacturers' requirements. A minimum of 8 psi should be allowed for other fixtures.) The pressure differential thus obtained will be available for overcoming pressure losses in the distributing system, and in overcoming the difference in elevation between the water main and the highest fixture.

The pressure loss, in pounds per square inch, caused by the difference in elevation between the street main and the highest fixture, may be obtained by multiplying the difference in elevation in feet by the conversion factor 0.43.

When water flows through a pipe, friction occurs as the result of the sliding of water particles past one another. If the pipe wall is rough, the roughness projections cause additional friction, owing to the development of increased turbulence in the flowing water. As the water flows along a uniform pipe, the pressure decreases as a result of a dissipation of

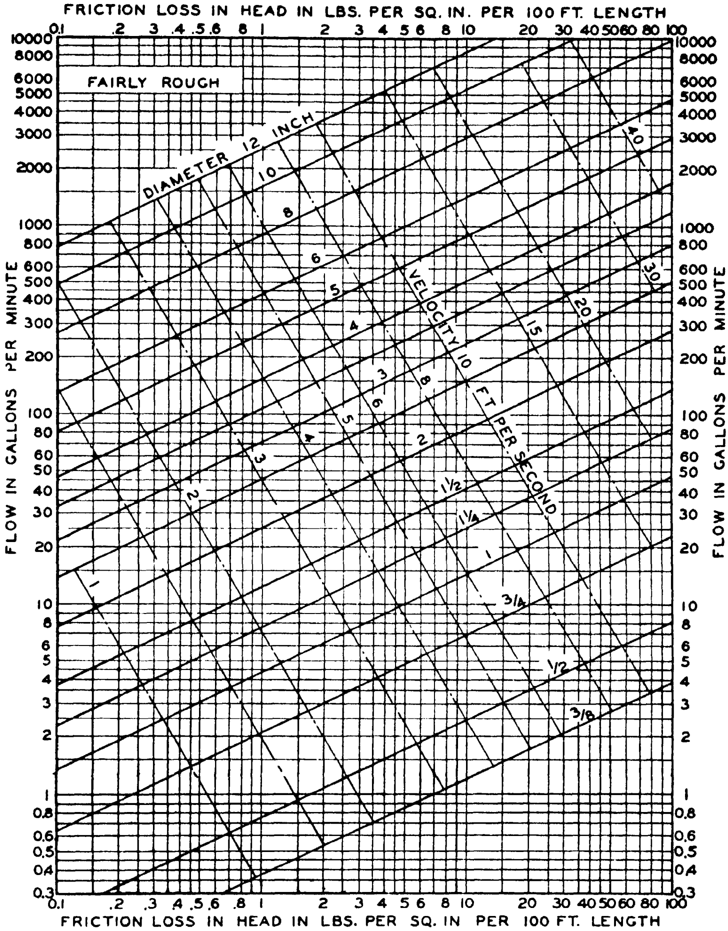


FIG. 4. FLOW CHART FOR FAIRLY ROUGH PIPE

energy arising from the internal friction set up by viscosity of the water. This loss in energy is shown by the loss of pressure. The pressure loss is proportional to the length of straight uniform pipes, and varies greatly with flow velocity, pipe diameter, and roughness of pipe.

On the basis of inside surface conditions, pipes may be classified as smooth, fairly rough and rough, as follows:

Smooth. The pipe surface shows no perceptible roughness. Pipes made of copper, brass, or lead may usually be classified as smooth.

Fairly Rough. All ordinary pipes, such as wrought iron, galvanized iron, steel and cast-iron, after a few years of usage, may be called fairly rough.

Rough. Pipes that have deteriorated fairly rapidly for some 10 or 15 years after being laid, are classified as rough.

Figs. 3, 4 and 5 give the pipe friction losses corresponding to these three types of pipes for various nominal diameters.[1]

Example 1: A 2½ in. fairly rough pipe supplies 100 gpm of water. Find the friction loss in head if the pipe length is 200 ft.

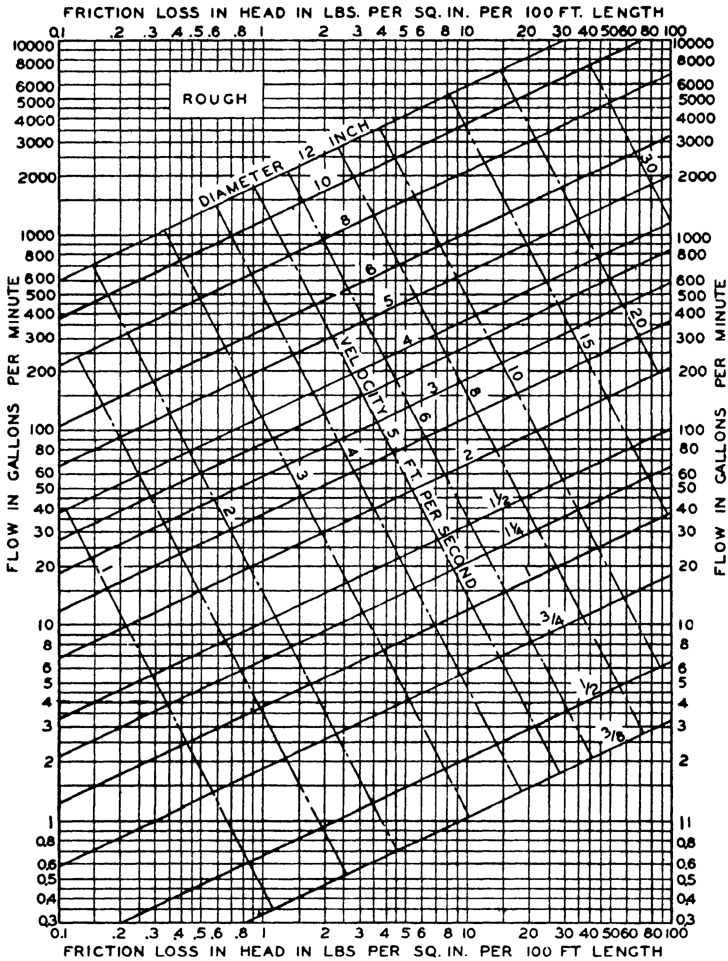


Fig. 5. Flow Chart for Rough Pipe

Solution: Enter Fig. 4 at 100 gpm, and move along this line until it intersects the 2½ in. diameter line. From this intersection point, move vertically down and read 4.5 psi friction loss per 100 ft of pipe length. Then the total friction loss will be 2 x 4.5 = 9 psi.

The pressure losses in the distributing system will consist of the pressure losses in the piping itself, plus the pressure losses in the pipe fittings, valves and the water meter, if any. Estimated pressure losses for disc-type meters for various rates of flow are given in Fig. 6.

Flow limits for disc-type meters, which may be regarded as the limits of recommended ranges in capacities, are given in Table 3. For information on other types of meters, the manufacturers should be consulted.

Fig. 7 shows the variation of pressure loss with rate of flow for various types of faucets and cocks, based on experimental data obtained at the *State University of Iowa.*

The loss of pressure through any fitting or valve can be expressed in

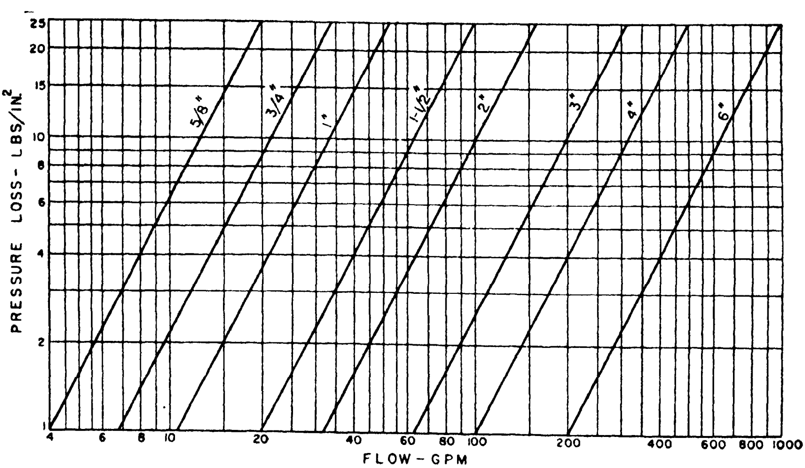


FIG. 6. PRESSURE LOSSES IN WATER METERS

pounds per square inch for any given rate of flow. Experience has shown, however, that the simplest method of expressing losses in fittings and valves is to use the concept of an equivalent length of straight pipe. Thus it has been found, for example, that a 1 in., 90 deg elbow introduces a loss equivalent to 2.2 ft of straight 1 in. pipe. Therefore, for each 1 in., 90 deg elbow, 2.2 ft of 1 in. pipe are added to the total length of 1 in. pipe.

Estimated pressure losses for pipe fittings and valves in terms of equivalent pipe length are shown in Table 4.

Table 5 lists the equivalent lengths for various special types of apparatus and fittings. The loss in water meters varies considerably with the design even in meters of the same nominal size. The values given in Table 5 are ample for the well-known meters now on the market.

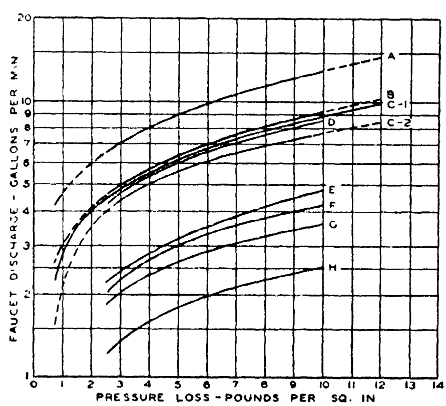


FIG. 7. VARIATION OF PRESSURE LOSS WITH RATE OF FLOW FOR VARIOUS FAUCETS AND COCKS

A. ½ in. laundry bibb (old style).
B. Laundry compression faucet.
C-1. ½ in. compression sink faucet (Mfr. 1).
C-2. ½ in. compression sink faucet (Mfr. 2).
D. Combination compression bath tub faucets (both open).
E. Combination compression sink faucet
F. Basin faucet.
G. Spring self-closing faucet.
H. Slow self-closing faucet.

(Dashed lines indicate recommended extrapolation)

TABLE 3. PERFORMANCE REQUIREMENTS OF WATER METERS[a]

SIZE IN.	NORMAL TEST-FLOW LIMITS GPM	MINIMUM TEST-FLOW GPM
5/8	1 to 20	1/4
3/4	2 to 34	1/2
1	3 to 53	3/4
1½	5 to 100	1½
2	8 to 160	2
3	16 to 315	4
4	28 to 500	7
6	48 to 1,000	12

[a] *American Water Works Association Standards:*
Registration. The registration on the meter dial shall indicate the quantity recorded to be not less than 98 percent nor more than 102 percent of the water actually passed through the meter while it is being tested at rates of flow within the specified limits herein under normal test flow limits: There shall be not less than 90 percent of the actual flow recorded when a test is made at the rate of flow set forth under minimum test flow.

The water demand for hose bibbs or other large demand fixtures taken off the building main is frequently the cause of inadequate water supply to the upper floor of a building. This condition may be prevented by sizing the distribution system so that the pressure drops from the street main to all fixtures are the same. It is good practice to maintain the building main of ample size (not less than 1 in. where possible) until all branches to hose bibbs have been connected. Where the street main pressure is excessive and a pressure reducing valve is used to prevent water hammer or excessive pressure at the fixtures, it is frequently desirable to connect hose bibbs ahead of the reducing valve.

The principles involved in sizing either up-feed or down-feed systems are the same. The principal difference in procedure is that in the down-feed system, the difference in elevation between the house tank and the fixtures provides the pressure required to overcome pipe friction.

Procedure for Sizing Cold Water Systems

The recommended procedure for sizing piping systems is outlined in following paragraphs 1 to 6, inclusive.

1. Draw a sketch of the main lines, risers, and branches, and indicate the fixtures to be served. Indicate the rate of flow of each fixture.

TABLE 4. ALLOWANCE IN EQUIVALENT LENGTH OF PIPE FOR FRICTION LOSS IN VALVES AND THREADED FITTINGS

DIAMETER OF FITTING IN.	EQUIVALENT LENGTH OF PIPE FOR VARIOUS FITTINGS						
	90 Deg Standard Ell Ft	45 Deg Standard Ell Ft	90 Deg Side Tee Ft	Coupling or Straight Run of Tee Ft	Gate Valve Ft	Globe Valve Ft	Angle Valve Ft
3/8	1	0.6	1.5	0.3	0.2	8	4
1/2	2	1.2	3	0.6	0.4	15	8
3/4	2.5	1.5	4	0.8	0.5	20	12
1	3	1.8	5	0.9	0.6	25	15
1¼	4	2.4	6	1.2	0.8	35	18
1½	5	3	7	1.5	1.0	45	22
2	7	4	10	2	1.3	55	28
2½	8	5	12	2.5	1.6	65	34
3	10	6	15	3	2	80	40
3½	12	7	18	3.6	2.4	100	50
4	14	8	21	4.0	2.7	125	55
5	17	10	25	5	3.3	140	70
6	20	12	30	6	4	165	80

2. Using Table 2, compute the demand weights of the fixtures in fixture units.
3. Determine the total demand in fixture units and, using Fig. 1 or Fig. 2, find the expected demand in gallons per minute.
4. Determine the equivalent length of pipe in the main lines, risers, and branches. Since the sizes of the pipes are not known, the exact equivalent length for various fittings, etc., cannot be made. Add the equivalent lengths, starting at the street main and proceeding along the service line, the main line in the building, and up the riser to the top fixture of the group served.
5. Determine the average minimum pressure in the street main and the minimum pressure required for the operation of the topmost fixture. This latter pressure should be 8 to 15 psi.
6. Calculate, by means of Equation 1, the approximate design value of the average pressure drop per 100 ft of pipe in the equivalent length determined in paragraph 4.

$$p = [P - 0.43H - 10]\frac{100}{L} \tag{1}$$

where

p = average pressure loss per 100 ft of equivalent length of pipe, psi.
P = pressure in street main, psig.

Table 5. Equivalent Lengths of Iron Pipe to Give Same Loss as Special Fittings and Apparatus

Fitting Apparatus	Nominal Diameter of Pipe—Inches			
	½	¾	1	1¼
30-gal Vertical hot-water tank, ¾ in. pipe.	4	17	56	—
30-gal Horizontal hot-water tank, ¾ in. pipe	1.2	5	16	—
Water meters (No valves included)				
⅝ in. with ½ in. connections	6.7	28	90	—
⅝ in. with ¾ in. connections	4.8	20	64	—
¾ in. with ¾ in. connections	3.4	14	45	—
1 in. with 1 in. connections	—	9	30	115
1¼ in. with 1 in. connections	—	4.4	14	54
Water softener	—	50–200	—	—

H = height of highest fixture above street main, feet.
L = equivalent length determined in paragraph 4, feet.

If the system is of the down-feed supply from a gravity tank, the height of water in the tank, converted to pounds per square inch by multiplying by 0.43, replaces the street main pressure, and the term 0.43 H in Equation 1 is added instead of subtracted in calculating the term p. In this case, H will be the vertical distance of the fixture below the bottom of the tank.
7. From the expected rate of flow, determined as in paragraph 3, and the value of p, calculated as in paragraph 6, choose the sizes of pipe from Figs. 3, 4 or 5.

Example 2: Assume a minimum street main pressure of 55 psig; a height of topmost fixture above street main of 50 ft; a developed pipe length from water main to highest fixture of 100 ft; a total load on the system of 50 fixture units; and that the water closets are flush-valve operated. Find the required size of supply main.
Solution: From Fig. 2 the estimated peak demand is found to be 51 gpm. From Table 3 it is evident that several sizes of meters would adequately measure this flow. For a trial computation choose the 1½ in. meter. From Fig. 6 the pressure drop through a 1½ in. disc-type meter for a flow of 51 gpm is found to be 6.5 psi.

Then the pressure drop available for overcoming friction in pipes and fittings is $55 - (15 + 50 \times 0.43 + 6.5) = 12$ psi.

At this point it is necessary to make some estimate of the equivalent pipe length of the fittings on the direct line from the street main to the highest fixture. The exact equivalent length of the various fittings cannot now be determined since the pipe sizes of the building main, riser, and branch leading to the highest fixture are not known as yet, but a first approximation is necessary in order to make a tentative selection of pipe sizes. If the computed pipe sizes differ from those used in determining the equivalent length of pipe fittings, a recalculation will be necessary, using

TABLE 6. COMPUTATION OF BRANCH SIZE IN EXAMPLE 2

FIXTURES NO. AND KIND	FIXTURE UNITS (FROM TABLE 2 AND NOTE C)	DEMAND (FROM FIG. 2) GPM	PIPE SIZE (FROM FIG. 4) IN.
3 flush valves	3 x 6 = 18		
2 bath tubs	¾ (2 x 2) = 3		
3 lavatories	¾ (3 x 1) = 2.25		
Total	23.25	38	1½

the computed pipe sizes for the fittings. For the purposes of this example assume that the total equivalent length of the pipe fittings is 50 ft.

Then the permissible pressure loss per 100 ft of equivalent pipe is $12 \times 100/(100 + 50) = 8$ psi.

Assuming that the corrosive and caking properties of the water are such that Fig. 4 for fairly rough pipe is applicable, a 2 in. building main will be adequate.

The sizing of the *branches* of the building main, the *risers*, and *fixture branches* follow the principles outlined. For example, assume that one of the branches of the building main carries the cold water supply for 3 water closets, 2 bath tubs, and 3 lavatories. Using the permissible pressure loss of 8 psi per 100 ft, the size of branch determined from Table 2 and Figs. 1 and 4 is found to be 1½ in. Items entering the computation of pipe size are given in Table 6.

COOLING WATER PIPING

Water is very frequently used in refrigeration systems, cooling towers and other similar installations. In designing the piping system of such installations, the principles of hydraulics, as already outlined, are employed. Nevertheless, there are several practical items having particular application to cooling installations. They are outlined in the following paragraphs, and it is important that the designer be familiar with them.

In choosing pipe material, the problem of corrosion should be kept in mind to prevent defects in the system. If the water does not have drastic corrosion characteristics, wrought iron or steel piping may be used; otherwise, galvanized steel piping may be preferred. If sea water is used as the circulating medium, it is advisable to use alloys such as admiralty metal in pipe and tubing. In refrigeration condensers where water is the cooling medium, iron pipe is commonly employed.

In regard to assembly, cast-iron flanges or welded joints are to be preferred to screwed joints wherever possible.

Valves used in circulating systems may be of the globe, gate, or angle

TABLE 7. PIPE SIZES FOR COOLING TOWERS[2]

RATED TONS OF REFRIG.	COOLING WATER GPM	PIPE SIZES (NOMINAL INCHES)	
		Inlet to Tower	Outlet from Tower
3 to 5	10 to 18	1½	1½
7 to 15	20 to 45	2	2
20	65	2½	3
25	86	2½	4
35	115	3	4
50	170	3	4
75	225	5	6
100	300	5	6
150	450	5	8
200	600	6	8
250	750	8	8

types. If water is the circulating medium, brass valves are usually used. However, if the circulating medium is an electrolyte, such as brine, then it is preferable to use valves made of the same material as the piping itself.

The friction loss in the piping may be determined from Fig. 4 for fairly rough pipe. If the coolant is brine, a correction for the proper density must be made. Experience indicates that in sizing piping for cooling systems, a pressure drop of the order of 2 to 3 psi per 100 ft of pipe length, and a fluid velocity of 3 to 8 fps, yield most economical results.

In the case of cooling towers, the amount of circulating water is about 3 gpm per ton of refrigeration, when based on a design wet-bulb temperature of about 76 F. Table 7 gives pipe sizes frequently used for various sizes of cooling towers, assuming a hot-water temperature of 95 F and a cold-water temperature of 85 F.[2]

TABLE 8. MAXIMUM DAILY (24-HR) REQUIREMENTS FOR HOT WATER IN GALLONS

	No. of Rooms	Number of Bathrooms				
		1	2	3	4	5
Apartments and Private Homes	1	60	—	—	—	—
	2	70	—	—	—	—
	3	80	—	—	—	—
	4	90	120	—	—	—
	5	100	140	—	—	—
	6	120	160	200	—	—
	7	140	180	220	—	—
	8	160	200	240	250	—
	9	180	220	260	275	—
	10	200	240	280	300	—
	11	—	260	300	340	—
	12	—	280	325	380	450
	13	—	300	350	420	500
	14	—	—	375	460	550
	15	—	—	400	500	600
	16	—	—	—	540	650
	17	—	—	—	580	700
	18	—	—	—	620	750
	19	—	—	—	—	800
	20	—	—	—	—	850
Hotels	Room with basin					10
	Room with bath—transient					50
	Room with bath—resident					60
	2 Rooms with bath					80
	3 Rooms with bath					100
	Public shower					200
	Public basins					150
	Slop sink					30
Office Buildings	White color worker (per person)					2.0
	Other workers (per person)					4.0
	Cleaning per 10,000 sq ft					30.0
Hospitals	Per bed					80–100

ESTIMATING HEATING LOAD AND STORAGE CAPACITY

The maximum daily and the maximum hourly hot water demand form the basis for the selection of the heater and the storage tank.

In general, two thirds of the total daily water consumption is hot water. For residential dwellings, a design value of about 20 to 30 gal per capita per day may be assumed, but it should be remembered that the hot water used will depend on the number of rooms and the number of bathrooms in any house or apartment. Table 8 gives estimates of the maximum hot water requirements in 24 hr in various types of buildings.

In estimating the size of hot water storage tank required, and the heating capacity to be provided, either from the boiler or from an independent domestic hot water heater, it is necessary to know the total quantity of water to be heated per day, and the maximum amount which will be used in any one hour, as well as the duration of the peak load.

In cases where the requirements for hot water are reasonably uniform, as in residences, apartment buildings, hotels, and the like, smaller storage capacity is required than in the case of factories, schools and office buildings, where practically the entire day's usage of hot water occurs during a very short period. Correspondingly, the heating capacity must be proportionately greater with uniform usage of hot water than with intermittent usage, where there may be several hours between peak demands during which the water in the storage tank can be brought up to temperature. As a general rule, it is desirable to have a large storage capacity in order that the heating capacity, and consequently the size of the heater, or the load on the heating boiler, may be as small as possible.

Table 9. Estimated Hot Water Demand Characteristics for Various Types of Buildings

Type of Building	Hot Water Required	Max. Hourly Demand in Relation to Day's Use	Duration of Peak Load Hours	Storage Capacity in Relation to Day's Use	Heating Capacity in Relation to Day's Use
Residences, apartments, hotels, etc.	40 gal per person per day[a]	1/7	4	1/5	1/7
Office buildings	2 gal per person per day[a]	1/5	2	1/5	1/6
Factory buildings	5 gal per person per day[a]	1/3	1	2/5	1/8
Restaurants				1/10	1/10
Restaurants 3 meals per day		1/10	8	1/5	1/10
Restaurants 1 meal per day		1/5	2	2/5	1/6

[a] At 140 F

In estimating the hot water which can be drawn from a storage tank, it should be borne in mind that only about 75 percent of the volume of the tank is available, since, by the time this quantity has been drawn off, the incoming cold water has cooled the remainder down to a point where it can no longer be considered hot water.

Where steam from the heating boiler is used to heat domestic hot water, the computed load on the boiler should be increased by 4 sq ft EDR (equivalent direct radiation) for every gallon of water per hour heated through a 100 deg rise. The actual requirement is $(100 \times 8.33)/240 = 3.48$ sq ft per gal heated 100 deg. The value of 4 allows for transmission losses.

There are two methods in common use for estimating the hot water requirements of a building: (1) by the number of people, and (2) by the number of plumbing fixtures installed. Where the number of people to be served can be reasonably estimated, the data in Table 9 may be used.

Example 3: Determine the heater size and storage tank capacity for a residence housing five people.

Solution: From Table 9, a residence housing five people would have a daily requirement of $5 \times 40 = 200$ gal. per day, and a maximum hourly demand of $200 \times 1/7 = 28.5$ gal. The heater should have a storage capacity of $200 \times 1/5 = 40$ gal, and a heating capacity of $200 \times 1/7 = 28.5$ gal per hr.

The conditions given in *Example 3* may be cited as average. It is possible to vary the storage and heating capacity by increasing and decreasing one over the other. Such a condition is illustrated in *Example 4*.

Example 4: Determine the required heater capacity for an apartment housing 200 people, if the storage tank has a capacity of 1000 gal. What heater capacity will be required if the storage tank is changed to 2500 gal capacity?

Solution: Assume an apartment house housing 200 people. From the data in Table 9: Daily requirements $= 200 \times 40 = 8000$ gal. Maximum hours demand $= 8000 \times 1/7 = 1140$ gal. Duration of peak load = 4 hr. Water required for 4-hr peak $= 4 \times 1140 = 4560$.

If a 1000 gal storage tank is used, hot water available from the tank $= 1000 \times 0.75 = 750$. Water to be heated in 4 hr $= 4560 - 750 = 3710$ gal. Heating capacity per hour $= 3710/4 = 930$ gal.

If instead of a 1000 gal tank, a 2500 gal tank had been installed, the required heating capacity per hour would be $\dfrac{4560 - (2500 \times 0.75)}{4} = 671$ gal.

Table 10 may be used to determine the size of water heating equipment from the number of fixtures. To obtain the *probable maximum demand,* multiply the total quantity for the fixtures by the *demand factor* in line 11. The heater or coil should have a water heating capacity equal to this *probable maximum demand.* The storage tank should have a capacity equal to the *probable maximum demand* multiplied by the storage capacity factor in line 12. *Example 5* will illustrate the procedure.

Example 5: Determination of heater and storage tank size for an apartment building from number of fixtures.

60 lavatories	$\times$ 2 =	120 gal per hr
30 bathtubs	$\times$ 20 =	600 gal per hr
30 showers	$\times$ 75 =	2250 gal per hr
60 kitchen sinks	$\times$ 10 =	600 gal per hr
15 laundry tubs	$\times$ 20 =	300 gal per hr
Possible maximum demand	=	3870 gal per hr
Probable maximum demand	$= 3870 \times 0.30$ =	1161 gal per hr
Heater or coil capacity	=	1161 gal per hr
Storage tank capacity	$= 1161 \times 1.25$ =	1450 gal

Although, in private dwellings a water temperature of 140 F is reasonable for dishwashing, in public places sanitation regulations call for 180 F water. Most of the dishwashing machines now available on the market require 180 F water. The amount of 180 F water needed in restaurants per day may be determined according to the method outlined by the *American Gas Association,*[3] in the following paragraphs:

1. Multiply the number of meals per day by the number of dishes per meal (6 for low-price restaurants, 8 for medium-price restaurants, and 10 for high-price restaurants) to determine the total number of dishes per day.
2. Divide the total number of dishes per day, as determined by method in paragraph 1, by the average number of dishes per rack to find the number of racks per day.
3. Multiply the number of racks per day by the gallons of 180 F water (using 1.5 for single tank machines and 0.75 for two tank machines). This product will give the gallons of 180 F water per day for rinse sprays.
4. Multiply the number of meal periods per day (one, two or three) by the dish-

washing tank capacity in gallons, giving the gallons of 180 F water per day necessary to fill the tanks.

5. Add values from paragraphs 3 and 4 to obtain the total number of gallons of 180 F water required per day.

For purposes other than dishwashing, a considerable amount of 140 F water is also used. To find the daily 140 F water requirement in a restaurant, multiply the total number of meals served per day by the gallons of 140 F water per meal. Low priced restaurants on the average utilize 0.9 gal of 140 F water per meal, while medium- and high-price restaurants use 1.2 and 1.5 gal per meal, respectively.[3]

METHODS OF HEATING WATER

Hot water may be heated either by the direct combustion of fuel, by an intermediate carrier such as steam or hot water, or by electrically heated surfaces. The simplest method is to have the fire on one side of a metal barrier and water on the other. In such a method, if the surfaces for transferring heat are small, and if the water carries a heavy proportion of precipitable salts, the water passages may soon become clogged with resultant cracking or burning of the surface. A familiar example of such trouble is the water back in the kitchen stove, or the pipe coil inserted into the firebox of a warm air furnace or small boiler. The critical water temperature at which the lime, magnesia, etc., collect on hot surfaces, varies with the character and proportions of the solids, but generally such deposits are not a serious trouble below 140 F.

TABLE 10. HOT WATER DEMAND PER FIXTURES FOR VARIOUS TYPES OF BUILDINGS
Gallons of water per hour per fixture, calculated at a final temperature of 140 F

	Apartment House	Club	Gymnasium	Hospital	Hotel	Industrial Plant	Office Building	Private Residence	School	Y.M.C.A.
1. Basins, private lavatory	2	2	2	2	2	2	2	2	2	2
2. Basins, public lavatory	4	6	8	6	8	12	6		15	8
3. Bathtubs	20	20	30	20	20	30		20		30
4. Dishwashers	15	50–150		50–150	50–200	20–100		15	20–100	20–100
5. Foot basins	3	3	12	3	3	12		3	3	12
6. Kitchen sink	10	20		20	20	20		10	10	20
7. Laundry, stationary tubs	20	28		28	28			20	..	28
8. Pantry sink	5	10		10	10			5	10	10
9. Showers .	75	150	225	75	75	225		75	225	225
10. Slop sink	20	20		20	30	20	15	15	20	20
11. Demand factor	0.30	0.30	0.40	0.25	0.25	0.40	0.30	0.30	0.40	0.40
12. Storage capacity factor[a]	1.25	0.90	1.00	0.60	0.80	1.00	2.00	0.70	1.00	1.00

[a] Ratio of storage tank capacity to probable maximum demand per hour.

Coal-burning, direct-fired water heaters may be constructed of cored cast-iron sections, or of steel. In some cases the external appearance of the cast-iron sections is the same as in heating boilers, but internally the cores are changed to enable the sections to withstand the city water pressure. In small capacity water heaters, efficiency is not considered so important as low first cost and ability to maintain a fire at a low rate of combustion, and consequently, such heaters are generally built with a *dry* section or fire-brick lining at the base of the fire-pot to prevent too much chilling of the fuel. While mud and scale will eventually clog the waterways of any direct-fired heater, increased life may be obtained by providing a three-way cock in the return line between the heater and the bottom of the storage tank, so that water can be blown through the heater or the tank separately, at full line pressure, to clean out loose sediment. Clean-out openings in the bottom of the heater are advantageous, if used by operators of water heaters for periodic cleaning out of sediment.

Oil-burning, direct-fired water heaters usually are of steel, and operate with higher flame temperature and better efficiency than commensurate sized coal-burning heaters. As they have the same tendency as coal boilers to accumulate lime deposits, the water passages should be large in cross-section and accessible for periodic cleaning.

Gas-burning, direct-fired water heaters may be of the instantaneous or storage type. Instantaneous heaters are generally constructed of spiral water tubes of copper, around which the products of combustion circulate upward from high capacity burners. Storage-type heaters may include

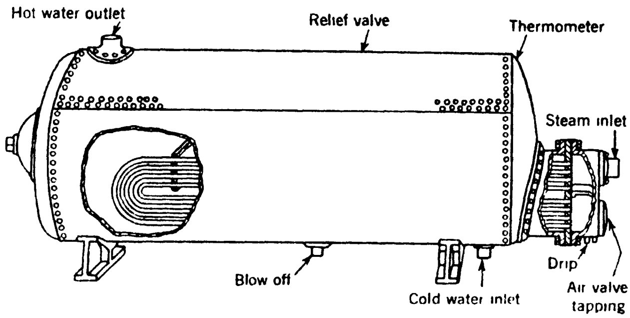


FIG. 8. INDIRECT WATER HEATER

in one unit an insulated storage tank, a combustion chamber, flues, burner equipment, and controls, or may consist of a separate storage tank and external direct-fired water heater, which may be a so-called *side-arm* heater for small capacity, or a gas-fired boiler for larger capacity. Gas boilers used for direct hot water supply must be able to withstand the city water operating pressure. While direct-fired gas heaters are used generally for residences and small installations of 100 gal storage capacity or less, indirect heaters are recommended for larger installations.

Chimney connections for all direct-fired, fuel burning water heaters are an important consideration. Refer to Chapter 16.

Electric water heaters for domestic hot water supply are described in the section Heating Domestic Water by Electricity in Chapter 41.

In the indirect method, either steam or hot water is used for heating the water. With steam, the water to be heated is preferably circulated around the outside of the steam tubes which are submerged within a tank. A typical indirect heater using steam is shown in Fig. 8. The coils usually are of copper, and are U-shaped to permit expansion and contraction. The shell may be of steel, with a protective coating or with a special inside protective lining, or may be of copper or copper alloy. Where straight heating tubes are used, one end of the tube is usually expanded into a *floating* head to take care of expansion. The coils should be capable of

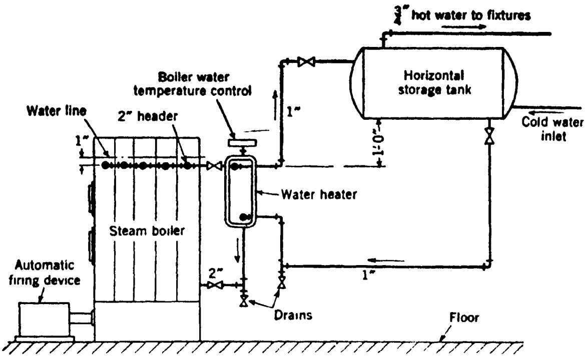


FIG. 9. INDIRECT WATER HEATER MOUNTED ON SIDE OF BOILER

easy withdrawal for inspection and for removal of scale. Instead of steam, the heating medium may also be hot water inside the tubes.

Another method of transferring heat from a heating boiler to the domestic water is illustrated in Fig. 9. The water heater is generally a cast-iron shell within which there is located a spiral copper coil. Hot water from the boiler circulates inside the shell and around the coil, and returns to the boiler, while domestic water from the storage tank circulates inside the coil. The storage tank should be installed with the bottom of the tank as far above the boiler as possible. Horizontal storage tanks of less than 18 or 20 in. diameter are not recommended because of the difficulty of preventing the hot and cold water from mixing, and especially is this an important consideration when large quantities of water are withdrawn. In Fig. 10 the heat transfer surface is placed inside the boiler instead of in a separate vessel, but otherwise the operation is similar to that of Fig. 9. This arrangement with vertical tank is commonly used for small domestic installations.

Sometimes the heating element is located inside of the larger type fire tube boilers and small residential boilers. In this case the heat transfer surface is in the form of a number of straight copper tubes, with rear U-bends or a floating head, inserted through a special opening in the boiler. While the coil may be placed in the steam space above the water line of a steam boiler, it is usually placed below the water line. Long coils of small diameter tubing, immersed in the water, are widely used without storage tanks. The rate of flow through the coil is limited by the friction loss in the coil,

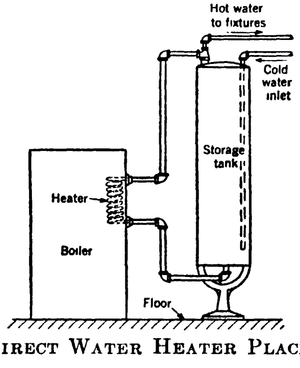


FIG. 10. INDIRECT WATER HEATER PLACED IN BOILER

and by fittings and restrictions, so that the water attains the desired temperature in one passage through the coil. This arrangement is frequently found in oil burner installations where the heating boiler, either steam or hot water type, is used to supply hot water during the summer. A thermostatic three-way mixing valve is frequently used to maintain a uniform temperature of the hot water supply to the plumbing fixtures.

In order to reduce clogging by precipitated solids, water heating plants sometimes develop steam in a closed circuit, transferring the heat through a tubular heater to the domestic water. The water in the primary heater, exposed to the high temperature of the fire, is repeatedly used and hence, has no appreciable tendency to deposit scale, while the domestic water, heated by steam at a much lower temperature than that of the fire, also exhibits a much reduced tendency to precipitate dissolved salts. Water characteristics, the effect of impurities and means of improving the quality of the water are important items, as brought out in Chapter 42.

COMPUTING HEAT TRANSFER SURFACE

The area of the inside surface of a heating coil may be determined from Equation 2.

$$A = \frac{Q \times 8.33(t_2 - t_1)}{U \times t_m} \tag{2}$$

where

A = surface area of coil, square feet.
Q = quantity of water heated, gallons per hour.
t_2 = hot water outlet temperature, Fahrenheit.
t_1 = cold water inlet temperature, Fahrenheit.
U = coefficient of heat transmission, Btu per (hour) (square foot) (Fahrenheit degree logarithmic mean temperature difference).
For copper or brass coils U = 240 (steam) and 100 (hot water).
For iron coils U = 160 (steam) and 67 (hot water).

t_m = logarithmic mean of the difference between the temperature of the heating medium and the average water-temperature, and is *approximately*:

$$t_s - \left[\frac{(t_2 + t_1)}{2}\right]$$

t_s = temperature of the heating medium, Fahrenheit.

Equation 2 may be used to check the heating coil ratings under temperatures other than those stated in the manufacturer's published ratings.

Example 6: What area of copper transfer surface will be required to heat 70 gal of water per hour from 40 to 180 F with boiler water at 220 F?

Solution:

$$t_m = \left[220 - \frac{(180 + 40)}{2}\right] = 110 \qquad A = \frac{70 \times 8.33(180 - 40)}{100 \times 110} = 7.42 \text{ sq ft}$$

For instantaneous submerged heaters, the surface required will depend upon (1) the velocity of water in the tubes, (2) the boiler water temperature, (3) the inlet water temperature, (4) the outlet water temperature, (5) the cleanliness of the coil surface, and (6) the condition of the boiler water surrounding the coil. If the heater is located in the water of an actively steaming part of a boiler, the heat transfer may be twice as great as would be obtained if the water surrounding the coil were circulating slowly. Ratings of instantaneous water heating coils will therefore vary greatly, depending upon the assumptions made regarding the conditions of operation. The values of the coefficient of heat transmission for instantaneous heaters, shown in Table 11, are conservative.

For a coil in which heat is transferred from steam to water, the value of $U = 300 \sqrt{v}$ may safely be used (v = velocity of water in feet per second).

The rate of heat transfer, between steam or water as the carrier, and the domestic water, is influenced by the rate of movement of both the carrier and the water which receives the heat. For this reason, where the transfer occurs from heating system water to domestic water, it is good practice to install a circulating pump to insure rapid movement of the boiler water.

In view of the high condensation rates obtained when steam is used with gravity circulation from the boiler, as when there is a sudden demand followed by an inflow of cold water, the bottom of a steam heating transfer element always should be at least 30 in. above the boiler water line, and the steam and condensate return pipes should be of liberal size. Otherwise,

TABLE 11. COEFFICIENT OF HEAT TRANSFER OF INSTANTANEOUS WATER HEATERS
U = Btu per (hr) (sq ft) (Fahrenheit degree logarithmic mean temperature difference)

Boiler water temperature	210	200	180
U	225	175	150

water hammer and reduced capacity may result, due to imperfect drainage of condensate.

When connecting a transfer-type hot water heater below the waterline of a cast-iron steam boiler having vertical sections, there should be a separate tapping for water circulation into every section of the boiler, as shown in Fig. 9, unless the boiler has large top nipple ports providing inter-sectional circulation. If the top nipples are entirely within the boiler steam space, no internal circulation occurs between sections. Steaming may then occur in the boiler sections not connected to the heater and, further, the unconnected sections will not deliver any heat to the water heater.

HOT WATER SUPPLY PIPING

It is common practice to provide circulating piping in all hot water supply systems in which it is desirable to have hot water available continuously at the fixtures. In average-sized and small residences and systems, in which the piping from the heater to the fixtures is short, return circulating piping is generally omitted in order to reduce installation cost, and to reduce heat loss from the piping, particularly during periods of no water demand.

The hot water supply may be distributed by either an up-feed or down-feed piping system. Three common methods of arranging the circulating lines are shown in Fig. 11. Although the diagrams apply to multi-story buildings, the arrangements (*a*) and (*b*) are sometimes used in residential designs.

A check valve should be provided in the run-out from each return riser to prevent temporary reversal of flow in the piping when a faucet is open. Proper air venting of a circulating system is extremely important, particularly if gravity circulation is employed. In Fig. 11 (*a*) and (*b*), this is accomplished by connecting the circulating line below the top fixture supply. With this arrangement, air is eliminated from the system each time the top fixture is opened.

Where an overhead supply main is located above the highest fixture as in Fig. 11 (*c*), an automatic float type air vent is installed at the highest point of the system, or a fixture branch is connected to the top of the main where air venting is desired, and then dropped to the fixture outlet.

It is sometimes necessary to make an allowance for pressure drop through the heater when sizing hot water lines, particularly where instantaneous hot water heaters are used and the available pressure is low.

The principles involved in the sizing of the hot water supply pipes are the same as those for the sizing of cold water supply lines. For small and medium sized installations a ¾-in. hot water return will be ample. For larger installations, the size of the hot water return may be computed from considerations of the heat losses in the hot water piping.[4] A throttling

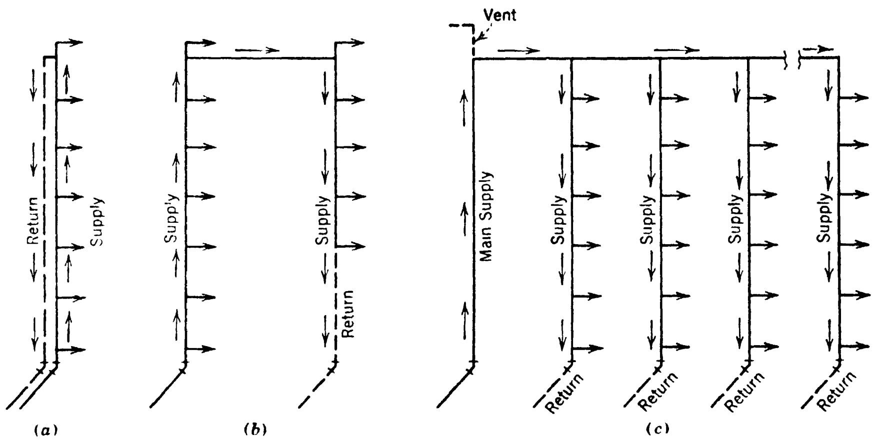


FIG. 11. METHODS OF ARRANGING HOT WATER CIRCULATION LINES

valve should be placed in the hot water return pipe so that the rate of circulation may be adjusted.

Where the hot water piping system is exceedingly long, a water circulator is frequently installed and controlled from an immersion thermostat (in the return line) set to start and stop the pump over approximately a 20 F deg temperature range.

CONTROL OF SERVICE WATER TEMPERATURE

Coal-fired boilers are usually controlled by an immersion thermostat (located in the heated water) which opens or closes draft dampers at the boiler to adjust the rate of fuel combustion. With oil or gas-fired boilers, the immersion thermostat controls the oil burner or the automatic gas valve. The gas pilot flame usually burns continuously. With electric heaters, the immersion thermostat operates a switch on the source of energy.

When steam or hot water is the medium for heating the water in the tank, an immersion thermostat is used to control a valve in the steam or hot water supply line. In small residence installations, using water as the carrier, a combined immersion thermostat and butterfly valve in one simple fitting may be installed in the transmitting circuit to prevent over-heating of the service water.

In residences heated by pump circulated hot water, the house temperature is controlled by operating the circulating pump intermittently, while domestic hot water is warmed by transfer from the house boiler, independent of the pump operation. The domestic water is heated from the heating boiler the year 'round. Under such an arrangement, to prevent overheating the house by thermal circulation when the pump is not running, it is usual to insert a weighted check-valve in the house heating main so that no circulation to the house heating system can occur unless the pump operates. In summer the fire may be controlled to maintain a boiler water temperature lower than when heating, and generally about 20 F warmer than that desired in the domestic hot water system.

The immersion thermostat in a hot water storage tank should be located no higher than the center of the tank, and possibly should be even closer to the bottom, since water in a tank stratifies proportionally to the temperature. When hot water is removed, the cold water entering to replace it quickly reduces the temperature in the lower parts of the tank.

SAFETY DEVICES FOR HOT WATER SUPPLY SYSTEMS

There are still numerous plumbing codes which do not have regulations for the prevention of hot water storage tank explosions.

An ordinary storage tank is under certain water pressure, depending on the static pressure in the system. When the water in the tank is heated by circulation through an external heater, or by heating units in the tank, it gradually expands. For instance, if the entire contents of a 30-gal storage tank are heated from 70 F to 160 F, there will be an increase in volume of about one-half gallon. If the tank is connected to some supply without any intervening check valve, the increase in volume causes part of the water to be pushed back into the supply line. If the hot water reaches the water meter, it may ruin the composition discs.

If back-flow cannot occur, as for example, due to the use of a check-valve or a pressure-reduction valve in the line, or because of temporary shut-off of the cold-water line, the pressure in the tank rises as heating continues. Such a pressure rise, if the heating continues for any length of time, may result in rupturing of the tank.

If the tank water is not confined, continued heating would, of course, cause no increase in pressure beyond ordinary line pressure. However, the temperature would continue to increase. If this should happen at elevated temperatures, the flashing of the water into steam might cause a serious tank explosion.

In order to guard against the development of excessive pressure inside a hot water storage tank (owing to the thermal expansion of the water), it is customary to install a spring-loaded pressure-relief valve which is set to open at a pressure about 20 psi higher than the normal line pressure.

The amount of water discharged by any pressure-relief device is usually quite small, since it takes only a small quantity of water to relieve any pressure rise due to the thermal expansion of the water. The rate of discharge should be such as to limit the pressure rise for any given heat input to 10 percent of the pressure at which the valve is set to open. For any given installation, the discharging capacity should be in excess of the water to be discharged by the heater. The heater discharge Q_m in gallons per minute may be computed by Equation 3:

$$Q_m = 0.000005b \tag{3}$$

where b is the heat input of heater in Btu per minute.

To prevent danger of overheating, temperature-relief devices should be used. One type of such a device has a fusible plug of metal which melts at about 212 F. The hot water then runs from the opening until the device is serviced. A better type of relief device is one which incorporates a bellows or bimetallic disc, which opens at a temperature of 210 F and closes when the temperature drops to approximately 160 F.

Another type of safety device is a fuel cutoff switch or valve in which a fusible plug is melted by excessive water temperature. The fuel supply is cut off until the fusible plug is replaced.

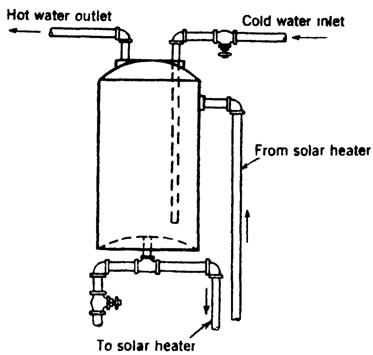


Fig. 12. Solar Heater Tank Connections

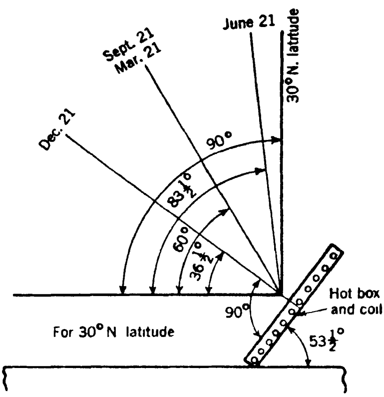


Fig. 13. Solar Heating Coil Inclination

The capacity of temperature relief devices may be calculated by Equation 4:

$$Q_m = 0.0008b \tag{4}$$

where Q_m and b denote the same quantities as in Equation 3.

Pressure relief valves should be installed in the cold water line near the tank out of contact with the hot water in order to prevent excessive corrosion and lime deposit on the valve seat. *Temperature relief* valves, in general, must be installed at the point of maximum water temperature.

SOLAR WATER HEATERS

Solar heaters utilize the energy of the sun for heating water. The successful operation of such heaters requires the availability of sunshine practically every day in the year, which has limited their use to Florida and the southern portions of California. When supplemented with some other means of gas, coal, or oil water heating, solar heaters may be used in climates where sunshine may be more or less intermittent. They have been used in summer homes as far north as Chicago. When properly installed and proportioned, solar water heaters render satisfactory service, especially in climates where the outside temperatures are high and extremely hot water is not necessarily desirable. Such installations consist essentially of a storage tank, heating coil, and hot box. The coil is installed in the hot box, and is arranged to circulate water to and from the storage tank. The advantage in the use of this type of heater is the fact that it requires no fuel. The same materials should be used for the coil, circulation lines, and tank. A copper coil is more efficient in absorbing heat in the box, but galvanized iron or steel may be substituted, depending on the local water conditions, cost, and other considerations.

The storage tank must be able to store sufficient heated water for the night period of about 16 hr when the coil is not functioning, or is operating under such poor sun conditions as to make its heating effect negligible. Due to the fact that the *no sun* period includes the night period when little or no hot water is used, an available storage of 50 percent of the average daily usage is considered adequate. Since about 25 percent of stored hot water cannot be drawn out of a storage tank before the incoming cold water reduces the temperature of all of the water in the tank to an unsatisfactory point for usage, the equation for calculating the storage capacity of the tank becomes:

$$S = \frac{Q_d \times 0.50}{0.75} = 0.666Q_d \quad (5)$$

where

S = storage capacity of tank, gallons. Q_d = average daily usage, gallons

Thus, for a family of four persons using an average of 40 gal of hot water per (person) (day), the size of the tank would be 4 persons x 40 gal x 0.666 or 106 gal, and the nearest standard size of tank would be used. The tank should be well insulated to prevent undue loss of heat during the 16-hour period when the coil is inoperative, and it should be located as high as possible in the building (under the peak of the roof if such exists) so as to secure a maximum circulation head from the coil. The hot water supply line to the house, as shown in Fig. 12, is connected to the top of the tank and serves to vent the air from the tank through the hot water faucets as fast as it accumulates.

The coil should be of the return-bend type (square or slightly rectangular in form), and should have the pipes running east and west, with the coil on the south side of the building where it can receive the full sun effect all day long without shadows from the building itself, or from adjacent obstructions such as trees or other structures. The coil should be placed as low as possible in relation to the storage tank level, such as on a porch roof, the roof of a one-story extension or, if necessary, even on the ground. Both the coil and the circulation lines should be designed to facilitate the circulation flow as much as possible, using long radius copper fittings or recessed galvanized iron fittings to match the materials of the coil, circulation lines, and tank. The coil should be inclined, as shown in Fig. 13, so that the north end is raised above the south end to secure an angle with the horizontal of about 53 deg. This will result in the inlet end of the coil being on the south side (or bottom), and the outlet end being on the north side (or top). This will satisfy conditions along the 30-deg N latitude, which includes the portions of Florida and Southern California where these heaters are most frequently used.

The hot box is usually constructed of wood on the four sides and bottom, and is insulated. Glass sash are placed over the top of the box, and the box should be constructed as air-tight as possible. The interior surfaces should be painted white to reflect the heat, while the coil should be painted black to absorb the heat. The box need not be deeper than necessary to house the coil and to protect it from the weather.

TABLE 12. SUGGESTED SOLAR HEATER DESIGN DATA[a]

Design Item	Based on Rate of 30 Gal per Day per Person								Based on Rate of 40 Gal per Day per Person							
No. of Occupants in Residence	1	2	3	4	5	6	7	8	1	2	3	4	5	6	7	8
Hot Water Used at Night, gal per person	15	15	15	15	15	15	15	15	20	20	20	20	20	20	20	20
Hot Water Used at Night, gal total	15	30	45	60	75	90	105	120	20	40	60	80	100	120	140	160
Retained in Tank, 25 per cent, gal	4	8	11	15	19	23	27	30	5	10	15	20	25	30	35	40
Tank Capacity Required, gal	20	40	59	75	94	113	130	150	25	50	75	100	125	150	175	200
Hot Water Used During Day, gal	15	30	45	60	75	90	105	120	20	40	60	80	100	120	140	160
Total Water to be Heated:																
Gal per 8 hr period	35	70	104	135	169	203	235	270	45	90	135	180	225	270	315	360
Gal per hour	4.5	9	13	17	21	26	29	34	6	12	17	23	28	34	39	45
Copper Coil Required:																
Surface area, sq ft	25	50	75	100	121	145	168	192	32	64	96	128	160	192	224	256
Equivalent length 1 in. coil, ft	100	200	300	400	484	580	664	768	128	256	384	512	640	768	896	1024
Box Size:																
Area, sq ft..	25	50	75	100	121	145	168	192	32	64	96	128	160	192	224	256
Width, ft ..	4	6	7	8	9	10	10	11	4	6	8	9	10	11	12	12
Length, ft ..	6	8	11	12.5	13.5	14.5	16.5	17.5	8	10	12	14	16	18	19	21

[a] Sun Effect and the Design of Solar Heaters, by H. L. Alt (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 131).

The addition, on the bottom of the box, of a light gage copper plate to which the pipe of the coil is soldered for good metallic contact, will add to the amount of heat received by the coil, due to the fact that this plate will receive all of the sun's rays which fail to directly strike the coil. The heat from this source is transmitted to the coil through the plate rather than from the heated air surrounding the coil. Otherwise, only part of the heat enters the coil, the balance being transmitted through the glass.

Design data given in Table 12 may be used with judgment in selecting the size of solar heater coil and box for a particular application. These data are based on consumptions of 30 and 40 gal of hot water per day per person.

DOMESTIC HOT WATER BY HEAT PUMP

Hot water may suitably be obtained by using a heat pump installation. The hot water heater may be either a heat exchanger installed just ahead of the compressor of a heat pump installation, or may be a self-contained domestic-water heat pump.

Various designs of self-contained domestic water heat pumps are available, and one particular arrangement is shown in Fig. 14.

Hot water heating by means of a heat pump is not yet advisable where

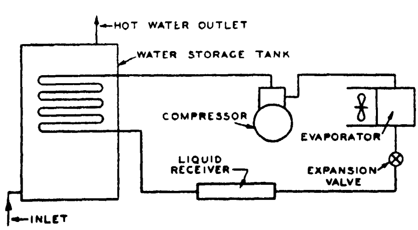


FIG. 14. HEAT PUMP ARRANGEMENT FOR HOT WATER SUPPLY

high water temperatures are desired. This is due to the fact that higher water outlet temperatures result in lower coefficients of performance.

For coefficients of performance of 4 or higher, the heat pump water heater may be more economical to operate than a conventional water heater.

Although the first cost of domestic hot water heat pumps is somewhat high, they have the advantages of eliminating products of combustion, odors and soot, and not needing a chimney. A further advantage is that they may be used for cooling purposes. With a coefficient of performance of 2½ to 3, water temperatures of 140 to 150 F may be obtained.[5]

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CHAPTER 49

INSTRUMENTS AND MEASUREMENTS

Temperature Measurement, Pressure Measurement, Air Flow Measurement, Air Change Measurement, Humidity Measurement, Heat Transfer Through Building Materials, Evaluation of Thermal Environment, Combustion Analysis, Smoke Density Measurements, Determination of Air Contaminants, Sound and Vibration Measurements

HEATING and air conditioning engineers and technicians require instruments for both laboratory and field use and somewhat more precision is attainable and essential in the laboratory, where research and development are undertaken, than in the field, where acceptance and adjustment tests are conducted. Some instruments have attained an adequate state of development, while others fall far short of the desirable. For instance, temperatures can now be measured readily with ample accuracy for most purposes, while a convenient and precise method for determining or comparing the dustiness of atmospheres is lacking, and improvement in existing hygrometers and humidity controllers is essential.

Codes and standards covering different types of heating and air conditioning devices and apparatus have been promulgated by various authoritative organizations, and instruments essential for performance or compliance testing are enumerated in the relevant publications.[1, 2, 3, 4] The present purpose is to discuss the use and characteristics of the more important instruments.

TEMPERATURE MEASUREMENT[5, 6]

Thermometers

Any device capable of indicating temperature is a thermometer, but in common usage the term *thermometer* without qualification has come to signify the ordinary mercury-in-glass temperature indicating device. This type has a useful range from −40 F, the freezing point of mercury, to about 1000 F, at or near which the glass usually softens. Lower temperatures can be measured with alcohol-filled thermometers for which the range is about −94 F to +248 F. The better thermometers have their scales, either Fahrenheit or Centigrade, etched with acid into the glass which forms their stems. The probable error for etched stem thermometers is plus or minus one scale division, and calibration is desirable for much test work. Thermometers are calibrated during manufacture at not less than two temperatures—the freezing and boiling points of water—and calibration is often accomplished with the instrument completely immersed in a bath at the known temperature. The intervening scale divisions are then applied to the stem. When such a thermometer is used with the stem incompletely immersed, a correction known as the stem correction is necessary for accurate measurements, and its magnitude is usually computed by means of the following formula:

$$K = 0.00009\ D\ (t_1 - t_2) \qquad (1)$$

where

K = correction to be added, Fahrenheit degrees.

D = number of degrees on the thermometer scale which are not immersed.
t_1 = temperature indicated on the thermometer, Fahrenheit degrees.
t_2 = temperature of the non-immersed mercury column, Fahrenheit degrees.
0.00009 = difference in the coefficient of expansion of the mercury and glass.

When a thermometer is used in a liquid or in air or gas near room temperature, the effects of radiation can often be ignored, but when the temperature of hot air or gas is desired, means are usually provided for minimizing the effect of radiation.[7] These include bright metallic shields around the thermometer bulb, and the use of aspirated thermometers in which a stream of the air or gas is drawn at considerable speed across the bulb and increases the influence of the gas temperature on the thermometer indication. In any case, to prevent errors in temperature measurements, there should be ample circulation so that the thermometer will indicate a true temperature of the medium under observation, and ample time should be allowed for the thermometer to reach the same temperature as the medium. In reading a thermometer the eye should be at the same level as the top of the liquid to avoid parallax.

Industrial-type thermometers are available for permanent installation in pipes or ducts. These instruments are fitted with metal guards to prevent breakage, and are useful for many purposes. However, the considerable heat capacity and conductance of the guards or shields prevent such thermometers from following closely the fluctuations in a medium of varying temperature.

Thermocouples

When two wires made of dissimilar metals are joined by soldering, welding or merely by twisting, a thermocouple or thermo-junction is formed and an electromotive force, which depends upon the temperature of the junction, is found to exist between the wires. When the wires are joined at two points a thermocouple circuit is formed, and if one junction is kept at a temperature different from the other, an electric current flows through the circuit due to the difference in emf developed by the two junctions. This phenomenon is employed for temperature measurements in thermocouple systems, one junction being ordinarily kept at a constant temperature, as in an ice bath, while the other junction is placed at a point at which it is desired to observe the temperature. In practice it is desirable to utilize emf to indicate temperature because, at small or zero current flow, the resistance of the circuit is unimportant. A high resistance millivolt meter is useful in some cases but the potentiometer yields better results. In the potentiometer the electromotive force generated by the thermocouples is balanced against an electromotive force from the battery so that observations are made with no flow of current through the thermocouple circuit. A conventional arrangement is illustrated in Fig. 1. The thermocouple leads A-B are so connected that their polarity opposes that of battery C. If the position of E on the graduated slide wire rheostat DF is adjusted until galvanometer G shows no current flowing, resistance DE will indicate directly the voltage generated by the thermocouple. In order to calibrate the instrument, switch S is thrown over to the standard cell circuit while rheostat R is adjusted so that the galvanometer shows zero current. Battery C then exerts the known voltage of the standard cell at DH.

The choice of materials for thermocouple wires is determined by the range of temperature to be measured. Up to about 600 F base metals such as iron-constantan or, preferably from the corrosion standpoint, copper-constantan are satisfactory and develop a relatively large emf of 40 to 60 microvolts per degree. Chromel-alumel couples are useful in the flue gas temperature range, while platinum (platinum-rhodium) couples are used for higher temperatures. Impurities make large differences in the performance of thermocouple wires and for this reason calibration of samples from each spool of wire is essential for precise work. Data on wire can usually be obtained from the manufacturer.

The act of adjusting rheostat D-F (Fig. 1) for zero current flow is known as balancing the potentiometer. Automatic self-balancing instruments of both the indicating and recording types are on the market. They usually contain an automatically compensating cold junction to avoid the use of an ice bath, and special thermocouple wire is furnished with them from the factory.

With a suitable potentiometer, small wires serve as well for thermocouples as large ones, and the fineness of the wires is limited only by consid-

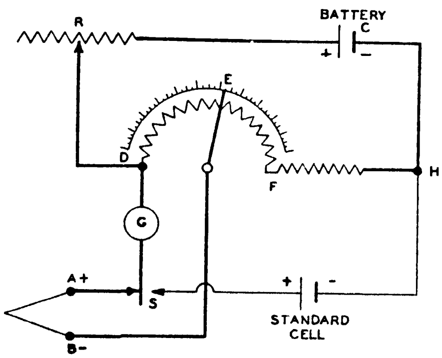

FIG. 1 BASIC CIRCUIT AND CONNECTIONS FOR THERMOCOUPLE AND POTENTIOMETER

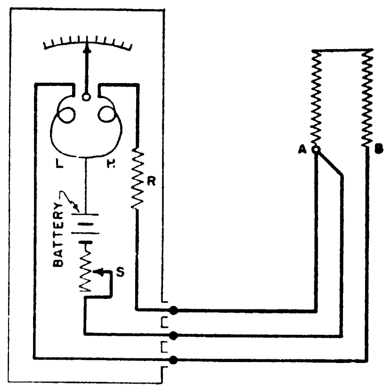

FIG. 2. TYPICAL RESISTANCE THERMOMETER CIRCUIT AND CONNECTIONS

eration of mechanical strength and convenience in handling. Small couples respond more promptly to changes in temperature and are less affected by radiation than large ones. For use in heated air or gases, thermocouples are often shielded, as are thermometers, and aspirated thermocouples are sometimes used. An arrangement has been described for avoiding error due to radiation. It involves several thermocouples of different sizes, the true temperature being estimated by extrapolation of readings to zero diameter.[8]

By the use of thermocouples, temperatures at remote points may be indicated or recorded on conveniently located instruments; average temperature may be readily obtained by connecting several couples in parallel or in series; and temperatures may be obtained within thin materials, narrow spaces, or otherwise inaccessible locations.

Thermocouples in series with every alternate junction maintained at a common temperature will give an emf which, divided by the number of couples to give the average emf[9] per couple, may be used to find the average temperature.

Thermocouples in parallel, having the similar metals of a number of couples connected together and run to a common cold junction, will cause an indication on a potentiometer which is the true emf only if the electrical resistances of the parallel junctions are the same.[9, 10]

Temperatures of surfaces below red heat are difficult to determine by any other means than thermocouples. For this purpose, a thermocouple of fine wires is preferable to minimize the possibility of error due to the conduction of heat along the wires. It may be attached to a metal surface in any of several ways. For permanent installations, soldering, brazing or peening may be desirable. A small hole is drilled for the peening operation; the thermocouple is inserted and the metal is peened to retain it. The fact that the thermocouple is in electric contact with the surface is unimportant in usual circuits. For temporary arrangements, couples may be attached by means of surgical or cellophane tape. For many boiler or furnace surfaces, furnace cement serves very well. The thermocouple may be attached by means of the cement when the surface is cold, and must be treated gently and usually supported until the cement dries, due to heat, and hardens—after which it has ample strength. It is good practice to use as little cement as possible, and also to plaster the wires to the surface for an inch or so from their junction to avoid errors due to heat conduction along the wire. Electric insulation between the wires should be perfect except at the junction since, otherwise, the indicated emf will be between those existing at the junction and at the short circuit

Resistance Thermometers

Resistance thermometers depend for their operation upon the change of electric resistance of metal with change in temperature. The resistance generally increases with rising temperature. Their use largely parallels that of thermocouples, although readings tend to be unstable above 950 F. Two-lead temperature elements are not recommended, since they do not permit correction for lead resistance. Three leads to each resistor are necessary to obtain consistent readings.

A typical circuit used by several manufacturers is shown in Fig. 2. In this design a differential galvanometer is used, in which coils L and H exert opposing forces on the indicating needle. Coil L is in series with the thermometer resistance AB, and coil H is in series with the constant resistance R. As the temperature falls, the resistance of AB decreases allowing more current to flow through coil L than through coil H. This causes an increase in the force exerted by coil L, pulling the needle down to a lower reading. Likewise, as the temperature rises the resistance of AB increases, causing less current to flow through coil L than through coil H. This forces the indicating needle to a higher reading. Rheostat S must be adjusted occasionally to maintain a constant flow of current.

As compared to the thermocouple, the resistance thermometer does not require a cold junction, and it can be simply scaled for more accurate measurements; but, generally because of its construction it is more costly and is apt to have considerable lag. It gives best results when used to measure steady or slowly changing temperature. For accurate results the entire thermometer coil must be exposed to the temperature to be measured.

Pyrometers

The pyrometer is the usual instrument for measuring high temperatures such as those of incandescent bodies or furnace interiors. There are two types. In the radiation pyrometer the radiant energy from an observed surface falls on a thermopile, and the emf generated by the pile, measured by a galvanometer or potentiometer, is an index of the surface temperature. With the optical pyrometer a narrow spectral band, usually red, emitted by the surface, is matched visually with the filament of a special electric lamp. The emf necessary to cause the filament to match the surface in brightness is the index of the temperature of the surface. Pyrometers are calibrated by means of various metals with known melting or freezing points. Portable as well as laboratory models are manufactured.

Color Indicating Crayons

Crayons are available, the marks of which change color or melt at specified temperatures. Such crayons have been sold in boxes covering the range from about 100 F to about 800 F in 100 deg steps, with a precision of some 10 deg. They are a rough but convenient means of determining temperatures, and of locating isothermal lines on surfaces below red heat.

PRESSURE MEASUREMENT

Pressure Gages

The Bourdon is the most common type of pressure gage, and its appearance probably is familiar to any one having an acquaintance with power plants or laboratories. The essential element of such a gage is the Bourdon tube, a metal tube of oval cross-section curved along its length to form almost a complete circle. One end is closed and the other is connected to the vessel in which the pressure is to be measured. With an increase of pressure, the tube tends to straighten, and vice versa, and the resulting motion of the closed end is communicated by suitable linkages to a needle moving over a graduated dial. If the range is above about 20 psi, such gages are usually calibrated by means of a *dead weight tester*, whereby known pressures are produced in a fluid by imposing known weights on a piston of known area. Gages are commonly set to read accurately at or near the pressure of probable use. Gages of several different types or qualities are available on the market.[11] Suction gages and pressure gages with ranges below about 20 psi are ordinarily calibrated against mercury manometers.

Manometers

The manometer is a simple and useful means for measuring partial vacuum and low pressure. It is, moreover, a primary instrument; it does not require calibration, and it is often used as a standard for the calibration of other instruments. It is so universally used that both the inch of water and the inch of mercury have become accepted units of pressure measurement. In its simplest form, the manometer consists of a U-shaped glass tube partially filled with a liquid. A difference in height of the two fluid columns denotes a difference in pressure in the two legs, which is proportional to the difference in height.

For converting manometer readings into other pressure units, certain proposed standard factors are applicable for precise work. These are based on a standard gravitational acceleration of 32.1740 ft per (sec) (sec) and are as follows:

1 Standard Atmosphere = 14.696 lb per sq in. = 29.921 in. mercury at 32 F
= 33.96 ft water column at 68 F

For most ordinary purposes, the following figures are of ample accuracy:

1 Atmosphere = 14.7 lb per sq in. = 29.9 in. mercury
= 34.0 ft (408 in.) water column

Manometer tubes should be chemically clean. The bore is not important, except insofar as it affects the meniscus through wetting or surface tension. Bores of at least $\frac{3}{16}$ in. for rough, and $\frac{1}{2}$ in. for more precise, measurements are recommended. Liquids other than water are sometimes used for low pressure measurement and, when this is done, the readings must be corrected for the density of the fluid used.

For measuring pressure differences of a few inches of water, or less, U-gages are often set at an angle for scale amplification. In many gages of this type, commonly termed draft gages or inclined manometers, only one tube of small bore is used and the other leg is replaced by a reservoir. The scale is calibrated to read in inches of water, and it is necessary to use a fluid having the same gravity as that for which the gage was originally calibrated, or to apply a correction if another fluid is used. Such gages may be checked one against another. For more accurate calibration the gage may be checked against a micromanometer or a calibrating device known as a hook gage.[12] The accuracy of a draft gage is dependent on the slope of the tube, and consequently the base of the gage must be leveled carefully. It is not desirable to use a slope of less than 1 in 10. Where pressures are read under extreme conditions of temperature, and calibration is possible only at normal temperature, it is necessary to correct for the change in density of the liquid in the manometer.[13] For measuring low pressure differences to within 0.001 in. of water, very sensitive micromanometers are available, such as the Illinois or Wahlen, the Meriam, the Trimount, and the Emswiler.[14, 15]

When using a manometer or other pressure gage for measuring air flow, by means of orifices, the type of duct openings used for manometer connections and their location are important. Where velocities are low, as in some plenum chambers, or where the flow is free of large eddies and parallel to the walls of a duct, a drilled hole cleared of burrs and at right angles to the stream, is satisfactory.[16] For higher velocities, it is common practice to provide four holes or taps around the periphery of the duct. Diametrically opposite pairs of taps are connected together, and then such pairs are manifolded together.

An alternate method involves the use of the Pitot tube, shown in Fig. 3. This instrument should be pointed up-stream, parallel with the air flow Where the flow is not axial or parallel to the side walls of the duct, a very close approximation of the static pressure and the flow direction can be obtained by a Fechheimer tube.[17]

Barometer

The simplest and earliest type of barometer consists of a glass tube somewhat more than 30 in. long filled with mercury, and inverted in a cup partially filled with mercury. The height of the mercury column in the tube above the mercury surface in the cup is a measure of the existing atmospheric pressure, except for the slight pressure of the mercury vapor in the space above the mercury in the tube. This can ordinarily be ignored.

Elaborate mercury barometers, fitted with vernier scales, are available. For precise work, corrections must be made for the thermal expansion of the mercury and of the scales.[18] The instruments are usually calibrated for 32 F mercury and 62 F scale temperature, and the correction C to be subtracted from the observed barometer's height is obtained by means of Equation 2.

$$C = \frac{h(t - 28.630)}{(1.1123\,t - 10978)} \tag{2}$$

where

C = correction to be subtracted, inches of mercury.

h = observed height, inches of mercury.
t = observed temperature of the barometer, Fahrenheit degrees.

Standard atmospheric pressure at sea level is 29.921 in. Hg, and since normal atmospheric pressure decreases about 0.01 in. Hg for each 10 ft increase in elevation, it is important to make a correction if the elevation of the barometer is not that of the test apparatus. In many cases the barometric reading may be obtained from a nearby Weather Bureau Station, in which case inquiry should be made as to whether the value is for *station* or *sea level* pressure.

Atmospheric pressure may also be measured by an aneroid barometer which is easily portable. In this type, variations in atmospheric pressure deflect the thin surface of a sealed diaphragm capsule. Most commercially available aneroid barometers are not as accurate as the mercurial type, and the best require occasional recalibration. Open-scale aneroid barometers are more expensive than common mercurial barometers. Most of the pressure gages used in engineering work indicate gage pressures, that is, the

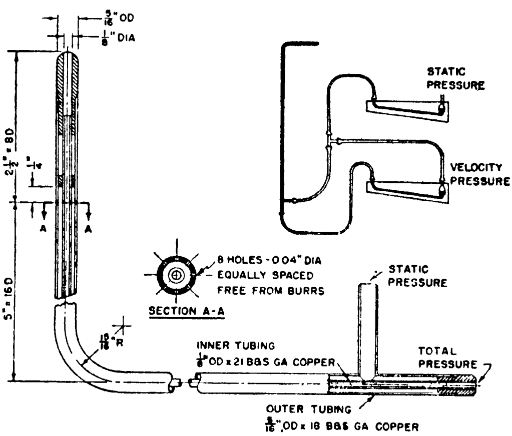

FIG. 3. STANDARD PITOT TUBE

difference between the pressure being measured and the atmospheric pressure. Such pressures are called gage pressures. Absolute pressure may be obtained by adding barometric pressure and gage pressure algebraically.

AIR FLOW MEASUREMENT

The theory of various means for measuring the flow of fluids is discussed in Chapter 4, Fluid Flow. Heating and air conditioning engineers are called upon to measure the flow of air more often than that of other gases, and usually the air is measured at or near atmospheric pressure. Under this condition, the air can be treated substantially as an incompressible fluid which implies that simplified formulas can be used with sufficient accuracy for the solution of many problems.[19]

The Pitot Tube

The construction of the Standard Pitot Tube[12] is shown in Fig. 3. The formula for velocity used in conjunction with it, is as follows:

$$V_m = 1096.5 \sqrt{\frac{h_{aw}}{\rho}} \tag{3}$$

where

V_m = velocity, feet per minute.
h_{sw} = velocity pressure (Pitot tube manometer reading), inches of water.
ρ = density of air, pounds per cubic foot.

Since the velocity in a duct is seldom uniform across any section, and since a Pitot tube reading indicates a velocity at only one location, a *traverse* is usually made to determine the average velocity so that the flow can be computed. Suggested Pitot tube locations for traversing round and rectangular ducts are shown in Fig. 4. In general, the velocity is lowest near the edges or corners, and greatest at or near the center. For this reason a large number of readings should be taken (in the case of round ducts not less than 20) along two diameters at centers of equal annular areas. In rectangular ducts the readings should be taken in the center of equal areas over the cross-section of the duct. The number of spaces should not be less than 16, and need not be more than 64. When less than 64 are taken, the number of equal spaces should be such that the centers of the areas are not more than

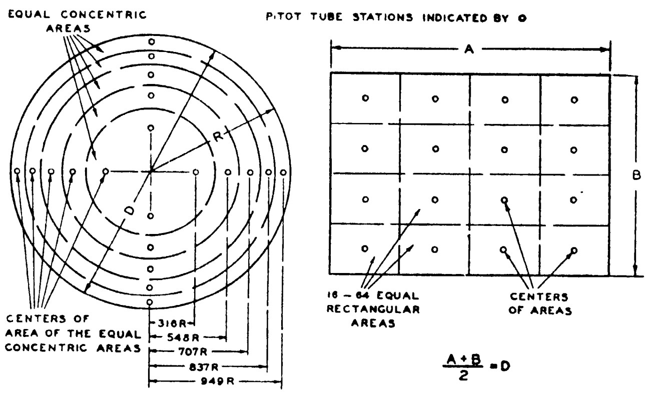

FIG. 4. PITOT TUBE TRAVERSE FOR ROUND AND RECTANGULAR DUCTS

6 in. apart. In determining the average velocity in the duct from the readings given, the calculated individual velocities or the square roots of the velocity heads must be averaged. It is incorrect to use the average velocity head for this purpose. Pulsating or disturbed flow will give erroneous results and therefore, if possible, the Pitot tube should be located at least 7½ diameters down-stream from a disturbance such as that caused by a turn; or a criss-cross type of flow straightener should be installed in the duct 1½ diameters ahead of the Pitot tube.[12] Flow straighteners do not equalize flow velocity across a duct. They merely serve to improve the precision of measurements. Equalization can be effected, if desirable, for measuring purposes by the use of wire netting, perforated plates or cloth screens across the duct.

Many forms of Pitot tubes, other than the one described, have been used and calibrated.[20] A double-ended tube,[21] one end pointing down-stream, and one up-stream, is sometimes used for low velocities, but it should be carefully calibrated for accurate results. A special form of this tube design consists of two straight ⅛ in. tubes soldered together, closed at the end, and with a 0.04 in. hole in each tube opposite the line of contact. This tube is useful in exploring velocities in exhaust inlets, such as hoods placed around grinding wheels. To meet special conditions, different sized Pitot tubes which are geometrically similar to the standard tube can be used.

Plate Orifices

Application of the Pitot tube is often inconvenient when velocities are low, because the resultant velocity pressures become so small that extraordinary means are necessary for measuring them. In addition, velocity surveys of the whole cross-sectional area of a duct are inexpedient when numerous test runs are in prospect. Chiefly for these reasons orifices are favored for much test work. There are two types: the plate orifice and the shaped orifice or nozzle. Plate orifices are simple to construct and convenient to use, in that a frame can be made to support them in the duct such that one can be removed and another inserted when it is desirable to use an orifice of a different size.

Formulas for Orifices

In the heating and air conditioning fields it is usually convenient to obtain orifice pressure drops in inches of water column, temperatures in Fahrenheit degrees and barometric pressure in inches of mercury. The air flow is usually desired in cubic feet per minute at the existing condition, so that velocities in various ducts can be computed, and in pounds per hour so that computations of heat transferred by the air can be based on weight, temperature change and specific heat. Equation 4 is applicable.

$$Q_M = 5.2KYD^2 \sqrt{\frac{T_f}{B_f} h_w} \tag{4}$$

where

Q_M = air flow, cubic feet per minute.
K = orifice coefficient.
Y = expansion factor, see Fig. 5.
D = orifice diameter, inches.
d = pipe diameter, inches.
T_f = temperature of air at orifice, Fahrenheit degrees, absolute.
B_f = absolute pressure ahead of orifice, inches of mercury.
B_a = absolute pressure after orifice, inches of mercury.
h_w = pressure drop through orifice, inches of mercury.

As most laboratories are less than 1000 ft above sea level, precision is adequate in many cases if standard atmospheric pressure, 29.92 in. Hg, is assumed. Equation 4 then becomes

$$Q_M = 0.95\, KYD^2 \sqrt{T_f\, h_w} \tag{5}$$

After the flow in cubic feet per minute is determined, it can be expressed in pounds of air per hour by means of the relation

$$W = 60\, \frac{PQ_M}{RT_f} \tag{6}$$

P = pressure, pounds per square inch, absolute.
R = 53.3, the gas constant for air.
T_f = temperature of the flowing air, Fahrenheit degrees, absolute.

The thin-plate square-edged orifice often has a discharge coefficient K near 0.60. The exact value depends on the location of the connections, the pressure drop, the diameter ratio of orifice to pipe, and the sharpness of the edge.[22,23] Other information on orifices and their use is contained in Chapter 4, Fluid Flow.

Shaped orifices or nozzles have the advantage, if well made, that their discharge coefficients are close to unity so that the probability of large errors is less. Orifices of this type have been adopted for several specific purposes, and designs are described in the A.S.H.V.E. Unit Heater[1] and Unit Ventilator Codes,[2] and in A.S.R.E. Circular 13[3] entitled "Standard Methods of Rating and Testing Air Conditioning Equipment". In some instances nozzles are used in multiple so that the capacity of the testing equipment can be changed by shutting off the flow through one or more nozzles. An apparatus designed for testing the air flow and capacity of air conditioning equipment is described by Wile[24] in an article in which pertinent information on nozzle discharge coefficients, Reynolds numbers, and the resistance

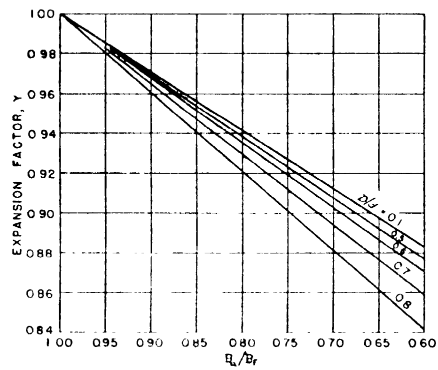

FIG. 5. EXPANSION FACTOR FOR AIR AND OTHER DIATOMIC GASES APPLICABLE TO FLANGE, RADIUS AND VENA CONTRACTA TAPS

of perforated plates is also presented. Such apparatus in some laboratories is commonly referred to as a *code tester*.

The Venturi meter is like the nozzle, except for the addition of a downstream transition section that reduces the pressure drop through the measuring apparatus.

In some cases air velocity through a duct, heater coil, or heating unit may be most conveniently estimated by computation from the heat given up by the coil, and the temperature rise (measured by thermocouples) of the air passing through. It is essential to have a uniform flow over the entire inlet and outlet of the heater at the plane of temperature measurement.

Propeller or Revolving Vane Anemometer

The propeller or revolving vane anemometer consists of a light revolving wind-driven wheel connected through a gear train to a set of recording dials that read the linear feet of air passing in a measured length of time. It is made in various sizes, 3 in., 4 in., and 6 in. being most common. Each instrument requires individual calibration. At low velocities the friction drag of the mechanism is considerable. In order to compensate for this, a gear train that overspeeds is commonly used. For this reason the correction is often additive at the lower range, and subtractive at the upper range, with the least correction in the middle range of velocities. Most of these are not sensitive enough for use below 200 fpm. Anemometers of this type are practically standard for wind measurements, and may be used in large ducts where the air flow is not seriously altered by the presence of the instrument itself.

Deflecting Vane Anemometer

The deflecting vane anemometer consists of a pivoted vane enclosed in a case. Air exerts a pressure on the vane as it passes through the instrument from an up-stream to a down-stream opening. The movement of the vane is resisted by a hair spring and a damping magnet. The instrument gives instantaneous readings of directional velocities on an indicating scale. With fluctuating velocities, it is necessary to average visually the swings of the needle to obtain average velocities. This instrument is very useful for studying motion of the air in a room,[25] and in locating objectionable drafts. Various attachments are available, such as the double tube arrangement for determining velocities in ducts, and a device for measuring static pressures. Each instrument, and the attachments for it, must receive individual calibration. For determining average velocity in a duct, it is necessary to *traverse* the duct as is done when using the Pitot tube.

Measurement of Velocities at Inlets and Outlets of Ducts

In the field it is often desirable to make volume measurements at the face of the supply openings. It is rare to have access to the interior of duct sections where the flow is sufficiently uniform for measurement. For accuracy, the instrument and its application should be checked on a similar approach and grille in the laboratory before use in the field.

Tests have shown that the propeller type anemometer can be used successfully on most of the common types of supply grilles.[26,27] The core area is divided into equal squares, and the anemometer is held against the face of the grille for the same length of time in each. To obtain the air volume in cubic feet per minute, the average corrected velocity in feet per minute thus obtained is multiplied by the average of the gross and net free area of the grille (core) in square feet.

On exhaust openings, the anemometer traverse is made as described previously. The air volume may be determined by multiplying the corrected velocity in feet per minute by the gross core area of the grille in square feet, and by a coefficient for average conditions of 0.85.[28]

When a propeller type anemometer is held in a stream of varying velocities, it tends to indicate higher than the true average, that is, the speed of the propeller is nearer to the top velocity in its area than it is to the minimum velocity. This is the main reason for the large difference in ratings of unit ventilators by the anemometer method, and by air volume measurements in a duct approach to the inlet.[29]

Anemometers can be used within their range at the face of supply grilles when properly applied. In principle, it is a case of finding the velocity at many points, and using the average thus found with the correct discharge area at that cross-section. The deflecting vane anemometer equipped with a jet on the end of a rubber tube has been found especially convenient and accurate on supply grilles.[30] On modern air conditioning grilles, the core area is used without a correction coefficient when the jet is held one inch away from the face of the grille. At this distance, the constriction due to the thin bars has disappeared, since the small air jets have reunited and the air stream has not yet spread beyond the core dimensions. With deflecting grilles the exploring jet should be turned to the angle giving a maximum reading. With suitable traversing tips and calibration, this instrument may also be used on exhaust grilles if proper grille factors are applied.[31] Those contemplating such measurements should consult the references cited.

Smoke is a qualitative tool which is very useful in studying air movements. Satisfactory smoke can be obtained from titanium tetra-chloride (which, however, is very irritating to nasal membranes) or by mixing potassium chlorate and powdered sugar (a non-irritating smoke) and firing the mixture with a match. This latter process evolves considerable heat, and it should be confined in a pan away from flammable materials. The titanium tetra-chloride smoke lends itself to spot determinations, particularly for leakage through casings and ducts, as it can be easily handled in a small pistol-like ejector. The fumes of aqua ammonia and of sulfuric acid, if permitted to mix, form a white precipitate which is useful for some purposes. Two bottles, one containing ammonia water and the other acid, are connected to a common nozzle by means of rubber tubing. Air is forced over the surfaces of the liquids in the bottles by means of a syringe, and the two streams, upon mixing at the nozzle, form a white cloud.

The Kata Thermometer

The measurement of air current velocities within enclosed spaces, such as the rooms of a house, is usually a tedious undertaking. However, useful data can be obtained by using the instruments described in following paragraphs if they are maintained in calibration, and the user understands the operation and limitations of the instruments.[32]

One of the instruments useful for determining the velocity of air currents in free spaces is the Kata thermometer which is essentially an alcohol thermometer with a large bulb. The instrument is heated above 100 F, and then the time in seconds required for it to cool from 100 to 95, when located in the air current, gives a measure of the air speed. It is important to have the Kata thermometer dry before taking the reading. Each Kata has its own factor etched on the stem, and this factor must be used with its cooling formula or chart for obtaining the velocity. The Kata thermometer is useful in exploring ventilated spaces to determine whether the proper air movement and distribution are being maintained. It is also used in determining the cooling power of the atmosphere, since it loses heat by radiation and convection when dry, and by radiation, convection, and evaporation when the bulb is equipped with a wetted cloth covering.[33]

Thermal Anemometers

If a suitable sensing element is heated electrically at a fixed rate and exposed to an air stream, the temperature difference between the element and the stream becomes a measure of velocity by calibration. In the *hot-wire anemometer*, a very fine heated wire is employed as a resistance-thermometer element whose temperature may be determined accurately.[34, 35] In the *heated-bulb thermometer* type, a heating wire is wound around the bulb of a mercury-in-glass thermometer, and the temperature difference between this thermometer and a similar unheated one serves as an index of air speed.[36] The *heated-thermocouple anemometer* is calibrated to give velocity in terms of the differential *emf* between heated and unheated thermo-junctions exposed to an air stream.[34, 37] Combined measurements of air temperature and velocity are particularly useful for air distribution studies, and automatic recording potentiometers or resistance-thermometer devices facilitate spatial traverses. A correct calibration of thermal anemometers requires consideration of the effects of temperature, humidity and pressure upon the air properties which influence convective heat transfer.[37] With sensing elements of simple shapes for which convection data are known, thermal anemometers may be designed, both thermally and electrically, for desired characteristics. Directional sensitivity is controllable. Thermal anemometers are convenient and practical for low velocities.

Infiltration or Air Change Measurement

In the past, efforts have been made to determine the rate of air change in buildings by impregnating the air with CO_2, hydrogen, water vapor or another substance, and then observing the rate of decrease in concentration[38] with an Orsat apparatus, a psychrometer or another suitable means. Success has not been attained in a satisfactory degree, chiefly because building materials absorb or reject the substances used to impregnate the air, thus impairing the precision of the tests. Recent experiments in England are described in a paper by Dick.[39]

HUMIDITY MEASUREMENT

Psychrometers

Any instrument capable of measuring the humidity or hygrometric state of the air is a hygrometer. A psychrometer is a particular kind of hygrometer which consists of two mercury thermometers, one of which has a cloth wick or *sock* applied to its bulb. For use, the wick is wetted with water and ventilated with air moving at a recommended rate of 900 fpm or more, relative to the instrument.[40] In the simpler and more common type, known as the sling psychrometer, the two thermometers are mounted side by side on a frame fitted with a handle by which the device can be whirled through the air. The motion is arrested for reading the thermometers, and continued until the thermometer readings become steady. Due to evaporation, the wet-bulb thermometer will indicate a lower temperature than the dry-bulb thermometer, and the difference is known as the wet-bulb depression. Charts and tables are available showing the relation between the thermometer readings and the humidity.[41,42] Data are usually based on a total pressure of one standard atmosphere. For precise work, a correction is necessary for barometric pressure and is usually made by multiplying the observed relative humidity by the ratio of the observed to the standard atmospheric pressure.

For air temperatures below 32 F, the water on the wick may either freeze or super-cool, and its state must be known and a proper table or chart used, since the wet-bulb temperature is different for ice and for water. Some operators remove the wick from the wet-bulb for freezing conditions and dip the bulb in water a few times, allowing the water to freeze on the bulb between dips and to form a film of ice. Since wet-bulb depression is slight at low temperatures, precise temperature readings are essential.

In the ventilated or aspirated psychrometer, the thermometers remain stationary, and a small fan or blower or a syringe is used to move the air across the thermometer bulbs. Various designs have been employed in the laboratories, and several commercial models are available.

The Dew-Point Hygrometer

In the usual form of these instruments, means are provided for cooling, and of observing the temperature of, a surface which is exposed to air. The temperature at which visible condensation occurs on the surface is considered the dew-point of the air. With the dew-point temperature known, the relative humidity and other properties of the air can be taken from tables and charts (See Chapter 3). A bright surface or metallic mirror is usually employed to improve the visibility of the dew deposit, and various means are used to cool the mirror from the back, including evaporating ether or another refrigerant, or a stream of air passed through dry ice. Dew-point temperatures, in some cases, are observed by means of thermometers in fluids in contact with the back of the mirror, but in modern instruments thermocouples are used, and are soldered or welded to the mirror itself. The dew-point apparatus is not so commonly used as the psychrometer, probably because it is less convenient. It is usable, however, for higher temperatures than the wet-and dry-bulb psychrometer, and should be considered for dew-points near or above the boiling point, as in the case of flue gases. Special apparatus for high precision has been constructed, in which the photronic cell and a light source are used for dew or frost detection instead of visual inspection.

Hair Hygrometers

Many materials, especially organic materials, change in dimensions with changes in humidity, and many devices have been designed in efforts to utilize this action in simple and effective humidity indicators, recorders and controllers. Unfortunately, no material has been found which can be relied upon to perfectly reproduce its action when exposed to repeated identical changes in humidity. The field has been well explored, and instrument and control manufacturers are practically unanimous in the selection of human hair for this service.

The hair hygrometer consists of from one to several strands of hair with a mechanism whereby changes in length of the strands, due to changes in humidity, cause an indicator to move across a dial. In the recording instrument, a pen is moved across and marks a moving paper ribbon, indexed in relative humidity. In a controller, or humidistat, the motion makes or breaks an electric contact governing the air conditioning equipment. Such devices require initial calibration and, for precise work, frequent recalibration or setting, especially if they are exposed to extremes of either high or low humidity. For continuous operation, with only slight changes in humidity, some operators report satisfactory reproducibility of results.

Electrolytic Hygrometers

The dampness, and therefore the electrical resistance of a salt film, varies with the humidity of the atmosphere to which the film is exposed, and at least two types of instruments based on this fact have been developed. The Dunmore hygrometer was originally designed for use in radio-sondes or small balloons, and means were devised whereby the device transmits humidity data back to earth in the form of a radio signal. In the more usual form, this hygrometer consists of a dual winding of small wire on a non-conducting tube. The whole is coated with an electrolytic film, usually containing a salt such as lithium chloride, which forms an electric connection between the windings. Means are provided for determining the electrical resistance of the film, which is an indication of the humidity. In the radio-sonde, variations in the resistance of the film affect the frequency of an oscillating circuit. These hygrometers are usually calibrated by comparison with a wet- and dry-bulb psychrometer. For calibration for some purposes, particularly for use at sub-zero temperatures, means have been provided for producing atmospheres of known humidity.[43] Advantages of instruments of this type for some scientific and industrial purposes are becoming apparent, and some forms of them are on the market.

The Weaver type hygrometer is particularly useful for determining the humidity of air or other gas in pipes or closed vessels at various pressures and temperatures. It consists of a threaded plug carrying a central electrode insulated from the plug, except for a gelatinous electrolytic film. The film is exposed to the air or gas from the pipe or vessel, and provision is made for determining its electrical resistance. The hygrometer, or detector, is mounted in a manifold equipped with valves whereby the film can be alternately exposed to the test gas, and to a standard gas having a known absolute humidity. The pressure of the standard gas is varied until contact with it establishes the same resistance in the film as the test gas, and the humidity of the test gas is then determined by computation based on the gas laws.

Chemical Hygrometry

The humidity of an atmosphere can be measured directly by extracting and weighing the water vapor from a known sample. For precise laboratory work, powerful desiccants such as sulphuric acid and phosphorus pentoxide are used for the extraction process, while for some purposes, calcium chloride, lithium chloride or silica gel are satisfactory. Freezing the water vapor out of a measured stream of air or gas with solid carbon dioxide, and weighing the resulting ice, is a similar operation. A thermal conductivity method for gas analysis can be used for temperatures above 212 F, or for very low humidities.[44]

HEAT TRANSFER THROUGH BUILDING MATERIALS

Thermal Conductivity

Use of the guarded hot plate apparatus for determining the thermal conductivity (k value) of homogeneous materials was adopted by A.S.H.V.E. in 1942, and has become practically universal, and the apparatus is described in an A.S.T.M. publication.[45, 46] It consists essentially of an electrically heated plate and two water cooled plates. Two identical specimens or slabs of a material are required for a test, and one is mounted on each side of the hot plate. A cold plate is then pressed against the outside of each specimen by a clamp screw. Hot plate apparatus accommodating specimens on the order of one foot square and an inch or more thick, is common. The apparatus at the *National Bureau of Standards* takes specimens 8 in. square, while plates as large as 3 ft square have been used. The heated plate is divided into two portions: the central or measuring section, and the outer or guard section. During tests the two sections are maintained as nearly as possible at the same temperature, and the purpose of the guard section is to minimize errors due to edge effects. The electric energy required to heat the measuring section is carefully observed and, converted to Btu per hour, is divided by the area and the temperature gradient to obtain the conductivity of a material tested.

Wall Conductances

The thermal conductances (C values) of many walls can be satisfactorily estimated from the conductivities of their components and their dimensions, but some walls are complicated by the inclusion of metal, for instance, and tests for conductance are required. The apparatus is required to accommodate large specimens representing actual construction. The shielded hot box apparatus was developed for this purpose.[47] Specimens for the apparatus at the National Bureau of Standards are 5 ft long and 8 ft high, while others require different sizes, some larger, others smaller.

The guarded hot box is described in the A.S.H.V.E. Standard Test Code for Heat Transmission through Walls.[47] The apparatus consists essentially of three boxes: a cold box, cooled by a refrigerating machine; a hot box, heated electrically; and a metering box also heated electrically. Each box has an open side to be placed against the specimen. The cold box is clamped against one side of the specimen, and the hot box against the other. The hot box encloses the metering box and is kept at the same temperature to minimize heat exchanges to or from the metering box, except through the specimen. The electric energy necessary to heat the metering box is measured, converted to Btu per hour, and divided by the area and the temperature difference through the wall, from surface to surface, to yield the conductance of the wall. The transmittance or U-value of the wall is then computed by means of the surface coefficients from Chapter 9.

The Nicholls heat flow meter is sometimes useful for measuring steady heat flow through a wall or other building member.[48] In essence, this meter consists of a plate or slab of material of known thermal resistance having attached thermocouples on both sides. For use, the device is pressed against or cemented to the wall to be tested. At steady state, the temperature difference through the slab, measured with the thermocouples, with the known thermal resistance of the slab, indicates the heat flow through the slab and hence, through the wall covered by it. For best results, such meters are calibrated by means of a guarded hot plate or other suitable apparatus. The chief precaution is to assure that the heat flow is steady at the time of measurement.

EVALUATION OF THE THERMAL ENVIRONMENT

Advocates of radiant heating emphasize the fact that comfort depends on radiant heat exchanges, as well as air temperature. For this reason several instruments have been devised to evaluate the comfort or warmth of rooms, taking radiant as well as convective effects into account. Prominent among these are the eupatheoscope, the globe thermometer, the thermal integrator and the heated globe.[49,50] Descriptions are contained in the references, and are omitted here because these devices are not widely used in America for several reasons, among which is the fact that radiant heating with high temperature sources is not a chief method of comfort heating in this country.

COMBUSTION ANALYSIS

There are two approaches to the problem of measuring the capacities of fuel burning devices, such as boilers and furnaces. The direct or calorimetric test consists in measuring the change in enthalpy or heat content of the fluid, air or water, heated by the device and multiplying by the flow rate in pounds per hour to arrive at the capacity in Btu per hour.[51] The indirect test consists in determining the heat lost in the flue gases and deducting it from the heat evolved by combustion of the fuel.[52] A heat balance consists in the simultaneous application of both tests to the same device. The indirect test almost invariably indicates the greater capacity, and the difference is credited to radiation from the boiler or furnace casing and *unaccounted for loss.*

In the case of some small equipment, the expense of the direct test is not considered justifiable, and the indirect test is relied upon with an arbitrary *radiation and unaccounted for* factor.[52]

Flue Gas Analysis

The Orsat apparatus is commonly used for analyzing flue gases. In its ordinary form, it consists of three pipettes and a means for isolating a sample of flue gas in a graduate. After measuring, the sample is expelled from the graduate into the first pipette where the carbon dioxide is extracted by potassium hydroxide. The sample is then remeasured and successively passed into the second and third pipettes, where the oxygen and the carbon monoxide are respectively extracted by potassium pyrogallate and cuprous chloride.

For field testing and burner adjustment, simpler portable devices are available for carbon dioxide determination only. From curves, based on typical hydrogen content of several common fuels, efficiencies may be estimated from the carbon dioxide value obtained. More elaborate laboratory equipment is sometimes provided for precise determination of carbon monoxide content by burning the carbon monoxide to carbon dioxide in presence of a catalyst.[53, 54] In large plants, carbon dioxide recorders are used to obtain a continuous indication of the plant's efficiency.[55]

TABLE 1. RINGELMANN SMOKE CHART SPACINGS

Number of Card	Thickness of Lines, mm	Distance in Clear Between Lines, mm
1	1.0	9.0
2	2.3	7.7
3	3.7	6.3
4	5.5	4.5

SMOKE DENSITY MEASUREMENTS

Ringelmann charts are widely used for evaluating the density of smoke discharged from chimneys or stacks, and smoke ordinances are based on them in some cities. Each chart is composed of a series of crossed black lines on white paper which, at a distance of about 50 ft, is visually compared with the smoke under observation. Four charts are used with different degrees of blackness as shown in Table 1. The smoke density is specified by Ringelmann numbers from 1 to 4.

The photoelectric cell is used in some apparatus developed for smoke density recording in large plants. The same device is included in the testing equipment for domestic oil burners described in *National Bureau of Standards*, Commercial Standard CS75-42.[56] Under *Laboratory Tests* this publication contains the following section: "Smoke Determination.—After combustion has reached equilibrium, the amount of smoke in the flue gases, when viewed lengthwise through 4 feet of the smoke pipe in accordance with the *Underwriters' Laboratories, Inc.*, Standard for Domestic Oil Burners (Subject 296), March 1934 and subsequent revisions, shall not reduce the output of a standard photoelectric cell from 9 microamperes, with a clear smoke pipe, to less than 8 microamperes." The Commercial Standard also requires that during a test after installation, the burner shall operate without visible smoke at the chimney top.

A method of evaluating smoke produced by pot type oil burners was developed for the *Institute of Cooking and Heating Appliance Manufacturers* by R. N. St. John. A glass rod is interposed between a light source and a photo-sensitive cell, both before and after being exposed to the flue gases from a heating device. The diminution of the light transmitted by the rod, due to the deposit of soot on its surface, causes a reduction in the cell emf which is taken as an index of the concentration of smoke in the flue gases. A description of the method is contained in National Bureau of Standards Commercial Standard CS104-46.[52]

DETERMINATION OF AIR CONTAMINANTS

Two measures of air dustiness are in use: particles per unit volume of air, and weight per unit volume of air. A comparative method consists in drawing known samples of air through a known area of filter cloth or paper, and comparing the density of the resulting spots with blackness charts or with spots from other sources.

For counting, particles are captured in a device such as the Smith-Greenburg impinger, the Owens jet dust counter, or in an electrostatic or a thermal precipitation device designed for the purpose.[57] Counting is done with a microscope, and the method yields important results when the nature or constituents of the dust are of interest.

For a weight determination, a known volume of air is drawn through a porous crucible or thimble, and the weight gained by the thimble during the operation is the weight of the dust captured from the air sample.[58] For precise work, the thimble must be dried in a desiccating chamber before weighing each time. The test method specified in the A.S.H.V.E. Code for Testing and Rating Air Cleaning Devices Used in General Ventilating Work is based on a weight method for evaluating the cleanliness of air after passing through an air cleaner.[59]

The Code has not filled all needs for an air cleaner testing method and the subject is now under investigation by the A.S.H.V.E. Research Laboratory. At the *National Bureau of Standards* a test method was developed for interested Government agencies, under which measured samples of the uncleaned air and of the air cleaned by a device under test are passed through filter papers.[60] The ratio of the flow rates through the two filter papers is adjusted during successive tests, until the resulting dust spots approach equality in density as shown by a photometer. The ratio of the flow rates is then indicative of the effectiveness of the cleaner in arresting dust. A statement of an air cleaner's efficiency by any test method, is meaningless unless the test dust is specified.

Other instruments for determining gaseous and particulate air contaminants are described in industrial hygiene literature.[57, 61, 62, 63]

Sound and Vibration Measurements

Approximate measurements of sound intensity can be made by aural methods. The ear is used to compare the measured noise with sounds of known strength.

Electrical devices, in which the ear plays no part, furnish the most satisfactory means of measuring noise intensities. The sound meter consists essentially of a microphone coupled to an amplifier designed with an ear-like response. The output of the microphone is read on a sensitive direct current milliammeter graduated to read directly in decibels. Instruments of this type, if connected with suitable band pass filters, can be used to study the intensity of the sound over its entire range of frequencies.

Electrical instruments are available for measuring the frequency, amplitude and acceleration of a vibrating mass. They are usually more convenient and accurate than the vibrating reed tachometer, the seismic type displacement meters or the accelerometers which can also be used for this purpose. Sound level meters are discussed in several text books[64, 65] and standards.[66]

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[43] Divided Flow Low Temperature Humidity Test Apparatus, by Arnold Wexler (*National Bureau of Standards*, Research Paper No. 1894).

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[46] Guarded Hot Plate Apparatus Complying With the Requirements of Section 4 of *A.S.T.M.* Method of Test for Thermal Conductivity of Materials by Means of the Guarded Hot Plate, *A.S.T.M.*

[47] A.S.H.V.E. Standard Test Code for Heat Transmission Through Walls (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 253), adopted 1928.

[48] A.S.H.V.E. RESEARCH REPORT No. 685—Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 65).

[49] Instruments and Methods for Recording Thermal Factors Affecting Human Comfort, by C. P. Yaglou, A. P. Kratz and C.-E. A. Winslow (Year Book, *American Journal Public Health*, 36–37).

[50] The Thermo-Integrator—A New Instrument for the Observation of Thermal Interchanges, by C.-E. A. Winslow and Leonard Greenburg (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 149).

[51] I = B = R Testing and Rating Codes for Low Pressure Heating Boilers, 1947 (*Institute of Boiler and Radiator Manufacturers*).

[52] Commercial Standard for Warm Air Furnaces Equipped with Vaporizing Pot-Type Oil Burners, C.S. 104-46 (*National Bureau of Standards*).

[53] Rapid Determination of Small Amounts of Carbon Monoxide, by Martin Shepherd (*Ind. Eng. Chem. Anal. Ed. 19*, 77: 1947).

[54] Determination of Small Amounts of Carbon Monoxide in Air by Various Reference Methods, by Martin Shepherd (*National Bureau of Standards Journal of Research* 38, 351–8, 1947 R. P. 1777).

[55] A Carbon Monoxide Recorder, by S. H. Katz, D. A. Reynolds, H. W. Frevert and J. J. Bloomfield (*U. S. Bureau of Mines, Technical Paper* No. 355, 1926).

[56] Commercial Standard for Mechanical Draft Oil Burners Designed for Domestic Installations, C.S. 75-42 (*National Bureau of Standards*).

[57] Industrial Dust, by Philip Drinker and Theodore Hatch (McGraw-Hill Book Company, New York).

[58] Testing and Rating of Air Cleaning Devices Used for General Ventilation Work, by S. R. Lewis (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 277).

[59] A.S.H.V.E. Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work, adopted January, 1934 (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 225).

[60] A Test Method for Air Filters, by Richard S. Dill (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 379).

[61] Analytical Chemistry of Industrial Poisons, Hazards and Solvents, by Morris B. Jacobs (Interscience Publishers, Inc., New York, 1941).

[62] The Determination and Control of Industrial Dust, by J. J. Bloomfield and J. M. Dalla Valle (*U. S. Public Health Bulletin* No. 217, 1935).

[63] Sampling and Analysis of Atmospheric Contaminants, by F. A. Patty (*Industrial Hygiene and Toxicology*, Vol. I, Interscience Publishers, Inc., New York, 1948).

[64] Electrical Engineer's Handbook, by Harold Pender and Knox McIlvaine (John Wiley and Sons, New York).

[65] Elements of Acoustical Engineering, by Harry F. Olson (D. Van Nostrand Co., New York).

[66] American Tentative Standards for Sound Level Meters for Measurement of Noise and Other Sounds, Z 24.3-1944 (*American Standards Association*).

CHAPTER 50

CODES AND STANDARDS

THE Codes and Standards listed in Table 1 represent accepted practice, methods, or standards prepared and accepted by the organizations indicated. They are valuable guides for the practicing engineer in determining test methods, ratings, performance requirements, and limits applying to equipment used in heating, ventilating, and air conditioning. Copies can usually be obtained *from the organization listed in the reference column.*

TABLE 1. CODES AND STANDARDS PREPARED AND ACCEPTED BY VARIOUS SOCIETIES AND ASSOCIATIONS

SUBJECT	TITLE	SPONSOR	REFERENCE
Acoustics (Terminology)	American Standard Acoustical Terminology (Proposed).	*AS of A*	*ASA* Z24.1
Air Conditioning	Code of Minimum Requirements for Comfort Air Conditioning (1938).	ASHVE *ASRE*	ASHVE
Air Conditioning (120,000 Btu/Hr or less)	Code and Manual for the Design and Installation of Warm Air Winter Air Conditioning Systems (1945).	*NWAH & ACA*	*NWAH & ACA* Manual No. 7
Air Conditioning (Above 120,000 Btu/hr)	The Technical Code for the Design and Installation of Mechanical Warm Air Heating Systems (1948).	*NWAH & ACA*	*NWAH & ACA* Manual No. 9
Air Conditioning	Standards of the *NBFU* for the Installation of Air Conditioning, Warm Air Heating, Air Cooling and Ventilating Systems.	*NFPA* *NBFU*	*NBFU* Pamphlet No. 90 Feb. 1950
Air Conditioning (Equipment)	*ASRE* Standard Methods of Rating and Testing Air Conditioning Equipment (1942).	*ASRE* ASHVE *NEMA* *RMA* *ACMA*	*ASRE* Circular No. 13-42
Air Conditioning (Equipment)	*ASRE* Standard Methods of Rating and Testing Air Conditioners (1949). (Supersedes *ASRE* Circular 13-42)	*ASRE*	*ASRE* Standard 16-R
Airplane	Aeronautical Recommended Practice for Heating and Ventilating Airplanes (1943).	*SAE*	*SAE* ARP 85
Airplane	Aeronautical Recommended Practice for Internal Combustion Type Airplane Heaters (1945).	*SAE*	*SAE* ARP 143A
Attic Ventilation	Residence Ventilation Guide (1950).	*PFMA*	*PFMA*
Boilers	I=B=R Testing and Rating Code for Low Pressure Heating Boilers (1950).	*IBR*	*IBR*
Boilers	Net Load Recommendations for Heating Boilers. Publ. semi-annually.	*HP & ACCNA*	*HP & ACCNA*
Boilers	Net Square Feet Radiation Loads in 70 Deg Fahr, Recommended for Low Pressure Heating Boilers (1948).	*HP & ACCNA*	*HP & ACCNA*
Boilers	Standard and Short Form Heat Balance Codes for Testing Low Pressure Steam Heating Solid Fuel Boilers (Codes 1 and 2)(1929).	ASHVE	ASHVE
Boilers	ASHVE Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3)(1929).	ASHVE	ASHVE
Boilers	ASHVE Standard Code for Rating Steam Heating Solid Fuel Hand Fired Boilers (Revised April 1930).	ASHVE	ASHVE

TABLE 1. CODES AND STANDARDS—(Continued)

SUBJECT	TITLE	SPONSOR	REFERENCE
Boilers	ASHVE Standard Code for Testing Steam Heating Boilers Burning Oil Fuel (1932).	ASHVE	ASHVE
Boilers	ASHVE Standard Code for Testing Stoker-Fired Steam-Heating Boilers (1938).	ASHVE	ASHVE
Boilers	*ASME* Boiler Construction Code for Low Pressure Heating Boilers (1946 with 1948 Addenda).	*ASME*	*ASME*
Boilers	*ASME* Boiler Construction Code (Combined Edition) (1946 with 1947 Addenda).	*ASME*	*ASME*
Boilers (Gas)	American Standard Approval Requirements for Central Heating Gas Appliances (1951)	*A.G.A.*	*ASA* Z21.13.-1951
Boilers (Miniature)	*ASME* Miniature Boiler Code (1946).	*ASME*	*ASME*
Boilers (Power)	*ASME* Power Boiler Code, Including Rules for Inspection (1946) with 1947 Addenda).	*ASME*	*ASME*
Boilers (Power)	Suggested Rules for Care of Power Boilers (1946).	*ASME*	*ASME*
Boilers (Steel)	Steel Boiler Institute Rating Code for Commercial Steel Boilers and Residential Steel Boilers (1948).	*SBI*	*SBI*
Boilers (Steel)	Simplified Practice Recommendation for Steel Firebox Boilers and Steel Heating Boilers (Commercial and Residential Types) (1950).	*BS* *SBI*	*BS* R157-50
Boilers (Steel)	SBI Code for Testing Oil-Fired Residential Steel Heating Boilers (1948)	*SBI*	*SBI*
Boilers (Steel)	SBI Rating Code for Scotch Type Boilers (Over 15 psi Working Pressure) (1949).	*SBI*	*SBI*
Building Code Standards	Building Code Standards of the *NBFU* for the Installation of Heat Producing Appliances, Heating, Ventilating, Air Conditioning Blower and Exhaust Systems.	*NBFU*	*NBFU*
Building Requirements	American Standard Building Requirements (1946).	*NHA* *USPHS*	*ASA* A53.1-1946
Buildings	Basic Building Code (Also published in Abridged form as Abridged Building Code) 1950.	*BOCA*	*BOF*
Burners (Anthracite)	Commercial Standard for Domestic Burners for Pennsylvania Anthracite (Underfeed Type) (1940).	*BS* *AIL*	*BS* CS48-40
Burners (Gas)	American Standard Requirements for Installation of Domestic Gas Conversion Burners (1948).	*A.G.A.*	*ASA* Z21.8-1948
Burners (Gas)	American Standard Requirements for Installation of Gas Burning Equipment in Large Boilers (1950).	*A.G.A.*	*ASA* Z21.33-1950
Burners (Gas)	American Standard Testing Requirements for Gas Conversion Burners (1948).	*A.G.A.*	*ASA* Z21.17-1948
Burners (Oil)	Commercial Standard for Mechanical-Draft Oil Burners Designed for Domestic Installations (1942).	*BS* *OBI*	*BS* CS75-42
Chimneys (Flue Linings)	American Standard Sizes of Clay Flue Linings (1947).	*AIA* *PC*	*ASA* A62.4-1947
Cleaners (Air)	ASHVE Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work (1934).	ASHVE	See ASHVE TRANSACTIONS, Vol. 39, 1933, p. 225

TABLE 1. CODES AND STANDARDS—(Continued)

SUBJECT	TITLE	SPONSOR	REFERENCE
Coils	Proposed Commercial Standard for Rating and Testing Air Cooling Coils Using Non-Volatile Refrigerants (1945).	*BCMI* *BS*	*BS* TS 4044
Color Scheme (Piping)	Scheme for Identification of Piping Systems (1945).	*HP & ACCNA*	*HP & ACCNA* Engrg. Stds., Sec. 2, Part V
Color Scheme (Piping)	Scheme for Identification of Piping Systems (1928).	*ASME*	*ASA* A13-1928
Compressors	*ASRE* Standard Methods of Rating and Testing Refrigerant Compressors.	*ASRE* ASHVE *ACRMA*	*ASRE* Standard 23
Condensers	*ASRE* Standard Methods of Rating and Testing Evaporative Condensers.	*ASRE* ASHVE *ACRMA*	*ASRE* Standard 20
Condensers	*ASRE* Standard Methods of Rating and Testing Water-Cooled Refrigerant Condensers.	*ASRE* ASHVE *ACRMA*	*ASRE* Standard 22
Condensing Units	*ASRE* Standard Methods of Rating and Testing Mechanical Condensing Units (1940).	*ASRE* ASHVE *ACRMA*	*ASRE* Standard 14-41
Conductivity	Standard Method of Test for Thermal Conductivity of Materials by Means of the Guarded Hot Plate (Tentative) (1942).	ASHVE *ASRE* *ASTM* *NRC*	ASHVE
Control Equipment (Industrial)	Underwriters' Laboratories, Inc., Standard for Industrial Control Equipment (July 1938, reprinted Sept. 1945).	*UL*	*UL* Subject 508
Controls	Underwriters' Laboratories, Inc., Standard for Temperature Indicating and Regulating Equipment (Jan 1947).	*UL*	*UL* Subject 873
Convector	ASHVE Standard Code for Testing and Rating Concealed Gravity Type Radiator (Hot Water Section) (1933)	ASHVE	ASHVE TRANSACTIONS, Vol. 39, 1933, p. 237
Convector	ASHVE Standard Code for Testing ing and Rating Concealed Gravity Type Radiation (Steam Code) (1931).	ASHVE	ASHVE TRANSACTIONS, Vol. 37, 1931, p. 367
Convector	Commercial Standard for Testing and Rating Convectors (1947).	*BS* *CMA* *IBR*	*BS* CS 140-47
Coolers (Air)	*ASRE* Standard Methods of Rating and Testing Forced Circulation and Natural Convection Air Coolers for Refrigeration (1945).	*ASRE* ASHVE *ACRMA* *REMA*	*ASRE* Circular No. 25-44
Coolers	*ASRE* Standard Methods of Rating and Testing Water and Brine Coolers.	*ASRE* ASHVE *ACRMA*	*ASRE* Standard 24
Ducts and Fittings	Simplified Practice Recommendation for Pipes, Ducts and Fittings for Warm Air Heating and Air Conditioning (1945).	*Mfrs.* *BS*	*BS* R207-49
Exchangers (Heat)	Standards of *Tubular Exchanger Manufacturers Association* (1941).	*TEMA*	*TEMA*
Exhaust Systems	American Standard for Grinding, Polishing, and Buffing Equipment Sanitation (1941).	*AFA*	*ASA* Z43-1941
Exhaust Systems	Tentative Code of Recommended Practices for Testing and Measuring Air Flow in Exhaust Systems (1937).	*AFA*	*AFA* Preprint 36-27
Exhaust Systems	Tentative Recommended Good Practice Code and Handbook on the Fundamentals of Design, Construction, Operation and Maintenance of Exhaust Systems.	*AFA*	*AFA*

Table 1. Codes and Standards—(Continued)

Subject	Title	Sponsor	Reference
Exhaust Systems	Standards for Blower and Exhaust Systems (1949).	*NFPA* *NBFU*	*NFPA* No. 91
Fans	Definitions and Terms in Use by the Blower Industry (1950) (Was *NAFM* Bulletin No. 105).	*NAFM*	*NAFM* Bulletin No. 110
Fans	Standard Test Code for Testing Centrifugal and Axial Fans (1950) (Was *NAFM* Bulletin No. 103).	*NAFM** ASHVE	*NAFM* Bulletin No. 110
Fans	Standards, Definitions, Terms and Test Codes for Centrifugal, Axial and Propeller Fans.	*NAFM* ASHVE**	*NAFM* Bulletin 110 1950
Fans	Standards for Fans (1947).	*NEMA*	*NEMA* Publ. 47-128
Fans	Test Code for Fans (1946).	*ASME*	*ASME* PTC 11-1946
Fire Prevention	Building Code Recommended by the *National Board of Fire Underwriters* (1943).	*NBFU*	*NBFU*
Fire Prevention	National Fire Codes (1951).	*NFPA*	*NFPA*
Fire Prevention	National Fire Code for the Prevention of Dust Explosions (1943).	*NFPA*	*NFPA*
Furnaces (Duct)	American Standard Approval Requirements for Gas-Fired Duct Furnaces (1942).	*A.G.A.*	*ASA* Z21.34-1942
Furnaces (Gas, Floor)	Commercial Standard for Gas Floor Furnaces—Gravity Circulating Type (1942).	*BS* *AGAEM*	*BS* CS99-42
Furnaces (Gas)	American Standard Approval Requirements for Central Heating Gas Appliances (1951).	*A.G.A.*	*ASA* Z21.13-1951
Furnaces (Forced Air, Solid-Fuel)	Commercial Standard for Solid-Fuel-Burning Forced Air Furnaces (1944).	*FHA* *N.W.A.H. & A.C.A* *AIL*	*BS* CS109-44
Furnaces (Oil-Fired)	Commercial Standard for Warm Air Furnaces Equipped with Vaporizing Type Oil Burners (1949).	*Mfrs.* *BS*	*BS* CS104-49
Furnaces (Oil)	Commercial Standard for Oil Burning Floor Furnaces Equipped with Vaporizing Type Burners (1951).	*BS* *OPA*	*BS* CS113-51
Furnaces (Oil)	A Tentative Code for Testing Oil-Fired Furnaces.	*NWAH & ACA*	*NWAH & ACA*
Garage Ventilation	Recommended Good Practice Requirements for the Construction and Protection of Garages (1932).	*NFPA* *NBFU*	*NFPA* No. 88
Garages	Code of Minimum Requirements for Heating and Ventilating Garages (1935).	ASHVE	ASHVE
Gas Equipment (Large Boilers)	American Standard Requirements for Installation of Gas Equipment in Large Boilers.	A.G.A.	*ASA* Z21.33-1950
Gases (Toxic) and Dust	American Standard Allowable Concentration of Harmful Gases:	*ASA*	*ASA*
	Carbon Monoxide		Z37.1-1941
	Hydrogen Sulfide		Z37.2-1941
	Carbon di-sulfide		Z37.3-1941
	Benzene		Z37.4-1941
	Cadmium		Z37.5-1941
	Manganese		Z37.6-1942
	Chromic Acid and Chromates		Z37.7-1943
	Mercury		Z37.8-1943
	Metallic Arsenic and Arsenic Trioxide		Z37.9-1943
	Xylene		Z37.10-1943

* Also endorsed by *PFMA*.
** Refers to Test Code for Centrifugal and Axial Fans.

TABLE 1. CODES AND STANDARDS—(Continued)

SUBJECT	TITLE	SPONSOR	REFERENCE
Gases (Toxic) and Dust (Continued)	Lead and Certain Inorganic Lead Compounds		Z37.11-1943
	Toluene		Z37.12-1943
	Oxides of Nitrogen		Z37.13-1944
	Methanol		Z37.14-1944
	Styrene-Monomer		Z37.15-1944
	Formaldehyde		Z37.16-1944
	Methyl Chloride		Z37.18-1949
	Trichloroethylene		Z37.19-1946
Heat Transfer (Walls)	ASHVE Standard Test Code for Heat Transmission Through Walls (1928).	ASHVE	ASHVE
Heaters (Room, Gas Fired)	American Standard Approval Requirements for Gas-Fired Room Heaters (formerly called Space Heaters) (1949 with Addenda 1950).	*A.G.A.*	*ASA* Z21.11-1949
Homes (Prefabricated)	Commercial Standard for Prefabricated Homes.	*PHMI* *BS*	*BS* CS125-47
Mineral Wool	Commercial Standard for Mineral Wool Insulation for Heated Industrial Equipment (1949).	*BS* *I.M.W.I.*	*BS* CS117-49
Mineral Wool	Commercial Standard for Mineral Wool Insulation for Low Temperature Installations (1948).	*BS* *IMWI*	*BS* CS105-48
Mineral Wool	Recommended Commercial Standard for Industrial Mineral Wool Products—All Types—Testing and Reporting (1946).	*IMWI* *BS*	*BS* CS131-46
Motors	Nema Motor and Generator Standards (June 1945).	*NEMA*	*NEMA* 45-102
Motors	Proposed Test Code for Single-Phase Motors (1941).	*AIEE*	*AIEE*
Panel System (Warm Air)	Code and Manual for the Design and Installation of Warm Air Ceiling Panel Systems.	*NWAH & ACA*	*NWAH & ACA* Manual No. 7-A
Pipe & Tubing (Copper & Brass)	Simplified Practice Recommendation for Copper Water Tubes and Brass Pipe.	*Mfrs.* *BS*	*BS* R217-49
Piping	American Standard Code for Pressure Piping (1942, with Supplement No. 2, 1947).	*ASME*	*ASA* B31.1-1942
Piping (Gas)	American Standard for Installation of stallation of Gas Piping and Gas Appliances in Buildings (1950).	*A.G.A.*	*ASA* Z21.30-1950
Pumps	Hydraulic Institute Test Code for Centrifugal Pumps. Hydraulic Institute Test Code for Rotary Pumps (1943).	*HI*	*HI* Section F
Radiation (Baseboard)	I=B=R Testing and Rating Code for Baseboard Type of Radiation (1950).	*IBR*	*IBR*
Radiators	Code for Testing Radiators (1927).	ASHVE	ASHVE
Radiators	Simplified Practice Recommendation for Cast Iron Radiators (1943).	*IBR* *BS*	*BS* R174-47
Refrigeration (Equipment)	Underwriters' Laboratories, Inc., Standard for Air Conditioning and Commercial Refrigerating Equipment (Feb. 1946).	*UL*	*UL* Subject 207A
Refrigeration (Mechanical)	American Standard Safety Code for Mechanical Refrigeration (1939).	*ASRE*	*ASA* B9-1939***
Refrigeration (Unit Systems)	Underwriters' Laboratories, Inc., Standard for Unit Refrigerating Systems (Feb. 1946).	*UL*	*UL* Subject 207C
Refrigerators (Gas-Fired)	American Standard Approval Requirements for Refrigerators Using Gas Fuel (1941).	*A.G.A.*	*ASA* Z21.19-1941

*** Also designated ASRE Circular No. 15.

Table 1. Codes and Standards—(Continued)

Subject	Title	Sponsor	Reference
Refrigerators (Household)	American Standard Test Procedures for Household Electric Refrigerators (Mechanically Operated) (1944).	*ASRE* *USDA*	*ASA* B38.2-1944
Sound (Measurement)	American Standard for Noise Measurement.	*AS of A*	*ASA* Z24.2-1942
Sound (Measurement)	American Standard for Sound Level Meters for Measurement of Noise and Other Sounds.	*AS of A*	*ASA* Z24.3-1944
Sound (Measurement)	American Standard Method for the Pressure Calibration of Laboratory Standard Pressure Microphones.	*AS of A*	*ASA* Z24.4-1938
Sound (Measurement)	Sound Measurement Test Code for Centrifugal and Axial Fans (1950) (Was *NAFM* Bulletin No. 104).	*NAFM*	*NAFM* Bulletin No. 110 (1950)
Space Heaters	Commercial Standard for Flue Connected Oil-Burning Space Heaters Equipped with Vaporizing Pot-Type Burners (1943).	*ICHAM*	*BS* CS101-43
Stokers	Code for Determination of Rated Capacities of Anthracite Underfeed Stokers (1944).	*SMA*	*SMA*
Stokers	Code for Determination of Rated Capacities of Bituminous Underfeed Stokers (1944).	*SMA*	*SMA*
Stokers	Recommended Minimum Firebox Dimensions and Base Heights (1944).	*SMA*	*SMA*
Stokers	Recommended Standards Governing Minimum Setting Heights (1944).	*SMA*	*SMA*
Tubing (Seamless Copper and Copper Alloy)	Simplified Practice Recommendation for Copper and Copper-Alloy Round Seamless Tube (1948).	*Mfrs.* *BS*	*BS* R235-48
Tubing (Seamless Copper Water Tube)	Standard Specifications for Seamless Copper Water Tube (1951).	*ASTM*	*ASA* H23.1-1951
Unfired Pressure Vessels	Unfired Pressure Vessel Code (1950).	*ASME*	*ASME*
Unit Heaters	American Standard Approval Requirements for Gas Unit Heaters (1940).	*A.G.A.*	*ASA* Z21.16-1940
Unit Heaters	Standard Code for Testing and Rating Steam Unit Heaters (1950).	ASHVE *IUHA*	ASHVE *IUHA* Bulletin 10
Unit Heaters	Proposed Standard Code for Testing Hot Water Unit Heaters (1942).	*IUHA*	*IUHA*
Unit Ventilators	A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators (1934).	ASHVE	ASHVE
Vacuum Pumps	A.S.H.V.E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps (1934).	ASHVE	ASHVE
Warm Air (Gravity)	Gravity Code and Manual for the Design and Installation of Gravity Warm Air Heating Systems (1947).	*NWAH & ACA*	*NWAH & ACA* Section No. 5
Water Heaters	American Standard Household Automatic Electric Storage-type Water Heaters.	*NEMA*	*ASA* C72.1-1949
Water Heaters	American Standard Approval Requirements for Gas Water Heaters (1950).	*A.G.A.*	*ASA* Z21.10-1950
Water Heaters	*NEMA* Standards for Electric Water Heaters (1945).	*NEMA*	*NEMA* 45-104

TABLE 1. CODES AND STANDARDS (Concluded)

SUBJECT	TITLE	SPONSOR	REFERENCE
Water Heaters	Testing and Rating Hand-Fired Hot Water Supply Boilers (1948).	*FHA*	*BS* CS145-47
Wiring	Interior Wiring Design for Commercial Buildings.	*AIEE*	*AIEE*
Wiring	National Electrical Code, Standard of NBFU and NFPA (1947, with 1949 Supplement).	*NBFU* *NFPA*	*NBFU* Pamphlet No. 70.

ABBREVIATIONS AND ADDRESSES

The Codes and Standards listed in preceding pages of this table can be obtained from the organizations listed in the *Reference Column*.

ACMA	Air Conditioning Manufacturers Association, superseded 1940 by ACRMA.
ACRMA	Air Conditioning and Refrigerating Machinery Association, Southern Bldg., Washington, D. C.
AFA	American Foundrymen's Association, 222 W. Adams St., Chicago, Ill.
A.G.A.	American Gas Association, 420 Lexington Ave., New York, N. Y.
AGAEM	Association of Gas Appliance and Equipment Manufacturers, superseded 1945 by G.A.M.A.
AIA	American Institute of Architects, 1741 New York Ave., Washington, D. C.
AIEE	American Institute of Electrical Engineers, 33 West 39th St., New York 18, N. Y.
AIL	Anthracite Industries Laboratory, 237 Old River Rd., Wilkes Barre, Pa.
ASA	American Standards Association, 70 East 45th St., New York, N. Y.
AS of A	Acoustical Society of America, 919 N. Michigan Ave., Chicago, Ill.
ASHVE	American Society of Heating and Ventilating Engineers, 62 Worth St., New York 13, N. Y.
ASME	American Society of Mechanical Engineers, 29 West 39th St., New York, N. Y.
ASRE	American Society of Refrigerating Engineers, 40 West 40th St., New York, N. Y.
ASTM	American Society for Testing Materials, 1916 Race St , Philadelphia, Pa.
BOCA	Building Officials Conference of America, 51 East 42nd St., New York 17, N. Y.
BOF	Building Officials Foundation, 51 East 42nd St., New York 17, N. Y.
BS	National Bureau of Standards, Washington, D. C.
CMA	Convector Manufacturers Association, 400 W. Madison Ave., Chicago, Ill.
FHA	Federal Housing Administration, Washington, D. C.
GAMA	Gas Appliance Manufacturers' Association, 60 East 42nd St., New York, N. Y.
HI	Hydraulic Institute, 90 West St., New York, N. Y.
HP & ACCNA	Heating, Piping and Air Conditioning Contractors National Association, 1250 Avenue of the Americas, New York, N. Y.
IBR	Institute of Boiler and Radiator Manufacturers, 60 East 42nd St., New York, N. Y.
ICHAM	Institute of Cooking and Heating Appliance Manufacturers, Shoreham Hotel, Washington, D. C.
IMWI	Industrial Mineral Wool Institute, 441 Lexington Ave., New York, N. Y.
IUHA	Industrial Unit Heater Association, 2157 Guardian Bldg., Detroit, Mich.
NAFM	National Association of Fan Manufacturers, 2159 Guardian Bldg., Detroit, Mich.
NBFU	National Board of Fire Underwriters, 85 John St., New York, N. Y.
NEMA	National Electrical Manufacturers Association, 155 East 44th St., New York, N. Y.
NFPA	National Fire Protection Association, 60 Batterymarch St., Boston, Mass.
NHA	National Housing Agency, Washington, D. C.
NRC	National Research Council, 2101 Constitution Ave., Washington, D. C.
NWAH & ACA	National Warm Air Heating and Air Conditioning Association, 145 Public Square, Cleveland, Ohio.
OBI	Oil Burner Institute, superseded 1942 by OHIA
OHIA	Oil Heat Institute of America, 6 East 39th St., New York, N. Y.
OPA	Office of Price Administration, Washington, D. C.
PC	Producers Council, 815–15th St., N.W., Washington, D. C.
PFMA	Propeller Fan Manufacturers Association, 2159 Guardian Bldg., Detroit, Mich.
PHMI	Prefabricated Home Manufacturers Institute, 908 20th St., N.W., Washington, D. C.
REMA	Refrigeration Equipment Manufacturers Association, 1346 Connecticut Ave., N.W., Washington 6, D. C.
RMA	Refrigerating Machinery Association. See ACRMA.
SAE	Society of Automotive Engineers, 29 West 39th St., New York, N. Y.
SBI	Steel Boiler Institute, 1308 Land Title Bldg., Philadelphia 10, Pa.
SMA	Stoker Manufacturers Association, 307 N. Michigan Ave., Chicago, Ill.
TEMA	Tubular Exchanger Manufacturers Association, 366 Madison Ave., New York, N. Y.
UL	Underwriters' Laboratories, 207 East Ohio St., Chicago, Ill.
USDA	United States Department of Agriculture, Washington, D. C.
USPHS	United States Public Health Service, Washington, D. C.

Table of Circumferences and Areas of Circles

Diameter in Inches	Area		Circumference		Diameter in Inches	Area		Circumference	
	Sq In.	Sq Ft	Inches	Feet		Sq In.	Sq Ft	Inches	Feet
¼	0.049	0.0003	0.785	0.0652	28	615.8	4.276	87.97	7.330
½	0.196	0.0014	1.571	0.1309	28½	637.9	4.430	89.54	7.462
¾	0.442	0.0031	2.356	0.1964	29	660.52	4.587	91.11	7.592
1	0.785	0.0054	3.142	0.2618	29½	683.5	4.747	92.63	7.725
1¼	1.227	0.0085	3.927	0.3273	30	706.8	4.909	94.25	7.854
1½	1.767	0.0123	4.712	0.3927	31	754.8	5.241	97.39	8.116
1¾	2.405	0.0167	5.498	0.4582	32	804.3	5.585	100.5	8.378
2	3.142	0.0218	6.283	0.5236	33	855.3	5.940	103.7	8.639
2¼	3.976	0.0276	7.069	0.5891	34	907.9	6.305	106.8	8.901
2½	4.909	0.0341	7.854	0.6546	35	962.1	6.681	109.9	9.163
2¾	5.939	0.0412	8.639	0.7200	36	1018.0	7.069	113.1	9.425
3	7.069	0.0491	9.425	0.7854	37	1075.0	7.467	116.2	9.686
3¼	8.296	0.0576	10.21	0.8510	38	1134.0	7.876	119.4	9.948
3½	9.621	0.0668	10.99	0.9160	39	1195.0	8.296	122.5	10.21
3¾	11.04	0.0767	11.78	0.9818	40	1256.0	8.727	125.6	10.47
4	12.57	0.0873	12.57	1.047	41	1320.0	9.168	128.8	10.73
4¼	14.19	0.0986	13.35	1.113	42	1385.0	9.621	131.9	10.99
4½	15.90	0.1104	14.14	1.178	43	1452.0	10.08	135.1	11.26
4¾	17.72	0.1231	14.92	1.243	44	1521.0	10.56	138.2	11.52
5	19.64	0.1364	15.71	1.309	45	1590.0	11.04	141.4	11.78
5¼	21.65	0.1504	16.49	1.374	46	1662.0	11.54	144.5	12.04
5½	23.76	0.1650	17.28	1.440	47	1735.0	12.05	147.7	12.30
5¾	25.97	0.1840	18.06	1.505	48	1810.0	12.51	150.8	12.57
6	28.27	0.1964	18.85	1.571	49	1886.0	13.09	153.9	12.83
6¼	30.68	0.2131	19.64	1.637	50	1963.0	13.64	157.1	13.09
6½	33.18	0.2304	20.42	1.702	51	2043.0	14.19	160.2	13.35
6¾	35.79	0.2486	21.21	1.768	52	2124.0	14.75	163.4	13.61
7	38.49	0.2673	21.99	1.833	53	2206.0	15.32	166.5	13.88
7¼	41.28	0.2867	22.78	1.899	54	2290.0	15.90	169.6	14.14
7½	44.18	0.3068	23.56	1.964	55	2376.0	16.50	172.8	14.40
7¾	47.17	0.3276	24.35	2.029	56	2463.0	17.10	175.9	14.66
8	50.27	0.3491	25.13	2.094	57	2552.0	17.72	179.1	14.92
8¼	53.46	0.3713	25.92	2.160	58	2642.0	18.35	182.2	15.18
8½	56.75	0.3942	26.70	2.225	59	2734.0	18.99	185.4	15.45
8¾	60.13	0.4175	27.49	2.291	60	2827.0	19.63	188.5	15.71
9	63.62	0.4418	28.27	2.356	61	2922.0	20.29	191.6	15.97
9¼	67.20	0.4668	29.06	2.422	62	3019.0	20.97	194.8	16.23
9½	70.88	0.4923	29.85	2.488	63	3117.0	21.65	197.9	16.49
9¾	74.66	0.5185	30.63	2.553	64	3217.0	22.34	201.1	16.76
10	78.54	0.5454	31.42	2.618	65	3318.0	23.04	204.2	17.02
10½	86.59	0.6010	32.99	2.750	66	3421.0	23.76	207.3	17.28
11	95.03	0.6600	34.56	2.880	67	3526.0	24.48	210.5	17.54
11½	103.9	0.7215	36.13	3.011	68	3632.0	25.22	213.6	17.80
12	113.1	0.7854	37.70	3.142	69	3739.0	25.97	216.8	18.06
12½	122.7	0.8520	39.27	3.273	70	3848.0	26.73	219.9	18.33
13	132.7	0.9218	40.84	3.403	71	3959.0	27.49	223.1	18.59
13½	143.1	0.9937	42.41	3.535	72	4072.0	28.27	226.2	18.85
14	153.9	1.069	43.98	3.665	73	4185.0	29.07	229.3	19.11
14½	165.1	1.146	45.55	3.796	74	4301.0	29.87	232.5	19.37
15	176.7	1.227	47.12	3.927	75	4418.0	30.68	235.6	19.63
15½	188.7	1.310	48.69	4.058	76	4536.0	31.50	238.8	19.90
16	201.1	1.396	50.27	4.189	77	4657.0	32.34	241.9	20.16
16½	213.8	1.485	51.84	4.321	78	4778.0	33.18	245.0	20.42
17	226.9	1.576	53.41	4.451	79	4902.0	34.04	248.2	20.68
17½	240.5	1.670	54.98	4.582	80	5027.0	34.91	251.3	20.94
18	254.5	1.767	56.55	4.712	81	5153.0	35.78	254.5	21.21
18½	268.8	1.867	58.12	4.845	82	5281.0	36.67	257.6	21.47
19	283.5	1.969	59.69	4.974	83	5411.0	37.57	260.8	21.73
19½	298.6	2.074	61.26	5.105	84	5542.0	38.48	263.9	21.99
20	314.2	2.182	62.83	5.236	85	5675.0	39.41	267.0	22.25
20½	330.1	2.293	64.40	5.367	86	5809.0	40.34	270.2	22.51
21	346.4	2.405	65.97	5.498	87	5945.0	41.28	273.3	22.78
21½	361.1	2.508	67.54	5.629	88	6082.0	42.24	276.5	23.04
22	380.1	2.640	69.12	5.760	89	6221.0	43.20	279.6	23.30
22½	397.6	2.761	70.69	5.891	90	6362.0	44.18	282.7	23.56
23	415.5	2.885	72.26	6.021	91	6504.0	45.17	285.9	23.82
23½	433.7	3.012	73.83	6.153	92	6648.0	46.16	289.0	24.09
24	452.4	3.142	75.40	6.283	93	6793.0	47.17	292.2	24.35
24½	471.4	3.274	76.97	6.415	94	6940.0	48.19	295.3	24.61
25	490.9	3.409	78.54	6.545	95	7088.0	49.22	298.4	24.87
25½	510.7	3.547	80.11	6.676	96	7238.0	50.27	301.6	25.13
26	530.9	3.687	81.68	6.807	97	7390.0	51.32	304.7	25.39
26½	551.6	3.832	83.25	6.938	98	7543.0	52.38	307.9	25.66
27	572.6	3.976	84.82	7.069	99	7698.0	53.46	311.0	25.92
27½	593.9	4.125	86.39	7.199	100	7854.0	54.54	314.2	26.18

CATALOG DATA SECTION

HEATING VENTILATING GUIDE AIR CONDITIONING 1952

INDEX TO ADVERTISERS

HEATING VENTILATING AIR CONDITIONING GUIDE, 1952

A

B

INDEX TO MODERN EQUIPMENT

OF ADVERTISERS APPEARING IN HEATING VENTILATING AIR CONDITIONING GUIDE, 1952

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

AIR DIFFUSERS AND VENTILATORS, CEILING, FLOOR AND WALL
Air Control Products, Inc., 1264-1265
Air Devices, Inc., 1178, 1263
Aladdin Heating Corp., 1234
Anemostat Corp. of America, 1266-1267
Barber-Colman Co., 1269
W. B. Connor Engineering Corp., 1188-1191, 1270-1271
Charles Demuth & Sons, Inc., 1272
Hart & Cooley Mfg. Co., 1274-1275
Independent Register Co., The, 1278
Jenn-Air Products Co., 1245
Minneapolis-Honeywell Regulator Co., 1316-1317
Pyle-National Co., The, Multi-Vent Div., 1280-1281
Register & Grille Mfg. Co., Inc., 1279
Standard Stamping & Perforating Co., 1282
Titus Mfg. Corp., 1284-1285
Tuttle & Bailey, Inc., 1286-1287
United States Register Co., 1288-1289
Universal Diffuser Corp., 1290

AIR DUCTS (*See Ducts*)

AIR ELIMINATORS
V. D. Anderson Co., The, 1410-1411
C. A. Dunham Co , 1415-1419
Illinois Engineering Co., 1424-1425
Jas. P. Marsh Corp., 1428-1429
Sarco Co., Inc., 1434-1435

AIR FILTER GAGE
Air Devices, Inc., 1178, 1263
Owens-Corning Fiberglas Corp , 1198
Research Products Corp., 1199

AIR FILTERS (*See Filters, Air; Air Cleaning Equipment*)

AIR MEASURING INDICATING AND RECORDING INSTRUMENTS
Anemostat Corp. of America, 1266-1267
Illinois Testing Laboratories, Inc., 1311
Minneapolis-Honeywell Regulator Co., 1316-1317
Parks-Cramer Co., 1114-1115
Powers Regulator Co., The, 1321
Taylor Instrument Co's , 1326

AIR MOISTENING APPARATUS (*See Humidifiers*)

AIR PURIFYING APPARATUS
Air & Refrigeration Corp., 1099
Air Filter Corp., 1177
American Solvent Recovery Corp., 1186-1187
V. D. Anderson Co., The, 1410-1411
W. B. Connor Engineering Corp., 1188-1191, 1270-1271
J. F. Pritchard & Co. (Puridryer Dept.), 1212
H. J. Somers, Inc., 1200-1201
Trion, Inc., 1202
Universal Diffuser Corp., 1290
Wilson & Co., Inc., 1205

AIR RECEIVERS (*See Receivers, Air*)

AIR RECOVERY, Method of
American Solvent Recovery Corp., 1186-1187
W. B. Connor Engineering Corp., 1188-1191, 1270-1271

AIR TRAPS (*See Traps, Air*)

AIR TUBING, Flexible Metal (*See Tubing, Flexible Metallic*)

AIR VELOCITY METERS (*See Meters*)

AIR WASHERS
Air & Refrigeration Corp., 1099
American Blower Corp., 1100-1101
Bahnson Co., The, 1102-1103
Bayley Blower Co., 1236
Bishop & Babcock Mfg. Co., The, (Massachusetts Blower Div.), 1237
Buensod-Stacey, Inc., 1104
Buffalo Forge Co., 1238
Carrier Corp., 1106-1107
Clarage Fan Co., 1105
Continental Air Filters, Inc., 1192
Marlo Coil Co., 1229
D. J. Murray Mfg. Co., 1163
New York Blower Co., The, 1250
Niagara Blower Co., 1112
Parks-Cramer Co., 1114-1115
Trane Co., The, 1168-1169
Western Blower Co., 1255
Westinghouse Electric Corp., Sturtevant Div., 1204, 1256
York Corp., 1119

ALARMS, Water Level
McDonnell & Miller, Inc., 1372-1375
Photoswitch, Inc., (Affiliate of Combustion Control Corp.), 1301
Yarnall-Waring Co., 1442

ALUMINUM DUCTS (*See Ducts, Aluminum*)

ALUMINUM FOIL, Insulation
Infra Insulation, Inc., 1482
Lockport Cotton Batting Co., 1474
Reflectal Corp., 1483
Silvercote Products, Inc., 1484

ALUMINUM FOIL VAPOR BARRIER (*See Aluminum Foil*)

ALUMINUM SHEETS (*See Sheets, Aluminum*)

AMMONIA COILS (*See Coils, Ammonia*)

ANCHORS, Pipe
H. W. Porter & Co., Inc., 1454

ANEMOMETERS
Illinois Testing Laboratories, Inc., 1311

ASBESTOS PRODUCTS (*See Insulation*)
Philip Carey Mfg. Co., The, 1448-1449
Johns-Manville, 1470-1471
H. W. Porter & Co., Inc., 1454
Union Asbestos & Rubber Co., 1456-1457
Grant Wilson, Inc., 1452

ATMOSPHERIC COOLING TOWERS (*See Cooling Towers, Atmospheric*)

ATOMIZING SPRAY NOZZLES (*See Spray Nozzles*)

ATTIC FAN COOLERS (*See Fans, Attic; Ventilators, Attic*)

AUTOMATIC FUEL BURNING EQUIPMENT (*See Burners, Automatic; Gas Burners; Oil Burners; Stokers*)

AUTOMATIC SHUTTERS (*See Shutters, Automatic*)

AXIAL FLOW FANS (*See Fans, Axial Flow*)

BASEBOARD HEATING
American Radiator & Standard Sanitary Corp., 1328-1331
Brown Products Co., 1333
Burnham Corp., 1335
Campbell Heating Co., 1124-1125, 1142
Crane Co., 1336-1337
C. A. Dunham Co., 1415-1419
Fedders-Quigan Corp., 1153
General Automatic Products Corp., 1338
Kritzer Radiant Coils, Inc., 1222
National Radiator Co., The, 1340-1341
Rome-Turney Radiator Co., The, 1223
United States Radiator Corp., 1344-1345
Vulcan Radiator Co., The, 1224
Warren Webster & Co., 1437-1441
Weil-McLain Co., 1351

BELLOWS
Chicago Metal Hose Div., Flexonics Corp., 1176
Fulton Sylphon Div., The, Robertshaw-Fulton Controls Co., 1304-1305

BENDS, Pipe, Ferrous and Non-Ferrous
Grinnell Co., Inc., 1154-1155
Arthur Harris & Co., 1295
Ladish Co., 1296
Parks-Cramer Co., 1114-1115

BENDS (*See Pipe, Returns*)

BLOCKS, Asbestos
Philip Carey Mfg Co., The, 1448-1449
Union Asbestos & Rubber Co., 1456-1457
Grant Wilson, Inc., 1452

BLOCKS, Glass
American Structural Products Co., 1460
Pittsburgh Corning Corp., 1462-1463

BLOWER HOUSINGS
Bishop & Babcock Mfg. Co., The, (Massachusetts Blower Div.), 1237
Champion Blower & Forge Co., 1239
Lau Blower Co., The, 1248
Meyer Furnace Co., 1130-1131
Westinghouse Electric Corp., Sturtevant Div., 1204, 1256

BLOWER MOTORS (*See Motors, Electric*)

BLOWERS, Centrifugal (*See Fans*)

BLOWERS, Fan (*See Fans, Supply and Exhaust*)

BLOWERS, Forced Draft
Aladdin Heating Corp., 1234
American Blower Corp., 1100-1101
Bayley Blower Co., 1236
Bishop & Babcock Mfg. Co., The, (Massachusetts Blower Div.), 1237
Buffalo Forge Co., 1238
Champion Blower & Forge Co., 1239
Clarage Fan Co., 1105
General Blower Co., 1241
Ilg Electric Ventilating Co., 1156, 1244
Joy Mfg. Co , 1246-1247
Herman Nelson Div., American Air Filter Co., Inc., 1164-1165
New York Blower Co., The, 1250
United States Air Conditioning Corp., 1116

Williams Oil-O-Matic Div., Eureka Williams Corp., 1393

BOILERS, Magazine Feed
Farrar & Trefts, Inc , 1359
Spencer Heater, Lycoming-Spencer Div., Avco Mfg. Corp., 1348-1349

BOILERS, Oil Burning
Air Devices, Inc., 1178, 1263
Airtemp Div., Chrysler Corp., 1126-1127
Aldrich Co., 1332
American Radiator & Standard Sanitary Corp., 1328-1331
Babcock & Wilcox Co., The, 1352
Brown Products Co., 1333
Bryan Steam Corp., 1334
Burnham Corp., 1335
Cleaver-Brooks Co., 1353
Combustion Engineering - Superheater, Inc., 1354-1355
Crane Co , 1336-1337
Cyclotherm Corp., 1356
Dewey-Shepard Boiler Co., 1357
Dutton Boilers, Division Hapman-Dutton Co , 1358
Farrar & Trefts, Inc., 1359
Fitzgibbons Boiler Co., Inc , 1360-1361
General Electric Co 1108-1109
International Boiler Works Co., The, 1362
S. T. Johnson Co , 1386-1387
Johnston Bros., Inc , 1363
Kewanee Boiler Corp., 1364-1367
National Radiator Co , The, 1340-1341
Norge Heat, Div. of Borg-Warner Corp., 1134-1135
Pacific Steel Boiler Div., U. S. Radiator Corp., 1342-1343
Petroleum Heat & Power Co., 1388-1389
Frank Prox Co., Inc., 1346
H. B. Smith Co., Inc., The, 1350
Spencer Heater, Lycoming-Spencer Div., Avco Mfg. Corp., 1348-1349
Titusville Iron Works Co , The (Div. of Struthers-Wells Corp.), 1368
United States Radiator Corp , 1344-1345
Vapor Heating Corp., 1369
Weil-McLain Co., 1351
Williams Oil-O-Matic Div., Eureka Williams Corp., 1393

BOILERS, Steel
Airtemp Div., Chrysler Corp., 1126-1127
Aldrich Co., 1332
Bryan Steam Corp., 1334
Burnham Corp., 1335
Cleaver-Brooks Co., 1353
Combustion Engineering - Superheater, Inc., 1354-1355
Cyclotherm Corp., 1356
Dewey-Shepard Boiler Co , 1357
Dutton Boilers, Division Hapman-Dutton Co., 1358
Farrar & Trefts, Inc., 1359
Fitzgibbons Boiler Co., Inc., 1360-1361
Foster Wheeler Corp., 1209
General Electric Co., 1108-1109
S. T. Johnson Co., 1386-1387
Johnston Bros., Inc., 1363
Kewanee Boiler Corp., 1364-1367
National Radiator Co , The, 1340-1341
Pacific Steel Boiler Div., U. S. Radiator Corp., 1342-1343
Petroleum Heat & Power Co., 1388-1389
Rheem Mfg. Co., 1136
Spencer Heater, Lycoming-Spencer Div., Avco Mfg. Corp., 1348-1349
Titusville Iron Works Co., The (Div. of Struthers-Wells Corp.), 1368

Williams Oil-O-Matic Div., Eureka Williams Corp., 1393

BOILERS, Unit Steam Generator
Cleaver-Brooks Co., 1353
Combustion Engineering - Superheater, Inc., 1354-1355
Cyclotherm Corp., 1356
Dutton Boilers, Division Hapman-Dutton Co., 1358
International Boiler Works Co., The, 1362
Johnston Bros., Inc., 1363
Titusville Iron Works Co., The, (Div. of Struthers-Wells Corp.), 1368
Vapor Heating Corp., 1369

BOILERS, Water Tube
American Radiator & Standard Sanitary Corp , 1328-1331
Babcock & Wilcox Co., The, 1352
Bryan Steam Corp., 1334
Combustion Engineering - Superheater, Inc., 1354-1355
Foster Wheeler Corp., 1209
International Boiler Works Co., The, 1362
H. B Smith Co., Inc., The, 1350
Titusville Iron Works Co., The (Div. of Struthers-Wells Corp), 1368

BRACKETS, Radiator
American Radiator & Standard Sanitary Corp., 1328-1331
Carty & Moore Engineering Co., 1414
Grinnell Co., Inc., 1154-1155
National Radiator Co., The, 1340-1341
United States Radiator Corp., 1344-1345

BREECHINGS
Farrar & Trefts, Inc., 1359

BURNER PROTECTION, Gas and Oil
Combustion Control Corp., 1301
Mercoid Corp., The, 1314
Milwaukee Gas Specialty Co., 1315

BURNERS, Automatic
Ace Engineering Co., 1382
Aldrich Co., 1332
Automatic Burner Corp., 1383
Enterprise Engine & Machinery Co., Burner Div., Sub. of General Metals Corp., 1384
S. T. Johnson Co., 1386-1387
H C Little Burner Co , 1385
Petroleum Heat & Power Co., 1388-1389
Ray Oil Burner Co., 1390-1391
C. L. Rayfield Co., 1392
Sonner Burner Co., Inc., 1378
United States Radiator Corp., 1344-1345
Webster Engineering Co., The, 1379

BURNERS, Combination for Natural and L.P. Gases
L. J. Mueller Furnace Co., 1132-1133

BURNERS, Combination Gas and Oil
Ace Engineering Co., 1382
Babcock & Wilcox Co., The, 1352
Cleaver-Brooks Co., 1353
Dewey-Shepard Boiler Co., 1357
Enterprise Engine & Machinery Co., Burner Div., Sub. of General Metals Corp., 1384
S. T. Johnson Co., 1386-1387
Petroleum Heat & Power Co., 1388-1389
Ray Oil Burner Co., 1390-1391
C. L. Rayfield Co , 1392
Webster Engineering Co., The, 1379

BURNERS, Gas (*See Gas Burners*)

BURNERS, Oil (*See Oil Burner*)

CALKING, Building
Philip Carey Mfg. Co., The, 1448-1449

CAPS, Vent Flue
G. C. Breidert Co., The, 1258

CARBON, Activated
American Solvent Recovery Corp., 1186-1187
W. B. Connor Engineering Corp , 1188-1191, 1270-1271

CASTINGS, Bronze and Dairy Nickel Silver Metal
Arthur Harris & Co., 1295

CELLULAR GLASS (*See Glass, Cellular*)

CEMENT, Asbestos
Philip Carey Mfg. Co , The, 1448-1449

CEMENT, Insulating
Philip Carey Mfg. Co , The, 1448-1449
Johns Manville, 1470-1471
Owens-Corning Fiberglas Corp , 1451
Union Asbestos & Rubber Co., 1456-1457
Z-Crete Div., Zonolite Co , 1458

CEMENT, Mineral Wool
Philip Carey Mfg. Co , The, 1448-1449

CEMENT, Refractory (*See Refractories*)

CHIMNEY TOPS
G. C. Breidert Co., The, 1258

CHIMNEYS (*See Breechings*)

CIRCULATORS, Hot Water Heating
Bell & Gossett Co., 1396-1397
Chicago Pump Co., 1403
Crane Co., 1336-1337
Hoffman Specialty Co., 1420-1423
Jas. P. Marsh Corp., 1428-1429
Minneapolis-Honeywell Regulator Co , 1316-1317
National Radiator Co , The, 1340-1341
Pacific Steel Boiler Div., U. S Radiator Corp., 1342-1343
Taco Heaters, Inc., 1400
H. A. Thrush & Co., 1398-1399
Trane Co., The, 1168-1169
United States Radiator Corp., 1344-1345

CLEANERS, Air (*See Air Cleaning Equipment*)

COAL BURNERS (*See Burners, Automatic*)

COATINGS, Protective
Philip Carey Mfg. Co , The, 1448-1449
Insul-Mastic Corp. of America, 1467

COILS, Aluminum
Aerofin Corp., 1217-1219
Marlo Coil Co., 1229
McQuay, Inc., 1158-1159
John J. Nesbitt, Inc., 1166
Niagara Blower Co., 1112
Patterson-Kelley Co., Inc., The, 1221

Barber-Colman Co., 1269
Johnson Service Co., 1312–1313
Minneapolis-Honeywell Regulator Co., 1316–1317
Powers Regulator Co., The, 1321
Young Regulator Co., 1291

CONTROL, Boiler Water Level
McDonnell & Miller, Inc., 1372–1375
Mercoid Corp., The, 1314
Penn Controls, Inc., 1319
Sarco Co., Inc., 1434–1435
Warren Webster & Co., 1437–1441

CONTROL EQUIPMENT, Combustion
Combustion Control Corp., 1301
Detroit Regulator Co , 1302
General Controls, 1306–1307
Illinois Engineering Co , 1424-1425
Mercoid Corp., The, 1314
Minneapolis-Honeywell Regulator Co , 1316–1317
National Radiator Co., The, 1340–1341
Penn Controls, Inc., 1319
Perfex Corporation, 1320
Simplex Mfg. Co., 1323
United States Radiator Corp., 1344–1345
Webster Engineering Co , The, 1379

CONTROL EQUIPMENT, Time
April Showers, Inc., 1459
Barber-Colman Co., 1269
General Controls, 1306–1307
Minneapolis-Honeywell Regulator Co., 1316–1317
Penn Controls, Inc., 1319
Perfex Corp., 1320

CONTROLLERS AND CONTROL EQUIPMENT *(See Humidity and Temperature Control)*
Alco Valve Co., 1299
American Moistening Co., 1214
Bahnson Co., The, 1102-1103
Barber-Colman Co , 1269
Combustion Control Corp., 1301
Detroit Regulator Co., 1302
C. A. Dunham Co., 1415–1419
Fulton Sylphon Div., The, Robertshaw-Fulton Controls Co., 1304–1305
General Controls, 1312–1313
Hoffman Speciality Co., 1420–1423
Hubbell Corp., 1310
Illinois Engineering Co., 1424–1425
Johnson Service Co., 1312–1313
Jas. P. Marsh Corp., 1428–1429
Mercoid Corp., The, 1314
Minneapolis-Honeywell Regulator Co., 1316–1317
Parks-Cramer Co., 1114–1115
Penn Controls, Inc., 1319
Perfex Corporation, 1320
Photoswitch, Inc., (Affiliate of Combustion Control Corp.), 1301
Powers Regulator Co , The, 1321
Sarcotherm Controls, Inc., 1433
Simplex Mfg. Co., 1323
Taylor Instrument Cos., 1326

CONTROLS, Water Level
Maid-O'-Mist, Inc., 1426–1427

CONVECTION HEATERS
American Radiator & Standard Sanitary Corp., 1328–1331
C A. Dunham Co., 1415–1419
Fedders-Quigan Corp., 1153
Kritzer Radiant Coils, Inc., 1222
Modine Mfg. Co., 1160–1161
National Radiator Co., The, 1340–1341
Rome-Turney Radiator Co., The, 1223
Shaw-Perkins Mfg. Co., 1347
Trane Co., The, 1168–1169
Warren Webster & Co., 1437-1441
Young Radiator Co., 1173

CONVECTORS & CONVECTOR ENCLOSURES
Acme Industries, Inc., 1216
Airtherm Mfg. Co., 1141
American Radiator & Standard Sanitary Corp., 1328–1331
Brown Products Co., 1333
Crane Co., 1336–1337
C. A. Dunham Co., 1415–1419
Fedders-Quigan Corp., 1153
G & O Mfg. Co., The, 1220
Kritzer Radiant Coils, Inc., 1222
Modine Mfg. Co , 1160–1161
National Radiator Co., The, 1340–1341
John J. Nesbitt, Inc., 1166
Rome-Turney Radiator Co., The, 1223
Shaw-Perkins Mfg Co., 1347
H. J Somers, Inc , 1200–1201
Trane Co , The, 1168–1169
Tuttle & Bailey, Inc., 1286–1287
United States Radiator Corp., 1344–1345
Warren Webster & Co., 1437-1441
Young Radiator Co., 1173

COOLING EQUIPMENT, Air
Aerofin Corp , 1217–1219
Airtemp Div., Chrysler Corp , 1126–1127
American Blower Corp , 1100–1101
American Moistening Co., 1214
American Radiator & Standard Sanitary Corp , 1328–1331
April Showers, Inc , 1459
Bahnson Co , The, 1102–1103
Baker Refrigeration Corp., 1225
Bell & Gossett Co., 1396–1397
Bishop & Babcock Mfg. Co., The, (Massachusetts Blower Div.), 1237
Buensod-Stacey, Inc , 1104
Buffalo Forge Co., 1238
Carrier Corp., 1106-1107
DeBothezat Fans Div , American Machine & Metals, Inc., 1240
Fedders-Quigan Corp., 1153
Frigidaire, Div. of General Motors, 1110
General Electric Co , 1108–1109
Ingersoll Products Div., Borg-Warner Corp., 1481
Jenn-Air Products Co., 1245
Joy Mfg. Co., 1246–1247
Kennard Corp., 1157
Marlo Coil Co., 1229
McQuay, Inc., 1158–1159
Meyer Furnace Co., The, 1130–1131
L J Mueller Furnace Co , 1132-1133
D. J Murray Mfg. Co., 1163
Niagara Blower Co., 1112
Parks-Cramer Co , 1114-1115
Ready Power Co., The, 1231
Refrigeration Appliances, Inc., 1232
Refrigeration Economics Co., Inc , 1167
Refrigeration Engineering, Inc., 1233
Servel, Inc., 1137
Trane Co., The, 1168-1169
United States Air Conditioning Corp., 1116
Utility Appliance Corp., 1140
Westinghouse Electric Corp., Air Conditioning Div., 1117
Worthington Pump & Machinery Corp., 1118
York Corp., 1119
Young Radiator Co., 1173

COOLING EQUIPMENT, Oil
Acme Industries, Inc., 1216
Aerofin Corp., 1217-1219
Bell & Gossett Co., 1396–1397
Marley Co , Inc., The, 1211
Niagara Blower Co., 1112
Patterson-Kelley Co., Inc., The, 1221
Ready Power Co., The, 1231
Refrigeration Economics Co., Inc., 1167
Refrigeration Engineering, Inc., 1233
Servel, Inc., 1137
Trane Co., The, 1168–1169
Worthington Pump & Machinery Corp., 1118
Young Radiator Co., 1173

COOLING EQUIPMENT, Water *(See also Water Cooling)*
Acme Industries, Inc , 1216
Aerofin Corp., 1217–1219
Airtemp Div., Chrysler Corp., 1126–1127
Baker Refrigeration Corp., 1225
Baltimore Aircoil Co., Inc., 1226
Bell & Gossett Co , 1396–1397
Binks Mfg. Co., 1206-1207
Carrier Corp., 1106–1107
Fedders-Quigan Corp., 1153
Fluor Corp , Ltd., The, 1208
Foster Wheeler Corp , 1209
Frick Co., 1228
Frigidaire, Div. of General Motors, 1110
General Electric Co., 1108–1109
Hastings Air Conditioning Co., Inc., 1111
Kennard Corp., 1157
Lilie-Hoffmann Cooling Towers, Inc , 1210
Marley Co., Inc., The, 1211
Marlo Coil Co., 1229
McQuay, Inc , 1158-1159
Modine Mfg. Co., 1160–1161
National Radiator Co., The, 1340–1341
John J Nesbitt, Inc., 1166
Niagara Blower Co , 1112
Patterson-Kelley Co., Inc , The, 1221
J. F Pritchard & Co , 1212
Ready Power Co., The, 1231
Refrigeration Appliances, Inc., 1232
Refrigeration Economics Co , Inc , 1167
Refrigeration Engineering, Inc., 1233
Servel, Inc , 1137
Trane Co , The, 1168–1169
Water Cooling Equipment Co , 1213
Westinghouse Electric Corp., Air Conditioning Div., 1117
Worthington Pump & Machinery Corp., 1118
Yarnall-Waring Co , 1442
York Corp , 1119
Young Radiator Co., 1173

COOLING TOWER FANS
Baltimore Aircoil Co , Inc , 1226
DeBothezat Fans Div., American Machine & Metals, Inc., 1240
Hartzell Propeller Fan Co., Div of Castle Hills Corp., 1242
Lilie-Hoffmann Cooling Towers, Inc , 1210
Joy Mfg. Co., 1246–1247
Marley Co., Inc., The, 1211
J F Pritchard & Co., 1212
United States Air Conditioning Corp., 1116
L. J. Wing Mfg. Co., 1170–1172

COOLING TOWERS, Atmospheric, Mechanical Draft, Forced Draft, Induced Draft, *(See also Cooling Equipment, Water)*
Acme Industries, Inc., 1216
Air & Refrigeration Corp., 1099
American Blower Corp., 1100–1101
Baker Refrigeration Corp., 1225
Baltimore Aircoil Co., Inc., 1226
Binks Mfg. Co., 1206–1207
Buensod-Stacey, Inc., 1104
Carrier Corp., 1106–1107

Trion, Inc., 1202
Wilson & Co., Inc., 1205

DUST COLLECTORS, Cloth Type
American Air Filter Co., Inc., 1183-1185

EJECTORS, Sewage
Chicago Pump Co., 1403

ELECTROSTATIC AIR CLEANERS
Air-Maze Corp., 1180-1182
American Air Filter Co., Inc., 1183-1185
Trion, Inc., 1202
Westinghouse Electric Corp., Sturtevant Div., 1204, 1256
Wilson & Co., Inc., 1205

ELECTROSTATIC FILTERS, Gas
Trion, Inc., 1202

ELIMINATORS, Air
Maid-O'-Mist, Inc., 1426-1427

EVAPORATIVE CONDENSERS (*See Condensers and Evaporators*)

EVAPORATORS
American Blower Corp., 1100-1101
Bell & Gossett Co., 1396-1397
Buffalo Forge Co., 1238
Farrar & Trefts, Inc., 1359
Foster Wheeler Corp., 1209
Arthur Harris & Co., 1295
Marlo Coil Co., 1229
Refrigeration Appliances, Inc., 1232
Refrigeration Engineering, Inc., 1233
Rome-Turney Radiator Co., The, 1223
Trane Co., The, 1168-1169
Westinghouse Electric Corp., Air Conditioning Div., 1117
Young Radiator Co., 1173

EXHAUST HEADS (*See Heads, Exhaust*)

EXHAUST TUBING, Flexible (*See Tubing, Flexible, Metallic*)

EXHAUSTERS
Air Devices, Inc., 1178, 1263
American Blower Corp., 1100-1101
Bayley Blower Co., 1236
G. C. Breidert Co., 1258
Buffalo Forge Co., 1238
Champion Blower & Forge Co., 1239
DeBothezat Fans Div, American Machine & Metals, Inc., 1240
General Blower Co., 1241
Hartzell Propeller Fan Co., Div. of Castle Hills Corp., 1242
Ilg Electric Ventilating Co., 1156, 1244
Jenn-Air Products Co., 1245
Joy Mfg., Co., 1246-1247
John J. Nesbitt, Inc., 1166
New York Blower Co., The, 1250
Swartwout Co., The, 1261
Trade-Wind Motorfans, Inc., 1254
Trane Co., The, 1168-1169
United States Air Conditioning Corp., 1116
Western Engineering & Mfg. Co., Inc., 1262
Westinghouse Electric Corp., Sturtevant Div., 1204, 1256
L. J. Wing Mfg. Co., 1170-1172

EXPANSION JOINTS
Badger Mfg. Co., 1293
Celotex Corp., The, 1466
Flexonics Corp., Expansion Joint Div., (Formerly Chicago Metal Hose Corp.), 1294
Foster Wheeler Corp., 1209
Fulton Sylphon Div., The, Robertshaw-Fulton Controls Co., 1304-1305
Arthur Harris & Co., 1295
Illinois Engineering Co., 1424-1425
Owens-Corning Fiberglas Corp., 1451
H. W. Porter & Co., Inc., 1454
Yarnall-Waring Co., 1442
Young Regulator Co., 1291

EXPANSION LOOPS
Durant Insulated Pipe Co., 1453
Ric-wiL Co., The, 1455
Z-Crete Div., Zonolite Co., 1458

EXPOSITIONS
International Exposition Co., 1395

FAN BLADES
Torrington Mfg. Co., The, 1252-1253

FAN MOTORS (*See Motors, Electric*)

FAN STACKS
Dutton Boilers, Division Hapman-Dutton Co., 1358

FANS, Attic
American Blower Corp., 1100-1101
American Coolair Corp., 1235
Buffalo Forge Co., 1238
Philip Carey Mfg. Co., The, 1448-1449
Champion Blower & Forge Co., 1239
DeBothezat Fans Div., American Machine & Metals, Inc., 1240
General Blower Co., 1241
Hunter Fan & Ventilating Co., Inc., 1243
Ilg Electric Ventilating Co., Inc., 1156, 1244
Jenn-Air Products Co., 1245
Lau Blower Co., The, 1248
H. J. Somers, Inc., 1200-1201
Torrington Mfg. Co., The, 1252-1253
Trade-Wind Motorfans, Inc., 1254

FANS, Axial Flow
American Blower Corp., 1100-1101
Bahnson Co., The, 1102-1103
Baltimore Aircoil Co., Inc., 1226
Buffalo Forge Co., 1238
Hartzell Propeller Fan Co., Div. of Castle Hills Corp., 1242
Hunter Fan & Ventilating Co., 1243
Ilg Electric Ventilating Co., 1156, 1244
Joy Mfg. Co., 1246-1247
New York Blower Co., The, 1250
Propellair Div., Robbins & Myers, Inc., 1251
Western Engineering & Mfg. Co., Inc., 1262
Westinghouse Electric Corp., Sturtevant Div., 1204, 1256
L. J. Wing Mfg. Co., 1170-1172

FANS, Centrifugal
Aladdin Heating Corp., 1234
American Blower Corp., 1100-1101
American Foundry & Furnace Co., 1120-1121
Bayley Blower Co., 1236
Bishop & Babcock Mfg. Co., The, (Massachusetts Blower Div.), 1237
Buffalo Forge Co., 1238
Campbell Heating Co, 1124-1125, 1142
E. K. Campbell Co., 1143
Champion Blower & Forge Co., 1239
Clarage Fan Co., 1105
General Blower Co., 1241
Hastings Air Conditioning Co., Inc., 1111
Ilg Electric Ventilating Co., 1156, 1244
Morrison Products, Inc., 1249
L. J. Mueller Furnace Co., 1132-1133
Herman Nelson Div., American Air Filter Co., Inc., 1164-1165
New York Blower Co, The, 1250
Niagara Blower Co., 1112
Torrington Mfg. Co., The, 1252-1253
Trade-Wind Motorfans, Inc., 1254
Trane Co., The, 1168-1169
Utility Appliance Corp., 1140
Western Blower Co., 1255
Westinghouse Electric Corp, Sturtevant Div., 1204, 1256

FANS, Electric
Philip Carey Mfg. Co., The, 1448-1449
Champion Blower & Forge Co., 1239
General Blower Co., 1241
Hunter Fan & Ventilating Co., Inc., 1243
Ilg Electric Ventilating Co., 1156, 1244
Propellair Div., Robbins & Myers, Inc., 1251
Torrington Mfg. Co., The, 1252-1253
Trade-Wind Motorfans, Inc., 1254
Western Engineering & Mfg. Co., Inc., 1262
L. J. Wing Mfg. Co., 1170-1172

FANS, Furnace
Aladdin Heating Corp., 1234
American Blower Corp., 1100-1101
Bishop & Babcock Mfg. Co., The, (Massachusetts Blower Div.), 1237
Champion Blower & Forge Co., 1239
DeBothezat Fans Div., American Machine & Metals, Inc., 1240
Joy Mfg. Co., 1246-1247
Meyer Furnace Co., The, 1130-1131
Morrison Products, Inc., 1249
L. J. Mueller Furnace Co., 1132-1133
United States Air Conditioning Corp, 1116
L. J. Wing Mfg. Co., 1170-1172

FANS, INDUCED DRAFT
American Blower Corp., 1100-1101
Buffalo Forge Co., 1238
Champion Blower & Forge Co., 1239
Clarage Fan Co., 1105
DeBothezat Fans Div., American Machine & Metals, Inc., 1240
General Blower Co, 1241
L. J Wing Mfg. Co., 1170-1172

FANS, Portable
Baltimore Aircoil Co., Inc., 1226
Champion Blower & Forge Co., 1239
General Blower Co., 1241
Hartzell Propeller Fan Co., Div. of Castle Hills Corp., 1242
Hunter Fan & Ventilating Co., Inc., 1243
Ilg Electric Ventilating Co., 1156, 1244
Lau Blower Co., The, 1248
Herman Nelson Div., American Air Filter Co., Inc., 1164-1165
Propellair Div., Robbins & Myers, Inc., 1251
Torrington Mfg. Co., The, 1252-1253
L. J. Wing Mfg. Co., 1170-1172

FANS, Propeller
American Blower Corp., 1100-1101
American Coolair Corp., 1235
Baltimore Aircoil Co., Inc., 1226
Bishop & Babcock Mfg. Co., The (Massachusetts Blower Div.), 1237
Buffalo Forge Co., 1238
DeBothezat Fans Div., American Machine & Metals, Inc., 1240
General Blower Co., 1241

FLANGES, Galvanized or Non-Ferrous
Simplex Mfg. Co., 1323

FLANGES, Lead, Roof
Simplex Mfg Co., 1323

FLANGES, Pipe, Alloy, Stainless Non-Ferrous
Ladish Co., 1296

FLANGES, Pipe, Blind
Grinnell Co., Inc, 1154-1155
Ladish Co., 1296
Taylor Forge & Pipe Works, Inc., 1297

FLANGES, Pipe, Reducing
Grinnell Co., Inc., 1154-1155
Ladish Co., 1296
Taylor Forge & Pipe Works, Inc., 1297
Tube Turns, Inc., 1298

FLANGES, Pipe, Steel
Grinnell Co., Inc., 1154-1155
Ladish Co., 1296
Taylor Forge & Pipe Works, Inc., 1297
Tube Turns, Inc., 1298

FLANGES, Welding
Grinnell Co., Inc., 1154-1155
Ladish Co., 1296
Taylor Forge & Pipe Works, Inc, 1297
Tube Turns, Inc., 1298

FLOATS, Ferrous and Non-Ferrous (*Seamless*)
Arthur Harris & Co., 1295

FLOATS, Welded
W. H. Nicholson & Co., 1430-1431

FLOOR PLATES
National Radiator Co., The, 1340-1341
Pacific Steel Boiler Div., U. S. Radiator Corp., 1342-1343
United States Radiator Corp., 1344-1345
United States Steel, 1292

FLUE GAS ANALYSIS
Minneapolis-Honeywell Regulator Co., 1316-1317

FOOD SAVERS
American Solvent Recovery Corp., 1186-1187
W. B. Connor Engineering Corp., 1188-1191, 1270-1271

FORCED-AIR DUCTS and FITTINGS (*See Ducts; Fittings*)

FORCED DRAFT COOLING TOWERS (*See Cooling Towers, Induced Draft, Mechanical Draft*)
Acme Industries, Inc., 1216
Air & Refrigeration Corp., 1099
Baker Refrigeration Corp., 1225
Fluor Corp., Ltd., The, 1208
Foster Wheeler Corp., 1209
Kennard Corp., 1157
Lilie-Hoffmann Cooling Towers, Inc., 1210
Marlo Coil Co., 1229
J. F. Pritchard & Co., 1212
Refrigeration Engineering, Inc., 1233
United States Air Conditioning Corp., 1116
Water-Cooling Equipment Co., 1213

FUEL BURNING EQUIPMENT, Automatic (*See Burners, Automatic; Gas Burners; Oil Burners; Stokers*)

FUEL OIL, Heating, Pumping and Straining Units
Ace Engineering Co., 1382
Bell & Gossett Co., 1396-1397
S. T. Johnson Co., 1386-1387
Ray Oil Burner Co., 1390-1391

FURNACE PIPE AND FITTINGS
Clayton & Lambert Mfg. Co., 1123
Meyer Furnace Co., The, 1130-1131
L. J Mueller Co, The, 1132-1133
United States Register Co., 1288-1289

FURNACES, Electric
Electromode Corp., 1152

FURNACES, Gas-Fired, Floor
American Radiator & Standard Sanitary Corp., 1328-1331

FURNACES, Oil Burning, Floor
Airtherm Mfg. Co., 1141
American Foundry & Furnace Co., 1120-1121
American Radiator & Standard Sanitary Corp, 1328-1331
Chicago Steel Furnace Co., 1146
Dravo Corp., 1144-1145
Lee Corp., 1147
H C. Little Burner Co, 1385
National Heater Co., 1148-1149
Arthur A. Olson & Co., 1150

FURNACES, Suspended
Chicago Steel Furnace Co., 1146

FURNACES, Wall
H. C. Little Burner Co., 1385
Rheem Mfg. Co., 1136
John Zink Co., 1380-1381

FURNACES, Warm Air, Heavy Duty
Air Devices, Inc., 1178, 1263
Airtherm Mfg. Co., 1141
Aladdin Heating Corp., 1234
American Foundry & Furnace Co., 1120-1121
E. K. Campbell Co., 1143
Campbell Heating Co., 1124-1125, 1142
Chicago Steel Furnace Co, 1146
Dravo Corp., 1144-1145
Hayes Furnace Mfg. & Supply Co., 1128
Lee Corp., 1147
H. C. Little Burner Co, 1385
Meyer Furnace Co., The, 1130-1131
L. J. Mueller Furnace Co., 1132-1133
National Heater Co., 1148-1149
Arthur A. Olson & Co., 1150
Ray Oil Burner Co., 1390-1391
Rheem Mfg. Co., 1136
Syncromatic Corp., 1129
United States Radiator Corp., 1344-1345
Waterman-Waterbury Co., The, 1138-1139
John Zink Co, 1380-1381

FURNACES, Warm Air, Residence
Airtemp Div., Chrysler Corp., 1126-1127
Aladdin Heating Corp., 1234
American Foundry & Furnace Co., 1120-1121
American Furnace Co., 1122
American Radiator & Standard Sanitary Corp., 1328-1331
Campbell Heating Co., 1124-1125, 1142
Chicago Steel Furnace Co., 1146
Electromode Corp., 1152
Fitzgibbons Boiler Co., Inc., 1360-1361
General Automatic Products Corp., 1338
General Electric Co., 1108-1109
Hayes Furnace Mfg. & Supply Co., 1128
Hoffman Specialty Co., 1420-1423
S. T. Johnson Co., 1386-1387
H. C. Little Burner Co., 1385
Meyer Furnace Co, The, 1130-1131
L. J. Mueller Furnace Co, 1132-1133
Norge Heat, Div. of Borg-Warner Corp., 1134-1135
Petroleum Heat & Power Co, 1388-1389
Ray Oil Burner Co, 1390-1391
Rheem Mfg. Co., 1136
Syncromatic Corp, 1129
United States Radiator Corp., 1344-1345
Utility Appliance Corp., 1140
Waterman-Waterbury Co., The, 1138-1139
Williams Oil-O-Matic Div, Eureka Williams Corp., 1393
John Zink Co., 1380-1381

GAGES, Altitude
Electric Auto-Lite Co., The, Instrument & Gauge Div., 1303
Jas. P. Marsh Corp., 1428-1429

GAGES, Compound
C A. Dunham Co., 1415-1419
Electric Auto-Lite Co., The, Instrument & Gauge Div., 1303
Jas P. Marsh Corp., 1428-1429

GAGES, Liquid Level
Minneapolis-Honeywell Regulator Co., 1316-1317
Rochester Mfg. Co., Inc., 1322
Taylor Instrument Cos, 1326
Yarnall-Waring Co., 1442

GAGES, Pressure
C. A. Dunham Co., 1415-1419
Electric Auto-Lite Co, The, Instrument & Gauge Div., 1303
Jas. P. Marsh Corp, 1428-1429
Minneapolis-Honeywell Regulator Co., 1316-1317
Perfex Corporation, 1320
Rochester Mfg Co., Inc., 1322
Taylor Instrument Cos., 1326

GAGES, Steam
Dole Valve Co., The, 1443
C. A. Dunham Co., 1415-1419
Electric Auto-Lite Co, The, Instrument & Gauge Div., 1303
Hoffman Specialty Co., 1420-1423
Jas. P. Marsh Corp., 1428-1429
Minneapolis-Honeywell Regulator Co., 1316-1317

GAGES, Tank
Minneapolis-Honeywell Regulator Co., 1316-1317
Rochester Mfg. Co., Inc., 1322

GAGES, Vacuum
Dole Valve Co, The, 1443
C. A. Dunham Co., 1415-1419
Electric Auto-Lite Co., The, Instrument & Gauge Div., 1303
Jas. P. Marsh Corp, 1428-1429
Minneapolis-Honeywell Regulator Co., 1316-1317

Propellair Div., Robbins & Myers, Inc., 1251
Skidmore Corp., 1408

PUMPS, Turbine
Aurora Pump Co., 1401
Chicago Pump Co., 1403
Peerless Pump Div., Food Machinery & Chemical Corp., 1405
Skidmore Corp., 1408

PUMPS, Turbine Vane Type
Peerless Pump Div., Food Machinery & Chemical Corp., 1405

PUMPS, Vacuum
Chicago Pump Co., 1403
C. A. Dunham Co., 1415-1419
Hoffman Specialty Co., 1420-1423
Joy Mfg. Co., 1246-1247
Nash Engineering Co., The, 1406-1407
Skidmore Corp., 1408
Sterling, Inc., 1325
Worthington Pump & Machinery Corp., 1118

PUMPS, Water Supply
Peerless Pump Div., Food Machinery & Chemical Corp., 1405

PURGERS, Refrigeration
Armstrong Machine Works, 1412-1413
York Corp., 1119

PURIFIERS AND SCRUBBERS, Air, Gas and Steam
V. D. Anderson Co., The, 1410-1411

PYROMETERS, Portable and Stationary
Illinois Testing Laboratories, 1311
Minneapolis-Honeywell Regulator Co., 1316-1317

RADIANT HEATING
Burnham Corp., 1335
Campbell Heating Co., 1124-1125, 1142
Crane Co., 1336-1337
Kritzer Radiant Coils, Inc., 1222
National Radiator Co., The, 1340-1341
Sarcotherm Controls, Inc., 1433
United States Radiator Corp., 1344-1345

RADIATION, Aluminum
Infra Insulation, Inc., 1482
Trane Co., The, 1168-1169
Young Radiator Co., 1173

RADIATION, Baseboard, Ferrous
American Radiator & Standard Sanitary Corp., 1328-1331
Burnham Corp., 1335
Crane Co., 1336-1337
C. A. Dunham Co., 1415-1419
Fedders-Quigan Corp., 1153
Kritzer Radiant Coils, Inc., 1222
National Radiator Co., The, 1340-1341
H. B. Smith Co., Inc., The, 1350
Trane Co., The, 1168-1169
Vulcan Radiator Co., The, 1224
Weil-McLain Co., 1351

RADIATION, Baseboard, Nonferrous
American Radiator & Standard Sanitary Corp., 1328-1331
C. A. Dunham Co., 1415-1419
Fedders-Quigan Corp., 1153
General Automatic Products Corp., 1338
Kritzer Radiant Coils, Inc., 1222
National Radiator Co., The, 1340-1341
Rome-Turney Radiator Co., The, 1223
Shaw-Perkins Mfg. Co., 1347
United States Radiator Corp., 1344-1345
Vulcan Radiator Co., The, 1224
Warren Webster & Co., 1437-1441

RADIATION, Cast-Iron
American Radiator & Standard Sanitary Corp., 1328-1331
Burnham Corp., 1335
Crane Co , 1336-1337
D. J. Murray Mfg. Co., 1163
United States Radiator Corp., 1344-1345
Weil-McLain Co., 1351

RADIATION, Copper
Airtherm Mfg. Co., 1141
C. A. Dunham Co., 1415-1419
Fedders-Quigan Corp., 1153
G & O Mfg. Co., The, 1220
Kritzer Radiant Coils, Inc., 1222
Modine Mfg. Co., 1160-1161
Rome-Turney Radiator Co., The, 1223
Shaw-Perkins Mfg. Co., 1347
Trane Co., The, 1168-1169
Young Radiator Co., 1173

RADIATION, Plain and Extended Surface
C. A. Dunham Co., 1415-1419
Fedders-Quigan Corp., 1153
G & O Mfg. Co., The, 1220
Infra Insulation, Inc., 1482
Kritzer Radiant Coils, Inc., 1222
Modine Mfg. Co., 1160-1161
Rome-Turney Radiator Co., The, 1223
Shaw-Perkins Mfg. Co., 1347
H B. Smith Co., Inc., The, 1350
Trane Co., The, 1168-1169
Vulcan Radiator Co., The, 1224
Warren Webster & Co., 1437-1441
Young Radiator Co., 1173

RADIATOR BRACKETS (*See Brackets, Radiator*)

RADIATOR ENCLOSURES AND SHIELDS
American Flange & Mfg. Co., Inc., 1480
National Radiator Co., The, 1340-1341
H. J. Somers, Inc., 1200-1201

RADIATOR HANGERS (*See Hangers, Radiator*)

RADIATOR HEAT REFLECTORS
Infra Insulation, Inc., 1482
Silvercote Products, Inc., 1484

RADIATORS, Cabinet
Airtherm Mfg. Co., 1141
American Radiator & Standard Sanitary Corp., 1328-1331
Crane Co., 1336-1337
C. A. Dunham Co., 1415-1419
Fedders-Quigan Corp., 1153
Kritzer Radiant Coils, Inc., 1222
Modine Mfg. Co., 1160-1161
National Radiator Co., The, 1340-1341
Shaw-Perkins Mfg. Co., 1347
United States Radiator Corp., 1344-1345
Young Radiator Co., 1173

RADIATORS, Concealed
Airtherm Mfg. Co., 1141
American Radiator & Standard Sanitary Corp., 1328-1331
Crane Co., 1336-1337
C. A. Dunham Co., 1415-1419
Fedders-Quigan Corp., 1153
Kritzer Radiant Coils, Inc., 1222
Modine Mfg. Co., 1160-1161
National Radiator Co., The, 1340-1341
Shaw-Perkins Mfg. Co., 1347
Warren Webster & Co., 1437-1441
Young Radiator Co., 1173

RANGES, Cooking, Hotel, Hospital, etc.
Air Devices, Inc., 1178, 1263
Ray Oil Burner Co., 1390-1391

RECEIVERS, Air
Curtis Refrigerating Machine Div., of Curtis Mfg. Co., 1227
Farrar & Trefts, Inc., 1359
International Boiler Works Co., The, 1362
Joy Mfg Co., 1246-1247
Worthington Pump & Machinery Corp., 1118

RECEIVERS, Condensation
Aurora Pump Co., 1401
Hoffman Specialty Co., 1420-1423
Illinois Engineering Co., 1424-1425
Worthington Pump & Machinery Corp., 1118

RECEIVERS, Refrigerants
Acme Industries, Inc., 1216
Worthington Pump & Machinery Corp., 1118
York Corp., 1119

RECORDERS, Humidity, Temperature
American Moistening Co., 1214
Electric Auto-Lite Co , The, Instrument & Gauge Div , 1303
Moeller Instrument Co., 1318
Powers Regulator Co., The, 1321
Taylor Instrument Cos., 1326

REFRACTORIES, Cement, Materials
Armstrong Cork Co. (Building Materials Div.), 1464
Babcock & Wilcox Co., The, 1352
Philip Carey Mfg. Co., The, 1448-1449
Johns-Manville, 1470-1471
Grant Wilson, Inc., 1452

REFRIGERATING EQUIPMENT, Centrifugal
Bell & Gossett Co., 1396-1397
Carrier Corp., 1106-1107
Trane Co., The, 1168-1169
Worthington Pump & Machinery Corp., 1118
York Corp., 1119

REFRIGERATING EQUIPMENT, Steam Jet
Ingersoll-Rand, 1404
Worthington Pump & Machinery Corp., 1118

REFRIGERATING MACHINERY
Airtemp Div., Chrysler Corp., 1126-1127
Baker Refrigeration Corp., 1225
Carrier Corp., 1106-1107
Curtis Refrigerating Machine Div., of Curtis Mfg. Co., 1227
Frick Co., 1228
Frigidaire Div., General Motors Corp., 1110
General Electric Co., 1108-1109
Niagara Blower Co., 1112
Ready-Power Co., The, 1231
Refrigeration Engineering, Inc., 1233
Trane Co., The, 1168-1169
Worthington Pump & Machinery Corp., 1118
York Corp., 1119

VACUUM REFRIGERATING SYSTEMS
Ingersoll-Rand, 1404

VALVES, Air
Crane Co., 1336-1337
Dole Valve Co., The, 1443
Hammond Brass Works, 1445
Hoffman Specialty Co., 1420-1423
Homestead Valve Mfg. Co., 1446
Jenkins Bros., 1447
Maid-O'-Mist, Inc., 1426-1427
Jas. P. Marsh Corp., 1428-1429
W. H. Nicholson & Co., 1430-1431
Ohio Brass Co., 1432

VALVES, Angle, Globe and Cross
Crane Co., 1336-1337
Fairbanks Co., The, 1444
Frick Co., 1228
Grinnell Co., Inc., 1154-1155
Hammond Brass Works, 1445
Henry Valve Co., 1309
Jenkins Bros., 1447
Ohio Brass Co , 1432

VALVES, Automatic
Fulton Sylphon Div., The, Robertshaw-Fulton Controls Co., 1304-1305
General Controls, 1306-1307
Hubbell Corp., The, 1310
Jas. P. Marsh Corp., 1428-1429
McDonnell & Miller, Inc., 1372-1375
Minneapolis-Honeywell Regulator Co., 1316-1317
Sarcotherm Controls, Inc., 1433
Sterling, Inc., 1325

VALVES, Back Pressure
Alco Valve Co., 1299
General Controls, 1306-1307
Hubbell Corp., The, 1310
Illinois Engineering Co., 1424-1425

VALVES, Balanced
General Controls, 1306-1307
Illinois Engineering Co., 1424-1425
Maid-O'-Mist, Inc., 1426-1427
McDonnell & Miller, Inc., 1372-1375
Minneapolis-Honeywell Regulator Co., 1316-1317

VALVES, Blowoff
Fairbanks Co , The, 1444
Homestead Valve Mfg. Co., 1446
Jenkins Bros., 1447
McDonnell & Miller, Inc., 1372-1375
Yarnall-Waring Co., 1442

VALVES, By-Pass
General Controls, 1306-1307
Jenkins Bros., 1447

VALVES, Check
Fairbanks Co., The, 1444
Grinnell Co., Inc., 1154-1155
Hammond Brass Works, 1445
Henry Valve Co., 1309
Hubbell Corp., The, 1310
Jenkins Bros., 1447
Ohio Brass Co., 1432

VALVES, Corrosion Resisting
Crane Co., 1336-1337
Homestead Valve Mfg. Co., 1446
Jenkins Bros., 1447

VALVES, Diaphragm
Crane Co., 1336-1337
General Controls, 1306-1307
Grinnell Co., Inc., 1154-1155
Henry Valve Co., 1309
Hubbell Corp., The, 1310
Johnson Service Co., 1312-1313

Powers Regulator Co., The, 1321
Taylor Instrument Cos., 1326

VALVES, Exhaust
Maid-O'-Mist, Inc., 1426-1427

VALVES, Expansion
Alco Valve Co., 1299
Crane Co., 1336-1337
General Controls, 1312-1313
Henry Valve Co., 1309

VALVES, Float
Alco Valve Co., 1299
Hubbell Corp , The, 1310
Illinois Engineering Co , 1424-1425
Maid-O'-Mist, Inc., 1426-1427
McDonnell & Miller, Inc , 1372-1375

VALVES, Flow Control
Bell & Gossett Co., 1396-1397
Crane Co., 1336-1337
General Controls, 1312-1313
Homestead Valve Mfg Co., 1446
Illinois Engineering Co , 1424-1425
Jas. P. Marsh Corp., 1428-1429
McDonnell & Miller, Inc., 1372-1375
Ohio Brass Co , 1432
Taco Heaters, Inc , 1400
H A Thrush & Co., 1398-1399

VALVES, Gas
Hammond Brass Works, 1445
Homestead Valve Mfg. Co., 1446
Jenkins Bros , 1447
Ohio Brass Co., 1432
Penn Controls, Inc., 1319
Perfex Corp., 1320

VALVES, Gate
Crane Co., 1336-1337
Fairbanks Co., The, 1444
General Controls, 1306-1307
Hammond Brass Works, 1445
Jenkins Bros, 1447
Ohio Brass Co., 1432

VALVES, Humidifier
Maid-O'-Mist, Inc., 1426-1427

VALVES, Hydraulic
Homestead Valve Mfg. Co., 1446
Jenkins Bros , 1447
W H. Nicholson & Co., 1430-1431
Yarnall-Waring Co., 1442

VALVES, Magnetic
Alco Valve Co., 1299
General Controls, 1306-1307
Hook & Ackerman, Inc , 1339
Hubbell Corp., The, 1310
McDonnell & Miller, Inc., 1372-1375

VALVES, Mixing, Thermostatic
Fulton Sylphon Div., The, Robertshaw-Fulton Controls Co., 1304-1305
Powers Regulator Co., The, 1321
Sarco Co., Inc., 1434-1435

VALVES, Motor Operated
Barber-Colman Co., 1269
Bell & Gossett Co., 1396-1397
General Controls, 1306-1307
Illinois Engineering Co., 1424-1425
Johnson Service Co., 1312-1313
W H. Nicholson & Co., 1430-1431
Sarco Co., Inc., 1434-1435
Taylor Instrument Cos., 1326
Warren Webster & Co., 1437-1441

VALVES, Non-Return
Fairbanks Co., The, 1444
Hammond Brass Works, 1445
Illinois Engineering Co., 1424-1425
Jenkins Bros , 1447
Ohio Brass Co., 1432

VALVES, Packless
C. A. Dunham Co., 1415-1419
Fulton Sylphon Div., The, Robertshaw-Fulton Controls Co., 1304-1305
General Controls, 1306-1307
Henry Valve Co., 1309
Hoffman Specialty Co., 1420-1423
Illinois Engineering Co., 1424-1425
Jas. P. Marsh Corp., 1428-1429
Minneapolis-Honeywell Regulator Co., 1316-1317
Ohio Brass Co., 1432
Powers Regulator Co., The, 1321
Sarco Co., Inc., 1434-1435

VALVES, Pressure Reducing
(*See Regulators, Pressure*)

VALVES, Radiant Heating
Homestead Valve Mfg Co., 1446

VALVES, Radiator
American Radiator & Standard Sanitary Corp., 1328-1331
Barnes & Jones, Inc , 1409
Crane Co., 1336-1337
C. A. Dunham Co , 1415-1419
Fairbanks Co , The, 1444
Fulton Sylphon Div., The, Robertshaw-Fulton Controls Co., 1304-1305
Grinnell Co , Inc , 1154-1155
Hammond Brass Works, 1445
Hoffman Specialty Co , 1420-1423
Illinois Engineering Co., 1424-1425
Jenkins Bros., 1447
Maid-O'-Mist, Inc , 1426-1427
Jas. P. Marsh Corp , 1428-1429
Minneapolis-Honeywell Regulator Co , 1316-1317
Ohio Brass Co , 1432
Sarco Co , Inc., 1336-1337
Sterling, Inc., 1325
Taco Heaters, Inc., 1400
Trane Co., The, 1168-1169

VALVES, Radiator, Humidifying
Maid-O'-Mist, Inc , 1426-1427

VALVES, Radiator Orifice
Barnes & Jones, Inc , 1409
C A. Dunham Co , 1415-1419
Grinnell Co., Inc , 1154-1155
Hammond Brass Works, 1445
Illinois Engineering Co , 1424-1425
Minneapolis Honeywell Regulator Co., 1316-1317
Ohio Brass Co., 1432
Sarco Co., Inc., 1434-1435
Warren Webster & Co , 1437-1441

VALVES, Radiator, Pneumatic Diaphragm
Johnson Service Co , 1312-1313
Minneapolis-Honeywell Regulator Co , 1316-1317
Powers Regulator Co., The, 1321

VALVES, Reducing
Bell & Gossett Co., 1396-1397
Crane Co., 1336-1337
C A Dunham Co , 1415-1419
Fulton Sylphon Div., The, Robertshaw-Fulton Controls Co , 1304-1305
Illinois Engineering Co., 1424-1425
Strong, Carlisle & Hammond Co., 1436
Taco Heaters, Inc., 1400
Taylor Instrument Cos., 1326
H. A. Thrush & Co., 1398-1399

VALVES, Refrigerant Line
Alco Valve Co , 1299
Baker Refrigeration Corp., 1225

MANUFACTURERS' CATALOG DATA

(PAGES 1099–1496)

On pages 1099-1496 will be found the Catalog Data of 277 manufacturers whose products are described and illustrated.

For the convenience of the user of THE GUIDE 1952 there are eight main divisions:

On pages 1073-1096, under each of the index headings—Air Cleaning Equipment, Fans, Humidifiers, Ventilators, etc., will be found a list of manufacturers of any desired products, fully cross-indexed, and the page numbers in the Catalog Data Section where the products are described.

By reference to these indices, the manufacturers' names and the page numbers, any item of equipment or materials, and the producers address, may be located quickly.

Air & Refrigeration Corporation

475 Fifth Avenue, **New York 17, N. Y.**

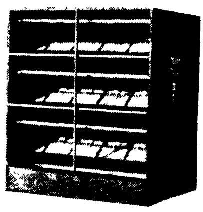
271 Milton Avenue, S. E., Atlanta, GA.

Representatives in all major cities

Air Conditioning, Humidifying, Dehumidifying, Cooling, Scrubbing, Air Washing and Purification Apparatus

Air & Refrigeration Corporation specializes in the design and manufacture of industrial and comfort-conditioning apparatus where maintenance of suitable humidity and temperature within closely controllable limits is essential. This specialization is based on technical knowledge and ingenuity born of extensive experience in the solution of the more difficult problems of air conditioning. A complete line of air conditioning equipment is available to contractors and owners for all phases of humidifying, dehumidifying, cooling and washing.

* **Capillary Air Washers** provide a superior type humidifying, dehumidifying, air washing, cleaning and cooling unit for central station apparatus. For most purposes the Capillary Washer requires 1/3 the volume of water at 1/3 the pressure used by conventional spray equipment. They are available with factory insulated casings and tank for central station applications. *For complete data, see Capillary Bulletin.*



Factory Insulated Class I Capillary Air Washer

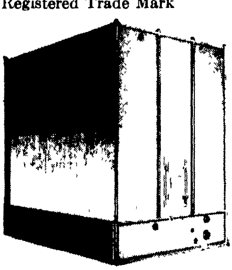
* **Capillary Unit Conditioners** are factory insulated and assembled, ready for use. They include fan, motor, drive, heating coils, Capillary Cells with suitable sprays, spray pump and mixing dampers. Units are designed for floor mounting or for ceiling suspension, and can be arranged for the reception of cooling coils, if required. *Complete description and engineering data will be found in Capillary Unit Bulletin.*

Spray Type Air Washers for washing, humidifying and dehumidifying air are all basically the same. A & R Spray Washers include special features of design developed to insure more efficient and dependable operation, lower maintenance costs, and, in many cases, lower installation costs. Such features relate especially to eliminators, collecting tanks, flooded baffles, nozzle arrangement, etc. Spray Washers can be supplied with factory insulated casings and tank for central stations applications. *For details, see Air Washer Bulletin.*

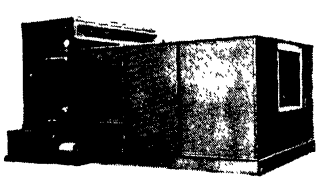
Sprayed Coil Dehumidifiers for year-round treatment of air are complete with cooling coils, sprays, circulating pump and glass mat eliminators. Sprayed coil dehumidifiers are factory insulated and complete, ready for assembly in the field. Special features in design insure continuous washing and cleaning of finned surfaces and easy accessibility to all parts. *For engineering information and detailed description, see Sprayed Coil Bulletin.*

A & R Insulated Panels consist of insulation between metal sheet on one side and hard fiber board on the other, the three laminated and cemented together. This unique panel design includes the structural frame to form units which require *only bolting together* to make enclosures of any required shape for plenum chambers and many other purposes. Panels are available in widths from 3 in. to 48 in., and in lengths to 12 ft. Their use insures tremendous economies in field labor. *For details, see Panel Bulletin. Write for catalog and engineering data.*

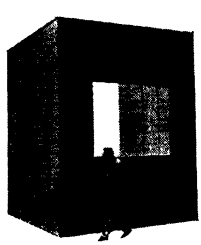
*Registered Trade Mark



Type 2S Air Washer Factory Insulated



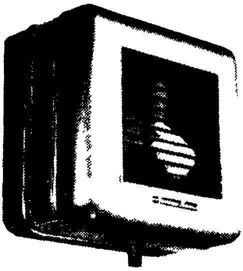
Factory Insulated Size #S-4 Capillary Unit Conditioner



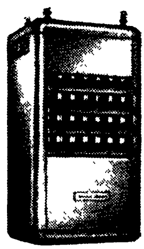
Factory Insulated Plenum Chamber

TYPES OF AMERICAN BLOWER CORPORATION AIR HANDLING AND CONDITIONING EQUIPMENT

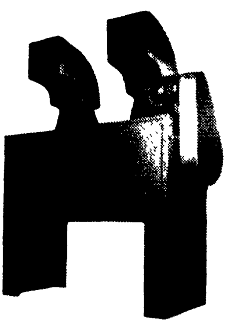
All types of air handling and air conditioning equipment for industrial applications, process work, drying, cooling; also equipment for stores, offices, shops, public buildings, power plants, etc., and attic and kitchen ventilation for homes.



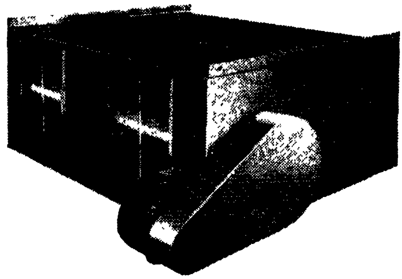
Unit Heaters—for many general purpose heating jobs. Wall or ceiling mounted. Streamline construction, rugged heating elements. Steam or hot water. Request Bulletin 6717.



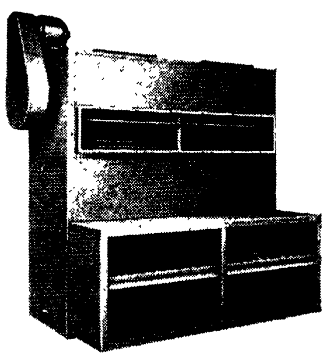
Gas Fired Unit Heater-self contained—available in 7 sizes for clean, automatic, instantaneous heating. Adjustable louvers assure efficient heat distribution throughout the work area. *A. G. A.* approved. Write for Bulletin 7117.



Centrifugal Type Unit Heaters—for use with or without duct systems, for large hard to heat areas. Ideal for floor mounting. Request Bulletin 5917.



Heating and Ventilating Units—with air filters and Aileron control. Ideal wherever attractive, quiet and economical heating and ventilation units are required. Wall, floor or ceiling mounting. Offer great flexibility of design and arrangement to meet specific needs. Bulletin 6017.



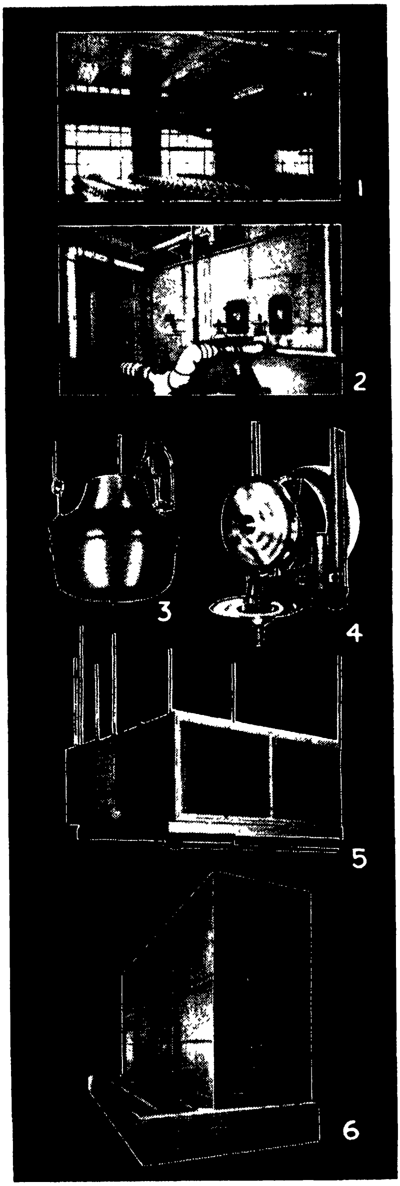
American Blower Air Conditioning Units. —*Type A* for all normal unitary type commercial and industrial applications. Cooling, heating, humidifying. Capacities 1000 cfm—13600 cfm. *Type S* for commercial and industrial applications desiring washed air or high relative humidities. Capacities 1000 cfm—13600 cfm. *Type M*, large capacity for central system installation with separately mounted fan. Cooling, dehumidification, heating, humidifying. Capacities 1000 cfm—41000 cfm. Bulletin 6527.

Bahnson Company

ENGINEERS AND MANUFACTURERS

WINSTON-SALEM, N. C.

HUMIDIFYING — VENTILATING — COOLING — FILTERING



COMPLETE SYSTEMS

1. **HUMIDUCT**—A unit system for industrial air conditioning—designed, manufactured, and installed by Bahnson. Systems provide ventilating, heating, cooling, humidifying, dehumidifying and filtering in any desired combination.

2. **CENTRAL STATION**—Systems for industrial and commercial applications —designed, manufactured and installed by Bahnson. Type Y or Centrispray Air Washers along with Bahnson manufactured components provide complete temperature and humidity control.

UNIT HUMIDIFIERS

3. **TYPE E HUMIDIFIER**—A self-contained unit for smaller commercial and industrial rooms. Evaporates up to 7 gph with excellent distribution and low sound level. Installed singly or in groups with individual or group automatic control. Requires only water supply and electrical connection.

4. **CENTRIFUGAL HUMIDIFIER**—A self-contained unit for industrial humidification. Evaporation up to 12 gph plus directional air flow with fractional horsepower motor. Installed singly or in groups with individual or group automatic control. Requires only water supply, drain, and electrical connection.

5. **TYPE BA-2 HUMIDIFIER**—A high-capacity unit for overhead suspension in commercial and industrial applications. Adjustable grilles give complete directional control of moist air delivered. Automatically regulated, the unit provides air flow, humidification and filtering. Requires only water supply, drain, and electrical connection.

SYSTEM COMPONENTS

6. **AIR WASHERS**—Bahnson Type Y Air Washers cleanse, cool and provide humidity control. Available in sizes from 6000 cfm up, they can be provided with spray systems best suited for particular applications. Bahnson FLEXI-DUR spray nozzles are used and automatic suction strainers are available.

Write for Further Information on the Equipment Shown Above.

Bahnson Company

ENGINEERS AND MANUFACTURERS

WINSTON-SALEM, N. C.

HEATING — DEHUMIDIFYING — AIR CLEANING — VAPOR ABSORBTION

SYSTEM COMPONENTS

7. TYPE S AIR CLEANER—An automatic self-cleaning filter for systems handling air which contains heavy concentrations of lint. Eliminates routine cleaning of recirculated air screens and requires no replacement of filter media.

8. CENTRISPRAY EVAPORATOR—A combined fan and evaporator unit available in capacities up to 40,000 cfm. Provides high humidification and air handling capacity with the inherent economy of the Bahnson centrifugal atomization principle.

9. GRILLES — Bahnson wide-blade grilles provide complete directional control of air flow. Special features include a removable core to facilitate cleaning as well as to provide access to the distribution duct.

10. MOTORIZED PSYCHROMETER—A psychrometer for accurate and rapid reading of wet and dry bulb temperatures—complete with humidity calculator and record pad holder.

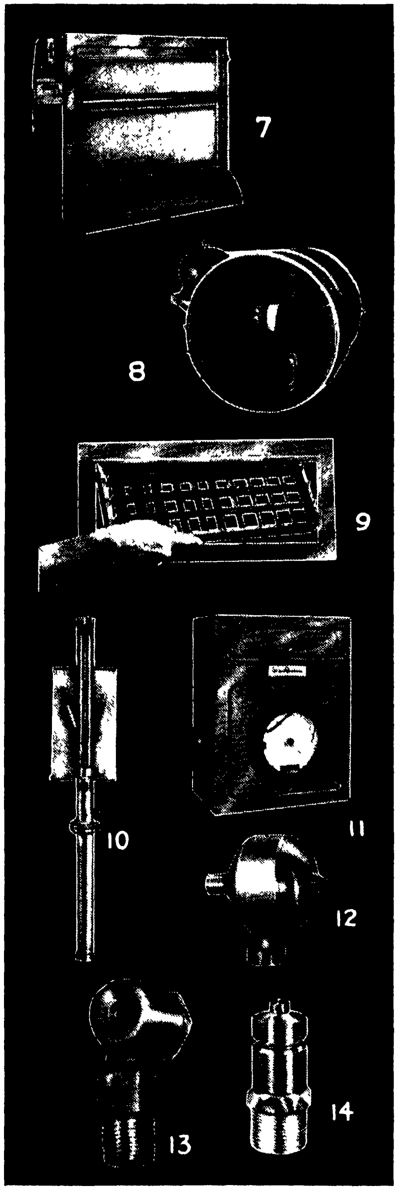
11. AUTOMATIC CONTROLS—Bahnson pneumatic or electric humidity and temperature controls are sensitive, accurate and dependable. Available in aspirated cabinets combined with recorders.

12. TYPE ESC ATOMIZER—A self-cleaning pneumatic atomizer of superior design for direct humidification or supplementary evaporation. Uses both air and water under pressure to provide high evaporation at low power cost. Modulating control regulates evaporation in direct proportion to the room requirements.

13. FLEXIDUR SPRAY NOZZLE—A brass spray nozzle with orifice stamped from stainless steel to eliminate erosion, prevent accumulation of foreign matter, and assure uniformity of spray pattern.

14. TYPE DC SPRAY NOZZLE—The Bahnson Type DC Spray Nozzle utilizes the principle of impingement of water jets to produce a fine uniform spray. This can result in a saving in pumping cost of 50 per cent over conventional spray nozzle systems.

Write for Further Information on the Equipment Shown Above.



BUENSOD-STACEY
INCORPORATED

221 Watson Building
Greensboro, N. C.

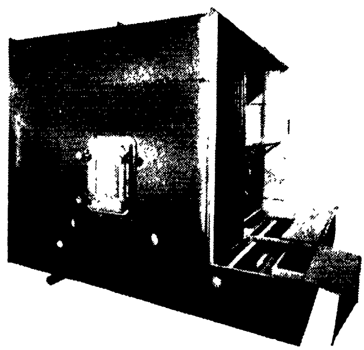
60 East 42 St.
New York 17, N. Y.

1001 N. Church St.
Charlotte 1, N. C.

Air Conditioning, Humidifying, Cooling, Ventilation Systems

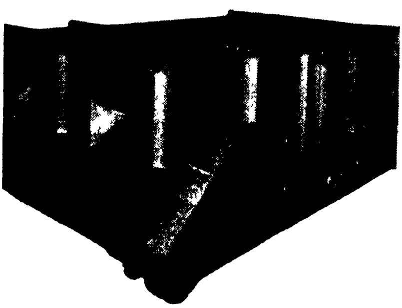
BUENSOD-STACEY AIR CONDITIONING SYSTEMS are based on long experience and great technical knowledge. Where close control of temperature and relative humidity is essential, the Buensod-Stacey engineering and contracting organization designs and installs systems to meet the most exacting requirements with the greatest economy. To facilitate the installation of systems for air conditioning, refrigeration, ventilation, etc. Buensod-Stacey maintains two factories, one in Charlotte, N. C. and the other in Long Island City, N. Y., with complete sheet metal and pipe shops in each. Buensod-Stacey also manufactures many components of the complete system, a few of which are listed below:

BUENSOD-STACEY HUMIDIFIERS AND DEHUMIDIFIERS incorporate many unique features to increase efficiency and decrease maintenance. Factory insulation reduces field labor expense. Special eliminators and spray systems insure proper performance for each application. All air washers are treated with protective coatings to reduce corrosive action.



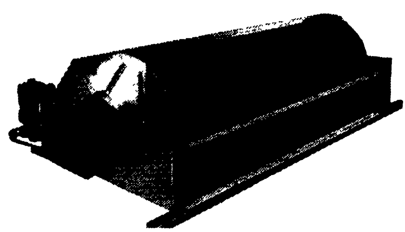
Buensod-Stacey Special Dehumidifier

BUENSOD-STACEY EVAPORATIVE COOLING UNITS have all the advantages of central station systems but are so designed that they can be suspended from the ceiling and, thus, save valuable floor space. They can be used to supplement and enlarge existing systems without disturbing their continuous operation.



Buensod-Stacey Evaporative Cooling Unit

BUENSOD - STACEY ROTARY STRAINERS save maintenance dollars. Constructed of stainless steel, they are especially useful where the air contains large quantities of solids such as lint in textile mills.



Buensod-Stacey Rotary Strainer

BUENSOD-STACEY DUAL DUCT SYSTEM* for air conditioning multi-story and multi-room buildings is a practical and economical way to obtain individually controlled temperatures in each enclosed space of such structures as office buildings, laboratories, apartment houses and hotels. This system, which is a simple combination of high pressure, high velocity, cold and warm air ducts, distributes the air at desired temperature through Buensod-Stacey Vertical and Horizontal Air Distributing and Mixing Units. It does not require widely scattered coils, fans or other mechanical apparatus.

***Protected by patent applications.**

Clarage Fan Company

Kalamazoo, Michigan

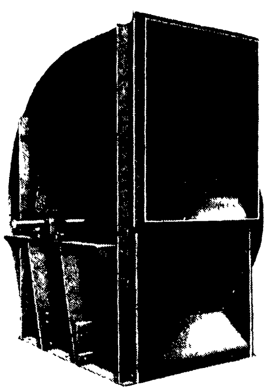
Application Engineering Offices **In Principal American Cities**

(Consult Telephone Directory)

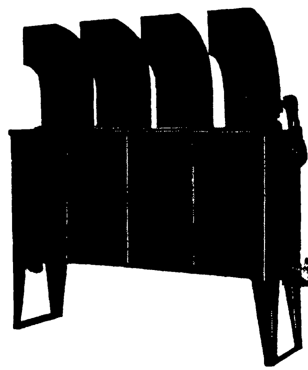
Clarage Air Handling and Conditioning Equipment

FANS AND BLOWERS
AIR WASHERS
CONDITIONING UNITS
UNIT HEATERS

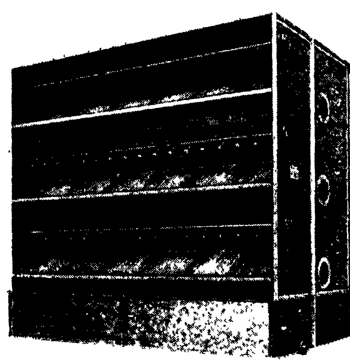
For over 35 years Clarage has been a leading manufacturer of equipment and units for ventilating, exhausting, cooling, air cleaning, humidifying and complete conditioning. Clarage equipment is designed to meet all types of industrial, commercial and public building requirements. Only a few examples of the complete Clarage line are shown on this page.



Clarage Fans are offered in many different standard types and arrangements. Sizes range from 200 to 200,000 cfm. Equipment is designed for slow speed (quiet) and high speed operation, either V-belt or direct motor driven. Most sizes quickly adjustable for any of 8 directions of air discharge. Special fans constructed for special industrial applications can be furnished.



Clarage Unit Heaters are available for either floor or suspended installation. 23 sizes range from 24,000 to 1,160,000 Btu. They can be operated on either steam or hot water.



Clarage Air Washers are built in both capillary and spray types. Sizes range from 2000 to 193,000 cfm. Used for air cleaning, humidifying and dehumidifying purposes.

We welcome your inquiry on any air handling or conditioning problem. Contact our nearest branch office, or write us at Kalamazoo, Michigan.

Carrier Corporation • Syracuse 1, N. Y.

MARINE DIVISION:
385 Madison Ave.
New York 17, N. Y.

Carrier

INTERNATIONAL DIVISION:
385 Madison Ave.
New York 17, N. Y.

Offices and Dealers in principal cities—refer to your telephone directory.

Carrier AIR CONDITIONING

Room Air Conditioners—for individual rooms and offices in compact modern styling and handsome finish to blend with finest of interior furnishings. Window sill models in $\frac{1}{3}$, $\frac{1}{2}$, $\frac{3}{4}$ and 1 hp capacity. Console models in 1 and 1 $\frac{1}{2}$ hp capacity.

Weathermakers—completely self-contained air conditioners for commercial and industrial applications. Six sizes from 3 to 20 hp cooling capacity.

Year-Round Weathermakers—for heating and cooling in residential and commercial applications. Compact, efficient gas heating combined with summer cooling. Four combination sizes.

Zoning Weathermakers—blow-through fan-coil units for air conditioning systems using remote sources of heat and refrigeration. In six sizes for comfort and industrial applications.

System Weathermakers—fan-coil unitary air conditioners for application to central cooling and heating systems for comfort and industrial use. Five sizes in vertical and horizontal models.

Central Station Air Conditioners—for application with fan and duct systems for large spaces such as stores, theatres, auditoriums and industrial plants.

Weathermaster Systems—for air conditioning of multi-story, multi-room buildings such as hotels, hospitals, office and apartment buildings. System consists of room units with individual temperature control, and central station air conditioning. Conditioned air distributed by high velocity conduits to individual Weathermasters.

Blast Freezers and Cold Diffusers—for food freezing and storage, meat packing operations, and other industries requiring low temperatures. Units available in suspension or floor models, for use within space to be refrigerated or remotely located and connected by ducts.

Write for descriptive literature.

Carrier REFRIGERATION

Centrifugal Refrigerating Machines—for large comfort and industrial air conditioning applications and for process cooling. High efficiency at peak or partial loads. Operate with any standard motor or turbine drive. Available in capacities from 100 to 2500 tons.

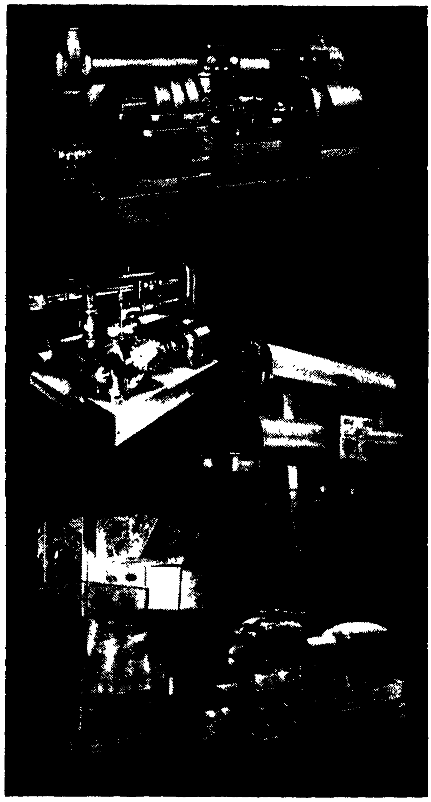
Absorption Refrigerating Machines—for producing chilled water at 36 F or higher in fully automatic operation from 10 to 100 per cent capacity. Uses high or low pressure steam for operation. Capacities 115, 150, 200, 270 and 350 tons.

Reciprocating Refrigerating Machines—for comfort and industrial air conditioning and for process cooling. Direct or belt drive, water or evaporative cooled types from 5 to 100 horsepower.

Commercial Refrigerating Machines—for storage refrigerators, display cases, walk-in coolers and similar duty. Complete with compressor, drive, air or water cooled condenser and controls. All sizes.

Reciprocating Compressors—for refrigeration needs of air conditioning and process cooling. Adaptable to all drives and available for "Freon" and ammonia refrigerants. Sizes from 75 to 200 hp.

Evaporative Condensers—for use with refrigerating compressors in place of water cooled condensers. Simplify water supply and disposal problems. For indoor or outdoor use. Capacities 10 to 85 tons.

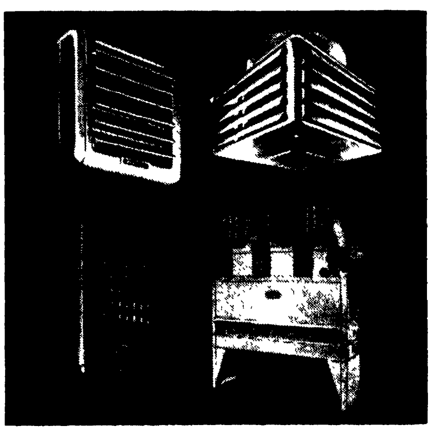


Carrier INDUSTRIAL HEATING

Unit Heaters—for commercial and industrial space heating using steam or hot water. Complete range of sizes in two types: Model 46U Horizontal Discharge in capacities from 21,000 to 200,000 Btu per hour, and Model 46S Four-way Directed-flo in capacities from 82,000 to 500,000 Btu per hour at 2 lb steam.

Gas-Fired Unit Heaters—for clean, economical heat in offices, stores, factories and similar spaces where gas is available. Approved for manufactured, mixed, natural and LP gases. Capacities: 70,000 to 230,000 Btu per hour.

Heat Diffusers—for ventilating as well as heating large commercial and industrial spaces. Floor, wall or ceiling mounted with coils for steam or hot water. Capacities 115,000 to 1,720,000 Btu per hour at 2 lb steam.

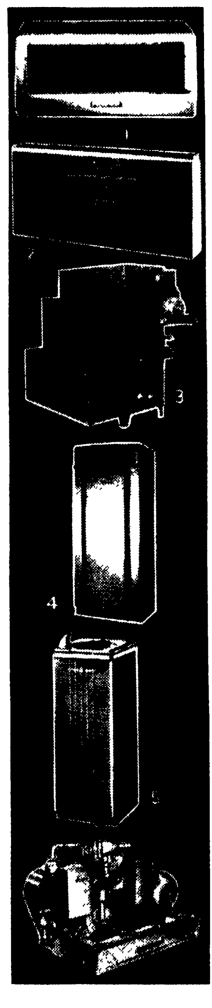


GENERAL ELECTRIC

HOME HEATING and YEAR 'ROUND AIR CONDITIONING
Air Conditioning Division, Bloomfield, New Jersey

REGIONAL OFFICES:
New York 22, N. Y.—570 Lexington Ave.
Chicago 54, Ill.—Merchandise Mart, Room 1144
New Orleans 12, La.—511 International Trade Mart
San Francisco 6, Cal.—235 Montgomery St.

See your classified telephone directory for local sources under:
G-E Home Heating Equipment—G-E Air Conditioning
G-E Refrigeration Machines—G-E Water Coolers
For information on equipment for large commercial or industrial installations·
Call or write Manager, Direct Sales, Air Conditioning Division, Bloomfield, N. J.



1. G-E ROOM AIR CONDITIONERS (WINDOW MOUNTED)

For offices, homes, apartments, hotels, and wherever small space cooling is required. Easily installed and serviced. Requires no plumbing—merely an electrical connection. Fits most standard windows. Performs same function as larger air conditioning units. Two handsome models—½ and ¾ hp.

2. G-E PERSONAL WEATHER CONTROL AIR CONDITIONING SYSTEMS

For office buildings, hotels, hospitals, and other multi-room structures. Room air conditioner (remote type) controlled by each tenant. No ductwork required. Quiet—refrigeration machines installed elsewhere. Compact, attractive cabinets—only 26 in. high, 9 in. deep—enclose fans, filters, coils, and controls. Three sizes: ¾ to 1½ tons cooling, corresponding heating capacities.

3. G-E CENTRAL PLANT AIR CONDITIONERS

For large space and multi-room building cooling and heating. G-E sectional design permits 12 vertical and 8 horizontal arrangements. Saves engineering and installation time, saves valuable space. Cooling, heating, year 'round air conditioning. Five sizes: 1,000 to 12,800 cfm.

4. G-E PACKAGED AIR CONDITIONERS

For industrial areas, office suites, stores, etc. Units cool, dehumidify, filter, ventilate, and circulate, (heat, if desired). Fast installation, economical operation. Muggy Weather Control on 3, 5, and 7½ hp sizes reduces clamminess without chilling. Five self-contained models: 2 to 10 hp.

5. G-E WATER COOLERS

For commercial and industrial installations. Sanitary top; angle-stream, non-squirt bubbler; automatic stream regulator; sturdy, attractive cabinet; foot pedal control; sealed refrigeration system; economical pre-cooler. Three air-cooled pressure type units—4, 7, and 10 gph; 10 gph water-cooled pressure type unit, and 3 gph bottle type cooler.

6. G–E CONDENSING UNITS AND COMPRESSOR UNITS

For air conditioning, process cooling, refrigeration; for self-contained or remote application. Open type units from ¼ to 60 hp; sealed units from ⅙ to ½ hp.

G-E HOME HEATING EQUIPMENT

7. G-E OIL FIRED BOILERS

Compact, integral unit. Outstandingly economical. High heat transfer rate and low water content mean fast heat. Built-in tankless-type hot water coil available. Five models: 100,000 to 450,000 Btu's per hour. Listed by *Underwriters' Laboratories, Inc.* Constructed in accordance with *ASME* Code.

8. G-E OIL FIRED WARM AIR FURNACES

Quick heat due to rugged steel, finned heat transfer surface. G-E engineered to retard chimney losses during shutdown. Four models:—60,000 to 155,000 Btu's per hour. Listed by *Underwriters' Laboratories, Inc.* Ideal for use in G-E Air-Wall® or conventional warm air systems. *For year 'round air conditioning, use in parallel with G-E Residential Air Conditioner.*

9. G-E GAS FIRED BOILERS

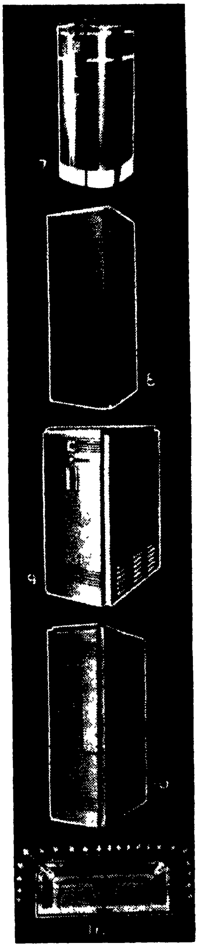
G-E specially designed, water filled, diamond-shaped projections on cast iron boiler sections and unique zigzag flue gas travel—to assure rapid heat transfer and maximum heat absorption. Clean, quiet, complete combustion because of raised port atmospheric burners. Seven models—96,000 to 432,000 Btu's per hour. Tested and approved by the *A.G.A.* Listed by *Underwriters' Laboratories, Inc.*, and constructed in accordance with *ASME* Code.

10. G-E VERTICAL GAS FIRED WARM AIR FURNACE

Rugged cast iron section with pin-point extended surfaces result in high heat transfer. Automatic controls for dependable operation. Burns natural, manufactured, or mixed gas. Five models: 60,000 to 210,000 Btu's hr input. Tested and approved by *A.G.A.* Listed by *U.L.* Ideal for use in G-E Air Wall® or conventional warm air systems. *For year 'round air conditioning, use in parallel with G-E Residential Air Conditioner.*

11. G-E *AIR-WALL* HEATING SYSTEM

The amazing new system of distributing heat from standard G-E Gas or Oil Fired Furnaces. Two-types of heat in one—*forced warm air* plus *radiant heat.* G-E standard *Air-Wall* Register directs heat in a fan-like pattern up and out in front of the normally cold walls of a room, warming them so that they actually radiate heat. G-E Automatic *Air-Wall* Register offers all these benefits plus *automatic individual room temperature control.* Thermostat element in automatic register modulates damper to admit warm air required to each room.



NEW PRODUCTS

G-E Horizontal Gas Furnace is designed for installation in attic, crawl space, basement, utility room. It may be suspended from rafters or floor joists to save floor space. This space-saver unit has all the famous G-E gas furnace features described above including the long-life, fast-acting pin-point heat transfer sections. Two models with 90,000 and 120,000 Btu's hr input. Tested and approved by *A.G.A.* Listed by *U.L.* Ideal for use in G-E Air-Wall® or conventional warm air systems.

G-E Kitchen White Oil Boiler is a compact, fully integrated unit, combining an oil-fired boiler, a built-in expansion tank, and a tankless-type hot water coil. Designed for radiant panel and forced hot water. Attractive white jacket stays comfortably cool. Oil burner listed by *Underwriters' Laboratories, Inc.*

G-E Year 'Round Air Conditioning System combines the new G-E Residential Air Conditioner with the G-E Oil or Gas Fired Warm Air Furnaces. With these units connected in parallel, this system heats, cools, filters, humidifies, dehumidifies, ventilates, and circulates the air. The G-E Residential Air Conditioner can often be installed in existing warm air heating systems with only minor changes in ductwork.

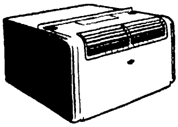
Frigidaire

Division of General Motors

Dayton 1, Ohio

Room Air Conditioners—Self-Contained Air Conditioners—Central System Air Conditioners

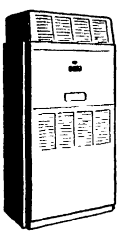
More railroad cars are air conditioned by Frigidaire



Room Air Conditioner

Room Air Conditioners

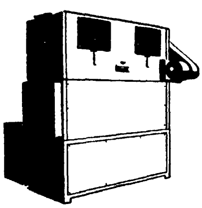
Compact, easily installed window units, providing all five summer air conditioning functions for rooms up to 500 sq ft. 4-way cool air outlet, fresh air control, exhaust air control, moisture disposal. All-steel, heavy-gage welded cabinet, bonderized. Thermostatic control optional. Quiet operation. Powered by sealed-in Meter-Miser, with 5-Year Warranty. ½ and 1 hp self-contained units.



Self-Contained Air Conditioner

Self-Contained Air Conditioners

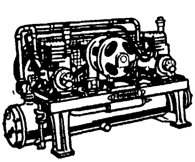
3, 5, 7½, and 10 ton capacity self-contained units. Quickly installed, fast, low-cost cooling. Little or no duct work needed. Use singly, or in multiple for a wide range of applications. Welded steel construction. Centrifugal, double-width fans. Thermostatically controlled. Water-cooled compressor, rubber-mounted for vibrationless operation. Uses safe Freon-12. Heating coil optional. Full *Underwriters'* approval.



Central System Air Conditioner

Central System Air Conditioners

Both horizontal and vertical central system air conditioning models, in capacities, types and sizes to meet practically any air conditioning need. 5 to 40-ton capacities. Heating coils and humidifier assembly optional. Quiet, vibrationless operation. Ball-bearing, oil-less type rubber-mounted motors. Aileron control adjusts air discharge. Frigidaire Multipath Cooling Units.



Heavy Duty Compressor

Heavy-Duty Compressors

Water-cooled compressors, including 7½ to 25 hp with step-control for regulating capacity. Designed for properly balanced operation with Frigidaire cooling units and controls. Precision manufacture reduces friction, drag, resistance to a minimum. Important features, such as long-wearing pistons, with floating piston pins, sealed crankcases, one-piece bodies, continuous lubrication.

Evaporative Condensers

10 to 60 ton capacities for use with evaporative type compressors. All steel reinforced cabinet, all surfaces fully protected from corrosion. Prime surface condenser coil, easily accessible. Low pressure, non-clogging spray nozzles.

Frigidaire reserves the right to change specifications, or discontinue models without notice.

Hastings Air Conditioning Co., Inc.

Hastings, Nebr.

Manufacturers of

Cooling, Heating and Ventilating Equipment.

Gas and Steam Unit Heaters.

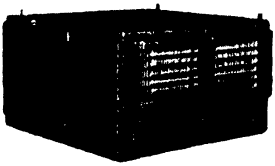
Water, DX and Steam Coils.

Dealers and Representatives in Principal Cities

A Complete Line of Air Conditioners For DX, Cold City Water, Chilled or Well Water Operation.

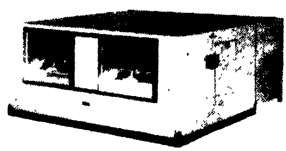
Coils are designed for utmost efficiency and constructed of copper tubing expanded and metallically bonded to pure copper fins.

FLOWMETERS (to visually indicate water-flow) are standard on all water equipment.



General Utility Models

Floor Models



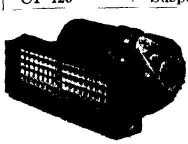
Central Plants

UTILITY MODELS: One or more units will handle any size job with or without ductwork. Attractively finished, these units are ideal for installation within the conditioned space. Four-way, stainless steel louver grilles are standard.

CENTRAL PLANTS: Sectional construction for ease of handling. Motors mounted inside to provide neat, compact units. Angle-iron base frame for suspension is standard.

SPECIFICATIONS

Model	Type	CFM	Capacity—tons: Water 6 Row Coil	Capacity—tons: DX 4 Row Coil	Capacity—tons: DX 6 Row Coil	Motor HP	Filters	Dimensions: Height	Dimensions: Width	Dimensions: Depth
Zephyr	Suspended	590	1-2	—	—	⅙	1	25"	24"	29"
Royal	Floor model	590	1-2	—	—	⅙	1	40	28	21
Majestic	Suspended	1,120	1½-3	2½	—	¼	2	26	28	38
Master	Suspended	2,240	3-6	5	—	½	4	27	46	48
Floormaster	Floor model	2,240	3-6	5	—	½	3	93	48	25
CP 30	Suspended	3,000	4-9	7½	10	1	5	30	51	60
CP 40	Suspended	4,000	6-12	10	15	1	8	31	64	73
CP 60	Suspended	6,000	9-18	15	20	2	10	31	70	74
CP 80	Suspended	8,000	12-24	20	30	3	12	36	75	85
CP 120	Suspended	12,000	18-36	30	40	5	20	39	95	90

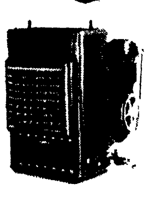


STEAM UNIT HEATERS.—Centrifugal Type for extreme quietness and efficiency.

Steam pressure—to 150 lbs per sq in.

Finish—Brown wrinkle enamel and stainless steel louvers.

Sizes—80,000 Btu/hr and 160,000 Btu/hr output.



GAS UNIT HEATERS.—Twelve models. 75,000 to 200,000 Btu capacity. Equipped with CENTRIFUGAL or PROPELLER type fans. *A.G.A.* approved for all gases.

Squirrel-cage blowers provide SILENT operation and permit air delivery thru duct systems up to ¼ in. S.P.

Stainless steel ribbon burners result in quiet efficient combustion.

Dual directional, individually adjustable stainless steel louvers permit complete control of air delivery.

Write for Catalogs, Literature, or Information

Niagara Blower Company

General Sales Office: 405 Lexington Ave. **New York 17, N. Y.**

CHICAGO—5: 37 W. Van Buren St. BUFFALO—7: 673 Ontario St. SEATTLE—4: 705 Lowman Bldg.
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Over 35 Years' Experience in Industrial Air Conditioning, Liquid Cooling and Air Drying

NIAGARA AERO HEAT EXCHANGER

For cooling industrial liquids, water, oils, solutions, chemicals, compressed air and gases, with Niagara "Balanced Wet-Bulb" temperature control to improve efficiency and obtain precise results. Patented (U. S. Nos. 2,296,946 and R. I. 22,553). Ask for Bulletin 120.

NIAGARA AIR CONDITIONING SYSTEMS

For human comfort and for all industrial applications requiring controlled conditions of temperature, relative humidity, air purity and air movement.

NIAGARA AIR CONDITIONER, TYPE A, AND CONTROLLED HUMIDITY METHOD

High precision apparatus using saturation to obtain control of R. H. to 1 per cent for laboratory work and control of hygroscopic materials. Ask for specific information.

NIAGARA AIR CONDITIONER, TYPE C

A year around air conditioning unit providing heating and humidifying or dehumidifying. Ask for Bulletin 80.

NIAGARA FAN COOLER AND DISK FAN COOLER

For comfort cooling, process cooling, low temperature storage for dairies, fruits, meats, food products, fur storage vaults, etc. Bulletin 72.

NIAGARA SPRAY COOLER

For all cooling applications requiring high humidity or high capacity in small space. Ask for Bulletin 110.

NIAGARA "NO FROST" SYSTEM

Using Niagara "No Frost" Liquid in spray coolers, prevents frosting of cooling coils, automatically keeps spray solution at proper concentration, gives freedom from brine troubles, corrosion. Constant, efficient operation. Temperature to −100° F. Ask for Bulletin 105.

NIAGARA AEROPASS CONDENSER (Illustrated)

Saves power and water cost utilizing atmospheric air to remove heat of condensation. Patented Duo-Pass prevents scaling, saves power. "OILOUT" positively removes oil and dirt from refrigerant lines, assuring always full capacity. Balanced Wet Bulb Control assures operation of refrigeration plant at minimum head pressure regardless of weather or load conditions. Ask for Bulletin 111.

NIAGARA "DUAL" COOLERS

Simultaneously cools a room and furnishes chilled water as a refrigerant. Saves equipment cost, operating expense. Patented. Ask for Bulletin 70.

NIAGARA INDUSTRIAL LIQUID COOLER

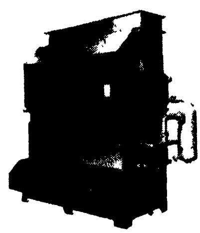
Furnishes refrigerated water or aqueous solution in any quantity up to 220 gpm. Positive control of temperature regardless of load variation. Delivers "sweet" water at 33° F without danger of freezing damage. Ask for Bulletin 104.

NIAGARA FAN HEATERS AND HIGH PRESSURE STEAM FAN HEATERS

For heating and ventilating large areas. Units of the highest quality in engineering, material and workmanship. Ask for Bulletins 73 and 109.

NIAGARA MOTOR BLOWERS

One, two and three-fan units. High and low static pressure models. Ask for Bulletin 89.

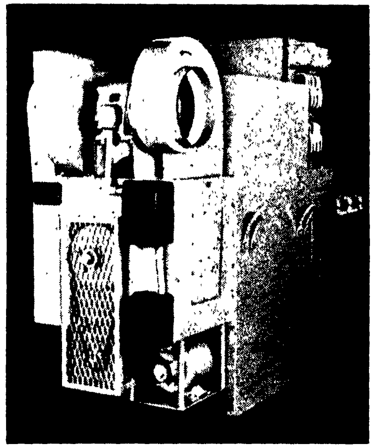


Niagara Aeropass Condenser with "Oil-out" and Balanced Wet Bulb Control

Pittsburgh Lectrodryer Corporation

Foot of 32nd Street **Pittsburgh, 30, Pa.**

This machine protects equipment in storage by maintaining a relative humidity of 35 per cent or lower.



Small automatic air conditioning type LECTRODRYER used for providing lowered relative humidities.

FOR INDEPENDENT CONTROL OF DEHUMIDIFICATION IN COMFORT AND INDUSTRIAL AIR CONDITIONING

The results of years of experience in the independent control of industrial dehumidification are now available for comfort air conditioning in the form of sturdy, dependable, thoroughly tested machines for controlled adsorption dehumidification.

LECTRODRYER equipment using Activated Alumina, a solid adsorbent, is widely used in maintaining lower than normal relative humidities in the chemical, pharmaceutical and other industries.

In comfort air conditioning these machines handle the latent heat load with only the sensible heat load left for refrigeration or water cooling. With this type system, only the air needed for the sensible heat load is cooled and no reheat is required.

Machines are available for steam, gas or electric operation, whichever the purchaser specifies. Standard machines are available in several sizes ranging from 350 cfm upward.

LECTRODRYERS are shipped complete as self-contained automatic units in that they require no regular manual attention except for starting. They are built for continuous operation with reactivation being carried on simultaneously with the drying operation.

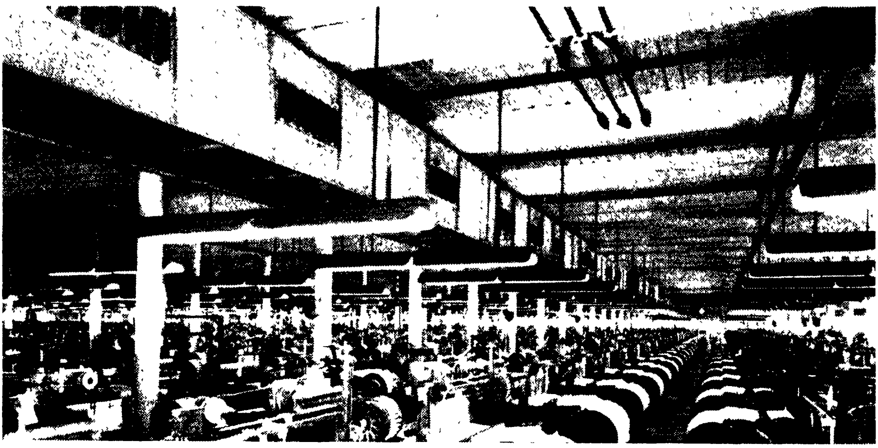
Write for full details.

Parks-Cramer Company

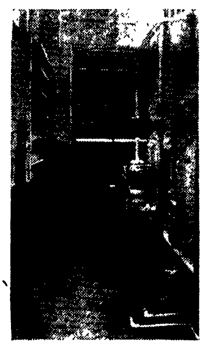
Fitchburg, Mass. **Charlotte, N. C.**

CERTIFIED CLIMATE

Complete Air Conditioning Systems including Humidifying or Dehumidifying, Cooling by Evaporation or Refrigeration, Ventilating, Filtering, Air Washing; with Automatic Control of Humidity, Temperature and Air Change.

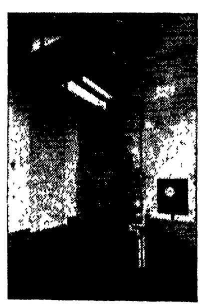


Central Station Air Conditioning



A complete system for conditioning air, with positive circulation and controlled ventilation. One or more air washer and fan units. High humidifying and evaporative cooling capacity. Heating, filtering, and refrigerated cooling optional. Ducts with adjustable outlets distribute conditioned air uniformly. Slight air pressure also improves uniformity. Centralized maintenance. Used with or without booster humidification. Under complete automatic control. Exclusive multi-speed fan motor control available.
Central Station Air Washer Units. Nozzles and Self-Cleaning Tank Screens for Central Station Air Washers.

Laboratory Air Conditioning



An essential of a good testing laboratory is adequate and dependable air conditioning. Accuracy in testing is maintained only when samples and testing equipment are free from fluctuations in both temperature and humidity. A vertical or horizontal unit is located within or adjacent to the laboratory. Includes equipment for heating, humidifying, dehumidifying, refrigeration, circulation, and adjustable recorder-controller. Heat by steam or electricity.

Parks-Cramer Company

Fitchburg, Mass. **Charlotte, N. C.**

Automatic Airchanger

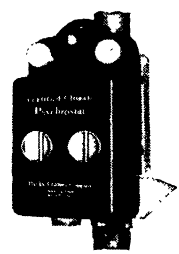
An improved system of forced air change and distribution used with a direct humidifying system. Insures fixed humidity and maximum evaporative cooling by controlling amount of air change and operation of humidifiers by humidity and temperature Psychrostat. Designed for either complete new installations or for supplementing existing direct humidifying equipment.

Gradumatic Humidifying System

New and advanced type of direct humidifying system for use alone, with Airchangers, or as booster for Central Station system. Air and water under pressure, water pressure always less than air. Safe. Economical. Few component parts, easy to take

apart and reassemble. Proper operation assured without testing. No adjustment required. Both air and water ports cleaned automatically. Heads operate continuously, evaporative output being varied gradually and automatically by Certified Climate Psychrostat to suit requirements for constant humidity at all times.

Certified Climate Psychrostat



Improved model more sensitive and accurate than ever before. Rugged and reliable. Use of the wet and dry bulb principle permits Psychrostat to perform the many and varied tasks in the field of humidity and temperature control which contribute to the success of Certified Climate systems.

The Pettifogger (not illustrated)

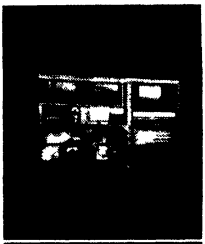
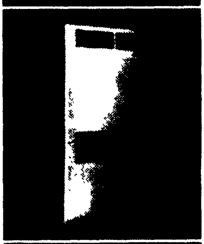
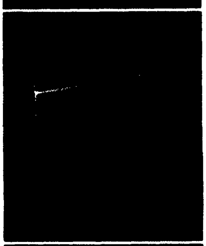
A compact centrifugal humidifier, with fan, for offices, storerooms, experimental rooms, hospitals, or other isolated departments. Self-contained in lacquered copper casing. Easily connected to water and electrical supplies. Automatic control. Adjustable capacity. Neutralizes drying effect of heating.

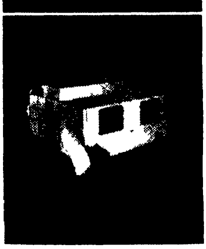
United States Air Conditioning Corporation

Engineers and Manufacturers of Air Conditioning, Refrigeration, Units Heaters, Coils and Ventilating Equipment

For Industrial, Commercial and Residential Applications

3321 Como Ave. Southeast **Minneapolis 14, Minnesota**



Refrigerated Kooler-aire—Central Station Air Conditioning plant with built-in evaporative condenser or water condenser. 3 to 50 ton sizes.
Other models available for water chilling with either evaporative condenser or water condenser.

Evaporative Condensers—A cooling unit that condenses refrigerants, available capacities from 3 through 100 tons. Permits water savings of 95 per cent.

Upright Store Conditioners—A completely packaged room air conditioner built in: 2, 3, 5, 7½ and 10 ton capacities.

Room Air Conditioner—One-half ton and three-quarter ton window type air conditioners with five year compressor guarantee.

Modu-aire—Individual room controlled coil and blower units which provide year around air conditioning without ductwork for multiroom buildings. Recessed, free standing and horizontal models built in capacities from 300 cfm **up.**

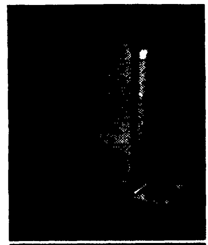
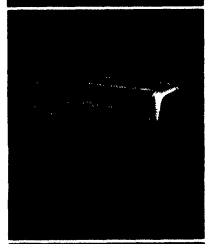
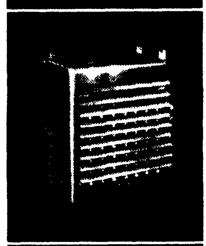
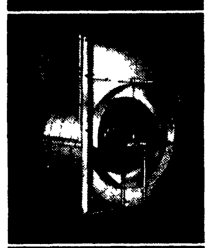
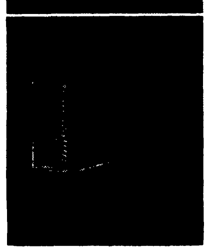
Unit Heaters—Suspension type heaters operating on steam, hot water or gas . . . built in a complete range of sizes.

Blower Type Unit Heater—Floor, ceiling inverted or wall mounted models from 1000 to 33,000 cfm for industrial heating.

Blowers—Backwardly or forwardly inclined blade blowers. Sizes and capacities for all cooling, heating, ventilating and air conditioning requirements.

Unit Air Conditioners—Coil-blower units for year around air conditioning. Made in ceiling and floor models, from 1000 cfm to 12,000 cfm.

Coils—Coils for every air conditioning requirement, including standard and steam distributing tube type steam coils, water coils for heating or cooling and direct expansion coils.

Worthington Pump and Machinery Corporation

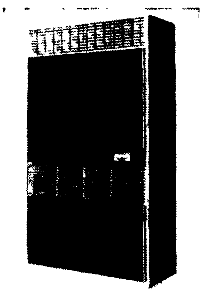
Air Conditioning and Refrigeration Division

General Offices: HARRISON, NEW JERSEY

Albany	Chicago	Fort Worth	Louisville	Phoenix	San Francisco
Atlanta	Cincinnati	Galveston	Milwaukee	Pittsburgh	Seattle
Baltimore	Cleveland	Greenville, S.C.	New Haven	Portland, Ore.	Springfield, Mass.
Birmingham	Dallas	Houston	New Orleans	Providence	Syracuse
Boston	Denver	Kansas City	New York	St. Louis	Tulsa
Buffalo	Detroit	Knoxville	Omaha	St. Paul	Washington, D. C.
Charlotte	El Paso	Los Angeles	Philadelphia	Salt Lake City	Wilmington, Del.

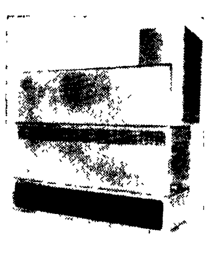
Representatives in all Principal Cities A.O.25

Packaged Air Conditioners



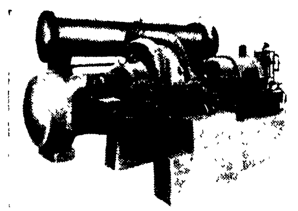
Provide cooling (or heating, if desired), dehumidification, ventilation, and air cleaning for commercial and industrial applications. 3, 5, 7½, 10, 15, 20, and 25 ton capacities.

Air Conditioning Units



Series AHY and AVY Central Station Air Conditioners for year-round air conditioning. AHY units for horizontal air flow, ceiling mounting; AVY units vertical, for floor mounting. 5 sizes, 2,000 to 12,000 cfm, 4 to 62 tons. With or without internal face and bypass dampers.

Centrifugal Refrigeration Water Cooling Systems

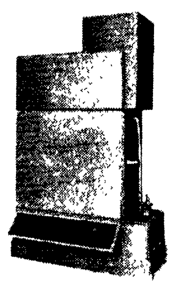


150 to 1,200 tons, 56 unit sizes. Freon centrifugal compressor, water cooler and water-cooled condenser in compact unit assembly. Electric motor or steam turbine drive.

Refrigeration Compressors Freon

Model HS-3 and 5 hp, 2-cylinder, vertical, splash lubrication. Model HF 7½ to 100 hp, 4-cylinder, B-type and 6-cylinder, W-type; full force feed lubrication.

Evaporative Condensers



Series ECZ, 10 to 150 tons. Sectionalized construction, all parts easily accessible. Galvanized steel coils for ammonia, bare copper coils for Freon-12.

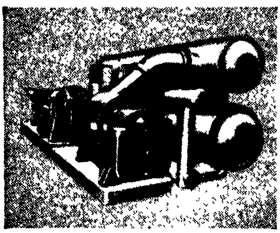
York Corporation

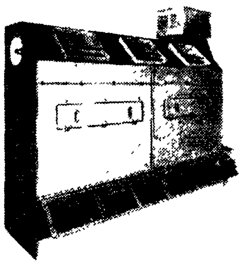
York, Pennsylvania

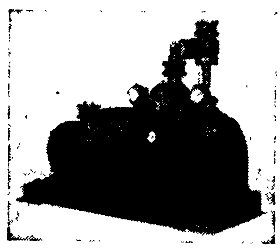
Factory Branches and Distributor Engineering and Sales Offices throughout the World.

Air Conditioning and Refrigeration for maintaining proper atmospheric conditions for industrial processes or comfort requirements. Installations of unit and central systems in a complete range of capacities and types for every design requirement.

Condensing and Water Cooling Systems—Turbo (centrifugal) brine and water cooling systems available over wide range of capacities—up to 1500 tons refrigeration for Freon-11 water cooling duty—suitable for steam turbine or motor drive.
Self-contained dynamically balanced, non-vibrating V/W type reciprocating compressors available in capacities up to 350 tons refrigeration in a single unit, with water cooled or economizer type condensers. Efficient automatic capacity reduction available for economical operation at reduced load.


York Turbo Compressor


York Sectional Economizer


York V-W Condensing Unit

The York Economizer—A combined force-draft cooling tower and refrigerant condenser, is available for installations where prohibitive water costs or inadequate drainage facilities preclude the use of a water cooled condenser. Standard factory constructed and built-up units may be used singly or in multiple for applications of any specified capacity. Economizers for use with Freon as the refrigerant are furnished, as standard, with a liquid sub-cooling coil. Economizers also designed for cooling of quench oil and other liquid coolants.
Air Conditioning Units: A complete line of finned coil, dry coil, wetted surface and spray type sectional air conditioners for horizontal or vertical applications, designed to facilitate installation and the distribution of air. Standard units can be equipped with by-pass feature and arranged for cooling and dehumidifying, heating and humidifying, for year-round processing.


Yorkaire Unit Air Conditioner

Yorkaire Unit Air Conditioner—A compact, self-contained model occupying but 21 x 42 inches of floor space and requiring only water, drain and electrical connections to operate. Special features provide utmost flexibility to meet varying conditions. Finger-tip dial control provides automatic and manual temperature and humidity control. Air volume and motion may also be adjusted by a special control and the directional grille provides *directed air flow*—up, down or from side to side. May be used with ducts if desired.
Yorkaire Conditioners are ruggedly built, quiet in operation, equipped with standard fan and compressor motors for AC or DC.

Dehumidifiers—For central station systems where a large volume of air is to be handled and where control of humidity is an essential requirement, the York dehumidifier is especially applicable. Construction features insure a minimum space demand and maximum performance conditions. Standard washers are available in a full range of capacities for industrial installation.

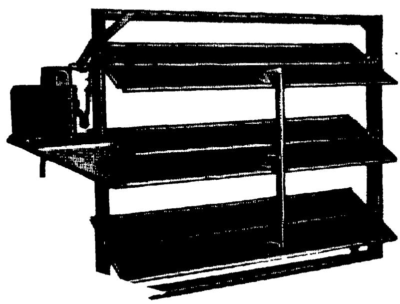
American Foundry and Furnace Co.

General Offices: Bloomington, Illinois
P. O. Box 904

Sales & Engineering Offices in Principal Cities

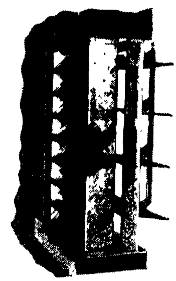
Auburn, Ind.	Lige Warm. & Vent. Co.	Milwaukee	Amer. Fdry. & Fur. Co.
Chicago	Temperature Equip. Corp.	Nashville	Cooper & Winters
Cincinnati	Walter A. Juergens	New Orleans	R. K. Rothrock
Cleveland	Amer. Warm. & Vent. Co.	New York City	The Demuth Co.
Dallas, Tex.	J. P. Ashcraft Co.	Oakland	Aladdin Heating Corp.
Denver	Kent Engineering Co.	Philadelphia	Amer. Htg. & Vent. Co
Des Moines	C. H. McGuiness Co.	Richmond, Ind.	Harry Hoff S.M. Works
Detroit, Mich.	H. J. Clemens	Richmond, Va.	Richmond Air Eq. Co.
Elmira, N. Y.	Amer. Warm. & Vent. Co.	St. Louis	M. F. Carlock
Ft. Worth, Tex.	J. P. Ashcraft Co.	Salt Lake City	Williams Gritton & Wilde
Grand Rapids	Bennett Heating Equip. Co.	San Antonio	Langhammer-Rummel Co.
Indianapolis	Elliott-Williams Company	Seattle	McPherson Fur. & Eq. Co.
Kansas City, Mo.	John H. Kitchen & Co.	Toledo	Amer. Warm. & Vent. Co.
Kenmore, N. Y.	Murton J. Rodman	West Lafayette, Ind.	F. H. Speaker & Son
Los Angeles	Harry F. Haldeman, Inc.		

F-12 LOUVER DAMPER



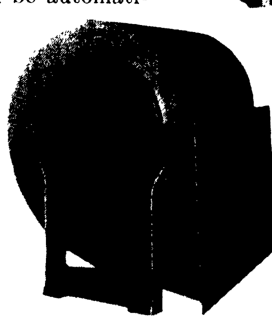
Made to fit any size opening. Adaptable to automatic or hand control. Blades of 16 gage steel. Channel frames 2 x ½ x ⅛ in. standard except in large sizes—optional ¼ x 2 in. bar iron frame. Standard steel painted aluminum—optional galvanized iron. Ball bearing blade pivots standard—optional brass trunions. Made for vertical or horizontal installation. For industrial plants, powerhouses, hotels, schools, theatres, etc. When F-12 is ordered with blades longer than 48 inches, dampers are made in multiple sections operating in unison. Motor brackets for internal or external mounting at extra charge. Motors and connecting linkage furnished by others unless specific arrangements are made. Standard is as illustrated with adjustable extended shaft.

S-454-F COMBINATION STORMPROOF LOUVER and DAMPER



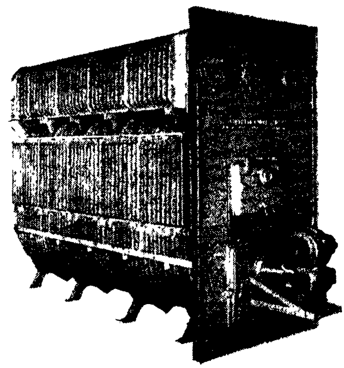
Consists of galvanized iron frame with 26 gage galvanized iron stationary horizontal stormproof louver blades riveted securely to outside frame. Apron extends over sill. Back of stormproof louver is No. 16 mesh, rust-proofed, insect screen in "U" type removable frame. Back of screen is multiple-blade ball bearing louver damper—similar to F-12 but with off-center axle—to control volume of air admitted. Louver damper blades of 16 gage steel galvanized. Frame of 2 x ½ x ⅛ in. galvanized channel iron. Dampers can be automatically or manually controlled. Blades all work in unison. Made to fit opening size specified. Standard for 8 in. deep wall. Entire assembly or any part can be furnished made of aluminum, copper or stainless.

SUPERIOR BLOWERS



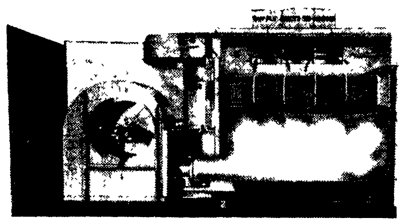
Forwardly Curved Multiblade Type
Heavy Duty Construction
Made in Single and Double Widths with wheel diameters ranging from 10 to 65 in., in 5 in. increments.
Capacity Range: 800 to 105,000 cfm

HEAVY DUTY HORIZONTAL HEATERS



Heavy Cast Iron Construction for long, dependable service and steadier heating even with automatic burners. *Sectional* for easy conveyance, assembly, and part replacement. *Tight Joints*—offset type packed with furnace cement and asbestos rope, then bolted. *Integrally Cast Fins* add strength and heating surface. *Long Fire Travel* saves heat and fuel. *Designed to Relieve Internal Stresses* set up by heating and cooling. *No Freezing—No Scaling* of heater possible since no water used to transfer heat. *Attractive Steel Panel Casing.*

CENTRAL TYPE FORCED WARM AIR

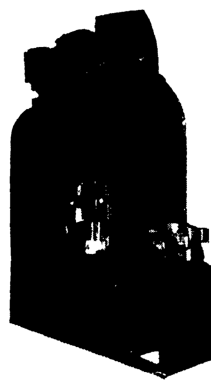


For schools, churches, theatres, auditoriums, gyms, drying plants, etc. Heats and ventilates with same system —uses part outside air to maintain air quality. Air filters, automatically controlled humidifier, and automatic temperature regulation optional. Summer cooling by adding compressor and coil. Output Capacity per heater, Btu per hr:

Hand Fired	
Coal—	278,000 to 1,942,000
Stoker Fired	
Coal—	278,000 to 2,440,000
Oil or Gas—	278,000 to 4,080,000

Two or more heaters may be set together to provide any desired output capacity. *Special Models* for hand fired coal, stoker fired coal, oil and gas. *Convertible Heater* (see picture top of page) designed so only minor changes necessary to switch from use of one fuel to another at any time and yet preserve same output capacity—a distinct advantage when fuel situation uncertain.

UNIT HEATER TYPE FORCED WARM AIR

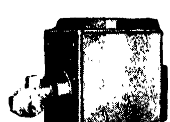


For Industrial Buildings, Warehouses, Factories, etc. For oil, gas or stoker fired coal. Use for heating and ventilating, or for tempering outside air supplied to replace air exhausted. Each unit is complete heating plant. Induced Draft Fan optional. Output Capacity Range per Unit:
Stoker Fired Coal - 557,500 to 2,028,000 Btu per hr.
Oil or Gas - 440,000 to 3,580,000 Btu per hr.

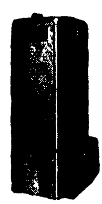
DOMESTIC HEATING EQUIPMENT

June-Aire Gas Fired

June-Aire Oil Fired



June-Aire Oil Fired



June-Aire Vertical Gas Fired

American Furnace Co.

1300 Hampton Ave., **St. Louis 10, Mo.**

Factory-Red Bud, Ill. Home Office—St. Louis, Mo.

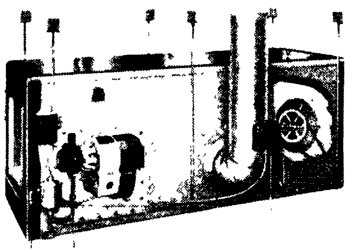
Manufacturers of Warm Air Heating Equipment

Distribution and Sales Offices in Principal Cities

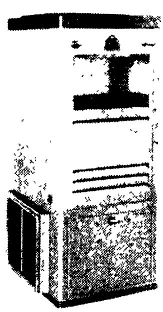
GAS - OIL - COAL FIRED HEATING UNITS

SPACE HEATERS—Suspended ceiling units for industrial and commercial installations—oil or gas fired. Space saving automatic heat—no furnace room needed. Functions effectively with minimum of duct work. Ratings up to 210,000 Btu output. Two sizes—oil input 1.35 gph and 2 gph.

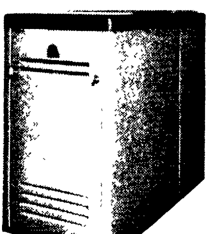
Unit Heater *AGA* approved—Gas fired—vented— propeller fan forced air type modernly styled, die-formed, welded steel cabinet—baked hammerloid finish. Heat exchange elements hidden from view and touch. Fan securely mounted at rear. Btu input, 90,000 and 150,000 per hour—air delivery cfm—1400 and 2300.



Model 23 OC



Model S-100BB



Model OM-90 MA

Model 700 Series MA

GAS FIRED UNITS—*A.G.A.* approved gravity, forced air, Base-Blo (hi-boy) and Counter-Flo models. Heat exchanger in Master Gas models—electrically welded steel—In Thermo design—sectional cast iron. Nationally known controls, double walled Hammerloid enamel finished cabinets. Hourly Btu input ratings, 70,000 to 200,000.

A.G.A. listed gas conversion units—single port upshot burners for all type gases. Ratings, 70,000 to 370,000 Btu input—for furnace or boiler installation.

OIL FIRED UNITS—Gravity, hi-boy, Counter-Flo and forced air models—Oil Master, Vapor-Fire and Air Stream designs.

Burners—Pressure gun type and oil vaporizing type.

Combustion unit—Heavy gage steel, designed for efficient transfer of heat.

Cabinets—Double walled construction, finished in two-toned baked Hammerloid enamel.

Models available in a range of 75,000 to 250,000 Btu output per hour.

Conversion Burners—Pressure gun type —Sizes 75,000 to 588,000 Btu output. Fuel burning capacity—0.75 to 6 gal per hour.

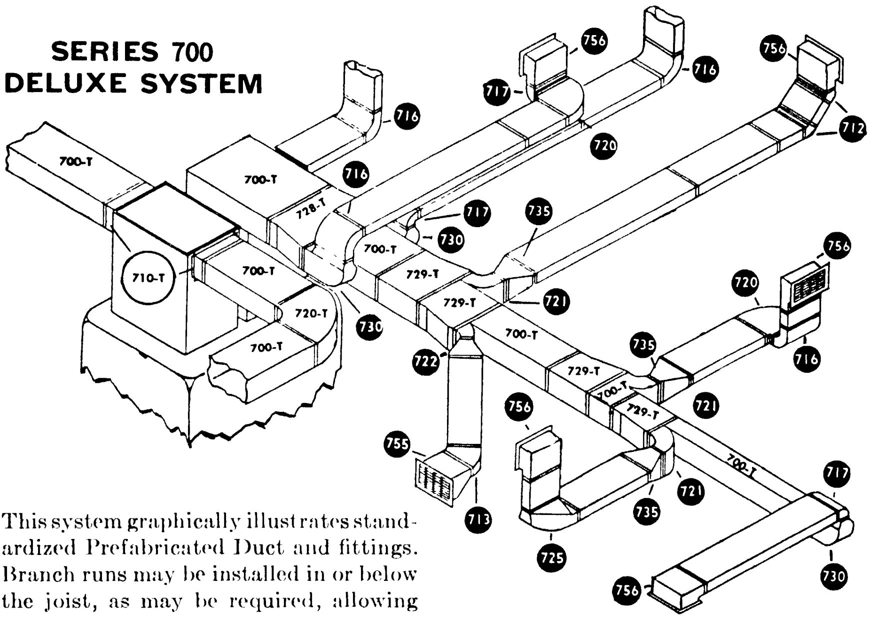
SOLID FUEL UNITS—AFCO gravity steel furnaces and THERMO gravity cast iron furnaces—Thermo pipeless cast iron and AFCO Modern Air (forced air) steel units. Gravity units available in conventional round galvanized casing or in square cabinet finished in two-toned Hammerloid enamel. All units adaptable to automatic heat with gas, oil, or stoker. Btu capacity at registers up to 236,000 per hour. Stoker—"Triple Seal" construction eliminates hopper smoke nuisance. Three sizes 20, 35 and 50 lb coal feed per hour. Capacity up to 480,000 Btu per hour.

Descriptive Bulletin Available on each Unit.

Clayton & Lambert Mfg. Co.

Louisville 10, Ky.

Manufacturers of C & L Blow Torches, HOFFMAN Water Heaters
Silver Shield Silos, LAMNECK Furnace Pipe and Fittings



This system graphically illustrates standardized Prefabricated Duct and fittings. Branch runs may be installed in or below the joist, as may be required, allowing maximum basement head room and freedom from exposed ducts in recreation rooms, etc. Flexibility, ease of assembly, reasonable original cost, efficiency and accurate control of air flow, combined with exceptional appearance, are inherent qualities of this DeLuxe System.

700-T	Duct sizes 4x8 to 36x8
710-T	Starting Collar, sizes same as duct
728-T & 729-T	Increaser-Reducer Sections, sizes same as duct (Max. Inc. 10″ for 728-T and 5″ for 729-T)
712-T	Main Trunk Angles, sizes same as duct
713-T	Main Trunk Angles, sizes same as duct
717-T	Main Trunk Ells, sizes same as duct
720-T	Main Trunk Ells, sizes same as duct
721	Side Takeoffs 4x8, 5x8, 6x8
722	Side Takeoffs 4x8, 5x8, 6x8
724–725–726	Reverse Stack Elbows 10x3¼, 12x3¼, 14x3¼
730	Top Takeoff 4x8–10x3¼, 5x8–12x3¼, 6x8–14x3¼
735	Stack Adapter 4x8–10x3¼, 5x8–12x3¼, 6x8–14x3¼
712	Angle 10x3¼, 12x3¼, 14x3¼
713	Angle 10x3¼, 12x3¼, 14x3¼
716	Elbow 10x3¼, 12x3¼, 14x3¼
717	Elbow 10x3¼, 12x3¼, 14x3¼
720	Elbow 10x3¼, 12x3¼, 14x3¼
700	Wall Stack 10x3¼, 12x3¼, 14x3¼
755	Stack Head 4, 5, 6, 8x10, 4, 5, 6, 8x12, 4, 5, 6, 8x14
756	Stack Head 4, 5, 6, 8x10, 4, 5, 6, 8x12, 4, 5, 6, 8x14

Write for L-49 Catalog showing our complete line of Standardized Ducts and Fittings.

Campbell Heating Company

3121 Dean, **Des Moines 17, Iowa**

SUMMER and WINTER AIR CONDITIONING

Industrial, Commercial—Institutions, Residences

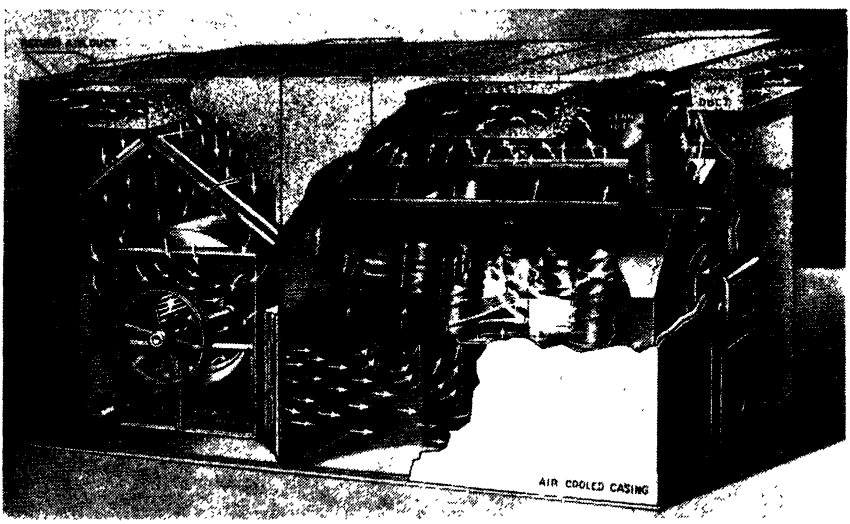
EASTERN REPRESENTATIVE: NEIL ADAMS, OLD YORK ROAD, LAMBERTVILLE, N. J., TELEPHONE NO. 665

CAMPBELL "WINTER-CHASER" AIR CONDITIONING SYSTEM

The Campbell "Winter-Chaser" System provides all the essentials of winter air conditioning: Simultaneous control of temperature, humidity, air circulation and air cleanliness, besides providing fresh air for ventilation, quick heating, flexibility; and a summer cooling effect. Campbell equipment is built of the best materials obtainable, and has been developed through over sixty years of experience. The system is designed by competent experienced engineers and installed by experienced mechanics. It is guaranteed as to results and for 10 years as to durability. We will be glad to help solve any heating or ventilating problems or help with layouts and specifications for churches, schools, garages, etc.

For Large Schools, Churches, Commercial and Industrial Buildings

GAS—OIL—STOKER OR HAND FIRED



8000 Series

Heater No.	Output Capacity BTU Per Hr (1)	Heating Surface Sq Ft (2)	Normal Blower CFM for 140° Reg. Temp.	Size Motor for ¼" SP HP	Dimensions Casing Inches	Addnl Space for Blower	Approx. Shipping Weight Including Burner & Comb. Chamber, lbs.
8075	725,000	280	8,850	¾	76 x 80- 88" high	54"	6,000
8100	893,000	320	10,900	1	76 x 93- 96" "	60"	7,200
8125	1,080,000	360	13,200	1½	76 x 105- 96" "	66"	8,500
8150	1,275,000	440	15,600	2	76 x 118-102" "	66"	9,700
8175	1,440,000	480	17,500	2	76 x 130-102" "	72"	11,500
8200	1,725,000	600	21,100	2	94 x 137-120" "	76"	12,700
8250	2,160,000	750	26,400	3	94 x 157-120" "	76"	15,000

Unit includes furnace, casing, blower, motor, V-flat drive.

(1) Allow 10 to 35 per cent over heat loss figures for pick up load and 10 to 15 per cent for duct and other radiation losses. The upper figures are preferable for best operating economy and life of the equipment.

(2) Heat emission per sq ft of heating surface is 3000 Btu per hr or less.

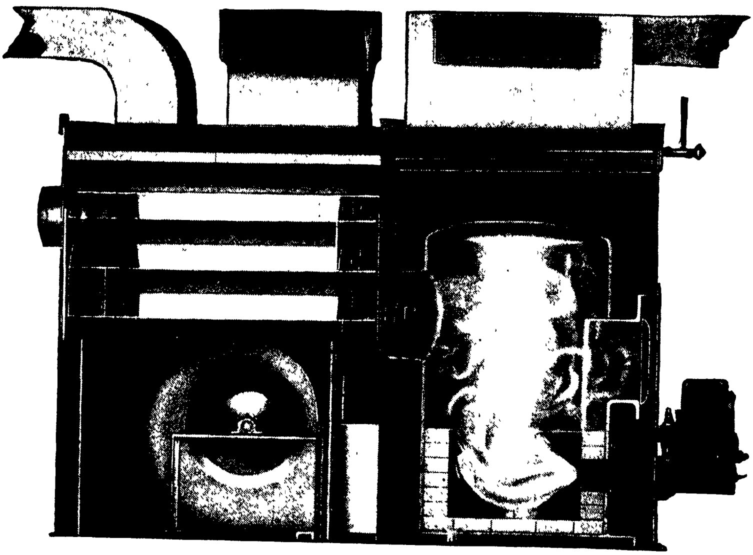
CAMPBELL "WINTER-CHASER" AIR CONDITIONING SYSTEM ENGINEERING SERVICE

Our Engineering Department will be glad to help solve any heating or ventilating problems or help with layouts and specifications for churches, schools, or any large building.

GUARANTEE

If the duct system is designed or approved by our Engineering Department, and heater and blower are furnished by us and are according to our ratings, we will guarantee any heater for ten years against repairs from any cause and will guarantee the heating of all rooms to which warm air is delivered to 70 deg in the coldest and windiest weather. The motor, humidifier, automatic burner and controls, and other parts made by others carry their manufacturers' guarantee.

GAS OR OIL FIRED



Unit No.	Output at Registers Btu per hr.	Heating Surface Sq Ft	CFM for 140° Register Temp.	Motor Size for ¼ SP HP	Gals. Oil per Hr.	Gas or Oil Input, Btu per hr.	No. of Filters and Size, inches	Casing Size, Inches L	Casing Size, Inches W	Casing Size, Inches Ht.	Approx. Shipping Wt. lbs.
For residences, stores and other buildings kept warm continuously											
1866	(1) 112,000	(3) 66	1400	1/6	1	140,000	4–16 x 20	78	41	52	1300
18100	168,000	100	2100	¼	1½	210,000	4–20 x 20	86	41	57	1700
27133	224,000	133	2800	¼	2	280,000	6–16 x 25	90	50	62½	2100
30166	280,000	166	3500	⅓	2½	350,000	6–16 x 25	92	60	74	2600
30200	336,000	200	4200	⅓	3	420,000	9–16 x 25	97	60	76	3100
30233	392,000	233	5000	½	3½	490,000	12–20 x 20	104	80	72	3500
30250	448,000	250	5600	½	4	560,000	12–20 x 20	104	80	72	3700
For churches, schools or buildings where rapid temperature raising is necessary											
133D	(2) 290,000	(4) 133	3700	⅓	3	365,000	6–16 x 25	90	50	62½	2100
166D	365,000	166	4600	½	3¼	455,000	6–16 x 25	92	60	74	2700
200D	440,000	200	5500	½	4	550,000	9–16 x 25	97	60	76	3200
233D	510,000	233	6400	¾	4½	635,000	12–20 x 20	104	80	72	3600
250D	580,000	250	7300	¾	5¼	730,000	12–20 x 20	104	80	72	3800
255D	660,000	255	8300	1	6	820,000	12–20 x 20	104	80	72	4000

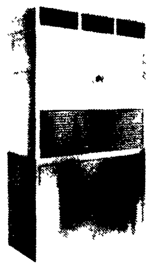
Campbell heaters are guaranteed to deliver full rated capacity

(1) Allow approximately 10 per cent over heat loss for pick up load.
(2) Allow up to 35 per cent over heat loss for pick up load.
(3) Heat emission per sqft of heating surface is 1800 Btu per hr or less.
(4) Heat emission per sqft of heating surface is 2600 Btu per hr or less.

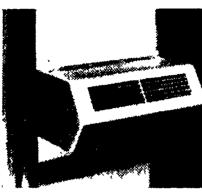
DIRECTOR, EASTERN SALES: NEIL ADAMS, OLD YORK ROAD, LAMBERTVILLE, N. J. TELEPHONE No. 665

CHRYSLER AIRTEMP

AIRTEMP DIVISION OF CHRYSLER CORPORATION, DAYTON 1, OHIO



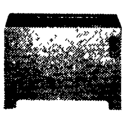
"PACKAGED" AIR CONDITIONERS 2-, 3-, 5-, 8-, 11- and 15-ton capacities COMPLETE—Assembled and tested at the factory. Cools, dehumidifies, filters and circulates air. Free air discharge or duct distribution. Heating coil for year-'round service optional. COMPACT—Entire unit is enclosed in "Bonderized" steel cabinet of modern design. Occupies very little floor space. EASILY INSTALLED—Needs only electric, water and drain connections. FLEXIBLE—Can be installed singly or in multiple to meet virtually every requirement. SEALED RADIAL COMPRESSOR—Quiet with all moving parts balanced and bathed in oil for long life, flexibly mounted to reduce vibration.



ROOM AIR CONDITIONERS ¾- and 1-hp Capacities Ideal for home or office use. Fits in window. Cools, ventilates, filters and circulates. The ¾ hp units operate with standard 115-volt current. The (1) one hp unit requires 230 volts. Adjustable grilles provide controlled air circulation. Adjustable outside air intake. Cooling mechanism can be turned off for mild weather air circulation.

DEHUMIDIFIERS ⅛-hp Capacity Low-cost, portable dehumidifier removes excessive moisture from the air. Can be plugged into any convenient electrical outlet. Protects against rust, mildew and corrosion. Very compact in size—15 in. high, 12 in. wide, 16½ in. long.

RADIAL COMPRESSOR UNITS 10 to 100-ton Capacities These heavy-duty units, for use with Freon, are especially designed for refrigeration and air conditioning. Radial compressors are direct connected and have force-feed lubrication. Automatic capacity-reduction device. Light weight, economical to operate.

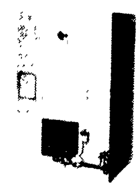


INDIVIDUAL ROOM AIR CONDITIONING UNITS Used for year-'round cooling and heating by connecting with central cold and hot water systems. Seven models in three types—floor, wall and ceiling—are available. Built-in controls make each unit independently operated. Two centrifugal fans provide circulation.

COMMERCIAL REFRIGERATION UNITS For refrigerator cases, cooler rooms, walk-in boxes and other commercial applications. Sealed and open types for self-contained or remote installation. Complete range of sizes from ⅙ hp to 7½ hp.

CHRYSLER AIRTEMP
AUTOMATIC HOME HEATING

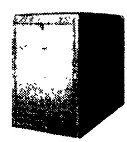
CHRYSLER AIRTEMP AUTOMATIC HOME HEATING GAS-FIRED FURNACE GRAVITY. . . .Sheet steel cabinet, "Bonderized" inside and out to resist rust. Heating surface is surrounded by an inner liner. Cooler air circulation between this and cabinet provides minimum heat loss. Burner is adaptable to all gases . . . gives quiet operation . . . and is easily removed for service. Complete with automatic controls. Capacity: 80,000 Btu.



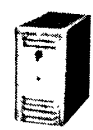
CONVERSION GAS BURNERS. . . .For quick, easy installation in existing furnaces. Combustion principle provides higher efficiency . . . minimum fuel consumption. Available in two capacities from 75,000 to 225,000 Btu.

CONVERSION OIL BURNERS. . . .New focused flame design. Four models available in capacities from ½ gal to 4½ gph. New type high velocity firing head secures exceptional efficiency, clean and quiet operation with both straight run and catalytic oils.

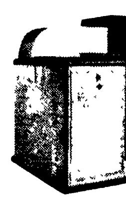
GAS-FIRED AUTOMATIC FURNACE. . . .Heats, humidifies, filters and circulates the air. Steel Models 60,000 to 185,000 Btu output. "Bonderized" and insulated jacket. The Airtemp Gas Burner starts, stops and operates quietly, has many exclusive features — no popping or flash-backs. Approved, *A.G.A. Laboratories*.



OIL-FIRED STEEL BOILERS. . . .12 models—from 81,000 to 324,000 Btu—for steam and hot water heating in every size and type of home. Available in flush and fully enclosed types, complete with Airtemp burner and all controls.



COMBINATION HEATING AND COOLING FOR THE HOME. . . .Combination of a 2-ton, 3-ton, 5-ton or 8-ton Chrysler Airtemp "Packaged" Air Conditioner and any of the larger Chrysler Airtemp automatic furnaces. The same blower, filters and ducts of the automatic heating system are employed for cooling in the summer.

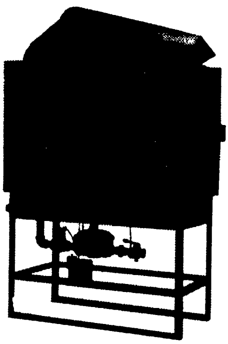


Hayes Furnace Mfg. & Supply Co.

2929 South Fairfax Ave., **Los Angeles 16, Calif.**

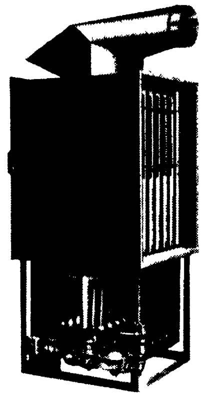
(Telephone Texas 0-3734)

STAINLESS STEEL GAS HEATING EQUIPMENT

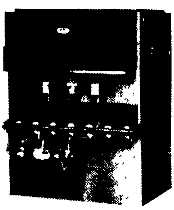


DUCT FURNACES

Constructed of Type 321 Stabilized Stainless Steel. Non-corrosive property of heat exchanger permits installation downstream of cooling coils or washer. Heat exchanger will not corrode when subjected to continuous ventilation during the summer season. Designed for continuous blower operation for offices, theaters, schools, churches, factories. Air throughput in either direction. Draft Hood and Vent Manifold reversible, adjustable for both horizontal and vertical connection. 14 sizes, 80,000 to 600,000 Btu per hr input in 40,000 Btu increments. *A.G.A.* certified for all gases.

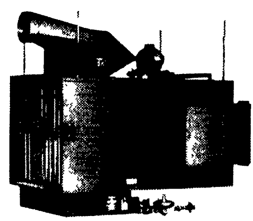


DUCT FURNACE MODEL SED-VF (Left). Tested and approved as a DUCT FURNACE—Constructed of type 321 stabilized stainless steel. Re-circulated or fresh air is introduced through bottom, sides or back of furnace. Equipped with air by-pass. Handles large cfm per Btu input. Vent in front of furnace. Made in 6 sizes from 70,000 to 245,000 Btu per hr input in increments of 35,000 Btu. For natural, manufactured and L.P. gases.



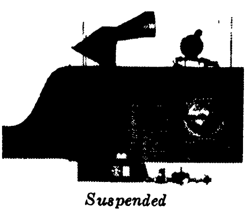
FORCED AIR SUSPENDED FURNACE, MODEL SES

(Right) Equipped to install in a suspended duct system. Inlet and outlet designed for sheet metal duct connection.



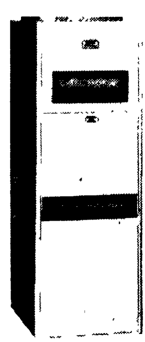
Suspended

UNIT HEATER MODEL SEU (Left, Below). Used where room air is to be recirculated. Diffuser outlet with adjustable vertical and horizontal vanes.



Suspended

Both types of Suspended units are designed for factories, commercial and other large Btu requirements—equipped with brackets for suspension, save floor space. Double inlet forward curve blower. Continuous duty variable pitch drives; cast iron, raised port, precision machined burners. Made in sizes 80,000 to 480,000 Btu per hr input in 40,000 Btu increments. For natural, manufactured and L.P. Gas.



FORCED AIR FURNACE MODEL SEC. Heat exchanger constructed of identical die formed sections of type 321 stabilized stainless steel. Sections are seam and arc welded. One piece cast iron burners with drilled ports. Oversize double inlet blower. Long hour motor. Equipped with glass filters. Finished in baked silver grey. 6 sizes 70,000 to 245,000 Btu per hr input in 35,000 Btu increments. For natural, manufactured and L.P. gases.

ALL HAYES FURNACES ARE *A.G.A.* APPROVED

THE MEYER FURNACE COMPANY

Manufacturers of Heating and Air Conditioning Equipment for Coal, Gas and Oil Burning

Peoria, Illinois

Branches and Distributors

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BALTIMORE, MD.	COLUMBUS, O.	KNOXVILLE, TENN.	MUSCATINE, IA.
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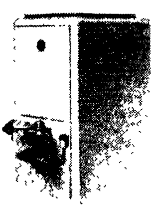
(Export Agent: Mid-States Export-Import Co., Inc., Peoria, Ill.)

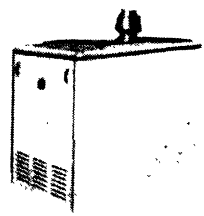
WEIR AND MEYER Steel Warm-Air Furnaces have a more than 80-year reputation for efficiency, dependability and durability. They are available for small and large requirements and for all fuels in a wide variety of firing applications.

MEYER GAS-FIRED EQUIPMENT—The MEYER gas-fired equipment is available in three types, the Q, M, the G Series ranging from 75,000 to 495,000 Btu/hr inputs. This entire gas line is *A.G.A.* approved.



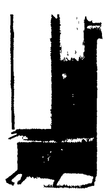
Q Series

Q SERIES GRAVITY gas-fired equipment is available in three sizes ranging from 75,000 to 150,000 Btu/hr. These units feature an extra large heating element and a Meyer fountain burner which produce amazing fuel economies.



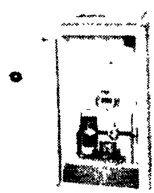
G Series

G SERIES FORCED AIR gas-fired is available in five sizes ranging from 110,000 to 495,000 Btu/hr inputs. These units employ extra heavy gage heating elements of the tubular design, which extract maximum heat from the fuel used.



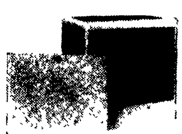
M Series

M SERIES FORCED AIR gas-fired Hi-boy has an input of 90,000 Btu/hr. It is designed to furnish winter air conditioning for those who want to save space in their small homes.



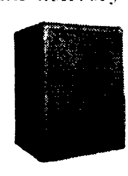
B-1 Series

B-1 SERIES FORCED AIR gas-fired Lo-Boy has an input of 110,000 Btu/hr and features the exclusive Meyer D-shaped Thermatic Radiator. Unit is shipped assembled.

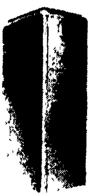


MEYER BLO-AIRE—These blower and filter units enable you to convert almost any gravity installation to forced air. They can be used for gas, oil, stoker, or hand-fired furnaces. Available in four sizes ranging from 1,000 to 2,600 cfm air delivery

MEYER SUMMER AIR CONDITIONER—MEYER Home Summer Air Conditioners are designed to fit neatly to the fan compartments of both oil and gas fired Meyer equipment. Available in 2, 3, 5, 6, 8 and 10 ton sizes these units are easy to install, and inexpensive to operate. They may be installed at the same time the heating equipment is installed or later.



MEYER OIL-FIRED EQUIPMENT.—The MEYER oil-fired air conditioners are available in four types, the D, B, E and K series ranging from 72,500 to 294,000 Btu,hr output at bonnet. The D and B Series are Hi-boys suitable for basement or first floor installation, and the E and K are designed for basement installations.

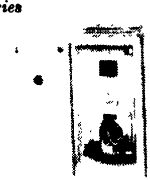


D Series

D SERIES FORCED AIR—oil fired Hi-boy has an output at the bonnet of 72,500 Btu/hr. This unit uses an *Underwriters' Laboratory* approved burner of the vaporizing type. Designed for small homes this unit is compact and smartly styled.

B SERIES FORCED AIR oil-fired Hi-boys are available in two sizes, 93,500 and 110,000 Btu/hr output at bonnet. These units use the high-pressure gun-type oil burner.

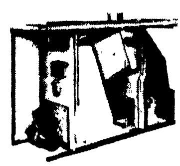
B Series



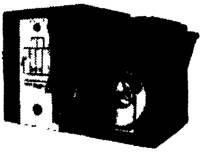
B-1 Series

B-1 SERIES FORCED AIR oil-fired Lo-Boy is available with pressure oil burner and output at bonnet of 105,000 Btu/hr or with vaporizing oil burner and output at bonnet of 96,000 Btu/hr. This series features D-shaped Thermatic Radiator. Unit is shipped assembled.

E SERIES FORCED AIR oil-fired equipment is available in two sizes, 110,000 and 165,000 Btu/hr output at bonnet. These units employ extra heavy gage heating elements of the tubular design, which extract maximum heat from the fuel used.



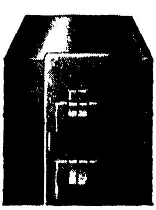
E Series



K Series

K SERIES FORCED AIR oil-fired equipment is available in two sizes, 203,000 and 294,000 Btu/hr output at bonnet. These units are designed primarily for larger homes and buildings.

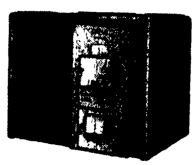
WEIR COAL-FIRED EQUIPMENT—WEIR coal-fired equipment is available in four types: the U, UC, R and 500 Series, ranging from 81,000 to 1,000,000 Btu/hr output at bonnet. Multiple installations will produce greater outputs. Welded and riveted construction is rugged, efficient and gas-tight. Equipment may be stoker-fired or adapted to oil or gas firing.



U Series

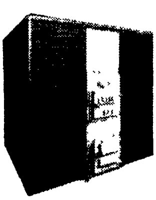
U SERIES GRAVITY coal-fired furnaces are available in five sizes ranging from 91,300 to 170,000 Btu/hr output at bonnet.

UC SERIES FORCED AIR coal-fired furnaces are available in five sizes ranging from 81,000 to 191,000 Btu/hr output at bonnet.



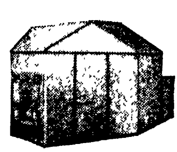
UC Series

R SERIES HEAVY DUTY coal-fired furnaces are available as forced air units in two sizes with maximum outputs at bonnet of 400,000 and 500,000 Btu/hr. The 400,000 Btu/hr unit is also available as a gravity unit. This equipment is available in round or rectangular casings.



R Series

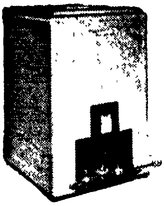
500 SERIES HEAVY DUTY coal-fired, forced air furnaces are available in five sizes ranging from 550,000 to 1,000,000 Btu/hr output at bonnet.



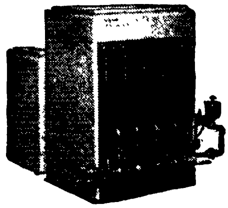
500 Series

L. J. Mueller Furnace Co. - Milwaukee 15, Wis.

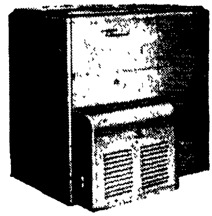
HEATING **Mueller Climatrol** AIR CONDITIONING



Type 101 (Gas) 201 (Oil)



Type 103

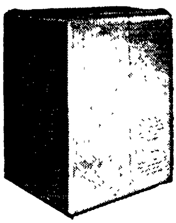


Type 105 (Gas) 202 (Oil)

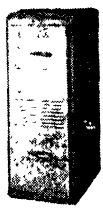
Type 101 (Gas) **201** (Oil) Gas-fired, steel, gravity furnace (convertible to, or available as oil-fired, Type 201). Available in four sizes with *A.G.A.* input ratings of 90-, 135-, 160- and 180,000 Btu.

Type 103 Gas-fired, steel, winter air-conditioner. Sectional construction. 8 sizes (in 45,000 Btu increments) with *A.G.A.* input ratings from 225,000 to 540,000 Btu.

Type 105 (Gas) **202** (Oil) Gas-fired, steel, winter air-conditioner (convertible to, or available as oil fired, Type 202). Available in nine sizes with *A.G.A.* input ratings of 100,000 to 675,000 Btu per hour.



Type 108



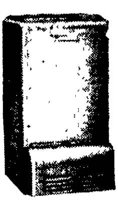
Type 109 (Gas) 209 (Oil)

Type 110

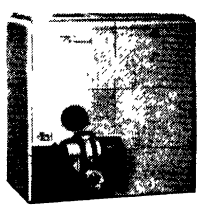
Type 108 Gas-fired, cast-iron winter air-conditioner. Sectional construction. Available in 67,500, 90,000, 112,500, 135,000 and 157,500 Btu input.

Type 109 (Gas) **209** (Oil) Gas-fired, steel, winter air-conditioner (convertible to, or available as oil-fired, Type 209). Two sizes: *A.G.A.* input ratings of 100,000 or 135,000 Btu per hour.

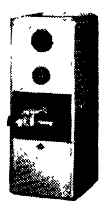
Type 110 Gas-fired, steel, winter air-conditioner for basements, closets or utility rooms. Available with *A.G.A.* input ratings of 60,000, 80,000, 100,000, 120,000 and 160,000 Btu per hour.



Type 111 (Gas) and 211 (Oil)



Type 112 (gas) and 212 (Oil)

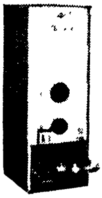


Type 114 (Gas) and 214 (Oil)

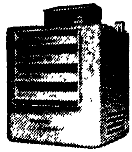
Type 111 (Gas) **and 211** (Oil) Gas-fired, steel gravity furnace. Available as, or convertible to oil-fired, Type 211. Shipped assembled. Available with *A.G.A.* input rating of 90,000 Btu per hour.

Type 112 (gas) **and 212** (Oil) Gas-fired, steel, winter air-conditioner (convertible to, or available as oil-fired, Type 212). Two sizes: 90- and 110,000 Btu input.

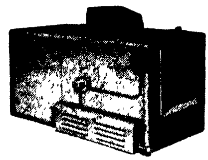
Type 114 (Gas) **and 214** (Oil) Gas-fired, steel, winter air-conditioner (convertible to, or available as oil-fired, Type 214). One size: *A.G.A.* input rating of 110,000 Btu.



Type 115 (Gas) and 215 (Oil)

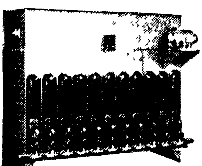


Type 150

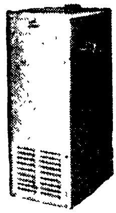


Type 155-151

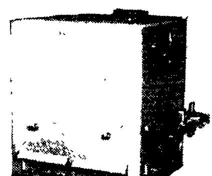
Type 115 (Gas) and 215 (Oil) Gas-fired, steel, winter air-conditioner (convertible to, or available as oil-fired Type 215). Counter flow design for perimeter heating, slab or crawl-space homes. *A.G.A.* rated at 110,000 Btu input.
Type 150 Gas, direct-fired, steel, suspended unit heater. Horizontal tubular design. Propeller-type fan. (Also with blower, Type 151.) Available with *A.G.A.* input ratings of 60,000, 90,000, 120,000 or 150,000 Btu per hour.
Type 155–151 Gas-fired horizontal winter air-conditioner. Heat exchanger as in Type 150 with blower. Two sizes with *A.G.A.* input ratings of 60,000 and 90,000 Btu per hour. Type 151 Blower unit heater available in four sizes 60,000, 90,000, 120,000 and 150,000. Also, Type 253 oil-fired horizontal winter air conditioner.



Type UH

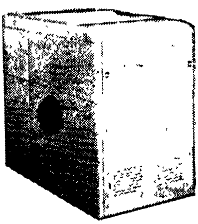


Type 10

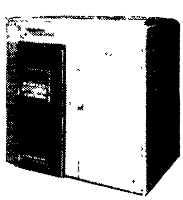


Type 20

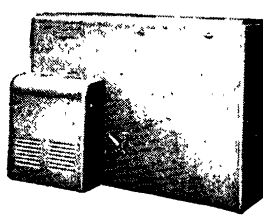
Type UH Gas-fired unit-heater. Steel sectional design heat exchanger with individual burners. Nine sizes (45,000 Btu increments) with *A.G.A.* input ratings from 180,000 to 540,000 Btu.
Type 10 Gas-fired, cast-iron boiler for residential heating, and hot-water supply. Controls enclosed. (Also with controls exposed, Type 11.) Approved for all types of gas. *A.G.A.* ratings of 290 to 2015 sq ft for water, and 180 to 1260 sq ft for steam.
Type 20 Gas-fired, cast-iron boiler for larger installations. Sectional construction (sizes increase in increments of 63,000 Btu). *A.G.A.* approved. Ratings of 1680 to 20,160 sq ft for hot water, and 1050 to 12,600 sq ft for steam.



Type 50



Type 702



Type 901

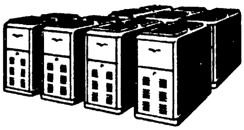
Type 50 Oil-fired, steel, winter air-conditioner. All-welded heat exchanger. Blower at rear. Pressure-atomizing burner. Available in four sizes with register output of 100,000, 150,000, 200,000 and 225,000 Btu output per hour.
Type 702 Coal-fired, steel, winter air-conditioner. Blower-filter cabinet may be installed on either side. Five sizes—20, 22, 24, 27 and 30 inches. (Gravity unit in same sizes, Type 701. Cast-iron gravity and forced air also available.)
Type 901 Summer air-conditioner available in 3-, 5- and 7-1/2 tons sizes. Shown installed in Type 105 gas-fired furnace which has *A.G.A.* inputs of 100,000, 150,000, 180,000 and 225,000 Btu.

Norge Heat
Division of Borg-Warner Corp.

672 E. Woodbridge, **Detroit 26, Michigan**

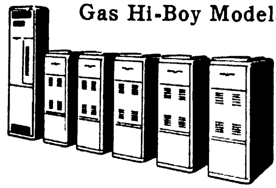
Shown here are pictures and brief descriptions of Norge Heat Products. More complete information on these products may be obtained by writing to Norge Heat Division, Borg-Warner Corporation, 672 E. Woodbridge, Detroit 26, Michigan.

Gas Lo-Boy Models



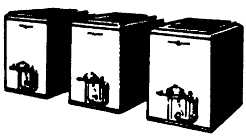
Fully automatic winter air conditioners for *any* gas—natural, mixed, manufactured, L.P., L.P.-Air. Vee-Sectional 12-gage steel heat exchanger with cleanout opening. Factory assembled. *A.G.A.* approved. 7 models, 80-170 M/Btu.

Gas Hi-Boy Models



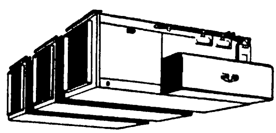
Compact winter air conditioners for any gas. Built-in draft diverter; filter frames for returns. Vee-Sectional 12-gage steel heat exchanger. Factory assembled. Also Hi-Boy wall-type. *A.G.A.* approved. 6 models, 62.5-120 M/Btu.

Gas Gravity Models



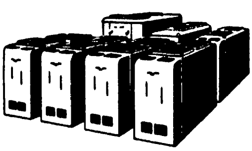
Fully automatic, easily installed. Drilled raised-port cast-iron burners with over 300 drilled jets provide cone-shaped flame. Round combustion chamber, baffled outer drum radiator. *A.G.A.* approved. 3 models, 80-120 M/Btu.

Gas Stowaway Models



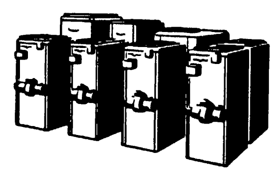
Compact forced-air units. Pressed-steel plate-type heat exchanger. Non-clogging, milled-slot, cast-iron burners. Sirocco-type blower. Dual flue outlets. Factory assembled and wired. *A.G.A.* approved. 3 models, 60-100 M/Btu.

Oil Lo-Boy Models



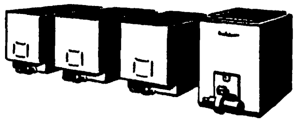
Winter air conditioners with pressure atomizing burner and jet-type combustion chamber. Tunnel firing design. 12-gage, all-steel heat exchanger. Rapid transfer for exceptional economy. Factory assembled. 7 models, 80-180 M/Btu.

Oil Hi-Boy Models



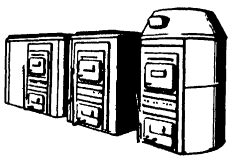
Fully automatic winter air conditioners. Pressure atomizing burner, jet-type combustion chamber. Corrugated plate-type steel heat exchanger. Tunnel firing. Hydraulically tested. Factory assembled. 8 models, 60-180 M/Btu.

Oil Gravity Models



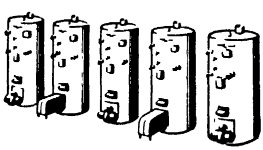
Fully automatic pressure-atomizing units. All-steel vertical combustion chamber, baffled outer-drum radiators. Pressure vaporizing and natural-draft floor furnaces available. Thermostat or manual operation. 4 models, 54-100 M/Btu.

Coal-Fired Models



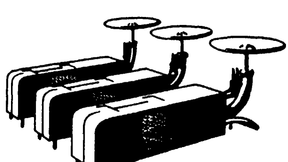
Forced air convertible furnaces for coal, oil, gas, *or* coal-fired gravity furnaces. Heavy gage, all-steel body; locomotive-type bar grates; waist-high shaker mechanism; pre-formed high-temperature fire brick. 12 models, 20 in.-27 in. dia.

Oil or Gas Round-Cased Boilers



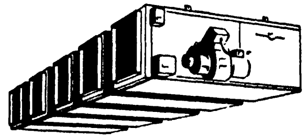
Boilers are round, dry-bottom, fully water legged. Corrugated plate-type flues. Tankless type coils—210 gal/hr—standard equipment. Storage tank coils optional. *SBI* rated; built to *ASME* code. 12 models to 880 sq ft.

Conversion Gas Burners



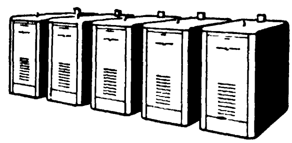
For boilers, warm air furnaces. Serve 95 per cent of domestic heating needs. Patented, single port, self-piloting, flame retention burner head—with non clogging features—insures safe, quiet operation. 7 models, 50–300 M/Btu.

Oil Suspended Models



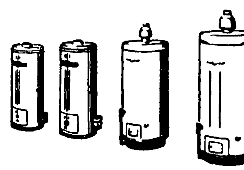
Horizontal winter air conditioners. May be installed on attic floor joists, suspended from rafters or overhead joists. Pressure atomizing burner; corrugated plate-type heat exchanger. Tunnel firing. 5 models, 80-180 M/Btu.

Oil or Gas Square-Cased Boilers



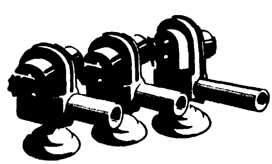
Round, dry-bottom, fully water legged boilers. Vertical corrugated plate-type flues for large heat transfer area in minimum space. Stainless steel combustion chamber. *SBI* rated; built to *ASME* code. 18 models to 1440 sq ft.

Gas and Oil Water Heaters



Automatic gas water heaters have spiral baffled internal flue. Heavy gage "Nor steel" tank. Raised-port cast-iron burner. Oil-fired heaters have vaporizing burner. 10-yr., 5-yr., 1-yr. warranties. 10 gas, 3 oil models; 20–66 gal.

Conversion Oil Burners



Dependable performance at minimum cost. Spinner for proper mixing of air and atomized oil; choke to govern angle of air mixture with oil spray; motor fan coupling for quiet operation. Easy to install and service. 8 models, 8-6.0 gph.

Rheem Manufacturing Company

570 Lexington Avenue, **New York 22, N. Y.**

Regional Offices:

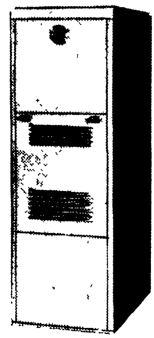
4361 Firestone Blvd., South Gate, Calif.
1025 Lockwood Drive, Houston 20, Texas
7600 S Kedzie Ave., Chicago 29, Ill.
29-28 41st Ave., Long Island City, N Y.
Sparrows Point 19, Md.

Every Rheem Furnace is given a 48-point test with pilot and burners ignited. Automatic controls, safety checks, operations and construction details must be 100 per cent.

GAS WARM-AIR HEATING EQUIPMENT

RHEEM GAS-FIRED WINTER-AIR CONDITIONER

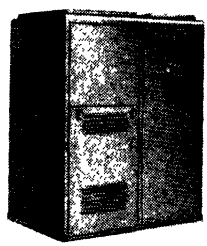
Series 3202—Highboy model, completely automatic, filtered and blower-circulated warmth. Electrically welded firebox is curved to eliminate expansion noises. Oversize blower is dynamically balanced for quiet, large capacity operation. Shipped completely assembled. Uses any type gas. Available with front vent. Easily installed in closet, utility room or basement.



Series 3202

Series 3402—*Lowboy* winter-air conditioner illustrates completeness of the Rheem line which embraces a model and size for any home. Vibration-free blower—motor equipped with thermal overload protection, insures true rated capacity. Exclusive high-efficiency sloped burner assures quiet extinction. Automatic shut off controls.

Series 3200—For large homes, stores, restaurants etc. Fully automatic; low-cost operation. Smart, graytone enamel jacket. Fully *A.G.A.* approved. Specified by the nation's leading architects and builders.



Series 3402 Lowboy

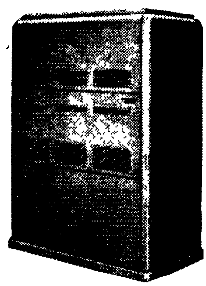
RHEEM GAS-FIRED GRAVITY FURNACE

Series 3300—Built for low-cost and economical operation. Electrically welded, air-tight, heating elements; no fumes can get into air stream. Corrugated liner prevents overheating and expansion noises. Large circulation area assures abundant warm-air flow. Automatic temperature controls. Humidifier available. For use with all gases.

RHEEM GAS-FIRED SPACE HEATERS

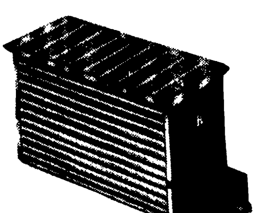
Series 1600—Rheem has developed these sturdy furnaces with the twin objectives of achieving the greatest possible heat output from units of the smallest practical size. No pit is required. Floor and wall furnaces are installed from floor level, thus saving costly construction details.

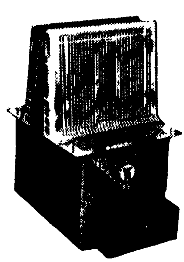
Gas tight, fumeless operation. Contours of die-formed steel heating elements keep warm-air flow at a maximum while baffles inside elements slow the flow of hot gases until all usable heat is transferred to the air. Interlocking safety valve. *A.G.A.* approved.

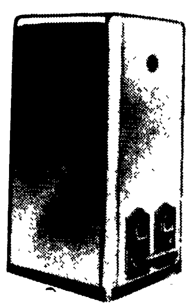


Series 3200

Series 1600







Series 3300

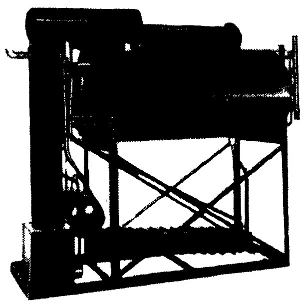
REGIONAL OFFICES
NEW YORK - ATLANTA
CHICAGO - DALLAS
LOS ANGELES

Servel Air Conditioning Division

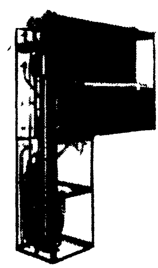
Evansville 20, Indiana

Air Conditioning equipment for residential, commercial and industrial applications

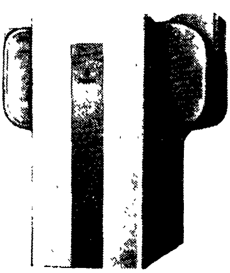
SERVEL ABSORPTION TYPE units are hermetically sealed, and operate without moving parts. Water is the refrigerant, lithium bromide the absorbent, and steam the source of energy. The most economical steam source may be used to operate these units. They may be operated at partial capacity without use of complicated controls. They may be installed in compact groupings in any convenient building location because the units are quiet and vibrationless.



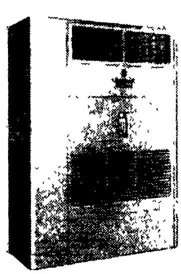
SERVEL 25-TON WATER CHILLER MODEL DUT. An economical source of chilled water for industrial processing and of air conditioning for office buildings, hospitals, factories and defense plants. 25-ton capacity *ASRE* rating.



SERVEL DIRECT EXPANSION UNIT 3 and 5-ton nominal capacities. May be used separately or in combination with other parts of SERVEL Air Conditioning equipment under "Incremental Assembly Plan."



SERVEL "ALL-YEAR" AIR CONDITIONER MODEL DE
Delivers 5-ton of refrigeration with 96,000 or 144,000 Btu/hr of heating. Operates on existing steam source or comes complete with gas or oil boiler and controls.



SERVEL "SELF-CONTAINED" AIR CONDITIONER MODEL SDE
Provides 5-tons of refrigeration. Heating can be added as optional equipment. Steam from any source supplies the energy for this unit.

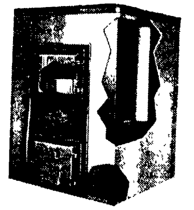
"CORROSIVE MASTER" EVAPORATIVE WATER COOLERS
Available for use with 3 and 5-ton SERVEL units.

SERVEL 5-YEAR WARRANTY—All SERVEL absorption equipment carries a 5-year factory warranty.

The Waterman-Waterbury Co.

Minneapolis 13, Minnesota

Manufacturers of WATERBURY Coal, Oil and Gas Fired Furnaces and Air Conditioning Equipment

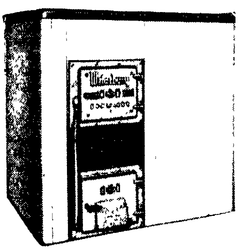


WATERBURY COAL-FIRED SEAMLESS FURNACE

Gas-tight, welded steel furnace body. A large combustion chamber and radiator with long fire travel insure efficiency and economy.

Air Conditioners for homes and larger buildings.

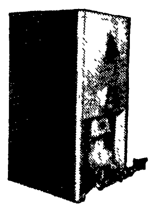
Size	Output BTU Per Hr.	Body Dia.	Body Hgt.	Round Galv. Dia.	Round Galv. Hgt.	Sq. in. Leader Pipe	Size Sq. Casing	Size Sq. Casing Outlet	Sq. Casing Hgt.	Heating Surface Sq. Ft.	Flue Pipe Dia.
A120	85,000	20″	52″	41″	65″	470	38x38	34x34	52″	35.7	8″
A122	94,200	22	52	43	65	524	40x40	36x36	52	39.6	8
A124	105,100	24	52	45	65	586	42x42	38x38	52	44.3	9
A127	136,500	27	58	49	72	758	46x46	42x42	59	57.4	9
A130	180,000	30	58	55	70	1000	53x53	49x49	57½	76.0	9
A133	225,000	33	66	58	80	1230	55x55	51x51	67	94.0	10
836R	265,000	36	63	58	80	1170	58x58	54x54	67	112.0	10



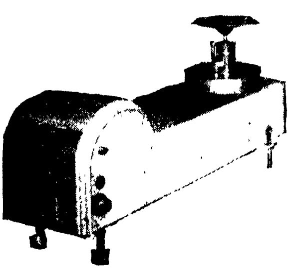
Specifications Waterbury Coal-Fired Air Conditioner

Size	Output BTU Per Hr.	Width Casing	Length Casing	Height Casing	Size Outlet Opening	Size Inlet Opening	No. of Filters	Size Each Filter	CFM Range
A120-10	95,400	38″	50″	52″	28x34	16x34	2	16x20	600–1400
A122-10	105,800	40	54	52	28x36	20x36	2	20x25	600–1400
A124-12	118,200	42	60	52	34x38	20x38	2	20x25	1000–2400
A127-15	153,000	46	62	59	34x42	22x42	2	20x25	1600–3200
A130-18	205,000	53	93	57½	49x49	36x36	4	16x25	2800–4800
A133-22	250,000	55	99	67	51x51	40x40	4	20x25	4000–6500
836R-22	300,000	58	102	67	51x51	40x40	4	20x25	4000–6500

Specifications Waterbury Gas-Fired Gravity Furnace



Size	Input Rating BTU Per Hr.	Output BTU Per Hr.	Width Casing	Length Casing	Height Casing	Size Outlet Opening	Flue Pipe Dia	Size Gas Connection Required
6413A	80,000	60,000	28″	28″	48″	24x21	5″	¾″
6415A	110,000	82,500	30	30	53	26x26	6	¾
6418	130,000	97,500	40	37	53	36x33	7	¾



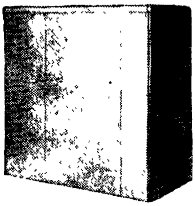
Waterbury Gas Conversion Burner

A real gas burner, the same as used in the Gas Furnace and Air Conditioner.

Number	Maximum BTU Input	Maximum Cubic Feet of Mixed Gas	Maximum Cubic Feet of Manuf'd Gas
G-200	200,000	250	360

The Waterman-Waterbury Co.

Minneapolis 13 Minnesota



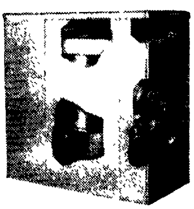
WATERBURY GAS-FIRED AIR CONDITIONER

Designed specifically for most efficient use of gas. Enclosed in the compact, baked enamel casing, it is completely automatic, providing filtered, humidified forced air. Can also be furnished for liquidified petroleum gas.

Size	Input Rating BTU Per Hr.	Output BTU Per Hr.	CFM Range	Width Casing	Length Casing	Height Casing	Size Outlet Opening	Size Inlet Opening	No. of Filters	Size Each Filter	Flue Pipe Dia.	Size Gas Connec. Required
6412-7*	60,000	48,000	250 to 550	22″	42″	42″	18x18	18x12	1	16x25	5″	½″
6413A-9*	90,000	72,000	400 to 1000	28	59	48	24x23	24x16	2	16x25	5	¾
6415A-10*	120,000	96,000	600 to 1400	28	61	53	24x23	24x18	2	16x25	6	¾
6418A-12*	150,000	120,000	1000 to 2400	34	71	53	30x30	22x30	2	20x25	7	¾
6420-15	185,000	147,000	1600 to 3200	36	60	53½	32x32	32x22	2	20x25	8	1

* Also available as a Hi-Boy.

WATERBURY OIL-FIRED AIR CONDITIONER 6300 SERIES

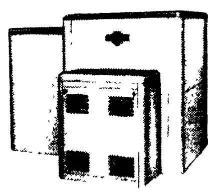


Completely automatic oil heat, plus the equipment to provide forced, filtered, humidified air, all enclosed in one compact casing finished in baked enamel.

Size	Input Rating Gal. Oil Per Hr	Output BTU Per Hr.	Width Casing	Length Casing	Height Casing	Size Outlet Opening	Size Inlet Opening	No. of Filters	Size Each Filter	Flue Pipe Dia.
6313A-9*	.75	84,000	28″	59″	48″	24x23	21x16	2	16x25	6″
6315A-10*	1.00	112,000	28	61	53	24x23	24x18	2	16x25	7
6318-12*	1.35	151,000	34	71	53	30x30	22x30	2	20x25	7
6320-15	1.50	168,000	36	60	53½	32x32	32x22	2	20x25	8

* Also available as a Hi-Boy.

WATERBURY OIL-FIRED AIR CONDITIONER B 300 SERIES



Designed especially for large installations, the B 300 Oil-Fired Air Conditioner has a unique furnace body. It features an extra large radiator with a baffle for unusually long fire travel, thus assuring efficient use of the fuel.

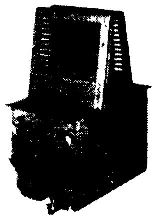
Size	Rating Gal. Oil Per Hr.	Output BTU Per Hr.	Width Less Front Hood	Length Casing	Height Casing	Size Outlet Opening	Size Inlet Opening	No. of Filters	Size Each Filter	Flue Pipe Dia.	CFM Range
B322-15	2.0	224,000	44″	72″	55″	40x40	24x28	2	20x25	8″	1600 to 3200
B324-18	2.5	280,000	48	88	55	44x44	36x36	4	16x25	9	2800 to 4000
B327-18	3.0	336,000	52	92	62	48x48	36x36	4	16x25	9	2800 to 4800
B330-22	3.5	392,000	56	100	62	52x52	40x40	4	20x25	9	4000 to 6500
B333-25	4.0	448,000	60	108	70	56x56	44x44	4	20x25	10	5000 to 8000
B336-25	4.5	504,000	66	114	70	62x62	44x44	4	20x25	10	7000 to 10,000

Utility Appliance Corporation

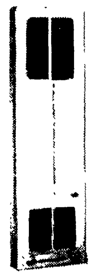
4851 S. Alameda St., **Los Angeles 58, Calif.** Cable address: UTILIFAN, Los Angeles

Utility Gas-Fired Heating Equipment, Evaporative Air Coolers and Blowers

Forced Air Furnaces



Floor Furnaces (Vented) Flat and Dual

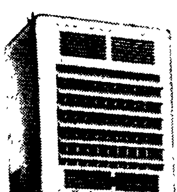


Utility Wall Heaters

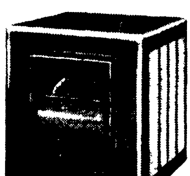
FORCED AIR FURNACES. Compact design permits use in basement or closet . . . dynamically-balanced blower, resilient mount . . . Fiberglass Dustop filters. . . heavy gauge steel, die-formed and welded . . . baked enamel finish. Models: from 70,000 to 200,000 Btu inclusive.

FLOOR FURNACES (Vented) FLAT and DUAL. Designed for quiet, efficient, trouble-free, long-life, satisfactory service. "FULL FLOW" performance. Highest quality materials. Guaranteed! Four basic models available—45,000 Btu and 60,000 Btu Floor and Dual Register. Controls: (1) Manual, with manual pilot; (2) Manual, with 100 per cent safety pilot; (3) Automatic (self-generating type) with safety pilot.

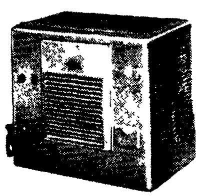
UTILITY WALL HEATERS. All units fit standard 4 in. stud walls without furring and are adjustable for finished wall thicknesses ranging from 4⅜ in.–5¼ in. Single and dual models. Optional Thermostat or 3-rate manual control. Vented. Models: from 27,500 to 50,000 Btu inclusive.



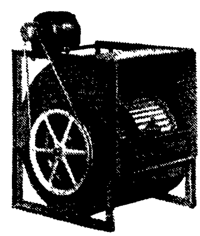
Unit Heaters



Evaporative Air Coolers



Evaporative Air Cooler Portable Room Model 1,000 P



Standard Blowers

UNIT HEATERS. Suspended type. Complete unit, burner, heat exchanger, draft diverter, motor, fan and all other parts housed in steel cabinet, baked enamel finish. Adjustable grilles. 65,000–90,000–150,000–225,000-Btu models.

EVAPORATIVE AIR COOLERS. Comfort cooling . . . residential, commercial, industrial . . . dynamically-balanced . . . centrifugal blower . . . uniform water distribution. "No-Sag" aspen-fiber pads. 24 cooling models 800 cfm–13,000 cfm.

EVAPORATIVE AIR COOLER PORTABLE ROOM MODEL 1,000 P. For home, office or trailer this portable model with 4-Way directional discharge grill is very popular. Equipped with Direct Drive Blower, Belt Drive Pump and Water Level Indicator. Seal Bonded against rust.

STANDARD BLOWERS. Dynamically-balanced, multiple-vane centrifugal blowers. Four-side angle iron frame increases rigidity, eliminates vibration . . . permits installation with any of four discharge positions. 9 in. dia.–26 in. dia. . . . Single and Double Width. Guaranteed Air Deliveries.

Utility Blowers and Coolers are tested in accordance with the A.S.H.V.E. Code

Utility heating appliances are *A.G.A.* approved.

Write for complete information, catalogs and price.

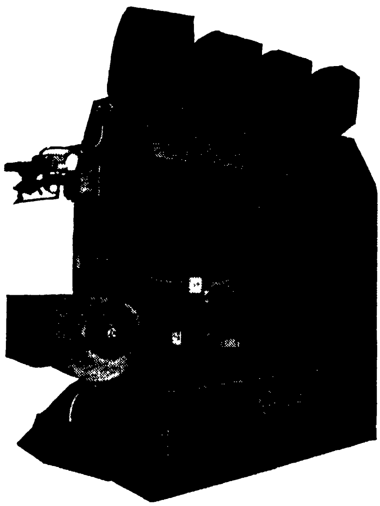
Airtherm Manufacturing Company

728 S. Spring Ave.

St. Louis 10, Mo.

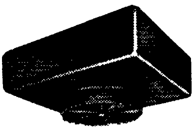
Representatives in Principal Cities

For Heating Satisfaction—Think first of AIRTHERM

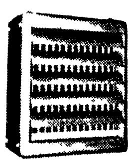


AIRTHERM Gas or Oil Fired Space Heaters

A complete heating, and heating and ventilating unit. Equipped with motors, burner and all safety and operating controls. Easy to install and operate Heavy duty steel combustion chamber lined with castable refractory. Capacities from 650,000 to 2,000,000 Btu per hour. Write for Catalog 802-A.



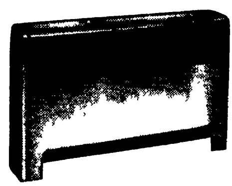
Vertical



Horizontal

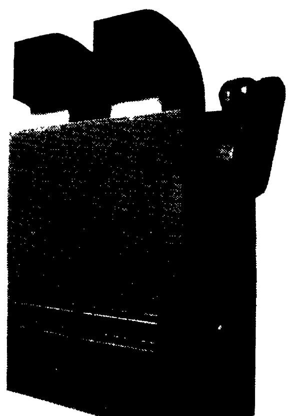
AIRTHERM Steam Unit Heaters, Horizontal and Vertical Discharge

Horizontal models are produced in capacities from 18,000 to 270,000 Btu. Vertical models 32,000 to 350,000 Btu per hour. Write for Catalog 1208-B.



AIRTHERM Convectors

Airtherm Convectors are designed for easy installation and outstanding performance. Copper coil for all hot water and two pipe steam systems. Produced in full range of types and sizes. Write for Catalog 702-A



AIRTHERM Centrifugal Fan Unit Heaters

For all types of heating, and heating and ventilating installations, in industrial and commercial buildings auditoriums, gymnasiums, schools. Available with dampers, filters and non-freeze coils. Capacities up to 1,440,000 Btu per hour. Write for Catalog 402-A.

Campbell Heating Company

3121 Dean, Des Moines 17, Iowa

DIRECT FIRED SPACE HEATERS

Oil, Gas, Stoker, Hand Fired or Combination Gas and Oil Fired

Heater Guaranteed for 10 Years from Any Cause, Blower, Motor, Burner and Controls, for One Year Against Defects.

The Campbell Direct Fired Space Heaters are designed to be located in the room to be heated but they can be connected to a duct system for heating any type of building and are quiet enough to be used in churches, schools, etc.

They are designed to be shipped as a complete unit ready to connect to fuel line, electric power and flue but they can be shipped in sections for assembling in a basement room. The steel heater is then assembled and welded on the job.

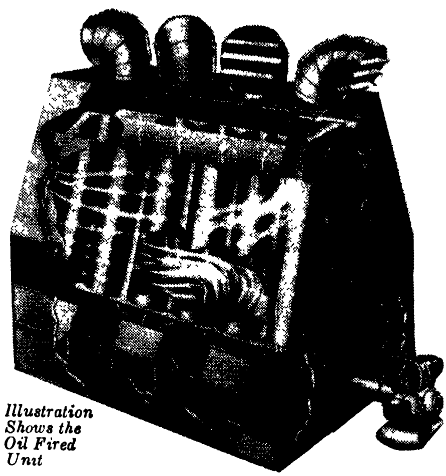


Illustration Shows the Oil Fired Unit

Furnished also for Gas, Stoker or Hand Firing

Unit Number	Output Capacity Btu per hr. (1)	65° Return Air—No Ducts: CFM for 140° Outlet Temp. (2)	65° Return Air—No Ducts: Motor HP (3)	Heating Surface SqFt (4)	Overall Dimensions Inches Not Incl. Burner: W	L	Ht.	Stack Connection Dia. In. (5)	Firing Rates: Oil GPH	Firing Rates: 1,000 Btu GAS C.F.H.	Firing Rates: Stoker Lbs/Hr	Approx. Shipping Weight
U8075	725,000	8,850	2-½ HP	280	80	80	114	12	7.0	900	80	5,000
U8100	893,000	10,900	2-½ HP	320	80	93	118	14	8.5	1,120	100	6,000
U8125	1,080,000	13,200	2-¾ HP	360	80	105	120	16	10.5	1,350	125	7,000
U8150	1,275,000	15,600	2-1 HP	440	80	118	120	16	12.0	1,600	150	8,000
U8175	1,440,000	17,500	2-1 HP	480	80	130	120	18	14.0	1,800	175	10,000
U8200	1,725,000	21,100	2-1 HP	600	94	137	136	20	16.5	2,150	200	11,000
U8250	2,160,000	26,400	2-1½ HP	750	94	157	136	20	20.5	2,700	250	13,000

(1) Allow 10 to 35 per cent above heat loss for pick up load. If heater is not located in room to be heated an allowance for duct and other radiation losses should be made of 10 to 15 per cent above heat loss figures.

(2) Rated air volume produces 75 deg air temperature rise through the heater. This air volume can be varied to suit other conditions.

(3) Motor size can be increased to provide for any duct system.

(4) Heat Emission per square foot of heating surface is 3000 Btu per hour or less.

(5) An induced draft blower can be furnished if a stack or chimney is not available. The gas passages of the heater are adequate so that an induced draft blower is not ordinarily necessary. Stack connection can be provided through the top of the casing to support a stack through the roof, or out the front as illustrated.

For data on other Campbell Products see pages 1124–1125

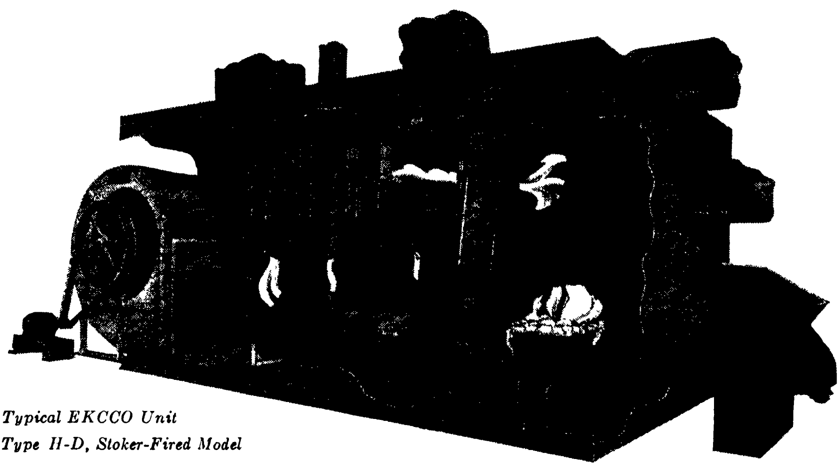
DIRECTOR, EASTERN SALES: NEIL ADAMS, OLD YORK ROAD, LAMBERTVILLE, N. J. TELEPHONE No. 665

E. K. Campbell Company

Kansas City 3, Missouri

HEAVY DUTY FURNACE FAN EQUIPMENT

Standard Units Up To 8,000,000 BTU/HR



Typical EKCCO Unit
Type H-D, Stoker-Fired Model

- Made in Units from 500,000 Btu/hr to 8,000,000 Btu/hr.
- Available for any fuel or in "All-Fuels" model.
- Operating efficiencies up to 84 per cent with resultant fuel economy.
- Counter-Flow heat transfer.
- Drive thru principle exclusively (no suction applications).
- Low Internal resistance to flue gases.
- Fiberglas insulated casing.
- Baked enamel exterior finish.
- Extra Heavy welded steel construction throughout.
- Available in any required arrangement of duct outlets.
- Extreme flexibility of equipment arrangements.
- Balanced job design—blower and furnace sized separately.
- Sold only on an engineered basis to fit job requirements.

Used in thousands of large buildings over country, the E. K. Campbell Co's Type H-D Furnace-Fan system is particularly suited for buildings containing large spaces, such as industrial plants, churches, schools, hangars, etc. High quality equipment, designed to last, giving unusually low cost on a year service basis.

The E. K. Campbell Company guarantees RESULTS as well as its equipment.
Inquiries invited regarding LARGE SPACE HEATING PROBLEMS.

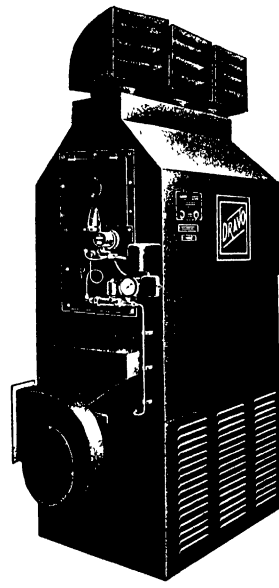
Manufacturing Engineers since 1910

DRAVO CORPORATION

HEATING DEPARTMENT

Dravo Bldg., Fifth and Liberty Avenues
PITTSBURGH 22, PA.

DRAVO *Counterflo* DIRECT-FIRED HEATERS



Gas-Fired Dravo Counterflo Direct-Fired Heater

Offer All these Important Advantages:

●**LOW FIRST COST** . . . Users report savings of 30 to 60 per cent compared with standard wet-type systems.

●**CONSERVES STEEL** . . . 50 to 70 per cent less steel required when using direct-fired hot air heating method with Dravo ***Counterflo*** Heaters.

●**WORKING-ZONE WARMTH** . . . . . . Units heat areas of 4,000 to 20,000 sq ft each. Warmth is concentrated in working zone; roof heat loss greatly reduced.

●**NO FUEL WORRIES** . . Burn gas or oil—readily converted from one to other.

●**LOW OPERATING COST** . . . 80-85 per cent efficiency at bonnet *plus* top efficiency in heat distribution holds cost down.

●**AUTOMATICALLY CONTROLLED** . . . On-off or modulating controls. The heater looks after itself—no continuous attention required.

●**EASY INSTALLATION** . . . Just hook up fuel, electric and exhaust connections —the heater is ready to go.

●**STAINLESS STEEL CHAMBER** . . Rugged mill-type construction, top-quality engineering, assure durability.

●**5 FUNCTION SERVICE** . . . Each unit can provide not only comfort heating, but year-around ventilating, tempered make-up air, process drying, and heat curing.

TESTED-APPROVED . . . *A.G.A.* and/or *UL* seal on all standard units. Each heater flame-tested at factory before shipment.

CAPACITY AND DIMENSION TABLE . . . Notes· A—CAPACITIES. *(1) Gas capacities based on Natural Gas having 1000 Btu/cu ft and 80 per cent efficiency. (2) Light oil capacities based on oil having 140,000*

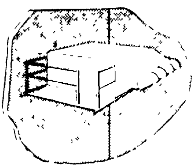
Model Number	Btu Output Capacity per hour	Approximate Fuel Burned			Equivalent Sq. Ft. Steam Radiation	Approx. C.F.M. Air at 70°F.
		SGA Cu. Ft. Nat. Gas per hr.	LO Gals. Light Oil per hr.	HO Gals. Heavy Oil per hr.		
40	400,000	500	3.6	3.5	1,670	4,500
50	500,000	625	4.5	4.3	2,080	5,500
75	750,000	940	6.7	6.5	3,130	8,500
100	1,000,000	1,250	8.9	8.6	4,170	11,000
125	1,250,000	1,560	11.1	10.8	5,210	14,000
150	1,500,000	1,875	13.4	13.0	6,250	17,000
175	1,750,000	2,190	15.6	15.0	7,280	19,000
200	2,000,000	2,500	17.8	17.2	8,330	22,000

DRAVO CORPORATION

•ATLANTA •BOSTON •CHICAGO •CLEVELAND
•DETROIT •NEW YORK •PHILADELPHIA •PITTSBURGH

Sales Representatives in Principal Cities

Manufactured and sold in Canada by Marine Industries, Ltd., Sorel, Quebec
Export Associates: Lynch, Wilde & Co., Washington 9, D. C.



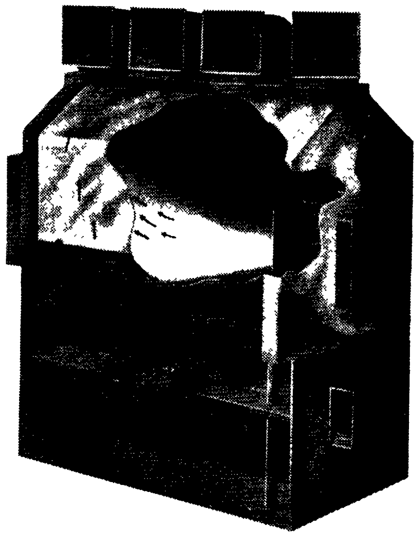
FLEXIBILITY OF APPLICATION

Although the Dravo ***Counterflo*** Heater attains its greatest space heating efficiency when operating on the floor, it can be suspended horizontally or inverted when floor space is not available. Because of its sturdy construction and the absence of fragile refractory, it can be placed in practically any conceivable position.

FIVE FUNCTION SERVICE WITH ONE UNIT:

- **COMFORT HEATING**—fast, economical heating of entire working-zone provides ideal comfort conditions.
- **YEAR-ROUND VENTILATION**—maintains positive circulation of clean, fresh air—improves hot weather plant conditions.
- **TEMPERED MAKE-UP AIR**—ideal for foundries, paint spray booths and similar applications; replaces contaminated air with fresh tempered make-up air.
- **PROCESS DRYING**—process applications include drying of ceramics, paint, hay, rugs, rubber toys, and many other items.
- **HEAT CURING**—the efficient design permits use for low temperature curing.

Write for bulletin HVG-523-A and specification sheets.



Cutaway view of heater

Btu/gal and 80 per cent efficieny. (3) Heavy oil capacities based on oil having 145,000 Btu/gal and 80 per cent efficiency B--Recommend a minimum height of stack 3 ft above peak of roof.

Fan Motor H.P.	Approx. Shipping Weight	Overall Heater Dimensions				Stack Diameter
		Width	Length	Height		
				With Nozzles	Without Nozzles	
1½	1,400	2'7"	4'11"	8'1"	7'0"	8"
2	1,500	2'7"	4'11"	8'1"	7'0"	8"
3	2,500	3'8"	7'3"	10'0"	8'8"	10"
5	2,600	3'8"	7'3"	10'0"	8'8"	10"
7½	3,400	4'3"	8'11"	11'2"	9'8"	12"
10	3,600	4'3"	8'11"	11'2"	9'8"	12"
15	4,200	4'9"	9'5"	12'9"	10'11"	12"
20	4,400	4'9"	9'5"	12'9"	10'11"	12"

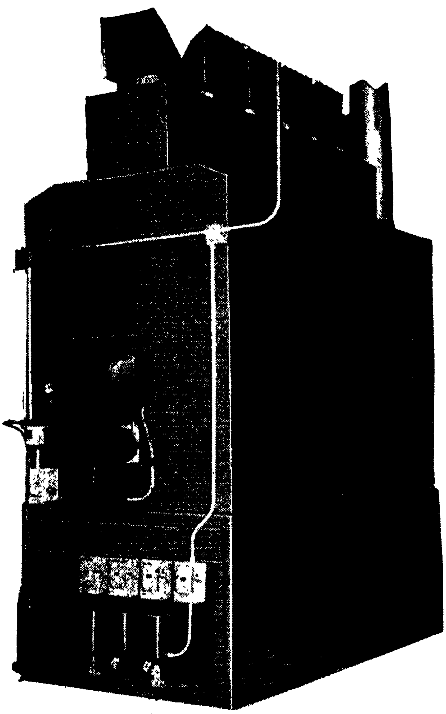
Chicago Steel Furnace Co.

9326 S. Anthony Ave., **Chicago 17, Ill.**

Manufacturers and Designers
Artcraft Heating & Air Conditioning Equipment

DIRECT-FIRED SPACE HEATERS

350,000 Btu to 2,000,000 Btu



Fired with Gas or Oil

UNIFORM AIR DELIVERY

BLOWERS—Three standard up-blast discharge, multi-vane Blowers, each individually powered, are provided to supply the correct amount of air to properly cool the Heat Exchanger. Individually-driven Blowers are used because of the great flexibility of operation obtainable. Uniform bonnet temperature may be enjoyed by the simple expediency of increasing or reducing the amount of air driven around any particular section of the Heat Exchanger.

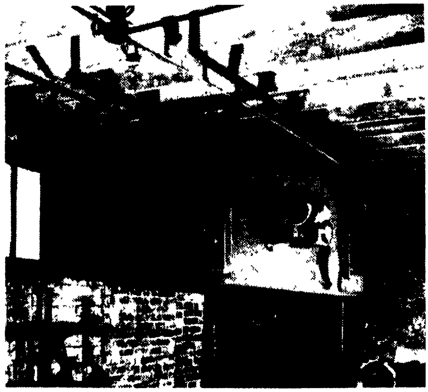
Performance Data
Direct Fired Model A Units

MODEL	OUTPUT BTU	AIR DEL. CFM	L	W	H
350-A	350,000	3000-4200	72	36	84
500-A	500,000	6000-7800	92	42	90
750-A	750,000	8200-9600	100	50	99
1000-A	1,000,000	12500 15500	114	54	110
1250-A	1,250,000	15000-17500	124	56	114
1500-A	1,500,000	17500-21000	128	62	115
2000-A	2,000,000	21000-24000	144	66	125

Ratings and Specifications
Suspended Units

MODEL	OUTPUT BTU	AIR DEL CFM-¼"S.P.	L	W	H
75-S	85,000	800-1000	63	24	34
75-SL	85,000	800-1000	78	22	22
100-S	100,000	1000-1600	76	24	34
150-S	150,000	1200-2000	79	31	36
200-S	200,000	2200-2600	86	31	40
250-S	250,000	2400-3200	90	35	42
350-S	350,000	3000-4200	104	38	53
400-S	400,000	4200-5200	107	40	53
500-S	500,000	4600-6200	114	45	62
750-S	750,000	8200-9600	146	52	66

ARTCRAFT SUSPENDED UNITS



Factory Installation Model 200S-2

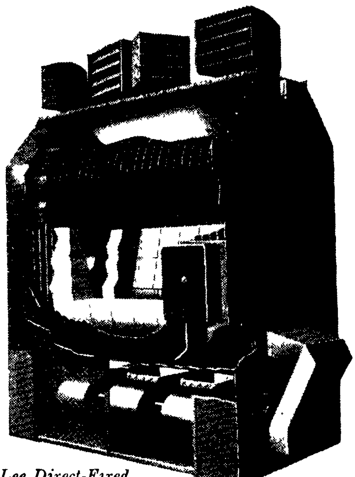
Lee Corporation

1001 Tatnall Street, **Wilmington, Delaware**

Manufacturers and Designers of Complete Line of Industrial and Commercial Warm Air Heaters

Representatives in all Principal Cities

Exclusive Representation to the Railroads: Railroad Supply and Equipment, Inc., First Federal Bldg., Scranton 3, Pa. In Canada: Comstock Lee Heating Division, Canadian Comstock Company, 206 Laird Drive North, Leaside, Toronto, Canada



Lee Direct-Fired Space Heater Model hd

LEE HEAVY DUTY HEATERS

LEE HEATERS have been in continuous operation since 1918. Lee Heater installations have been made in most of the States of the United States, in Canada, Alaska, Yukon Territory, Northwest Territory, Labrador, Iceland, Norway, Mexico and South America.

Firing Equipment manufactured specially for us by leading Companies with International service organization. There is a Service man near you if and when you require assistance. In addition Lee maintains a competent field serving staff.

Replacement and Spare Parts Service is maintained. Most items are carried in stock.

Lee Heater Number	300	400	500	600	750
Output Capacity (Btu per hr)	300,000	400,000	500,000	600,000	750,000
Total Heating Surface in sq ft	92	106	126	202	235
Cu ft of Air per Min. (Approx.)*	3500	4500	6000	7000	8500
Fan Motor HP	1	1½	2	3	5
External Static Pressure (Std.)	¼"	¼"	¼"	¼"	¼"
Maximum External Static Press Available for Duct Work (Larger Motor Required)	1"	1"	1"	1"	1"
Temperature Rise (Degree F)	79°	82°	77°	79°	82°
Fuel Consumption**					
A-cu ft Gas per hr (App.)	380	500	630	750	940
B-gal Oil per hr (Approx.)	2.6	3.5	4.3	5.2	6.5
Shipping Weight (Approx.) Lbs					
Refractory Comb. Chamber	3000	3100	3300	4500	4700
Stainless Steel Comb. Chamber	2100	2200	2400	4000	4200

Lee Heater Number	1,000	1250	1500	1750	2000
Output Capacity (Btu per hr)	1,000,000	1,250,000	1,500,000	1,750,000	2,000,000
Total Heating Surface in sq ft	278	333	375	423	470
Cu ft of Air per min. (Approx.)*	11,500	14,000	17,000	20,000	22,000
Fan Motor hp	5	7½	10	15	15
External Static Pressure (Std.)	½"	¼"	¼"	½"	¼"
Maximum External Static Press. Available for Duct Work (Larger Motor Required)	1"	1"	1"	¾"	¾"
Temperature Rise (Degree F)	80°	83°	82°	81°	84°
Fuel Consumption**					
A-cu ft Gas per hr. (App.)	1250	1560	1880	2190	2500
B-gal Oil per hr. (Approx.)	8.6	10.8	13.0	15.0	17.2
Shipping Weight (Approx.) Lbs					
Refractory Comb. Chamber	6000	6300	7500	8600	9800
Stainless Steel Comb. Chamber	5200	5500	6200	7400	8600

* Based on Standard Air at 70 F Free Delivery.

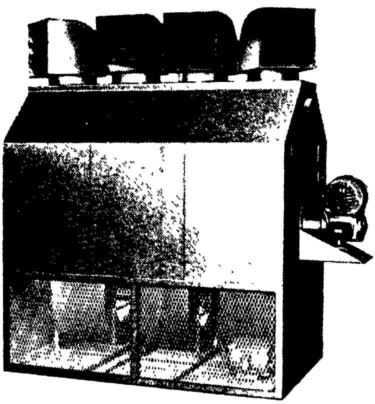
** Fuel Consumption Based on 1000 Btu per cu ft Gas, 145,000 Btu per Gal Oil, and on 80 per cent Efficiency for the heater.

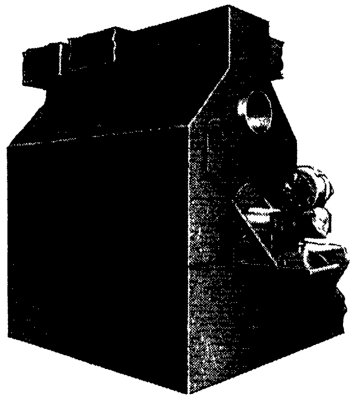
National Heater Company

NATIONAL Champion DIRECT FIRED

2182 Cleora Avenue **St. Paul 4, Minnesota**

OIL, GAS and COMBINATION GAS-OIL FIRED UNITS 200,000 to 1,500,000 Btu

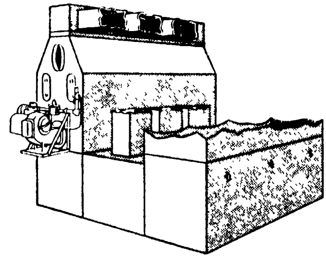




MODEL "H"—Fully self-contained, automatically operated, direct fired unit heaters. Provides working level warm air circulation for factories, foundries, warehouses, garages, hangers and terminals.

MODEL "D"—Meets specific requirements of large central heating systems necessary in theaters, auditoriums, schools, churches, offices, and super markets. Multiple oversize blowers insure positive air delivery against resistance of long supply and return duct runs.

MODEL "S"—Similar to Model "D," equipped with side or rear mounted blower cabinets, these styles are particularly adapted to modulating type of face and by-pass temperature control systems and where head room is limited.

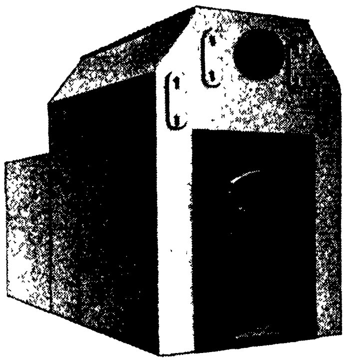


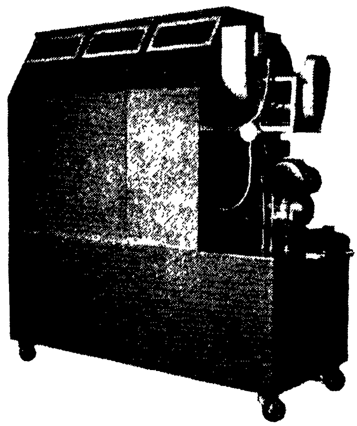
Heater Number	Btu Output Capacity	CFM at ⅝ in. S.P.	HP Blower Motor	Overall Dimensions			Smoke Outlet	Shipping Weight
				Width	Length	Height		
TD-20	200,000	2,400	½	28	52	66	10"	950
TD-25	250,000	3,200	¾	28	52	66	10"	1020
TD-30	300,000	4,000	¾	32	60	81	10"	1250
TD-40	400,000	5,400	1	32	60	81	10"	1360
TD-50	500,000	6,600	1½	32	80	81	12"	1600
TD-70	750,000	8,800	2	32	80	81	12"	1850
TD-80	800,000	10,200	3	48	80	81	12"	2310
TD-100	1,000,000	12,500	5	48	80	81	12"	2680
TD-125	1,250,000	15,300	5	54	100	103	14"	3155
TD-150	1,500,000	19,400	7½	54	100	103	14"	3340

National Heater Company

NATIONAL Champion DIRECT FIRED

2182 Cleora Avenue St. Paul 4, Minnesota





HEAVY DUTY "ALL FUELS" HEATERS

A durable, welded, one-piece steel constructed heater suitable for coal, oil or gas firing and easily converted from one fuel to another. Designed chiefly for central heating systems, drying, and make up air applications. Large firebox with ample flue travel ensures complete fuel combustion with minimum stack loss—giving an unusually high efficiency factor whether stoker, oil or gas fired. Provisions made for relief of internal stresses due to repeated expansion and contraction. Rigid panel type insulated casings are shaped to contour of heaters. Fan capacities listed provide an average 75 deg temperature rise through heater. Blowers or blower-filter-cabinet units can be sized to meet exact requirements of each installation.

PORTABLE HEATING UNIT

A "plug in" portable heating system, self-contained and designed for immediate heat delivery when and where needed. Shipped fully assembled with induced draft blower and built-in oil storage tank, if desired. Smaller models are 28 in. and 32 in. wide. Wired and flame tested at factory.

Heater No.	BTU	CFM	Motor HP	Shipping Wt.
P-200	200,000	3,000	½	840
P-300A	300,000	4,200	¾	950
P-300	300,000	5,000	1	1400
P-500	500,000	7,000	2	1600
P-800	800,000	10,500	3	2060
P-1200	1,200,000	15,000	5	2900
P-1800	1,800,000	23,000	7½	3800

Heater Number	Btu Output Capacity	Sq Ft Heating Surface	CFM at ⅝ in. S.P.	HP Blower Motor	Overall Dimensions Length	Width	Height	Blower	Smoke Outlet	Weight
#904	400,000	138	5,400	1	64	38	69	50	10"	2,370
#905	500,000	184	6,600	1½	70	48	72	50	10"	2,640
#907	700,000	234	8,800	2	82	48	78	52	12"	3,020
#908	800,000	268	10,000	3	94	48	78	52	12"	3,280
#910	1,000,000	351	12,500	3	118	48	78	70	14"	3,730
#915	1,500,000	519	18,000	5	118	66	96	52	16"	4,880
#920	2,000,000	667	24,000	7½	142	66	96	70	16"	5,920

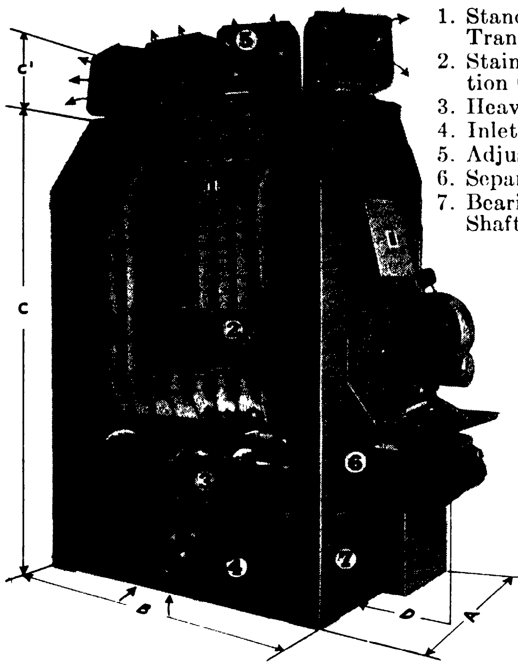
Arthur A. Olson & Company

Canfield, Ohio

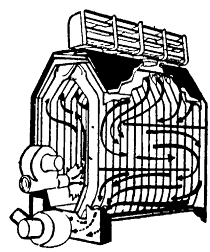
Manufacturers of Direct Fired Heaters

Olson

A completely automatic direct-fired unit heater. Olson heaters operate on gas, oil, dual gas-oil or coal. These units may be hung, suspended horizontally or inverted. All heaters can be adapted to filtering, humidifying and air conditioning Use of standard accessories and easy access to bearings and other vital parts simplify maintenance. Also special adaptions for make-up air, drying, high-temperature and process work.



1. Standard 10 Gage Boiler Tube Heat Transfer Surface.
2. Stainless Steel or Refractory Combustion Chamber.
3. Heavy Duty Fans and Shaft.
4. Inlet Screens or Frame for Filter Box.
5. Adjustable, Deflecting Outlets.
6. Separate Induced Draft Fan.
7. Bearings Outside Heater at End of Shaft.



Four Pass Gas Travel-Counter Current.

Engineering Data—Gas, Oil, or Dual Gas-Oil Heaters.

Heater Model Number	BTU/HR Output Capacity	CFM Blowing Fans	General Dimensions A	B	C	C1	D	No. of Blowing Fans	Motor HP Blowing Fans	Stack Dia.	No. of Nozzle Outlets
U-300	300,000	3,500	3′- 1″	2′-11″	7′-10″	1′-1″	1′- 6″	1	1	8″	2
U-400	400,000	4,500	3′- 1″	3′- 7″	7′-10″	1′-1″	1′- 6″	1	1½	8″	3
U-500	500,000	5,500	3′- 1″	4′- 3″	7′- 3″	1′-3″	1′- 7″	2	2	8″	3
U-600	600,000	7,000	3′- 1″	4′-11″	7′- 3″	1′-1″	1′- 7″	2	3	8″	4
U-750	750,000	8,500	3′- 8″	5′- 3″	8′- 9″	1′-1″	1′- 8″	2	3	8″	4
U-1000	1,000,000	11,000	3′- 8″	5′-11″	8′- 9″	1′-3″	1′-11″	2	5	10″	4
U-1250	1,250,000	14,000	4′- 4″	7′- 0″	9′- 9″	1′-5″	1′-11″	2	7½	10″	4
U-1500	1,500,000	17,000	4′- 4″	8′- 0″	9′- 9″	1′-5″	2′- 0″	2	10	12″	5
U-1750	1,750,000	20,000	4′-10″	8′- 0″	10′- 9″	1′-5″	2′- 1″	2	15	12″	5
U-2000	2,000,000	22,000	4′-10″	9′- 0″	10′- 9″	1′-5″	2′- 1″	2	15	12″	5

Write for complete data on stoker fired units and large central systems.

Automatic Gas Equipment Company

Brushton and Thomas St., **Pittsburgh 21, Pa.**

Manufacturers of Pittsburgh Gas Unit Heaters and Duct Furnaces

Cast Iron Heating Elements

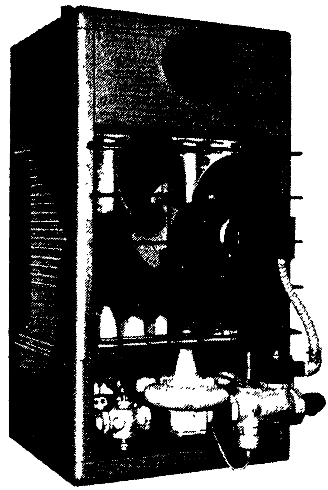
Cast iron is considered to be the best material to withstand the corrosive effects of the products from combustion of gases. For this reason, both the heat exchanger and combustion chamber in a Pittsburgh Gas Unit Heater are made of cast iron. Furthermore, they are cast in one piece and the extended heating surface fins on the heat exchanger are cast integral.

Pittsburgh Gas Unit Heaters have been designed to consume exactly the right amount of air to support complete combustion but without permitting an excess of air to lower heating efficiency. This is accomplished by means of a built in draft hood which absorbs all excessive chimney action and thereby conserves heat. The heater does not depend upon forced draft from the fan for either the primary or secondary air supply. For this reason there is no possibility of variation in the air supply to the burners resulting from changes in fan speed or louver adjustment. By the use of adjustable horizontal louvers, warm air can be directed to any desired level.

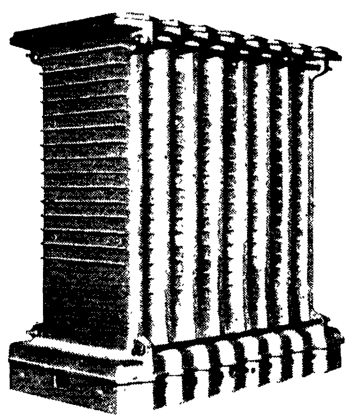
Safety Features

A tested and proved safety pilot is used on these heaters to automatically turn off the gas if the pilot light goes out or if it burns too low to insure positive ignition. The draft diverter is absolute protection against any possible down drafts through the chimney. Write for folder containing complete details, including installation measurements.

Approved by *American Gas Association and Underwriters' Laboratories.*



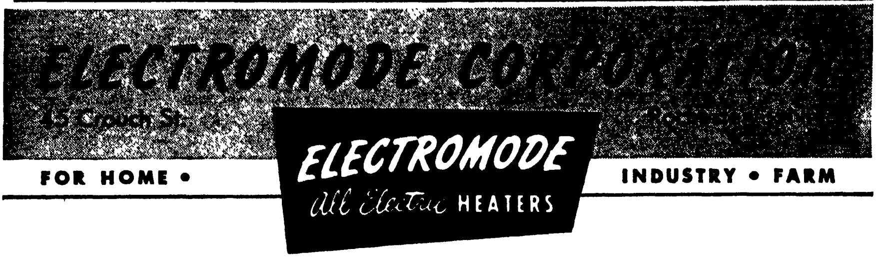
Cut-away view of heater, showing the cast iron heat exchanger in place.

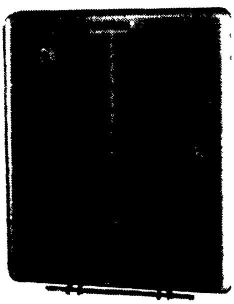


Heat exchanger and combustion chamber.

SIZES AND CAPACITIES (also made in five sizes in Blower Type Units)

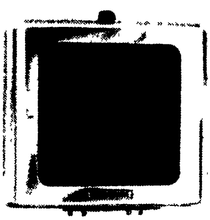
Unit No.	Input B.T.U. Per Hour	Output—AGA B.T.U. Per Hour	Sq. Ft. E.D.R.	Air Del. C.F.M.	Motor H.P.	Speed R.P.M.	Approx. Wts.
215 C	215,000	172,000	744	3500	1/4	1140	525
175 C	175,000	140,000	605	2900	1/6	1140	475
160 C	160,000	128,000	553	2600	1/8	860	450
140 C	140,000	112,000	484	2320	1/12	860	400
110 C	110,000	88,000	381	1820	1/20	860	350
85 C	85,000	68,000	294	1350	1/30	1000	300





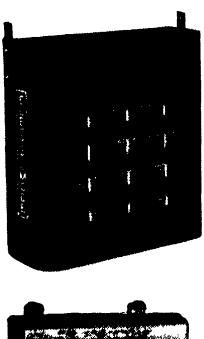
BILT-IN-WALL Heaters for Homes and Offices

Using Electromode's Down-Flo principle, these heaters fan-circulate warm air at floor level for greater comfort and improved heating efficiency. Quick and easy to install. No duct work required. Famous lifetime heating element eliminates all danger of fire, shock or burn. Capacities from 1500 to 4000 watts—with manual, wall thermostat or built-in thermostat control. All have thermal safety switches. Silver gray enamel finish.



BILT-IN-WALL BATHROOM Heater

Designed to build into the wall, this 115 volt, 1320 watt heater is designed for small rooms such as baths, bedrooms, kitchens, etc. Employs the famous lifetime heating element that is completely safe. Available with or without built-in thermostat. Thermal cut-off prevents overheating. Two-way switch permits use of fan without heat. Available in white baked-on enamel or chrome.

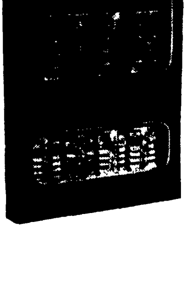


UNIT HEATERS for Auxiliary Warmth

These fan-circulating units are used as auxiliaries to a main heating system or, where electric rates permit, as main heating sources. Require no plumbing or duct work—only circuit wiring. Employ the completely safe lifetime heating element that has no exposed hot or glowing wires, gives high thermal conductivity and resists corrosion. Safety switch prevents overheating. Thermostat control available on all models.

Suspension Type (left) for wall or ceiling mounting in factories, stores, offices. 10,000 to 45,000 watts.

Combination Portable and Suspension Type plugs in where needed or mount permanently. 1500 to 7500 watts. Automatic thermostat control available on 1500 to 5000 watt models.



Explosion-Proof Heaters (left) for areas charged with inflammable vapors. Convection type. Entire electrical input provides useful heat. Three models range from 2000 to 6000 watts. Listed by *Underwriters' Laboratories* for Class 1-Group D Hazardous Industries.

Engineering help is at your disposal. For more information, see your supplier or write Dept. HVG-12, Electromode Corporation.

Electromodes are approved by *Underwriters' Laboratories* and are fully guaranteed.

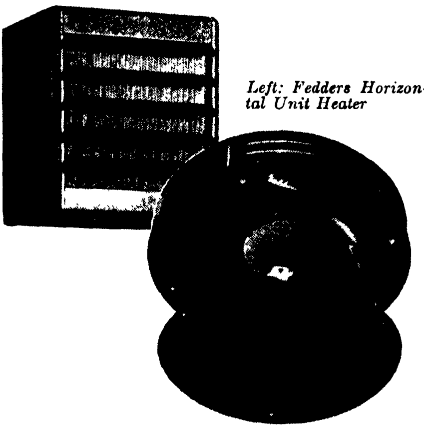
Fedders-Quigan Corporation

57 Tonawanda Street
Buffalo 7, N. Y.

Manufacturers of Convector-Radiators, Wall Radiation, Baseboard Radiation, Unit Heaters, Railroad Car Convectors, Unit Coolers, Refrigeration Coils, Air-cooled Fin and Tube Condensers, Clip-on Thermometers, Room Air Conditioners, Automotive Radiators, Car Heater Cores, Electric Water Coolers.

fedders

A Great Name in Comfort

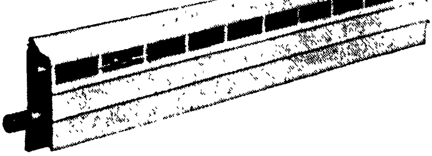


Left: Fedders Horizontal Unit Heater

Right: Fedders Downblow Unit Heater

FEDDERS UNIT HEATERS

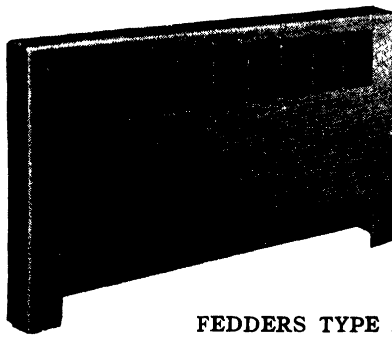
Horizontal Unit Heaters are built in capacities from 100 to 1000 EDR and Downblow models from 155 to 2050 EDR. Advanced design assures maximum heating efficiency with minimum use of piping and materials. Quick response to manual and thermostatic controls contributes to comfort and fuel economy. Circulating air in summer season adds to year 'round comfort. Write for catalog.



FEDDERS BASEBOARD RADIATION

Patent Applied For

Complete line for semi-recessed and flush mounted installations in homes, apartments, offices, hospitals and institutions. Anti-streak design adds to cleanliness. Warm air is directed out into the room by Fedders built-in angular louvers in front face of enclosure. Cool air flows down from the wall and is directed out into the warm air stream by means of curved top of enclosure.

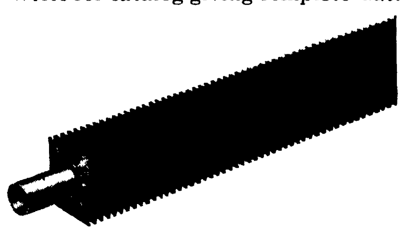


FEDDERS TYPE F CONVECTOR-RADIATORS

For installation in homes, apartments, offices, institutions and other locations. They can be used with steam, and forced or gravity hot water systems.

In addition to Type F stock models, Fedders offers a complete line of Convector-Radiators for special applications.

Write for catalog giving complete data.



FEDDERS WALL RADIATION

Well adapted for industrial, commercial, institutional and household use. Cold drawn, seamless steel pressure type tubing is bullet expanded into self-spacing, collared, die-formed steel fins.

Covers available in expanded mesh or solid front with flat or sloping top with die cut grille.

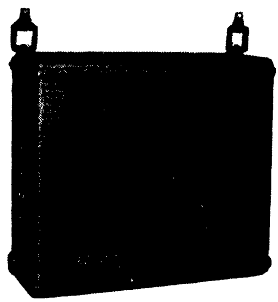
GRINNELL COMPANY, INC.

Heating, Power and Process Piping, Pipe Hangers, Pipe Fittings, Welding Fittings, Valves, Unit Heaters, Piping Supplies

Executive Offices: **Providence, R. I.**

Branch Offices in Principal Cities of United States and Canada

THERMOLIER GRINNELL UNIT HEATERS



Thermolier is a quality unit heater. It is backed by Grinnell's 60 years of varied heating experience. It is so soundly designed and engineered that no changes in its basic construction or operation have been necessary since its introduction years ago.

Basically, all models incorporate the same quality features. However, the heating element of the Horizontal and Vertical delivery types consist of a single header with a series of copper, pitched, finned U-tubes, while the Textile type utilizes a single header with a series of copper, pitched, webbed U-tubes—parallel to one another.

Thermolier motors are not standard stock motors but are built to exact specifications for unit heater duty. This assures maximum life at maximum operating efficiency with the minimum of electrical input. Fans have aluminum blades with special steel hubs and are made to specifications to insure accurate balance, minimizing noise and vibration.

Supply and return pipes are both on the same side of the unit, assuring compactness, neatness and economy in installation. Each Thermolier is furnished with adjustable swivel couplings which provide an easy and practical method of hanging the unit and facilitating adjustment after erection.

The housing of all units are formed from heavy gage steel and are finished in attractive artmetal slate grey.

THERMOLIER CONSTRUCTION FEATURES

that save trouble and money

Use of a plain thermostatic trap, the simplest and least expensive kind of trap, is practical because of Thermoliers's exclusive internal cooling leg.

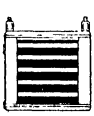
Maximum capacity provided at all times and annoying destructive water hammer eliminated by built-in pitch of tubes and internal cooling leg which assures continuous drainage of condensate.

Damaging strains caused by expansion and contraction eliminated by "U" type expansion tubes.

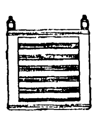
Safety and durability assured with leak-proof tube-to-header construction.

Five other important features. Write for Thermolier Catalog.

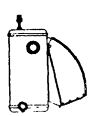
4 MODELS..................18 SIZES



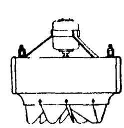
Horizontal Delivery



Textile Model



Velocity Nozzle



Vertical Delivery

CAPACITIES—GRINNELL THERMOLIER

All based on Standard Basis of Rating (2 lb Steam Pressure and 60 deg Entering Air Temperature)

Horizontal delivery models—for constant speed operation, use bold face figures only
—for two speed operation, use both figures

model	rpm at normal speeds	total heat delivered Btu per hr	equivalent direct radiation edr	cfm at exit air temp	exit air temp °F	condensation lb per hr	air velocity at exit—linear ft per min		
							louvers wide open	louvers set at 45°	velocity nozzle max
D21	**1725**	**35,600**	**148**	**637**	**116**	**37**	**786**	**912**	**1336**
	1250	29,700	124	482	122	31	595	690	1011
D31	**1725**	**48,700**	**203**	**689**	**133**	**50**	**851**	**987**	**1447**
	1250	38,400	160	513	138	40	633	734	1076
D37	**1140**	**62,200**	**259**	**1062**	**119**	**64**	**753**	**949**	**1160**
	950	54,200	226	874	123	56	620	781	955
D41	**1725**	**71,000**	**295**	**1271**	**116**	**73**	**901**	**1135**	**1388**
	1250	56,000	233	914	122	58	648	816	998
D44	**1140**	**84,100**	**350**	**1250**	**129**	**87**	**887**	**1118**	**1366**
	750	63,000	262	852	137	65	604	761	930
D57	**1140**	**101,300**	**422**	**1433**	**133**	**105**	**1016**	**1280**	**1565**
	875	84,400	352	1127	138	87	799	1007	1230
D66	**1140**	**128,700**	**536**	**2166**	**120**	**133**	**779**	**982**	**1332**
	625	90,000	375	1340	129	93	482	607	824
D71	**1140**	**151,700**	**632**	**2716**	**116**	**157**	**977**	**1231**	**1671**
	625	103,000	423	1592	126	107	573	722	980
D91	**1140**	**196,000**	**817**	**2738**	**134**	**203**	**985**	**1241**	**1684**
	625	127,500	532	1551	147	132	558	703	954
D111	**1140**	**275,300**	**1147**	**5095**	**114**	**285**	**1048**	**1415**	**1803**
	640	195,700	815	3342	119	203	688	929	1183

Capacity tables for hot water Thermolier on request.

Vertical delivery models—for constant speed operation, use bold face figures only
—for two speed operation, use both figures

model	rpm at normal speeds	total heat delivered Btu per hr	equivalent direct radiation edr	cfm at exit air temp	exit air temp °F	condensation lb per hr	air velocity at exit ft./min.
VA1042	**1725**	**50,800**	**212**	**1483**	**93**	**53**	**1399**
	1250	40,100	167	1078	96	41	1017
VA1045	**1725**	**73,600**	**307**	**1363**	**114**	**76**	**1287**
	1250	59,900	249	995	121	62	939
VA1065	**1140**	**109,400**	**456**	**2869**	**97**	**113**	**1354**
	625	74,500	310	1630	105	77	769
VA1075	**1140**	**145,600**	**607**	**2609**	**116**	**151**	**1231**
	625	94,200	392	1402	129	98	662
VA1101	**1140**	**185,000**	**770**	**4510**	**100**	**191**	**1495**
	600	122,200	508	2517	108	126	835
VA1111	**1140**	**257,000**	**1071**	**4921**	**112**	**266**	**1631**
	640	187,000	729	3105	121	194	1030

Textile models—constant speed operation

model	rpm at normal speeds	total heat delivered Btu per hr	equivalent direct radiation edr	cfm at exit air temp	exit air temp. °F	condensation lb per hr	air velocity at exit—linear ft per min		
							louvers wide open	louvers set at 45°	velocity nozzle max
TX70	**1140**	**69,800**	**291**	**2297**	**89**	**72**	**826**	**1041**	**1412**
TX110	**1140**	**113,700**	**474**	**2438**	**106**	**118**	**877**	**1105**	**1509**

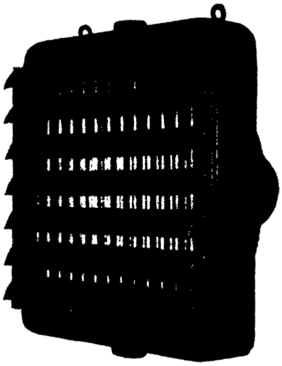
Textile Thermolier not available for hot water systems.

Grinnell Thermoliers are tested and rated in accordance with rules of the *Industrial Unit Heater Association* adopted January 1930.

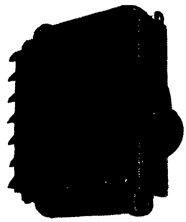
ILG Electric Ventilating Co.

2880 North Crawford Ave., **Chicago 41, Ill.**

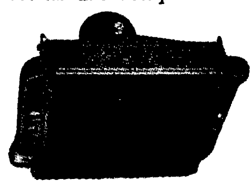
Offices in more than 40 Principal Cities



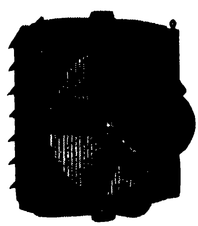
Horizontal Type Unit Heaters—have ILG-built Self-Cooled Motor to counteract coil heat—never "slow roasts." Graduated 2-piece, cast iron header gives "balanced" steam distribution. Brass orifice bushings expand tubes uniformly in header plate. Copper fins are *pressed* into round copper tubes for permanent union—no brazing, soldering, or welding. Bottom header "floats" to permit expansion and contraction of coil independent of casing. Tested and Rated according to codes of *I.U.H.A.* and A.S.H.V.E. Ratings certified by *I.U.H.A.*—"One-Name-Plate" Guarantee. Wide range of sizes, capacities.



Low-Ceiling Type—for mounting where there is little head-room. Side inlets and outlets assure compact installation.

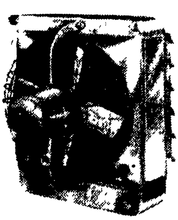


Vertical Type—for installations with extremely high or low ceilings. Diffusers or deflectors available to direct air flow

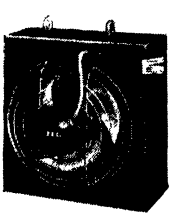


Textile Type—more tubes, no fins—for applications where lint, etc. normally adheres to fin surfaces, clogs up coil.

ILG Electric Unit Heaters



Standard Type—for instant, clean, safe, dependable heating. Coil is of black heat type which operates below 400 degrees. Protected against excessive temperature rise by patented automatic thermal cut-out and magnetic starter. Sizes 5 to 15 KW.



Type "HT"—for installations requiring a small volume of heat. Exceptionally efficient. Suitable for constant duty. Non-overheating black heat type coil with individually interchangeable elements. Sizes 1-1/2 to 4 KW.

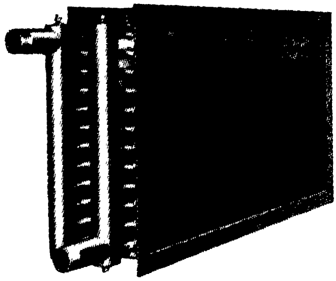
For ILG Propeller and Centrifugal Fans, see page 1244.

KENNARD Corporation

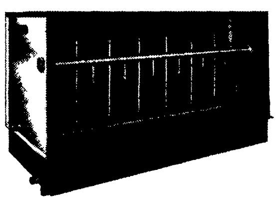
1819 So. Hanley Rd., St. Louis 17, Mo.

Manufacturers of
HEAT TRANSFER EQUIPMENT

Blast Finned Coils—Heating and Cooling Air Conditioning Units—Regular—Multi-Zone Sprayed Coil Dehumidifiers—Heating & Ventilating Units Evaporative Condensers and Cooling Towers



WATER, STEAM AND DE COILS—Complete Range of Sizes—9 in. x 12 in. to 36 in. x 120 in.



SPRAYED COIL DEHUMIDIFIERS—56 sizes—6 sq ft to 81 sq ft. Air Volumes up to 48,600 cfm.

MULTI-ZONE UNITS—(Not Illustrated). Sizes range up to 36 sq ft coil face, and 19,800 cfm.

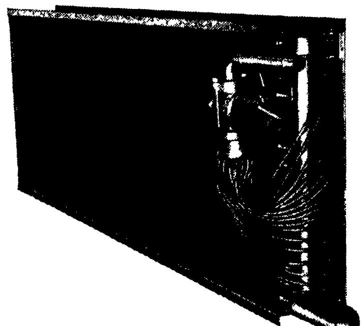
HEATING AND VENTILATING UNITS—(Not Illustrated). Air volumes from 400 to 21,000 cfm.

*EVAPORATIVE CONDENSERS—3 to 75 tons. All refrigerants. All prime surface coils. Indoor or outdoor units.

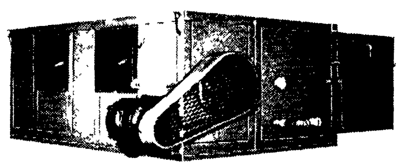
*COOLING TOWERS—3 to 75 tons. Triple tier wetted deck. Indoor or Outdoor units.

Write for catalogs and name of nearest representative

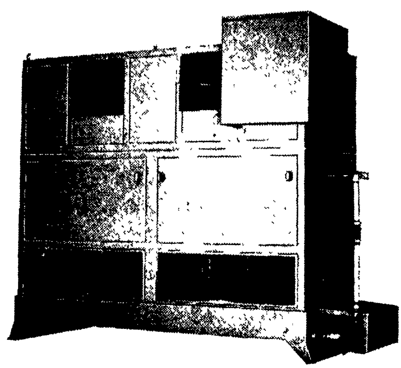
* **Hot-Dipped Galvanized after Fabrication**



ALL KENNARD AIR UNITS—Pentapost construction; Multi Access Panels—flush mounted with latches.



BLOWER UNITS—Ceiling and floor type air-conditioning units. 1 to 50 tons nominal capacities in 12 sizes. 400 to 20,000 cfm. Various arrangements of discharge, filter box and motor drive. (Horizontal and Vertical).



McQuay, Inc.

1602 Broadway, N.E., Minneapolis 13, Minn.

MANUFACTURERS OF AIR CONDITIONING EQUIPMENT

Sales Offices in all Principal Cities

- **Air Conditioners**
- **Air Conditioning Coils**
- **Blast Heating Coils**
- **Refrigeration Coils**
- **Unit Heaters**
- **Unit Coolers**
- **Seasonmakers**
- **Cooling Towers**

PROVEN McQuay PRODUCTS

- **Comfort Coolers**
- **Blower Coolers (Suspended & Floor Type)**
- **Ice Cube Makers (Automatic)**
- **Icy-Flo Accumulators**
- **Zeropak Low Temp. Units**
- **Evaporative Condensers**

THE EXCLUSIVE McQUAY NEW RIPPLE FIN COILS

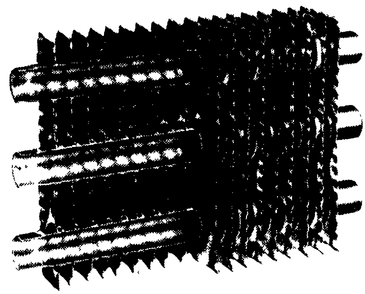
Heat transfer surface with higher efficiency and greater durability is assured when you select a McQuay New Ripple Fin Coil.

The expansion of all tubes into fins having wide smooth collars, without the use of any "low conductivity" bonding material, provides a permanent mechanical bond, for long life high heat transfer efficiency.

Ripple fin coils produce a rippled air flow pattern assuring a closer and longer contact between the air stream and the coil surface, thereby preventing air bypass and producing more rapid heat transfer.

Ripple fin coils have higher flexible strength with minimum air friction and cleaner operation. The copper tube headers provide inherent flexibility to accommodate unequal contraction and expansion during operation. Ripple fin coils easily and effectively drain off condensed moisture, with water hang-up being sharply reduced on coils requiring vertical (up) air flow.

Ripple fin coils permit increased face velocities without danger of moisture carry-over from the fin surface to the air stream.

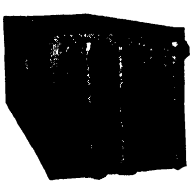


New Ripple-Fin Coil

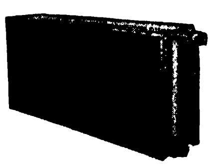
McQuay's numerous header sizes and tube lengths provide greater flexibility for sizing jobs.

McQuay coils are available in a wide variety of styles and sizes; both standard and special coils for steam, hot water, cold water, brine, direct expansion, refrigerant condensing, and other applications.

McQuay coils are proved and preferred —proved by service under the most rigid conditions, and preferred because of their exclusive fin construction.



Combination Cooling Coil

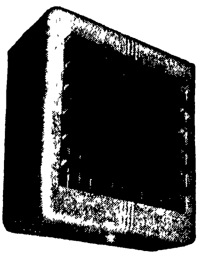


Water Coil

Steam Blast Coil

MORE THAN 1,000,000 COIL TYPES AND SIZES

McQUAY manufactures a complete line of Standard Coils for the Industry.
Coils for Heating—1 to 10 rows deep using low or high pressure steam or hot water.
Jet-Tube (Non-Freeze steam inner tube) type coils 1 and 2 rows deep.
Cleanable Tube—Removable plug type water coils 1 to 10 rows deep.
Water Coils for Cooling—1 to 10 rows deep.
Direct Expansion Coils for Cooling—1 to 8 rows deep.
Refrigeration Coils—all types and sizes.
Special Coils—of various materials furnished on order for special applications.



Horizontal Unit Heater

Down Flow Unit Heater

HORIZONTAL UNIT HEATERS

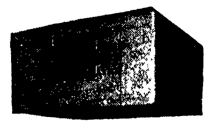
Three basic types available—Standard, High cfm, and Textile. 13 Standard models ranging in size from 21,600 to 360,000 Btu; 11 High cfm models 20,300 to 248,000 Btu; 5 Textile models 38,100 to 196,300 Btu. Write for Catalog 321 and 322.

DOWN FLOW UNIT HEATERS

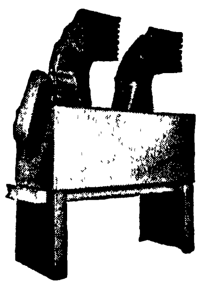
Two basic types available—Standard and High cfm. 11 Standard models ranging in size from 39,300 to 500,000 Btu; 11 High cfm models 25,400 to 289,000 Btu. Write for Catalog 760.

"RH" AIR CONDITIONER
(For Small Commerical Jobs)

Four basic sizes: 1000, 1500, 2000, and 3000 cfm. Nominal capacities range from 2 to 10 tons. Freon and water cooling coils—steam and water heating coils. Write for Bulletin 86A.



Air Conditioner
(Year-Round)



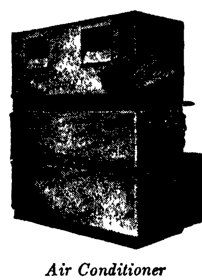
Blower Type Unit Heater

BLOWER TYPE UNIT HEATERS

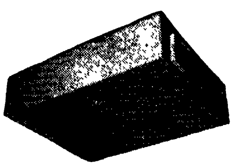
Made in 20 basic sizes covering the entire range from 1000 cfm and 20,600 Btu to 21,000 cfm and 1,600,000 Btu. Filters, face and by-pass dampers, mixing boxes, humidifiers, and all styles of discharge nozzles are available. Write for Catalogs 340 and 342.

CENTRAL STATION AIR CONDITIONER
(For Large Commercial and Industrial Jobs)

Horizontal and Vertical types. Cools, dehumidifies, filters, and circulates air in summer; heats, humidifies, filters, and circulates air in winter. Freon and water cooling coils—steam and water heating coils. Cooling capacities from 2 to 70 tons, cfm from 1000 to 15,675 in both suspended and floor type. Write for Catalog 502.



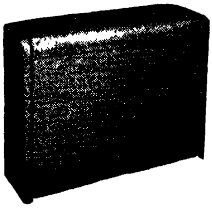
Air Conditioner
(Year-Round)



Seasonmaker (Ceiling Type)

SEASONMAKERS
(For Multi-Room Buildings)

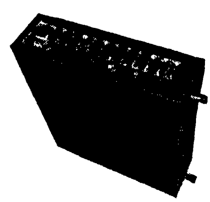
Room air conditioner, three types: floor, basic, and ceiling. Cooling and heating medium supplied from central plant. Cools, dehumidifies, and filters in summer; heats and filters in winter. Three basic sizes: 200, 400, and 600 cfm (up to 25 per cent of cfm may be fresh air). The Seasonmaker's compactness, attractiveness, quiet operation, and ease of installation make it ideal for hotels, apartment buildings, hospitals, etc. Write for Bulletin 700.



Seasonmaker (Floor Type)

ICY-FLO ACCUMULATORS

The new practical "Storage-Battery" for refrigeration effect is now available for handling heavy loads of short duration. Ideal for churches, lodges, mortuaries, noon cafeterias, and many industrial applications. Write for Bulletin 108.



Accumulator

Modine Manufacturing Company

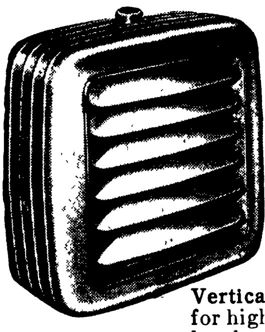
Heating and Air Conditioning Division

General Offices: 1515 Dekoven Ave., **Racine, Wis.**

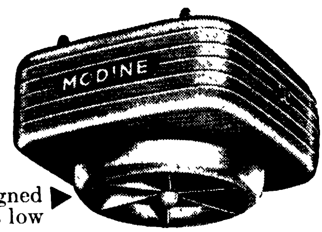
Factories at Racine, Wis., LaPorte, Ind., Paducah, Ky., and Whittier, Calif.

Sales Representatives in all Principal Cities

MODINE UNIT HEATERS FOR HOT WATER AND STEAM



◀ **Horizontal Delivery**—Modine gives you 23 Models to choose from. Built for general industrial and commercial applications.



Vertical Delivery—16 Models designed ▶ for high overhead installation or at low levels.

A fully coordinated line of Modine Unit Heaters offers greatly expanded opportunities for correct unit heater application. Used individually or in combination, they meet the exacting engineering demands of any space heating application.

Rugged Condenser—Tubes and headers are cylindrical and brazed at the joints for greater pressure-resisting strength. Individual expansion bends absorb differential stresses.

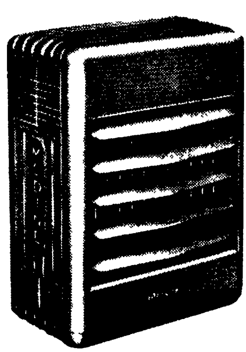
One-Piece Construction—Tubes, header, inlet and outlet connections are *brazed* into a rugged, pressure-resisting unit.

Safety Fan Guard—On all Horizontal models, a built-in fan guard offers constant protection from exposed fan.

Bonderized Casings—All Unit Heater casings are Bonderized to prevent rust and bond paint to steel.

Efficient Motors—Nationally known makes of continuous-duty, totally enclosed fan type. Rubber mounted to prevent vibration noise.

Easy Installation—Direct-from-pipe line suspension for low cost, fast installation of all Horizontal types. Accurately rated in strict accordance with the Standard Test Code for Unit Heaters.



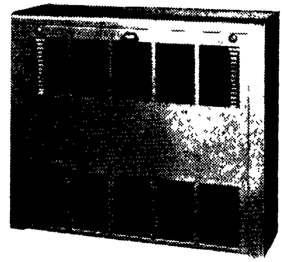
NEW MODINE GAS-FIRED UNIT HEATERS

New Modine Gas-Fired Unit Heaters in five sizes . . . from 217 to 550 EDR . . . feature light weight, smaller size, greater durability and faster response to automatic controls.

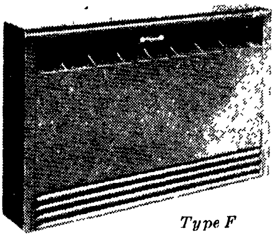
Heat exchangers and burners are corrosion-resistant stainless steel. Tubes are individually fired for most effective heat transfer. Elongated burner ports having approximately four times the free area of conventional drilled ports, discourage clogging and minimize cleaning. All models are *A.G.A.* approved for natural, manufactured, mixed, LP and LP-air gases.

MODINE CONVECTOR RADIATION

Available in Standard enclosure styles and in heavy-duty Institutional models. Enclosures are bonderized. Accurate ratings are determined in conformance with Commercial Standard CS-140-47 as developed by the *National Bureau of Standards* and approved by the Convector Rating Committee.

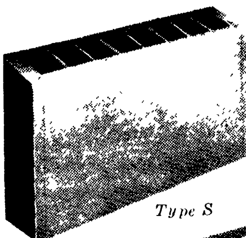


TYPE IF—Institutional style for recessing or free standing installation. Optional special tamper-proof fronts and dampers. Institutional models also available in the wall types IW and IS.



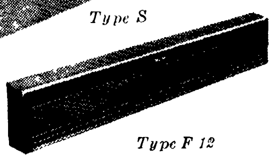
Type F

TYPE F—Shown with attractively louvred lower grille and fingertip control damper—both optional. Ideal for either complete or partial recessing in a wall or for flush installation against a wall.



Type S

TYPE S—Wall type with air outlet grille incorporated in sloping top. Also available in heavy gage metal for institutional use. Type W, another wall hung enclosure resembling the Type F, can also be furnished in heavy-duty models.



Type F 12

TYPE F-12—Designed for free-standing or recessed installation beneath a picture window, convector has an over-all height of only 12 in. Damper and lower grille are optional.

MODINE CABINET UNITS

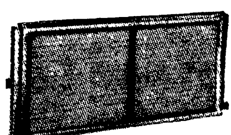
Modine Cabinet Units were intended for heating with steam or hot water or cooling with chilled water. Water models may be used for both heating and cooling. Available in five models from 120 to 640 EDR. Various combinations with and without optional equipment permit free standing, recessed and concealed installations, wall mounting in upright and inverted positions as well as ceiling mounting. This quiet, high capacity unit for quick, positive distribution of heated or cooled air is ideal for offices, lobbies, commercial and public buildings.

Important new design and performance improvements make it possible to closely match the specific requirements of the application.

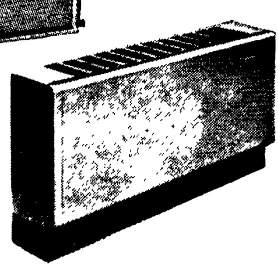
Casings fully coordinated in over-all size, size of flanged edges and diameters and location of mounting holes, simplify installation in a single duct. Smaller duct connections are possible because all surface within casings is heat transfer surface.

MODINE HEATING COILS

The extensive line of Modine Heating Coils is engineered to meet the diversified requirements of modern air-handling systems. In addition to more than 1200 catalogued heating coils for use with steam and hot water, Modine produces many custom built coils for installation in air-handling equipment produced by other firms. Types include standard and non-freeze heating and booster coils, and hot water heating coils.



Standard Heating Coil



Cabinet Unit

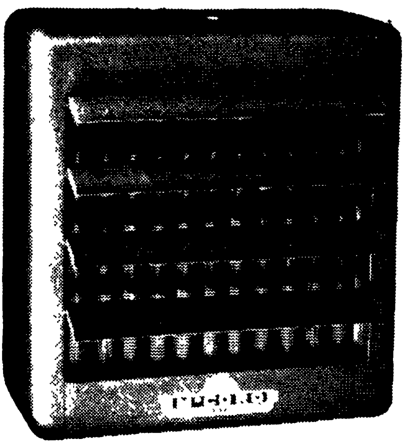
McCord Corporation

AIR CONDITIONING AND REFRIGERATION DIVISION

Detroit 11, Michigan

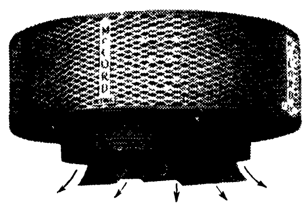
FACTORIES: DETROIT, MICH., WASHINGTON, IND., WINDSOR, CANADA

MANUFACTURERS OF UNIT HEATERS, CONDENSERS FOR DOMESTIC AND COMMERCIAL REFRIGERATORS



MCCORD HORIZONTAL TYPE UNIT HEATERS

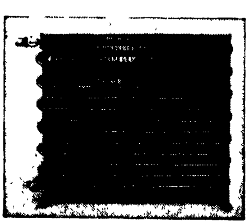
16 models for industrial, commercial general office, and store use—a type size and design for every application. Features: modern design, high capacity.



MCCORD VERTICAL TYPE UNIT HEATERS

12 models for overhead installation—up near the ceilings, in bays, or at low levels in offices and stores.

McCord unit heaters incorporate every unit heater advancement that McCord engineers and the experience of thousands of users could suggest. The spiral fin tube surface as used in McCord horizontal and vertical types provides a type of heat transfer surface of maximum efficiency. Constructed entirely of copper, the fins are bonded by solder. The solder protects the fins from external corrosion, prolonging the life of the heating element. Individual tubes prevent stresses due to unequal expansion. Tube headers are rounded for strength. Improved fan blades deliver large quantities of air with minimum noise or vibration. The design of the die-formed air inlet is an improvement that increases air quantity and decreases noise and horsepower. The motor support is nonair-resrticting. Standard base type motors are used, making replacement easy if it should ever be necessary.



CONDENSERS FOR DOMESTIC AND COMMERCIAL REFRIGERATORS

McCord refrigeration condensers are fabricated by the most modern and economical methods, resulting in a high quality product at low cost. The McCord construction permits use of continuous tubing without joints, eliminating possibility of leaks. Fins are permanently copper-brazed to the tube. Complete range of sizes.

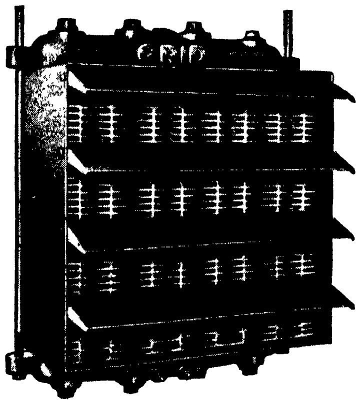
D. J. Murray Manufacturing Co.

T. M. REG.
U. S. PAT. OFF.

Wausau, Wisconsin

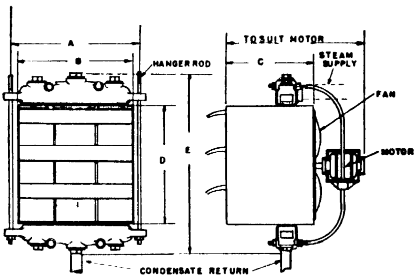
Offices in Principal Cities

MANUFACTURERS OF GRID UNIT HEATERS AND GRID BLAST COILS



One piece construction "fin" heating sections of high test cast iron—no soldered, brazed, welded or expanded connections. Patented.

Designed and tested to operate with steam or hot water systems—for steam pressures from 2 lbs to 250 lbs. Engineered along the same lines as the standard GRID Unit which had aluminum heating sections and has been on the market since 1929.



Overall dimensions for installation of Cast Iron GRID Unit Heater

CI (CAST IRON) SERIES GRID UNIT HEATER DATA

Model No.	Dimensions					Motor		Vol. Fan CFM	Capacities 5 PSI Steam 60°F Air		Pipe Size		Support Rod Dia.	Approx. Ship. Weight Lbs.
	A	B	C	D	E	HP	RPM		Btu/Hr	Final Temp. °F	Supply	Return		
CI-1000	11⅝	13⅜	12½	9¾	15⅛	1/25	1550	572	29,080	106	1¼	1¼	⅜	150
CI-1200	14⅞	16⅞	12¾	12¾	18⅛	1/15	1550	798	45,450	112	1¼	1¼	⅜	210
CI-1500	17⅛	15⅜	11½	16	23⅞	1/8	1750	1500	76,500	107	1½	1¼	½	280
CI-1520	17⅛	15⅜	11⅛	21	28⅜	1/8	1750	1700	101,500	114	1½	1¼	½	390
CI-2000	22⅛	20 9/16	11⅞	21 1/16	28⅜	1/6	1150	2600	148,000	110	2	1¼	½	490
CI-2025	22⅛	20 9/16	11⅞	25⅞	35½	1/6	1150	2875	173,640	115	2	1¼	½	520
CI-2500	27⅛	25⅞	13	25⅞	35½	1/2	1150	4850	224,000	107	2	1¼	⅝	700
CI-2504	27⅛	25⅞	13	25⅞	35½	1/4	1150	3800	206,000	117	2	1¼	⅝	660
CI-2530	27⅛	25⅞	13	31	40⅛	1/2	1150	4650	275,800	114	2	1¼	⅝	900
CI-3000	32⅝	31	13	31	40⅛	1/2	850	6300	332,000	108	2½	1¼	⅝	1020
CI-3000	32⅝	31	13	31	40⅛	1½	1150	8000	380,000	103	2½	1¼	⅝	1070

NO ELECTROLYSIS TO CAUSE CORROSION

Low maintenance expense.
More air changes per hour.
Positive "directed" heat.
No leaks—no breakdowns.

Lower outlet temperature.
Larger air volume.
No soldered, brazed or expanded joints.
Open design that keeps units clean.

Send for complete catalog information
Send for information on Blast coils and radiation.

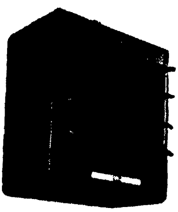
HERMAN NELSON

Division of the American Air Filter Company, Inc.

Louisville, Kentucky • Moline, Illinois

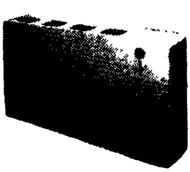
Branch Offices and Product-Application Engineers in the Following Cities:

Atlanta, Ga.	Detroit, Mich.	Milwaukee, Wis.	Portland, Ore.
Baltimore, Md.	Duluth, Minn.	Minneapolis, Minn.	Richmond, Va.
Boston, Mass.	Grand Rapids, Mich.	Missoula, Mont.	Saginaw, Mich.
Buffalo, N. Y.	Greensboro, N. C.	Moline, Ill.	St. Louis, Mo.
Cape Elizabeth, Me.	Houston, Tex.	Nashville, Tenn.	Salt Lake City, Utah
Chicago, Ill.	Indianapolis, Ind.	New Orleans, La.	San Antonio, Tex.
Cincinnati, O.	Jackson, Miss.	New York, N. Y.	San Francisco, Calif.
Cleveland, O.	Kansas City, Mo.	Oklahoma City, Okla.	Seattle, Wash.
Columbus, O.	Los Angeles, Calif.	Omaha, Neb.	Spokane, Wash.
Dallas, Tex.	Louisville, Ky.	Philadelphia, Pa.	Springfield, Mass
Denver, Colo.	Memphis, Tenn.	Phoenix, Ariz.	Syracuse, N. Y.
Des Moines, Iowa	Miami, Fla.	Pittsburgh, Pa.	Washington, D. C.

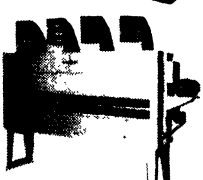


HERMAN NELSON HORIZONTAL SHAFT PROPELLER-FAN TYPE UNIT HEATERS. Designed for ceiling suspension, these unit heaters project warm air downward in an angular direction. Copper heating element for use with steam or hot water, incorporates patented stay tube which maintains proper relationship between headers without increasing strain on loops thus prolonging life of unit. A wide variety of models, sizes and arrangements.

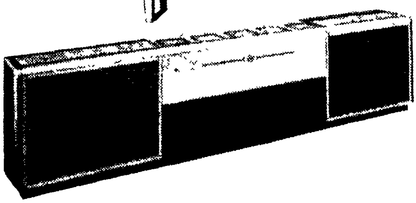
HERMAN NELSON VERTICAL SHAFT PROPELLER-FAN TYPE UNIT HEATERS. For high ceiling installations. Discharge air vertically downward, or at an angle to vertical in various directions. Long life copper heating element for use with steam or hot water incorporates patented stay tube. Units available with either high or low velocity discharge, each with a wide range of capacities.



HERMAN NELSON DE LUXE UNIT HEATERS. Efficient economical, compact, quiet and attractive, these Unit Heaters provide the ideal method for heating offices, showrooms, corridors, markets, stores, etc. Copper heating element incorporates patented stay tube. Units may be placed on floor, wall or suspended from ceiling. Eighteen models, sizes and arrangements.



HERMAN NELSON CENTRIFUGAL-FAN TYPE UNIT HEATERS. The Herman Nelson Centrifugal-Fan Type Unit Heater can be applied to solve a multitude of heating and ventilating problems. With 1,890 combinations of models, sizes and speeds available, there is a unit to fit the average requirements of commercial and industrial buildings of all types.



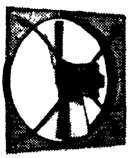
HERMAN NELSON DRAFT/STOP UNIT VENTILATORS . . .The DRAFT/STOP Unit Ventilation System developed by Herman Nelson employs a new design principle. Drafts are trapped before they can reach classroom occupants. Cold drafts are prevented from circulating throughout the room to create constantly changing temperature levels. DRAFT/STOP produces abundant fresh air of even temperature . . . a truly healthful atmosphere that minimizes the possibility of health hazards. The system is quiet, has a smart modern appearance and is economical, being almost entirely free of maintenance problems.

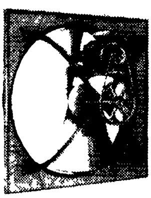
Herman Nelson Unit Heaters and Unit Ventilators are tested and rated in accordance with the Standard Test Code adopted jointly by the *Industrial Unit Heater Association* and THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

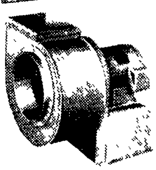
HERMAN NELSON

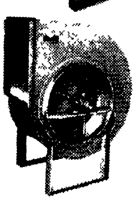
Division of the American Air Filter Company, Inc.

Louisville, Ky. • Moline, Ill.

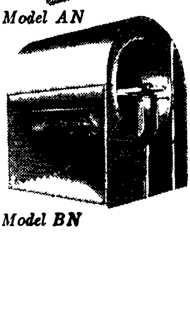








Model AN



Model BN

HERMAN NELSON DIRECT DRIVE PROPELLER FANS. Provide most economical form of quality ventilation obtainable for industrial buildings of all types. 7 standard sizes available with wheel diameters from 14 to 36 in. and capacities from 655 to 12,400 cfm. There are 7 high powered models to operate against static resistance of ⅜ in., with wheel diameters from 14 to 36 in. and capacities from 1,200 to 14,600 cfm. Also three models especially adapted to small store and office applications. Standard Models available with two speed motors.

HERMAN NELSON BELT DRIVE PROPELLER FANS. For public and commercial building installations where slow speed, quiet operation are required. Twelve sizes of the standard model with wheel diameters from 24 in. to 54 in. Also six sizes of the High Powered model with the same wheel diameters. Capacities: 5,650 to 36,150 cfm. Due to quiet operation of Herman Nelson Belt Drive Propeller Fans, use of two speed motor is unnecessary.

HERMAN NELSON DIRECT DRIVE UNIT BLOWERS. Designed for many applications, such as fume hoods, toilet ventilation, chemical laboratories, industrial processing and drying problems. Compact, direct connected, motor driven units have universal discharge and mount on floor, wall or ceiling. Available in four sizes with 9 speed combinations. Wheel diameters from 6$\frac{1}{16}$ in. to 11 in. and capacities from 360 to 2,265 cfm.

HERMAN NELSON BELT DRIVE UNIT BLOWERS. Fully self-contained unit including motor, drives and housing; slow speed or non-over-loading type wheels available; adjustable motor pedestal with vibration dampers; universal discharge; nine sizes with 70 drive combinations. Available with any rotation and discharge. Wheel diameters from 11 in. to 30 in. and capacities from 980 to 16,892 cfm.

HERMAN NELSON CENTRIFUGAL FANS. Herman Nelson Centrifugal Fans are designed and constructed for smooth, efficient, long-life operation on any system requiring the use of a Class I or II centrifugal fan. These fans are available in either slow speed or non-over-loading type; 17 wheel diameters from 12¼ to 73 in.; single or double width; 8 arrangements for direct or belt drive; and any rotation or discharge.

Herman Nelson Propeller and Centrifugal Fans are tested and rated in accordance with the Standard Test Code adopted jointly by the *National Association of Fan Manufacturers* and THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

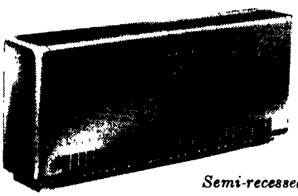
JOHN J. NESBITT, INC.

Philadelphia 36, Pa.

Manufacturers of

THE NESBITT SYNCRETIZER Heating and Ventilating Unit, and
NESBITT WIND-O-LINE RADIATION, and
THE NESBITT PACKAGE, sold by Nesbitt and American Blower Corporation;
NESBITT SERIES D HEATING SURFACE with Steam-distributing Tubes,
NESBITT SERIES H HEATING SURFACE, and
NESBITT SERIES W COOLING SURFACE, sold by John J. Nesbitt, Inc.;
NESBITT CONVECTORS, sold by plumbing and heating wholesalers
WEBSTER-NESBITT UNIT HEATERS **(See page 1441),**
distributed in U.S.A. by Warren Webster & Company

NESBITT SYNCRETIZER—Series 500



Semi-recessed Mode

For heating and ventilating schoolrooms, offices, etc. where the continuous introduction of outdoor air is desired. Incorporates exclusive Nesbitt features: *Comfort Control* automatically regulates temperature of air stream's protective blanket; *Air Volume Stabilizer* prevents more than designed percentage of outdoor air from entering the unit, saves fuel; *Uniform Air Discharge Temperatures* assured by Nesbitt Dual Steam-distributing Tube radiator; *Directed-Flow Adjustable Outlet* permits air discharge pattern best suited to individual classrooms. For data on Syncretizer and schoolroom ensemble consisting of the Syncretizer, Convector, and storage units, Pub. 261.

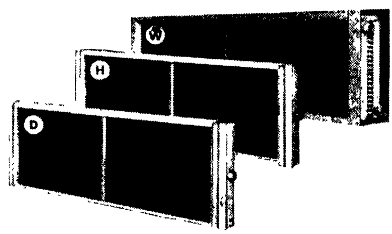
Wind-O-Line Radiation

New auxiliary heating for down-draft protection under long window exposures. Available *wall-hung* in attractive casing for use with free-standing Nesbitt Syncretizer, or *recessed* in component units of The Nesbitt Package. Publication 264. Engineering Pub. 261, Section W.

Nesbitt Series B Thermovent

For heating and ventilating auditoriums, gymnasiums, assembly halls, and similar gathering places. Pub. 227.

NESBITT SURFACE

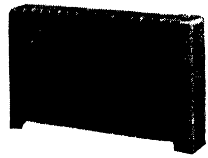


SERIES W. Continuous or cleanable tube water surface for air-cooling, dehumidifying or heating. Copper tubes, aluminum fins. Wide range of sizes in three types: Type WD with exclusive *freeze-proof* drainability feature and surface pitched in the casing. Pub. 246. Type WB sections for booster-heating or air-cooling relatively small air volumes; without drainability feature. Type WC employs standard Series W cores pitched in the casing; cast iron headers and removable cover plates. Single or double serpentine circuits. Pub. 255.

SERIES H. General blast coil surface for heating, ventilating, air-conditioning and drying in both high and low pressure systems. Copper tubes, aluminum fins. Seven types, full range of sizes. Pub. 248.

SERIES D. Steam-distributing tube surface. *Freeze-proof.* Ideal for preheating outdoor air; uniform discharge temperatures; precise controllability with modulating valves. Type DS: Single supply header with single steam-distributing tubes for normal heating and ventilating applications, or DUAL tubes in finned lengths up to six feet. Type DD: Two supply headers; DUAL steam-distributing tubes. Finned lengths from 78 in. to ten feet. Pub. 247.

NESBITT CONVECTORS

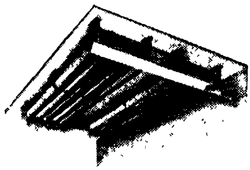


Designed for two-pipe steam, forced or gravity hot water heating of offices, residences, apartments. Available in 23 stock sizes: heights 20 and 24 in.—lengths 16 in. to 88 in.—capacities 13 to 88.5 sq ft EDR. Universal cabinets for free-standing or semi-recessed installation. Pub. 262.

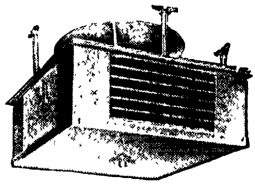
Refrigeration Economics Co., Inc.

1231 Tuscarawas St. E., **Canton 2, Ohio**

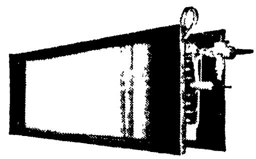
RECOY PRODUCTS



C. T. Coils

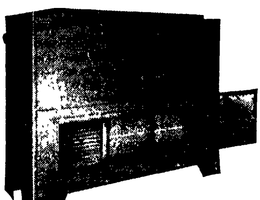


Ceiling Diffusers



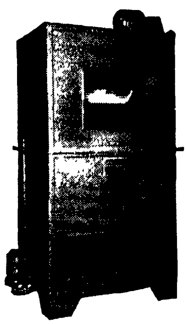
C. F. Coils

C. T. Coils—Continuous-tube down-draft fin-coils are still first choice for meat coolers, or other products requiring high humidity and gentle air circulation.
Ceiling Diffuser—Ceiling diffusers distribute the cooled air across the ceiling, so the draft does not strike the products stored or occupants.
C. F. Coils—Continuous fin coils for unit coolers, blast heaters, air conditioning and condensers.

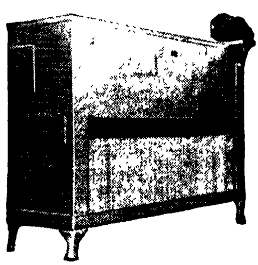


Air Conditioning

Shell Condenser

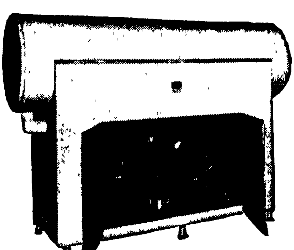


Evaporative Condensers

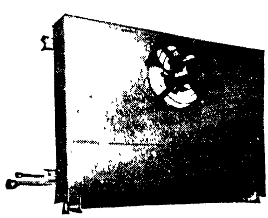


Floor Units

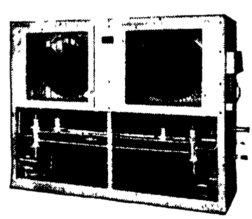
Air Conditioning –Air Conditioning units of ceiling or floor type in all capacities, for cooling, heating, or both.
Shell Condenser—Shell and tube, also shell and fin coil condensers.
Evaporative Condensers—Evaporative condensers from 2 to 100 tons. Brine spray cooling to 25 tons.
Floor Units—Floor units with cooling surface exposed to view have a definite advantage over those with coils hidden.



Water Cooling



Wall Units



Automatic Defrost Units

Water Cooling—Self contained complete ice water and brine coolers complete with high and low sides, circulating pumps, controls, and insulation. 1½ to 50 hp.
Wall Units—Recoy "All Seasons" wall units provide a damper for deflecting the cold air down along the wall or out horizontally into the room, thus providing proper air circulation for "All Seasons."
Automatic Defrost Units—Complete, ready for electric, liquid, suction, and hot gas connections. One coil working always, both 98 per cent of time.

The TRANE Company

2021 Cameron Avenue, La Crosse, Wisconsin

In Canada: Trane Company of Canada, Ltd., Toronto, Ontario—Offices in 14 Canadian Cities

COMPLETE LINE HEATING AND CONDITIONING

Over 80 Trane Sales Offices in U. S.

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Allentown, Pa.
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Appleton, Wis.
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Aurora, Ill.
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Cincinnati, Ohio
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Cleveland, Ohio
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Miami, Fla.
Milwaukee, Wis.
Nashville, Tenn.
Newark, N. J.
New Orleans, La
New York, N. Y.
North Tarryton, N.Y.
Oklahoma City, Okla
Omaha, Nebr.
Philadelphia, Pa.
Phoenix, Ariz.
Pittsburgh, Pa.
Portland, Maine
Portland, Ore.
Providence, R. I.
Richmond, Va.
Roanoke, Va.
Rochester, N. Y.
St. Louis, Mo.
St. Paul, Minn.
Saginaw, Mich.
Salt Lake City, Utah
San Antonio, Texas
San Francisco, Calif.
Seattle, Wash.
Sioux City, Iowa
South Bend, Ind.
Spokane, Wash.
Syracuse, N. Y.
Toledo, Ohio
Trumbull, Conn.
Washington, D. C.
West Hartford, Conn.
Wichita, Kans.
Wilkes-Barre, Pa.
Wilmington, Del.
Worcester, Mass.

A COMPLETE LINE. The Trane Company fabricates a complete line of heating, cooling, air conditioning and air handling equipment. So comprehensive is the Trane Line that any number of complete heating and air conditioning systems can be designed in which all the major parts are made by Trane.

TRANE CONVECTORS—Attractive, efficient, easy-to-install units. Available for either steam or hot water heating systems in a variety of cabinet types.

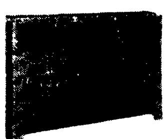
TRANE COILS—There are Trane Extended Surface Coils for every heating or cooling application. Types include coils for steam, hot water or booster heating, direct expansion or water cooling.

TRANE PROJECTION UNIT HEATERS—A Trane development, the Model P taps the reservoir of wasted ceiling heat, offers ideal solution to multitude of heating problems. Illustrated here with the new adjustable Louver Cone Diffuser for uniform heat distribution.

TRANE HORIZONTAL UNIT HEATERS—Combines the Trane broad blade propeller fan, extended surface coil and many other features into a heater that is rugged, efficient and handsome. Available in 24 sizes 22,000 to 352,000 Btu. New Louver Fin Diffuser completely controls direction of air flow.

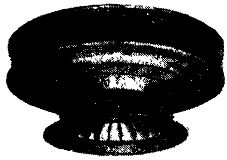
TRANE BLOWER TYPE UNIT HEATERS—Better known as Torridors they combine Trane Centrifugal Fans, Extended Surface Coils, rugged casings to provide the ideal unit for heating large spaces and for process applications.

TRANE GAS UNIT HEATERS—The built-like-a-boiler unit that's available in seven sizes. These new units are suitable for every heating application. Capacities from 50,000 to 210,000 Btu.

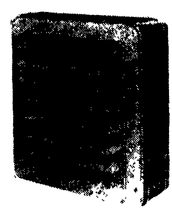


Convector

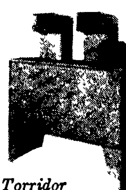
Type E Heating Coil



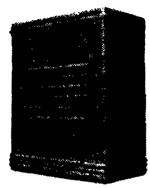
Projection Heater with Louver Cone Diffuser



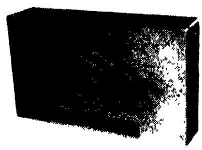
Horizontal Propeller Unit Heater with Louver Fin Diffuser



Torridor Unit Heater



Gas-Fired Unit Heater



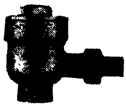
Force-Flo Heater

Wall-Fin Heater

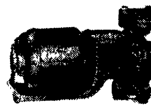
Unit Ventilator

Roof Ventilator

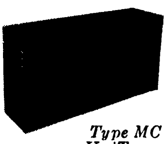
Lifetime Valve



Thermostatic Trap



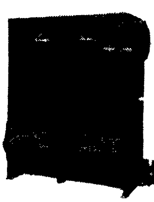
Circulator



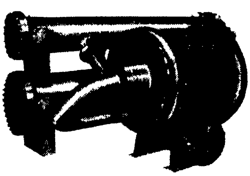
Type MC UniTrane

Climate Changer

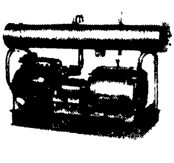
Self-Contained Air Conditioner



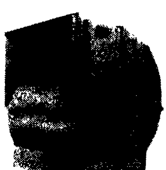
Evaporative Condenser



CenTraVac



Condensing Unit



Centrifugal Fan

TRANE FORCE-FLO HEATERS—A deluxe unit heater with centrifugal fan. A powerhouse of heat in a neat appearing cabinet. Capacities from 17,700 to 105,000 Btu.

TRANE WALL-FIN HEATERS—Finned radiation in both ferrous and nonferrous construction. For industrial and commercial applications. Lengths 2 ft to 12 ft in 6 in. increments. Expanded metal grilles and cabinets available.

TRANE UNIT VENTILATORS—Meets every requirement of ventilation in school rooms and similar installations. Units available with matching shelving and auxiliary convectors.

TRANE ROOF VENTILATORS—A complete line of powered ventilators for supply and exhaust and vertical units for exhaust only. Capacities from 355 to 25,000 cfm.

TRANE STEAM SPECIALTIES—Over 150 items including three types of valves, thermostatic traps, inverted bucket traps, float traps, strainers, vents and direct traps.

TRANE HOT WATER HEATING SPECIALTIES—Included are the Trane Circulator, FloValves and Fittings. Combined with Trane Convectors they provide an ideal warm water heating system.

TRANE MULTI-ROOM AIR CONDITIONING SYSTEMS—Includes Custom-Air which provides separate control of temperature and moisture, UniTrane for control of temperature and moisture without the use of ducts.

TRANE CLIMATE CHANGERS—A unit type air conditioner, designed for year-round air conditioning. Available in various coil combinations with or without humidification equipment. Capacities from 450 to 23,000 cfm.

TRANE SELF-CONTAINED AIR CONDITIONERS—Packaged units, ideal for office, shop or home air conditioning.

TRANE EVAPORATIVE CONDENSERS—For condensing Freon and Methyl Chloride refrigerants. Uses minimum amount of water. Capacities from 3 to 100 tons.

TRANE REFRIGERATION EQUIPMENT—The CenTraVac is a hermetic centrifugal refrigeration unit in sizes from 50 tons up. Trane Reciprocating Compressors and Condensing Units are available from 3 to 50 tons.

TRANE FANS—Centrifugal in Class I and II. Backwardly inclined or forward curved types, all arrangements. Wheels from 12 in. to 108 in. Capacities from 668 to 491,000 cfm. Utility and Propeller Fans with direct or belt drive.

TRANE EDUCATIONAL MATERIAL—The Trane Air Conditioning Manual ($5.00), unbiased textbook for the engineering profession. The Trane Refrigeration Manual ($1.50), a reference for servicing and installing all types of refrigeration systems.

OTHER TRANE EQUIPMENT—1. Condensation and Centrifugal Pumps; 2. Dry Type Water Chillers; 3. Product Coolers; 4. Fluid Coolers; 5. Brazed Aluminum Heat Transfer Units; 6. Transportation Air Conditioning Equipment; 7. Shell-and-Tube Heat Exchangers; 8. Evaporative Coolers; 9. Air Washers; 10. Multi-Zone Air Conditioners.

Office and Factory
LINDEN, N. J.

L. J. Wing Mfg. Co.

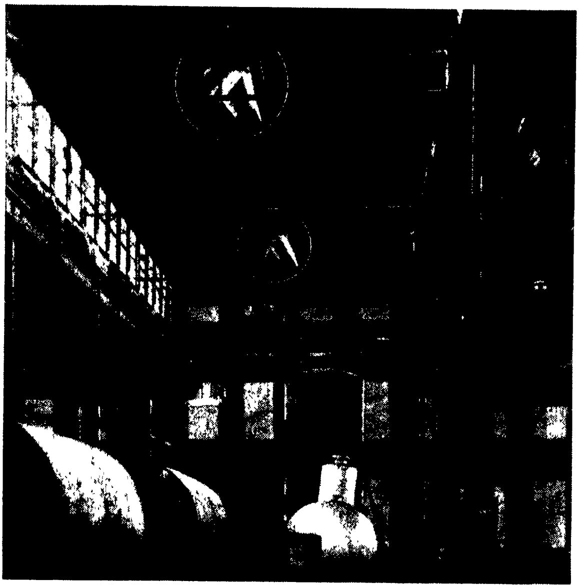
59 Vreeland Mills Road, Linden, N. J.

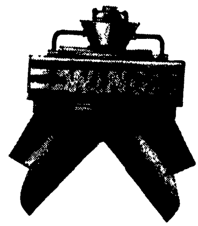
Canadian Factory:
MONTREAL

Branch Offices in Principal Cities

European Representative for Heaters: Wanson, Haren-Nord, Brussels, Belguim

WING REVOLVING UNIT HEATERS





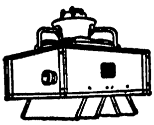
Discharge Outlet Design No. 8

Discharge Outlet Design No. 4

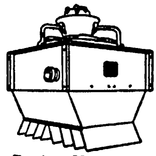
Discharge Outlet Design No. 6

Wing Revolving Unit Heaters draw the air from the roof or ceiling, pass it through the Wing Featherfin Heating Section, and project the resultant heated air through slowly revolving discharge outlets to the working level below. As the moving warm air reaches the working level, it mixes uniformly with the cold air, circulates around bulky machinery or other obstacles and penetrates into cold damp corners, repair pits, etc. Thus is assured a comfortable, uniform temperature throughout the entire plant, a condition not attainable with the single-direction discharge of the ordinary unit heater. The effect on the worker, because of the gentle air motion, is a pleasing sensation of fresh, live, invigorating warmth.

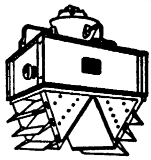
The Wing Revolving Unit Heater is available in three different types of revolving discharge outlets, as illustrated above, to suit the varying requirements of buildings and rooms of different heights and shapes. An additional feature of the Wing Revolving Unit Heater is its use for summer cooling. With the steam turned off and the fans on, the revolving discharge outlets provide an equally pleasant cooling effect. Wing Revolving Unit Heaters are also available in Gas-Fired models. *Bulletin HR-6.*



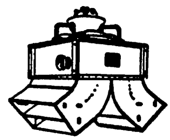
Design No. 1



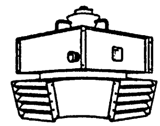
Design No. 1-HV



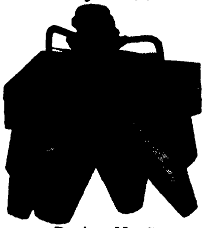
Design No. 3



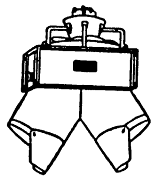
Design No. 4



Design No. 5



Design No. 7

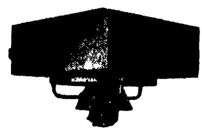


Design No. 8

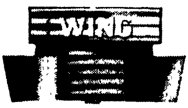
WING FIXED DISCHARGE UNIT HEATERS

Eight different designs of outlets meet the requirements of every type, size and height of building or occupancy. Located near ceiling or roof, the accumulation of hot air in the upper spaces, with the accompanying costly waste of heat, is prevented. *Bulletin HR-6.*

DOOR HEATERS, GARAGE HEATERS



Applicable for heating the inrush of cold air at large doorways and for garage heating. Often cuts heating costs in half. *Bulletin HR-6.*

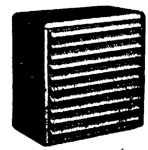


Type "LC"

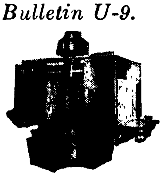
FOR LOW CEILINGS

Position of fan and motor are reversed to meet conditions of ceiling or roof height, form and shape of building, coverage, etc. *Bulletin HR-6.*

WING UTILITY UNIT HEATERS



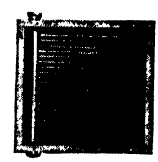
A lightweight suspended unit heater for delivering heated air in one general direction. Has the same powerful fan and rugged heating element as WING Featherweight Unit Heaters. *Bulletin U-9.*



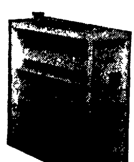
WING GAS FIRED UNIT HEATERS

For natural or manufactured gas. Combines gas burners, heat exchanger and combustion chamber with motor driven Wing fan and discharge outlets. The revolving discharge outlet distributes the heat continuously in constantly changing directions. *Bulletin GH-1.*

FEATHERFIN HEATER SECTIONS



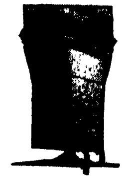
For heating or cooling air for any purpose by steam, hot or cold water or refrigerant. The heating element is extremely light and, for equal heat transfer, offers little resistance to air flow. Available for any desired final air temperature. *Bulletin HS-3.*



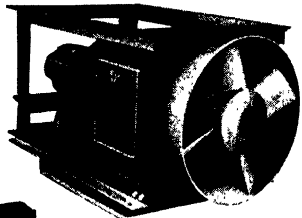
VARIABLE TEMPERATURE SECTIONS

Invaluable in supplying fresh air of varying temperatures for space heating or process work. Close control of the delivered air temperature. Positively will not freeze. Manual or automatic control. *Bulletin HS-3.*

WING INDUSTRIAL FOG ELIMINATORS



Eliminate fog, odor and fumes in dyeing, bleaching and finishing plants, creameries, pasteurizing, bottling, canning and packing plants, chemical works, paper mills, steel pickling plants, etc. No ducts are required. *Bulletin FE-12.*

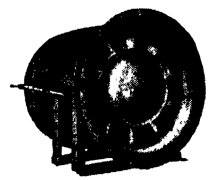


WINGFOIL DUCT FANS

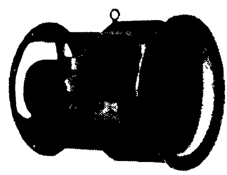
A compact, economical "housed fan" designed to operate against static pressure. Available in either the elbow or straight-line type. The motor is entirely outside the housing—always cool, clean and easily accessible—and out of the path of hot or dirty air or gases which might be injurious to it.

Delivers large air volumes with low power consumption efficiently against static-pressure. Straight line type furnished for horizontal or vertical operation and is driven by V-belt. Elbow type may have either belt or direct drive. Complete range of sizes from 200 to 85,000 cfm. *Bulletin F-11.*

WING MOTOR-DRIVEN BLOWERS



(1) Simple and Rugged Construction (2) Compact Design (3) Low Installation Cost (4) High Efficiency, Quiet Performance (5) Built-in Voltrol Vanes (capacity regulating dampers) actuated by an external balanced lever. Can be adjusted manually or connected to any standard combustion control regulation. Precise in construction. Reduction in volume is accompanied by a reduction in horsepower over a wide range. *SW-51.*

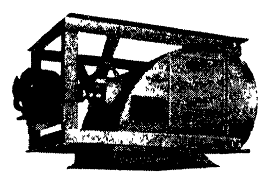


WING TURBINE-DRIVEN BLOWERS

Applied to hand, stoker, oil or pulverized fuel fired boilers, increase boiler capacity, maintain constant steam pressure and permit complete combustion of low-cost fuels. The exhaust steam, free from oil, can be used for heating or processes. *Bulletin SW-51.*

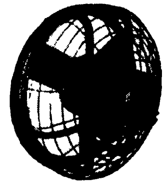
WING DRAFT INDUCERS

Provides positive, uniform draft regardless of weather conditions and assures thorough and efficient combustion with high CO_2 content. Eliminates high, costly, unsightly stacks. *Bulletin I-51.*



WING SYSTEM OF CONTROLLED COMBUSTION

For low pressure heating boilers and small power boilers. Increases capacity and permits use of lowest cost fuel. Eliminates necessity of frequent firing, allowing intervals as great as 24 hours even in zero weather. *Bulletin M-496.*



WINGFOIL SAFETY VENTILATING FANS

An axial flow fan that will deliver air against static pressure, quietly and efficiently.

Capacities to 100,000 cfm. *Bulletin F-11.*

Young Radiator Co.

Dept. 541, Racine, Wis.

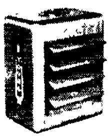
Sales and Engineering Offices in Principal Cities

YOUNG HEAT TRANSFER PRODUCTS

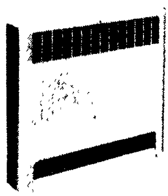
T M Reg U S Pat Off.

Heat Transfer Products for Automotive and Industrial Applications. Heating, Cooling, Air Conditioning Products for Home and Industry.

HEATING, COOLING & AIR CONDITIONING EQUIPMENT

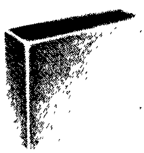


Left: Type "SH" unit heaters for horizontal air discharge. Capacities from 19,000 to 325,000 Btu per hour.

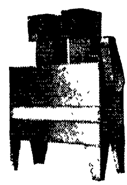


Right: Convector-Radiators in Standardized Types—circulate rather than radiate heat. Used with steam or hot water systems. Cabinets blend with architecture and room decor.

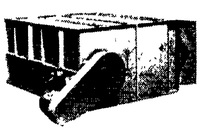
Left: Type "V" or Vertiflow unit heaters for vertical air discharge. Capacities from 52,600 to 552,000 Btu per hour.



Left: Cabinet Type unit heaters offer unit heater performance plus the beauty of modern cabinet styling. Capacities from 26,200 Btu to 115,000 Btu per hour.

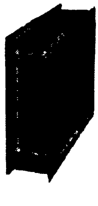


Right: Type "BH" blower unit heaters for floor, wall or ceiling mounting. Capacities from 109,400 to 1,047,000 Btu per hour.



Young "YAC" Air Conditioning Units provide year around air conditioning. Available in eight sizes, in capacities ranging from 450 cfm to 15,750 cfm, horizontal (left) or vertical (right) units. Heating coils furnished for either steam or hot water; cooling coils for water or direct expansion operation. Possible to provide a unit for heating only, cooling only, or for heating and cooling. Compact, easy to handle, and install.

HEATING & COOLING COILS



Left: Type "W" water coils for cooling or heating with either unit or central plant systems. Five widths, 11 to 35 in.; 2 to 8 rows of tubes; many lengths.

Right: Types "B" and "A" blast coils for either unit or central plant heating and air conditioning systems. Steam distributing tube type available.

Other types in Young's complete line of coils include: **Type "E," evaporator coils for direct expansion cooling systems; Type "C," unencased coils for use in factory built unit air conditioners available in either standard or steam distributing tube types.**

The American Brass Company

General Offices: Waterbury 20, Conn.

District Offices in Principal Cities

ANACONDA
from mine to consumer

IN CANADA: ANACONDA AMERICAN BRASS LIMITED, NEW TORONTO, ONTARIO

PRODUCTS—Anaconda Deoxidized Copper Tubes and Fittings; Anaconda "85" Red Brass Pipe; Everdur Metal for storage heaters, storage tanks, ducts and air conditioning equipment

ANACONDA COPPER TUBES AND FITTINGS

For Heating, Plumbing and Air Conditioning

Anaconda Types K and L Deoxidized Copper Water Tubes, assembled with solder type Fittings, offer an unusual combination of advantages for hot water and low pressure steam heating systems, including radiant panels. These advantages may be summarized briefly as follows:

Low Friction Loss—Because the inside surfaces of copper tubes are inherently smoother than those of pipe and tubes made of ferrous materials and also because they do not become roughened by the formation of rust, these tubes offer a lower resistance to flow. In addition, the long radius turns of elbows and the smooth inside surface of wrought copper fittings further reduce friction losses.

These factors naturally increase the efficiency of the system, particularly when it includes a forced pressure circulator.

Ease of Installation—In many places the flexibility of copper tubes simplifies connections that ordinarily would be awkward and expensive to make with rigid pipe and threaded fittings. Solder fittings are compact. They can be installed in restricted space where the use of a wrench would be impossible.

They meet the requirements for these types of tubes in Federal Specification WW-T-799a and *ASTM* Specification B88. Type K, the heavier, is recommended for heating lines and general piping.

Anaconda Copper Water Tubes, in standard sizes are furnished soft in 60 and 100-ft coils; also hard and soft in 20-ft straight lengths.

Accuracy of Dimensions—Anaconda Copper Water Tubes are all finished to the close size tolerances required by the *ASTM* and Federal Specifications, which have been found essential for efficient assembly with solder fittings.

REFRIGERATION TUBING

Anaconda Dehydrated Copper Refrigeration Tubes are manufactured in accordance with *ASTM* Specification B68, in all standard sizes up to and including ¾ in. O.D., in 50-foot coils. Longer lengths are made to special order. These tubes are manufactured under exceptionally clean mill conditions and close technical control to assure clean, smooth inside surfaces, unusual accuracy in size and shape, and uniform softness. The tubes are sealed immediately after annealing and dehydrating.

The American Brass Company

VIBRATION ELIMINATORS

American Vibration Eliminators—Compressor vibration and noise are muffled in a line equipped with an American Vibration Eliminator. What's more, the line that absorbs vibration is far less apt to fail from fatigue. An American Vibration Eliminator is a deluxe product. The corrugated bronze tubing is *seamless*. Copper ferrules and tube ends are *bronze welded*. Each unit is pressure-tested under water, is spotlessly clean and dry, with ends firmly sealed.

ANACONDA "85" RED BRASS PIPE

Anaconda "85" Red Brass Pipe, in standard pipe sizes, is considered the highest quality corrosion-resistant pipe commercially obtainable at a moderate price and is recommended for steam return lines.

Anaconda "85" Red Brass Pipe contains 85 per cent copper and conforms to Government specifications for Grade "A" water pipe. The mark "Anaconda 85" is stamped in the metal at one-foot intervals throughout each length.

EVERDUR*

Everdur Metal is the original copper-silicon alloy group. It is manufactured by The American Brass Company in five standard compositions and in practically all commercial forms.

This high strength engineering metal is resistant to a wide range of corroding agents. Because of a versatile combination of useful properties, Everdur has become standard as a materal for equipment in many fields of engineering and industry.

In addition to their non-rusting properties and high strength, Everdur alloys possess many qualities not usually found in metals of this character. They are unusually resistant to general atmospheric conditions and other normally corrosive factors. Everdur alloys have excellent machining and working characteristics and can be fabricated into a variety of forms and shapes. Everdur alloys are available for oxy-acetylene, carbon and inert-gas-shielded arc welding.

Everdur Tanks—Everdur copper-silicon alloy is an ideal material for durable, rustless water tanks of every description—from domestic range boilers to large storage heaters for hotels, laundries, hospitals, textile plants, schools or breweries.

Everdur is made in all commercial shapes including annealed tank plates which have physical properties as given in *A.S.T.M.* Specification B96.

Minimum specification requirements for hot-rolled-and-annealed tank plates are: Tensile Strength, 50,000 psi.; Yield Strength (at 0.5 per cent elongation under load) 18,000 psi.; Elongation, 40 per cent in 2 inches.

Welds made with annealed Everdur tank plates meet the requirements for U68 and U69 construction in the *A.S.M.E.* Code for Unfired Pressure Vessels.

For additional data and names of fabricators address our nearest District Office.

EVERDUR FOR AIR CONDITIONING EQUIPMENT

Because of its strength and welding properties, Everdur may be substituted for steel and fabricated by substantially the same methods and with much the same equipment as steel.

Everdur metal has been used with marked success for fans and blowers, ducts, humidifiers, cast and wrought parts of other equipment items subject to corrosive influences.

EVERDUR LITERATURE

Descriptive literature containing much pertinent tabular data will be sent on request.

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* "Everdur" is a trademark of The American Brass Company registered at the U. S. Patent Office.

RESTRICTOR TUBE • FORMED TUBE PARTS • HARD COPPER TUBE CUT TO LENGTH • COPPER WATER TUBE IN COILS AND STRAIGHT LENGTHS • FITTINGS FOR TYPES K AND L TUBES • VIBRATION ELIMINATORS • CHARGING HOSE • FLEXIBLE REFRIGERATION TUBING CONDUIT • DIE PRESSED FORGINGS • COPPER, BRASS, BRONZE IN SHEETS, WIRE, RODS, TUBES AND SPECIAL SHAPES

Chicago Metal Hose Division

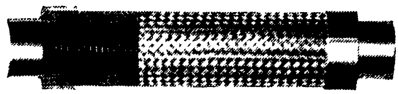
Flexonics Corporation

Maywood, Illinois

District Offices
Atlanta Boston Cincinnati Cleveland Detroit Ft. Worth
Los Angeles New York Philadelphia St. Louis San Francisco
Distributing Outlets in Principal Cities
In Canada: Flexonics Corporation of Canada, Ltd., Brampton, Ontario

REX Super Service

Vibra-Sorbers



Rex Vibra-Sorbers control vibration and reduce noise in refrigeration and air conditioning machinery. All-metal construction is liquid- and gas-tight, does not age, and has high corrosion resistance. Available in copper bearing alloy for use with Freon or Methyl Chloride; or steel for Ammonia systems.

SIZES: $\frac{3}{16}$ in. to 4 in. inside diameters.
BURST PRESSURES: 1,000 to 3,700 psi.
LENGTHS: Standard stock lengths. Special lengths available on order.
COUPLINGS: Stock units with male or female sweat fittings; also available with male pipe thread fittings.

REX-TUBE

Flexible Metal Hose

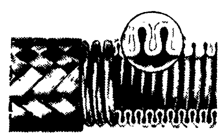
FOR

Diesel engine exhaust lines
Refrigeration tubing armor
Air blower ducting
Ventilating ducts
Control wire casing
Wiring conduit
Suction hose
General utility hose

Rex-Tube Convoluted Flexible Metal Hose Types have three basic formation patterns: square-locked, ball-bearing (or double-groove), and fully interlocked. Made of stainless steel, brass, steel, aluminum, bronze and other alloys. Packless and packed types. Sizes range from $\frac{3}{16}$ in. to 12 in. inside diameters, and lengths to suit requirements.

REX-WELD

Flexible Metal Hose



FOR

Steam Hose
Reciprocating flexible connections
Refrigerant loading, unloading and charging
Oil burner connections
Pressure lubricating lines
Conducting searching gases and liquids
Diesel engine exhaust lines
Misalignment correction

Rex-Weld hose types are manufactured from uniform wall tubing by a special CMH corrugation-forming process, Metals used are steel, bronze, and other alloys. Rex-Weld sizes range from $\frac{3}{16}$ in, to 12 in. inside diameter; with lengths and couplings to fit specific requirements. Especially designed for use under high temperatures and pressures, and where corrosive action is present.

Revere Copper and Brass Incorporated

Executive Office: 230 Park Avenue, **New York 17, N. Y.**

MILLS—Baltimore, Md.; Chicago, Ill.; Clinton, Ill.; Detroit, Mich.; Los Angeles and Riverside, Calif.; New Bedford, Mass.; Rome, N. Y.

DISTRICT SALES OFFICES—Atlanta, Ga.; Boston, Mass.; Buffalo, N. Y.; Cincinnati, Ohio; Cleveland, Ohio; Dallas, Tex.; Dayton, Ohio; Grand Rapids, Mich.; Hartford, Conn.; Houston, Tex.; Indianapolis, Ind.; Milwaukee, Wis.; Minneapolis, Minn.; New York, N. Y.; Philadelphia, Pa.; Pittsburgh, Pa.; Providence, R. I.; St. Louis, Mo.; San Francisco, Calif.; Seattle, Wash.

REVERE PIPE AND TUBE OF COPPER AND COPPER ALLOYS

For Heating, Air Conditioning, Plumbing, Revere Copper Water Tube, Types K, L, and M, meets Federal and *ASTM* specifications.

Types K and L furnished in hard and soft tempers.

Type M, ⅜ in. and above, furnished in hard temper only.

Type K soft temper tube is recommended for underground water service or fuel lines.

Hot water lines of Copper Water Tube lose very little heat to ambient air, hence the use of copper saves fuel.

Revere Red-Brass Pipe (Gov't Grade A) or Copper Pipe (both SPS) are recommended for piping systems where threaded connections are required.

Revere Dryseal Copper Tube is dehydrated and sealed. It is commonly used for Refrigeration and Air Conditioning Systems, fuel lines, compressed air lines, and general service work.

Furnished in dead soft temper, ⅛ in. to ¾ in. OD, it is easily bent and flared.

For Radiant Heating

Revere Copper Water Tube, furnished in 60 ft coils is easily bent to form sinuous coils for heating panels.

Long, one-piece lengths of copper tube reduce the number of couplings or joints required.

Small diameters of copper tube require less thickness of embedment in plaster.

For Condensers and Heat Exchangers

Revere Cupro-Nickel condenser tube has definitely been found superior for condensers, aftercoolers, and similar heat exchangers.

Similar tubes of Revere Admiralty Metal are widely used.

Revere Seamless Copper Tube is commonly used for finned tube coils.

For Industrial Piping and the Process Industry

Revere produces a wide range of pipe and tube made of copper and copper alloys for industrial use where high resistance to corrosion is required.

Solicitations for assistance in selecting piping material best suited to specific conditions for **Fuel Lines, Compressed Air Lines, etc.,** are welcome.

Silver-Brazed Joints

Revere ed-Brass Pipe or Copper Pipe is recommended where Silver-Brazed joints are required with standard pipe sizes.

Revere Copper Water Tube and standard soldered type fittings can also be Silver-Brazed satisfactorily and generally at less cost than heavier pipe.

Technical Advisory Service

Revere maintains a staff of technical men to assist engineers, designers, and contractors in the selection of suitable Revere products for various applications. Their services are available without obligation.

Technical Literature

Literature relating to many fields of application for **Revere** pipe and tube products is available upon request.

Two booklets on Radiant Panel Heating cover design procedure, and a third is in the form of a non-technical and unbiased discussion for lay readers.

Revere Copper Water Tube

STANDARD DIMENSIONS AND WEIGHTS

Size In In.	Type K			Type L		Type M	
	O.D in In.	Wall Thickness In.	Wt. Lb per Ft	Wall Thickness In.	Wt. Lb per Ft	Wall Thickness In.	Wt. Lb per Ft
¼	.375	.032	.134	.030	.126		
⅜	.500	.049	.269	.035	.198	.025	.145
½	.625	.049	.344	.040	.285	.028	.204
⅝	.750	.049	.418	.042	.362		
¾	.875	.065	.641	.045	.455	.032	.328
1	1.125	.065	.839	.050	.655	.035	.465
1¼	1.375	.065	1.04	.055	.884	.042	.682
1½	1.625	.072	1.36	.060	1.14	.049	.940
2	2.125	.083	2.06	.070	1.75	.058	1.46
2½	2.625	.095	2.93	.080	2.48	.065	2.03
3	3.125	.109	4.00	.090	3.33	.072	2.68
3½	3.625	.120	5.12	.100	4.29	.083	3.58
4	4.125	.134	6.51	.110	5.38	.095	4.66
5	5.125	.160	9.67	.125	7.61	.109	6.66
6	6.125	.192	13.9	.140	10.2	.122	8.92

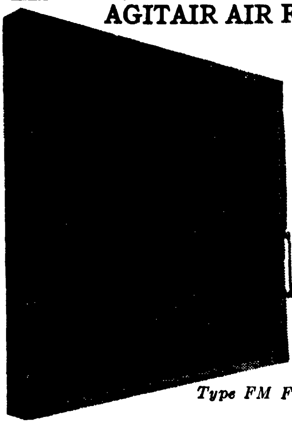
Air Devices, Inc.

Air Diffusers Exhausters Air Filters Filter Holding Frames Hot Water Generators

**17 East 42nd St.
New York 17, N. Y.**

**Agents in All
Principal Cities**

AGITAIR AIR FILTERS



Type FM Filter

**HIGH
VELOCITY
●
ALL
METAL
●
PERMANENT
●
CLEANABLE**

Designed along entirely new air filtering principles, the high velocity Agitar Type FM permanent, cleanable air filter assures an amazingly high dust arresting efficiency and dust-holding capacity coupled with sustained low resistance to air flow. This permits the Type FM to remain in service from two to three times as long as ordinary 2 in. permanent, cleanable filters. Although these new filters do not have to be cleaned as often, particular attention has been paid to their design to make cleaning easier and more thorough. They can be restored to top efficiency easily and quickly. Ruggedly constructed to withstand the mechanical abuse of cleaning. Panels and frames are accurately designed to prevent leakage around the filters.

HOW IT WORKS

High turbulence of many finely divided air streams is the keynote of Type FM air filter's new design. The media divides the air into countless fine streams and throws those streams into violent cyclonic turbulence. Each little "cyclone" centrifuges its dirt particles against countless viscous-coated "wiping surfaces" which virtually scrub the air clean by catching and holding the dirt. There is no straining action, hence no clogging.

HIGH VELOCITY

The Agitair FM Filter is designed to perform at highest efficiency at an approach velocity of 432 fpm—or 1200 cfm through a 20 x 20 in. filter panel. The efficiency of the FM is higher than conventional filters when operating at the lower design velocity of 288 fpm.

1/3 LESS SPACE REQUIRED

The ability of the FM to filter, with greater efficiency, 50 per cent more air at the high velocity of 432 fpm reduces the number of filter panels required. Now TWO FM's will do the work of THREE ordinary filters . . . ⅓ less space required . . . fewer units to be installed . . . fewer units to be serviced . . . overall installation and maintenance costs reduced to a minimum.

HIGHER EFFICIENCY

At the recommended velocities of other leading all metal viscous type filters, the Agitair FM has a higher dust arresting efficiency, which increases as velocities are stepped up.

LOWER RESISTANCE

The sustained low resistance of the Agitair FM means sustained peak volume of air for longer periods of time. . .no loss in air volume. . .no danger of unloading. . .clean filtered air at all times.

HIGHER DUST HOLDING CAPACITY

Employing a new formula for air filtration the new Agitair FM holds more dirt, from two to six times as much as ordinary 2 in. permanent, cleanable filters. No early clogging of air passages. . .less frequent servicing. . .lower maintenance cost.

LONGER SERVICEABLE LIFE

The Agitair FM Filter with its greater dust holding capacity, stays in service for longer periods of time; gives efficient performance for months instead of weeks or weeks instead of days.

Less frequent servicing—and rugged construction combine to give the Agitair FM Filter a much longer serviceable life.

TWO TYPES OF HOLDING FRAMES

Individual type: Designed and constructed for easy handling in single unit installations, and to facilitate "on the job" assembly, into a multiple unit bank.

Pre-Fabricated Type: Delivered completely knocked down for easy assembly, this Agitair holding frame has been especially designed for installations requiring an unusually large number of filter panels and where cramped and unusual conditions place a limitation on available space

GREASE FILTERS

An efficient, all-metal grease catching, grease holding media. Available in all sizes.

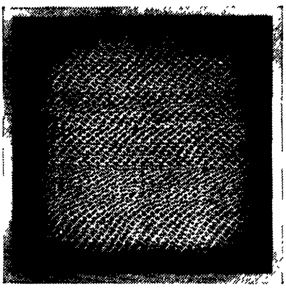
Air Filter Corporation

108G North Water St. Canadian Representative Milwaukee 2, Wis.
DOUGLAS ENGINEERING CO., Ltd. Montreal

Airsan

(formerly Aircor)

AIR FILTERS Permanent-Cleanable GREASE FILTERS



AIRSAN VIRO-CRIMP FILTER

The specially designed high velocity Airsan Viro-Crimp filter core is viscous type and constructed of horizontal layers of galvanized wire mesh so arranged as to assure a large filter area with no appreciable pressure drop.

Its exclusive Viro-Crimp is designed to operate at face velocities of 300-500 fpm at minimum resistance. Viro-Crimp edges are hemmed, an important safety feature that provides a smooth surface, extra rigidity and filtering area.

Airsan Viro-Crimp filters are easily cleanable. Drain slots are provided, hasten drying and aid in cleaning. Airsan Viro-Crimp Air Filters are of all metal construction throughout, the formed galvanized steel frames having full bronze welded corners and joints. Bulletin W 801.

ENGINEERING DATA

Initial Resistance			*Rated Efficiency*
Type F_1	(1 in thick)	.060 in. w.g. at 288 fpm	98.5%
Type F_2	(2 in. thick)	.065 in. w.g. at 288 fpm	98.5%
Type D_2	(2 in. thick)	.09 in. w.g. at 288 fpm	98.5%
Type D_4	(4 in. thick)	.10 in. w g. at 288 fpm	98.5%
Type W	(2 in. thick)	.07 in. w.g. at 300 fpm	98.5%

AIRSAN HOLDING FRAMES

Made of heavy gage metal complete with fireproof felt seal and locking device. Available in straight and V-banks. Prefabricated with Airsan slip-groove construction—eliminates felt between filter frames and cuts installation costs. Built to your specifications. Bulletin L601.

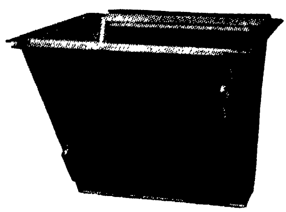
AIRSAN AIR FILTERS

Industrial *Domestic* *Commercial*

Airsan's expanded metal face plate acts as a lint arrestor to provide easier cleaning and servicing. It distributes air evenly over entire filtering area providing high filtering efficiency and dust-holding capacity with low resistance. Media is viscous type, permanent, cleanable, and is constructed of multiple layers of galvanized wire mesh to give maximum air resistance. All Airsan Filters have full bronze welded corners, galvanized steel frames and drain slots for quicker, easier cleaning.

Airsan Air Filters are available in standard 1 in. and 2 in. thickness—Bulletin L301. Also HEAVY DUTY filters for industrial and special applications in 2 in. to 4 in. thickness—Bulletin L401.

AIRSAN GREASE FILTER



Permanent cleanable type **Airsan** Grease Filters specially designed for range canopies, galleys, kitchens.

Removes grease at source, reduces fire hazard in exhaust ducts and prolongs life of fans, motors and other mechanical equipment. Assemblies for mounting on ceiling or wall, single or multiple units—includes holding frames, supporting angles and end seals. Bulletin L503.

Initial Resistance: .07 in. w.g. at 216 fpm
Efficiency Rating: 98.5% Stand.
Thickness: 2 in.

Write AIR FILTER Corp. for Complete Bulletins

AIR-MAZE CORPORATION

5200 Harvard Ave., **Cleveland, Ohio**

THE FILTER ENGINEERS

Representatives in all principal cities

TYPES AVAILABLE—During the past 27 years, Air-Maze engineers have encountered and solved nearly every air filtering problem. As a result, a wide variety of engineered filter designs are available, including Electromaze® electrostatic dirt precipitator, special panel air filters for ventilating, grease, railroad, aircraft and marine applications.

Efficiency—Efficiency of dirt arrestance of various filter panels varies with filter design, type of dust, amount of dust, method of charging filter with adhesive, etc. Specific information on any type of filter furnished on request.

Resistances—Filters available with initial resistances as low as .045 in. water at face velocity of 300 fpm. Resistance versus velocity curves available on request.

Face Velocities—Air-Maze offers filters designed to operate at peak efficiency at velocities ranging from 100 fpm to 700 fpm. Recommended velocities for each type of filter furnished on request.

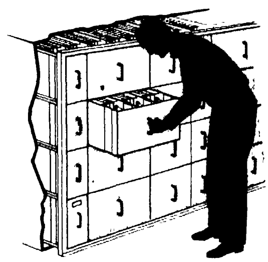
Sizes—Filter panels furnished in accepted standard sizes and in special rectangular sizes if required. Thickness varies with filter type. Most types furnished in both 2 in. and 4 in. thicknesses but available in certain special thicknesses if required.

Holding Frames—Holding Frames, complete with neoprene seals and choice of locking devices, available for use either singly or drilled for assembly into panel bank.

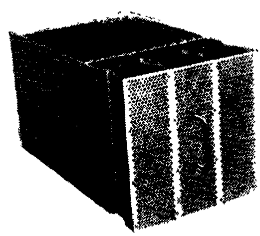
Cleaning and recharging filters easily cleaned by washing in hot water containing commercial detergent or with steam. Recharged by immersing in special adhesive or SAE 30–50 oil. Complete instructions available on request.

Write factory for name of nearest representative on information on any type of filter. Representatives are in most principal cities. See classified section of your telephone directory.

ELECTROMAZE ELECTRIC AIR CLEANER



For truly super-clean air, the Electromaze electrostatic dirt precipitator is the choice. It removes particles as small as 0.0000039 in. diameter (tenth of a micron), has a rated efficiency exceeding 90 per cent, as tested by the *National Bureau of Standards* Discoloration Method. Collects smoke, fumes, pollens, and soot.



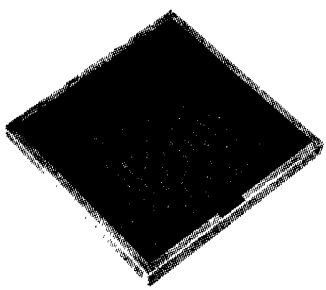
Electromaze consists of individual precipitator cells 4 in., 8 in., or 12 in. in width by 12 in. high. Each cell is a complete filter with built-in ionizing section. These cells slide into a framework, automatically making electrical connection by means of a plug in the rear.

This unique "desk drawer" construction provides greater flexibility of size, reduced erection cost, easier cleaning. Electromaze cells can be formed into banks for any cfm requirement to fit any space in increments of 4 in. wide and 12 in. high. The cells are easily cleaned in a separate tank when no duct drain or water connections are available. When duct drains are installed, cells may be flushed clean in place with ordinary city water pressure.

AIR-MAZE CORPORATION

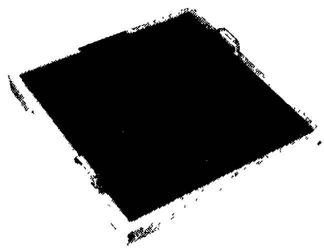
All the necessary power pack equipment, indicator lights, door interlock equipment, etc., is furnished with each installation.

Air-Maze Corporation manufactures a complete line of viscous impingement type ventilating air filters to meet your requirements within a wide range of limits of pressure drop, velocity, efficiency, weight, dirt holding capacity and price. A few types are illustrated on this page. THESE FILTERS ARE ALL-METAL, WASHABLE, FIRE RETARDANT.



KLEENFLO

For average-duty residential and commercial ventilating applications. Kleenflo panels are baffle impingement type, constructed of layers of crimped galvanized screen. Low initial cost. Available in 1 in., 2 in., and 4 in. thicknesses.

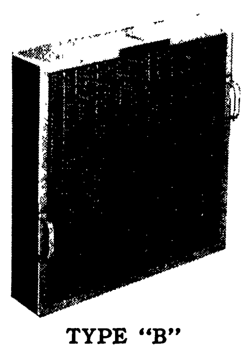


TYPE P-5

High velocity panel for handling a large volume of air at low resistance. Particularly recommended where space is limited. Available in heavier construction for railroad application (Model P-5RR). Also in bronze for maritime use. Available in 2 in. and 2⅛ in. thicknesses.

GREASTOP

Widely used in restaurant and hotel kitchens to trap airborne grease and prevent duct fire hazards. Progressive density of media provides large accumulation without undue restriction of air flow. Available in either "V" or wall angle assemblies or as individual filters.



TYPE "B"

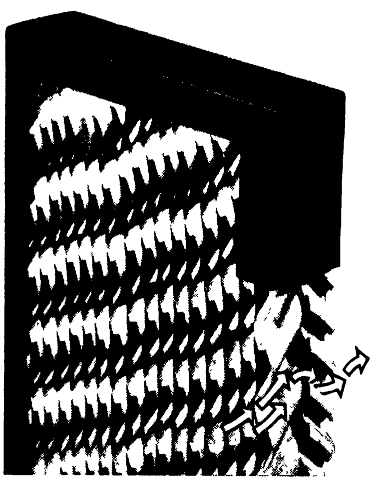
Heavy duty industrial filter for fresh air intakes. Has large dirt holding capacity, high efficiency, low pressure drop. All-metal construction. Easy to clean. Holds approximately 2 to 2½ times more dust than conventional filters. Available in 2 in. and 4 in. thicknesses.

AIR-MAZE CORPORATION

DETROIT AIR FILTER DIVISION

DUSTAY® DISPOSABLE AIR FILTER PANELS

Engineered and constructed to provide greater capacity and longer life. Filter medium is constructed of highly absorbent fiber board and designed to change direction of air flow twice as it passes through an outer and inner wafer. This two-wafer design, as shown below, causes a scrubbing action in the air, forcing it to contact the adhesive coated passageways.



"WICK ACTION" PROVIDES GREATER DIRT-HOLDING CAPACITY

Dust is collected on the inside cellular passages of a Dustay filter, not just the face surface alone. Dustay's highly absorbent filter medium holds more than a pound of adhesive compared to only 3 to 6 ounces for other disposable filters. As the dust is collected it absorbs adhesive from the reserve by capillary action. This constantly maintains a dust-collecting film on the wafer surfaces and substantially increases the filter's life. Full air flow is maintained because the dust's volume decreases when it is wetted out by the adhesive.

RESISTANCES

Filters available with resistances as low as .155 in. water at 400 ft per min. (Graphs showing resistance at various velocities are available for each type of filter.)

SIZES

Dustay filter panels are available in the following standard sizes:

20 x 25 in.	16 x 20 in.
20 x 20 in.	16 x 23 in.
16 x 25 in.	15 x 20 in.

Four separate types, varying in the size of the cellular passages, meet a wide range of applications. Three panel thicknesses, 1 in., 2 in., and 4 in. provide varied capacities to suit individual requirements. Specific information available upon request.

ADHESIVE

Odorless and stable. Has extremely high "wetting out" properties which maintain a tacky surface even after large amounts of dust have been collected.

For full details see your nearby Air-Maze representative, or write Air-Maze Corporation, Cleveland 5, Ohio.

American Air Filter

COMPANY, INC.

673 Central Avenue, Louisville 8, Ky.

HERMAN NELSON DIVISION

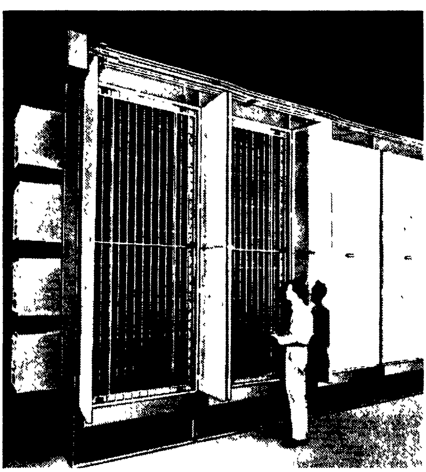
In Canada: Darling Brothers, Ltd., Montreal, Quebec

THE COMPANY: The American Air Filter Company, Inc. is recognized internationally as an authority on air filtration and dust control. In 30 years, its leadership and scientific "know how" have been responsible for installations of AAF equipment in every industrialized part of the world. Because its continuous research, high engineering standards and exclusive specialization have been consistently maintained, AAF equipment is used by leading companies in nearly every industry and is specified by leading architects and engineers for use in commercial and industrial air conditioning.

The American Air Filter Company Inc. manufactures a complete line of air-filtering and dust control equipment.' A few representative types are shown here. AAF's years of experience and the scope of its knowledge dealing with the elimination of air-borne particles in every form insures maximum efficiency in application, design, manufacture and installation . . . with a minimum of field engineering. Write AAF on all problems of air filtration and dust control, or its Herman Nelson Division (see pages 1164–1165) on heating or ventilating problems.

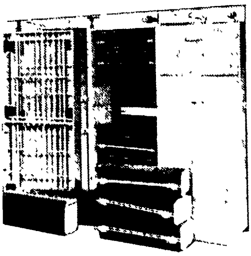
REPRESENTATIVE AAF PRODUCTS

ELECTRONIC PRECIPITATORS: For air cleaning problems requiring super-clean air AAF has developed three types of high-efficiency electronic filters. The complete line now available is the result of more than ten years of basic research and experimentation with electronic precipitation. It includes the self-cleaning Electro-Matic, the washable Electro-Cell with removable collector plates, and the Electro-Pl with the replaceable Airmat Paper medium.

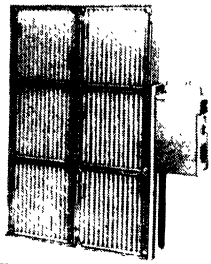


These three electronic filters are now used extensively in both industrial and commercial installations. The wide range of applications has made available engineering data and performance characteristics covering innumerable air cleaning and air volume requirements. This data simplifies materially the problem of specifications and selection of filter type.

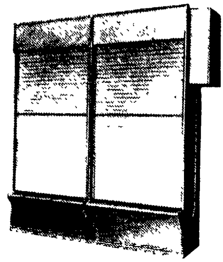
Electro-Matic Self-Cleaning Electronic Precipitator—The Electro-matic Precipitator is an automatic electronic precipitator combining advanced principles of electronic air cleaning with an exclusive self-cleaning principle. It eliminates the necessity of shutting down the filter for manual cleaning, minimizes the need for personal attention and permits continuous high-efficiency operation. It also allows the Electro-Matic filter to be built in standardized self-contained sections, easy to install and with all exposed parts of the filter casing electrically grounded for the protection of operating personnel. Send for Bulletin No. 250.



Electro-Cell Electronic Filter



Electro-PL Electromatic Filter



Multi-Duty Automatic Filter

Electro-Cell Electronic Filter—Differs in concept and design from all other plate type electronic precipitators. Installation has been simplified, performance improved and maintenance advantages provided. Built in vertical sections of two widths—2 ft and 3 ft over-all. Collector plate assemblies are removable; ionizers are hinged and extend the full height of the sections, preventing current loss. A choice of washing collector plate assemblies while in place, or removing assemblies for individual cleaning. Write for Bulletin No. 252.

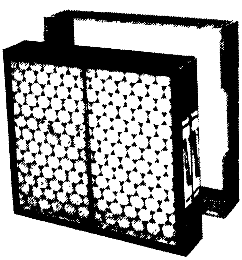
Electro-PL: An exclusive dry-type electronic air filter with charged Airmat collector element which combines air filtration and electronic preipitation in a single unit. Gives intermediate efficiency at lower cost, when the need for super-clean air is not indicated. However, it has twice the efficiency of uncharged dry-type filters. Available in Straight Bank or "V" arrangement to meet virtually any space or capacity requirement. Send for Engineering Bulletin No. 257.

Multi-Duty Automatic Filter provides outstanding features of performance and design and will accomodate either armored screen panels or die stamped louver panels available in three types. Offers advantage of uniformly constant air supply, fixed operating resistance and automatic operation. Ideal for ventilation and air conditioning service. Available in any size or capacity. Send for Bulletin No. 241-A.

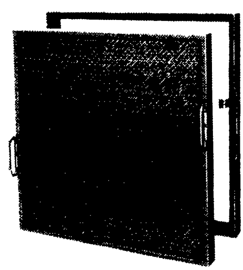
AMER-glas Replaceable Unit Filters. A new viscous impingement type for eliminating atmospheric dust from forced air heating systems. The highly efficient filtering media consists of continuous, curled and interlaced, extremely fine glass filaments, bonded with thermoplastic to form a thick resilient pad. The pad is sprayed with a special Viscosine and placed in a fiberboard casing between perforated metal grilles. The AMER-glas is nonflammable, sanitary and odorless and the Viscosine remains in a fluid-jell state for the life of the filter. Available in 15 sizes. Write for Bulletin No. 211-A.

Airmat Type PL-24—Airmat filters use standard Airmat medium, renewable after collecting dust load. Used both for comfort and industrial air conditioning. Available with unit frames to be set up to meet any capacity requirement or space condition. Send for Bulletin No. 230.

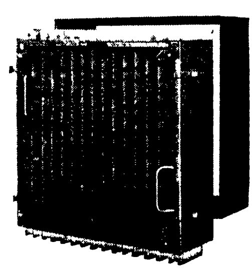
Type HV-2 Filter: A high capacity, low-resistance unit designed for velocities up to 500 fpm. Exclusive pyramid pocket media design eliminates through-air passages and gives uniformly high efficiency over wide range of air velocities. HV-2 has large dust capacity, long life, and decided advantages where space is limited. Available in three designs. Write for Engineering Bulletin No. 203.



AMER-glas Disposable Filter



American Type HV-2

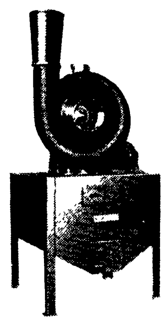


Airmat Type PL-24

AAF ROTO-CLONE DYNAMIC DUST PRECIPITATORS

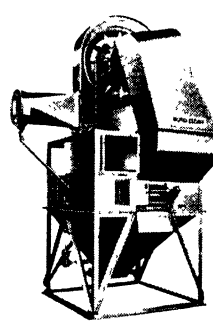
THE ROTO-CLONE is an exclusive, patented development of AAF combining the functions of exhausting, separating and storing dust in one simple, compact, self-contained unit. Most types of dust collectors or separators are so large that they are usually installed outdoors, which requires long pipe runs, increased power consumption, and higher installation costs. The Roto-Clone, however, eliminates expensive ductwork and reduces both installation and operating costs to a minimum. Available in a wide variety of types and sizes for either individual unit or central system installations and with wet or dry type collectors, the Roto-Clone has a proven record of economy and efficiency covering thousands of production line and individual applications. Write for profusely illustrated 35-page application and Engineering Bulletin No. 274-A.

Type D Roto-Clone—a dynamic precipitator which has been successfully applied to the dry collection of nearly every kind of granular industrial process dust. Combines functions of exhauster, dust separator and storage facilities in one compact, space-saving unit with turbine-like impeller as the only moving part. The impeller's numerous hyperboloid blades produce high mechanical and collection efficiency, which is unaffected by changes in air volume or operating speeds, remaining constant over entire pressure volume range. Adapted to floor or overhead installations and easily relocated, the Type D's low power consumption and trouble free performance effect real economies in space and operating costs. It is also adapted to individual unit applications at isolated dust sources or to a central system with main dust and branch connections for in-line production. Available in 13 sizes with capacities ranging from 100 to 15,000 cfm. Write for Bulletin No. 272A.



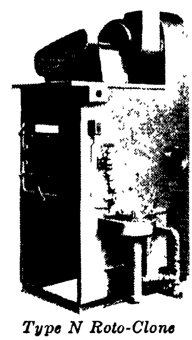
Type D Roto-Clone

Type W Roto-Clone provides the high dust separation efficiency where extreme fines and exceptionally heavy dust loads are involved. Widely used in the ceramic, chemical and mining industries and particularly in foundry dust control. Combining dynamic precipitation with integral sprays, which maintain a flowing film of water on all collecting surfaces, the Type W functions continuously at peak efficiency without interruption for reconditioning or servicing of any kind. It also delivers a constant exhaust air volume, while dust discharged as sludge creates no secondary dust problems. The compact, self-contained unit including exhauster, collector, and hopper requires minimum space and has a low installation cost. Type W is available in 12 sizes and three arrangements for capacities from 1,000 to 50,000 cfm. Write for Engineering Bulletin No. 274A.



Type W Roto-Clone

Type N Roto-Clone is a hydro-static type dynamic precipitator with dual washing and scrubbing action, engineered originally for efficient control of combustible dust. The Type N Roto-Clone has no moving parts, pumps or auxiliary equipment, but cleans by forcing air twice through an inverted S-shaped water curtain induced by the forced flow of air through a stationary impeller. Water is constantly reused and water level automatically maintained while entrained moisture in clean air is removed by special wide-spaced eliminators. There are no interior horizontal surfaces or narrow air passages to allow dust accumulation. Like all Roto-Clones, the Type N combines the functions of exhausting, separating and storing dust. In the latter case three hopper designs have been developed for different requirements in sludge disposal. Type N is available in 11 sizes with capacity ratings from 750 cfm to 48,000 cfm. Write for Engineering Bulletin No. 277A.



Type N Roto-Clone

American Solvent Recovery Corp.

Cassady and 8th Aves. **Columbus 3, Ohio**

Pur Air ACSC*

Air Recovery • Odor Removal • Air Purification

AIR RECOVERY

Air Recovery conserves costly conditioned air and reduces the amount of heating and/or cooling capacity required on new installations. Pur Air AIR RECOVERY equipment enables existing systems to handle increased ventilation load or a larger space without additional heating or cooling equipment. Substantial savings in fuel and power readily amortizes all installation costs.

ODOR REMOVAL

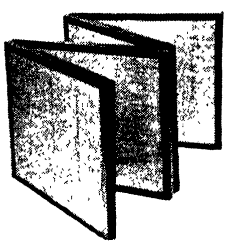
Pur Air ACSC Filters continuously remove odors and gases by positive adsorption. By physically removing the odors—stale, stuffy "used" air is revitalized for fresh use. Pur Air ACSC Filters will eliminate odor nuisances from fresh air intake, recirculated air, as well as air exhaust when air decontamination is desired.

AIR PURIFICATION

Air Purification removes airborne, vaporous impurities. Industrial wastes, processing odors, combustion odors, atmospheric irritants and contaminants are removed from the air with ACSC Air Purification.

PUR AIR ACSC FILTERS

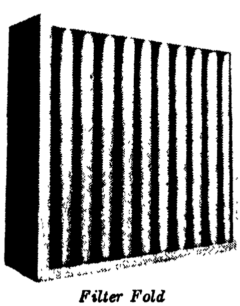
Special, high activity (increased adsorptive and retentive capacity) *activated coconut shell carbon* has been developed especially for air purification purposes. This high quality ACSC is available in a complete line of perforated, compact, easy to handle Pur Air Filters designed to purify all types of enclosed spaces. The manufacturer of Pur Air ACSC, Filters and Units produces *both* the activated coconut shell carbon and the complete air purification equipment. This combined, integrated function permits Pur Air to offer the greatest protection in better quality activated coconut shell carbon at lower initial and maintenance costs.



Filter Plate

PUR AIR FILTER PLATE

Pur Air ACSC Filter Plate permits installing the most carbon in the smallest space. For precision results, install one (1) Filter Plate per 100 cfm. Air resistance only 0.1125 in. (w.g.) Available in assembly frame components and also self-contained units.



Filter Fold

Activated Coconut Shell Carbon

PUR AIR FILTER FOLD "FF"

The Pur Air ACSC Filter Fold is a "package filter" that packs a lot of punch. In a small space—24 in. x 24 in. x 8¾ in.—it recovers (deodorizes) 1000 cfm of "used" air. For Recirculated Air that has tobacco odors, body odors, etc., in homes, public buildings, theaters and all types of enclosed spaces.

PUR AIR SALES REPRESENTATIVES

ALBANY, NEW YORK, F. R. Foote Co., Inc.
ALBUQUERQUE, N. M., Boyd Engineering Co., Inc.
AMARILLO, TEXAS, Snook & Aderton, Inc.
ATLANTA, GEORGIA, Crawley-Gorbandt Co.
BALTIMORE, MD., Lancaster, May & Co.
BINGHAMTON, N. Y., Dudley W. Gregg
BIRMINGHAM, ALA., S. C. Bratton
BOSTON, MASS., L. R. Geissenhainer
BUFFALO, N. Y., Van Ness Harwood
BUTTE, MONT., Sullivan Valve & Engr. Co.
CHARLESTON, W. VA., Engineering Products Co.
CHARLOTTE, N. C., Robert E. Mason Co.
CHICAGO, ILL., Zintel, Byfield & Co.
CINCINNATI, O., Russell R. Gannon Co.
CLEVELAND, OHIO, Dan B. Billington
COLUMBUS, OHIO, Russell R. Gannon Co.
CORPUS CHRISTI, TEX., L. S. Pawkett & Co.
DALLAS, TEX., W. E. Lewis & Co.
DAYTON, OHIO, Russell R. Gannon Co.
DENVER, COLORADO, E. P. Murr
DETROIT, MICH., George Q. McNamara, Inc.
EL PASO, Tex., Boyd Engineering Co.
HOUSTON, TEX., Jack Thomas Davis
INDIANAPOLIS, IND., Russell R. Gannon Co.
KANSAS CITY, MO., Manufacturers' Sales Co.
LITTLE ROCK, ARK., J. C. Lewis Co.
LOS ANGELES, CAL., Hess, Greiner & Polland
LOUISVILLE, KY., Russell R. Gannon Co.
LUBBOCK, TEX., Snook & Aderton, Inc.
MEMPHIS, TENN., J. B. Lammons
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MILWAUKEE, WIS., Zintel, Byfield & Co.
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NEWARK, N. J., John B. Hewett Co., Inc.
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OKLAHOMA CITY, OKLA., J. M. O'Connor Co.
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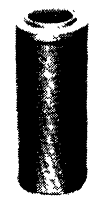
WICHITA, KANSAS, J. M. O'Connor Co.

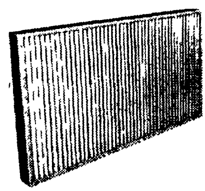
M-15 CANISTER

For replacement on existing systems. For new installations where canister-type ACSC equipment is specified. →



PARTIAL AIR BYPASS

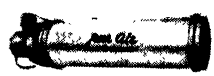
For application where available space is limited. Existing ventilation and air conditioning systems can be equipped without requiring extensive or costly alterations.



Partial Air By Pass

AIR FRESH'NER

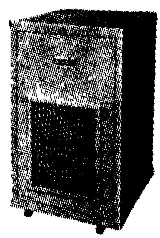
Purifies and removed odors from offices, toilets, hospital rooms, small laboratories, darkrooms, recreation rooms, etc. 65 cfm capacity serves up to 1500 cubic feet.



Air Fresh'ner Unit

PORTABLE UNIT

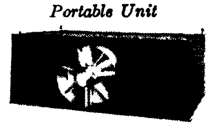
Suitable for offices, recreation rooms, hospital rooms, laboratories, animal rooms, toilets, locker rooms, libraries, vaults, food coolers, etc. 450 cfm and 660 cfm models available.



Portable Unit

FIXED MOUNT UNIT

Complete line of self-contained units available for warehouses, industrial rooms, cold storages, etc. 12 sizes from 220 cfm to 2640 cfm.



Fixed Mount Unit

STANDARD EQUIPMENT

Pur Air Filters and Units are available to purify air and to remove odors in practically all occupied spaces. Efficient air recovery and removal are functions of selecting the proper Pur Air filter for each application and available space.

CUSTOM DESIGN

Custom design and fabrication service is available for special requirements. Abnormal odor problems, industrial solvent recovery and compressed air filters are among the many projects handled. Inquiries are invited on any phase of air recovery, odor removal and air purification. Key personnel have security clearance for handling classified military documents.

W. B. CONNOR ENGINEERING CORP.

WHERE TO APPLY AIR RECOVERY

Air Recovery is simply the conversion of foul or stale air to fresh air. It has been used to advantage wherever air is conditioned to enhance comfort, raise production efficiency, extend food preservation or protect product quality. Depending on the source of contamination, Dorex Air Recovery Equipment has been installed to remove odors and other gaseous impurities from intake air, from recirculated air or from exhaust air.

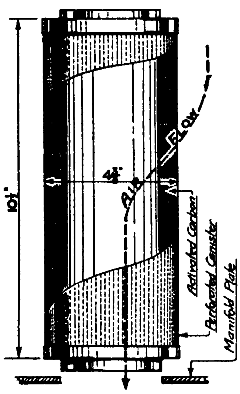


Fig. 1 Dorex Canister

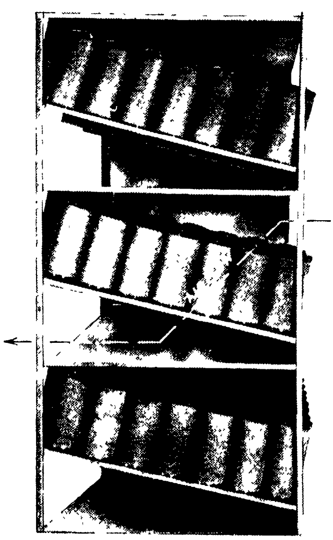


Fig. 2 Typical Canister Arrangement—side view

When applied to recirculated air, Dorex Adsorbers reduce the amount of unconditioned outdoor air needed for ventilation and effect savings in installation and operation costs. For example: Given an air conditioning requirement of an area of 20,000 cfm, of which it is assumed 14,000 cfm would be recirculated and 9,000 would be outdoor ventilation before installation of Air Recovery Equipment, it may be possible to cut the amount of unconditioned outdoor air intake to 2,000 cfm by converting 4,000 cfm of used, already conditioned recirculated air to fresh air. Figured for average temperate zones, this 33⅓ per cent load reduction would lower the installation and operating cost substantially because each 1,000 cfm of heated or cooled air that is converted saves: (1) 100,000 Btu of installed heating capacity, (2) 2.6 tons of installed refrigeration, (3) 1,800 kw hours of current per cooling season, (4) 1,500 gallons of fuel oil or 9 tons of coal, and (5) incidental water consumption and maintenance.

In existing systems, the application of Dorex Air Recovery Equipment will enable the system to serve a larger space or satisfy a greater conditioning load without increasing cooling or heating equipment and without consuming more fuel or power.

Activated Carbon Traps Gases and Odors

Activated carbon removes gases and odors by adsorption—a natural phenomenon which takes place when airborne gases or vapors come in contact with it. An instantaneous condensation occurs and the condensed impurities are held tenaciously until the carbon is forced to give them up in reactivation. For air conditioning purposes, however, the carbon must be especially processed, activated, and impregnated to meet the following specifications: (1) High activity (adsorptive capacity) for a wide range of gases and vapors; (2) High retentivity over an entire range of normal operating conditions; (3) No retentivity for water vapor; (4) Extreme hardness to avoid dusting in handling and in service; (5) High apparent density (in the granular form) of not less than 0.45; (6) Adaptability to repeated reactivation without appreciable loss in activity or retentivity. In actual use, Dorex activated carbon has removed and retained 95 per cent of all gaseous impurities from the air passed through it and maintained that efficiency from six months to two years, depending on the air contamination.

Equipment to Suit Individual Requirements

Dorex Air Recovery Equipment is available in a range of types and sizes to suit individual requirements. Each type is designed to hold the correct amount of activated carbon in a manner to provide a maximum area for decontamination, a minimum of air resistance and uniform air flow through the carbon. The average resistance to air flow ranges only from 0.15 to 0.2 in. wg.

TYPE H—Adaptable to Most Central Systems for Recovering the Freshness of Intake Air and Recirculated Air and for Eliminating Exhaust Nuisances

Type H Equipment—for complete decontamination of all air passed through it—consists of removable, perforated, carbon - filled canisters which are mounted in multiple on one or more supporting manifold plates. Fig. 1 shows a canister and its function; Fig. 2 shows a typical arrangement of canisters as installed. The flexibility of this arrangement makes Type H Equipment readily adaptable to a wide variety of space limitations.

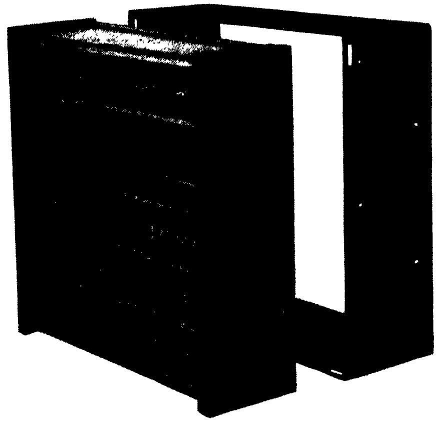


Fig. 3. Dorex Type C Air Recovery Cell

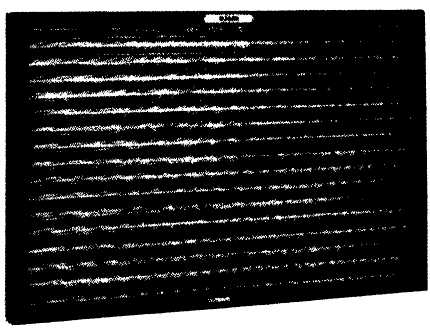


Fig. 4. Dorex G Panel

TYPE C—Equipment for Recovering the Freshness of Recirculated Air.

Dorex Type C Air Recovery Cells were developed to meet a need for a large capacity, easily handled and installed air purification unit. Each cell measures only 24 in. x 24 in. x 8¾ in. deep and completely purifies 1,000 cfm. They require no more engineering than that required for ordinary dust filters and can be mounted right along with them in either flat or "V" arrangement. (Fig. 3)

TYPE G—for "Package" Conditioners, Unit Heaters, Refrigerated Spaces, Airplane, Bus, Railway Car, and Marine Installations and Other Systems Where Space Is at a Premium.

These compact panels consist of sturdy metal frames, each housing a battery of exposed perforated metal tubes which contain the activated carbon. Standard units of one, two or three tube rows in depth are available in a range of stock sizes for arrangement in air ducts. (Fig. 4)

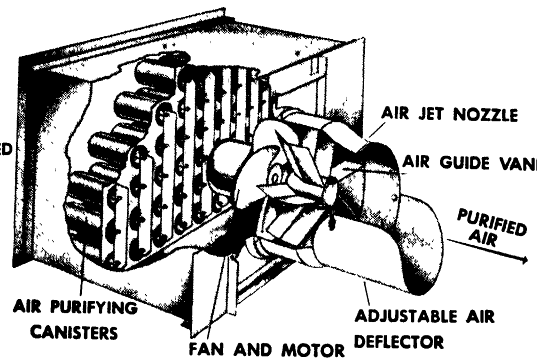


Fig. 5. Dorex Type D Storage Unit

TYPE D—For Extending Storage Life and Preserving Produce Quality in Refrigerated Storage

Type D units are designed to remove ripening gases, disease-causing gases and flavor-impairing odors from the air in cold storages, thus extending storage life and generally preserving produce quality. In apple storages, for instance, they have added 3 to 8 weeks to the keeping time of the fruit. In order to control storage atmosphere adequately and economically, Type D equipment was engineered to the following specifications: (1) Constant purification and recirculation of *all* storeroom air, (2) Thorough mixing of purified air with storage room air, (3) Continuous operation independent of other equipment in the storage space, (4) Flexibility in location, and (5) Self-contained unitary design to eliminate costly duct work or alteration.

Dorex units are portable and can be floor mounted or hung from walls or ceilings. The directional air jet creates an individual pattern for air mixing and distribution and avoids undue air impact on stored produce or fixtures. With the straightening vanes, the amount of "throw" can be adjusted up to a tight jet that reaches 90 feet away from the unit. The large quantity of air thus handled and the high aspiration it creates (five to six times the volume of supply air) results in a very efficient mixing of room and supply air. Dorex Storage Units are built in sizes and capacities to fit any storage space. All parts are either of non-corrosive metal or protected with corrosion-resistant coating. (Fig. 5)

TYPE PL—For Purifying Compressed Air

The Type PL Dorex Vapor and Gas Adsorber is designed specially to extract oil vapors, fermentation odors and other gaseous impurities from compressed air. It is especially designed to effectively remove air-entrained gaseous odors and impurities not eliminated by commercial filters, separators, after-coolers or receivers. (Fig. 6)

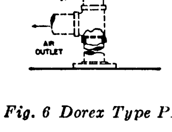


Fig. 6 Dorex Type PL

FOOD SAVER—for Refrigerated Coolers, Storage

The Food Saver is designed to extract gases (odors) from refrigerated coolers and other food storage spaces. Compact, sturdy construction. One unit serves up to 1000 cu ft of space. (Fig. 7)

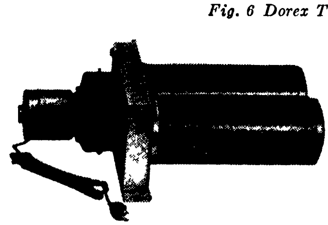


Fig. 7 Food Saver Unit

All Dorex equipment is covered by U. S. Patents Nos. 2,214,737; 2,303,331; 2,303,332; 2,303,333; 2,303,334 and others pending; Canadian Patents Nos. 383,986; 429,206; 395,611; 404,855; 410,088; 418,787; 443,235.

Nation-wide Sales and Engineering Service

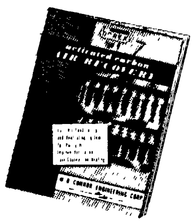
The W. B. Connor Engineering Corporation maintains a research laboratory, a staff of trained specialists, and district representatives in leading cities. Their services are at the disposal of consulting engineers, architects, air conditioning dealers, and plant engineers. Our staff can assist you in determining whether or not it would be to your advantage to install Dorex in a system you may be designing or improving.

Among Thousands of DOREX Users

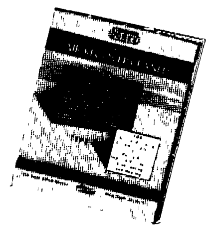
American Tel. & Tel. Co.	General Motors Corp.	Radio Corp. of America
Anheuser-Busch, Inc.	Hammermill Paper Co.	E. R. Squibb & Sons
Boeing Aircraft Co.	Lever Bros.	Union Carbide & Carbon Corp.
Bristol Myers Co.	Monsanto Chemical Co.	Western Electric Mfg. Corp.
E. I. du Pont de Nemours & Co., Inc.	Pennsylvania R. R. Co.	Western Union Telegraph Co

FREE LITERATURE

Shows How to Save Money on Air Conditioning

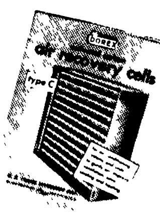


Bulletin 105 A



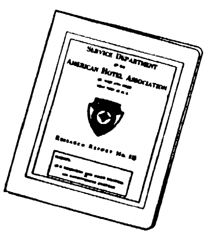
Bulletin 106 A

Bulletin 105A on Type H Equipment and **Bulletin 106A** on Type G Equipment are handbooks containing all the detailed drawings, charts and text necessary for the selection and application of Air Recovery Equipment and some typical applications. They also cover pertinent information on ventilation, oxygen requirements and recommended fresh air volumes for offices, stores, apartments, hotels, restaurants, night clubs, theaters, hospitals, and schools.



Bulletin 117-C

Bulletin 117-C contains complete information on Dorex Type C Air Recovery Cells.

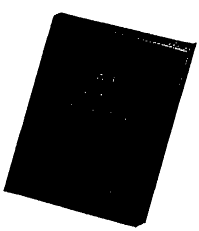


York Research Report

Air Recovery and Odor Control in Air Conditioning Systems is an unbiased research report prepared for the American Hotel Association by the York Research Corporation of Connecticut on their investigation into the effect of Air Recovery on air quality and conditioning costs.

New and Complete Textbook for Only $2

Completely revised and brought up to date, the new edition of **Air Conservation Engineering** goes thoroughly into the economics, functions and mechanics of Air Recovery. It can help you figure requirements to a wide variety of uses. Contains technical data needed in designing most applications, including valuable air conditioning tables and charts. Cites actual cases and describes the advantages to Design Engineers, Architects, Plant Engineers and Contractors.



Air Conservation Engineering

For your copies of the FREE literature outlined above or the textbook, Air Conservation Engineering, at $2, please send your request to our Engineering Dept. at Danbury, Conn.

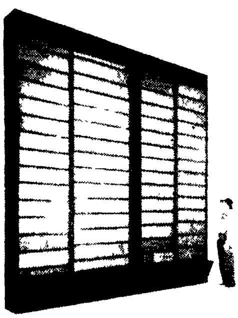
(See pages 1270 and 1271 for data on KNO-DRAFT Adjustable Air Diffusers.)

Continental Air Filters, Inc.

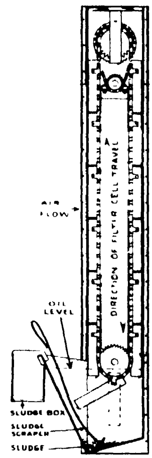
P.O. Box 1647 — **Louisville, Kentucky**

REVOLUTIONARY NEW FILTER MEDIA USED IN CONTINENTAL AIR FILTERS

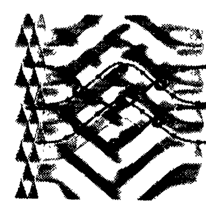
AUTOMATIC, SELF-CLEANING Air Filters range in rated capacities from 3,220 cfm to 153,900 cfm (larger units are available) having an efficiency of 91.3 per cent and a resistance of 0.28 W.G. Filters are the viscous impingement type with a continuous, rotating filter curtain arranged so that the air passes through both curtains in the same direction, made possible by the "Ferris Wheel" action at the top and bottom of the curtain.



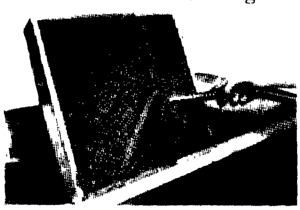
At the bottom each cell soaks in a semihorizontal position in the oil bath to loosen collected dust, then falls to a vertical position, causing a flushing action counter to the air flow. The dust in the oil settles as sludge to the bottom of the oil tank. The rotation of the curtain is controlled by an automatic electric timer. Power is transmitted from a fractional horsepower motor through a speed reducer and a chain and sprocket drive, with shear pin protection.



MEDIA as used in Continental Filter Cells is composed of die-formed, double corrugated metal strips placed under pressure in non-nesting relationship in sturdy holding frames. Efficiency of performance results from changes in direction and turbulence created as air streams cross and recross each other while passing through the media, insuring contact of all dust particles with oil coated surfaces. Low resistance results from relatively large, non-clogging passages.



UNIT FILTERS utilizing the same type media have the same advantages, with the additional advantage that they may be thoroughly cleaned by a cold water spray. No hot water, detergents, or special equipment are required. Filters are available with handles, latches and holding frames for bank installations, and in 2 in., and 4 in. thicknesses of popular sizes and capacities. Used without oil, these filters make excellent grease filters, or replacements for glass cells in air washers.



Glasfloss

155 E. 44th St.,
New York 17, N.Y.

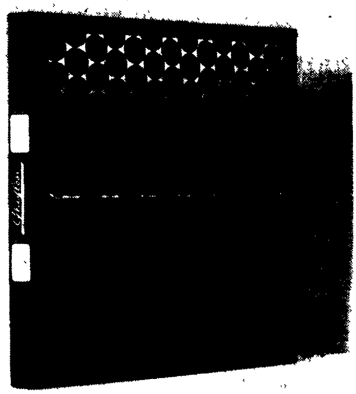
Glass Fiber, Disposable Air Filters For All Forced Air Heating, Ventilating and Air Conditioning Systems

← GLASFLOSS I-S AIR FILTER

Recommended for maximum air cleaning efficiency. To the GLASFLOSS Standard Filter pad of long, fine glass fibers has been added a strainer mat (see sectional illustration at left). The combination of these two elements brings air filtering efficiency to as high as 95 per cent. Resistance to the passage of air is normal.

GLASFLOSS STANDARD FILTER →

This type of filter is most widely used for home heating equipment and normal industrial installations (see illustration at right). The long, fine glass fibers provide a greater surface area to hold more dust. Extremely low resistance to passage of air. Available in all standard sizes.



← GLASFLOSS ROLL-PAK

Carefully planned for bulk users and for economy. Filter pads are cut to size from standard rolls, 40 in. wide 10 ft long and either 1 in. or 2 in. thickness, then fitted into permanent frames. The result in saving is as much as 30 per cent of air filter costs. Changes can be made quickly and the fluffy, fine glass fibers are easy on the hands.

For Complete details, write
Dept. HVG-52

Dollinger Corporation

Filters for Building Ventilation,
Air Conditioning, Engine Intake, Pipelines and
Many Other Special Applications.

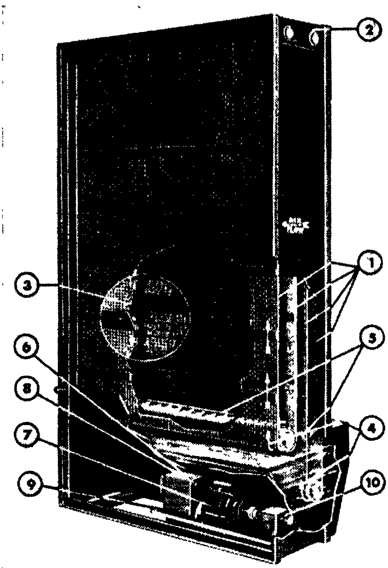
Representatives in Principal Cities

6 Centre Park — Rochester 3, N. Y.

STAYNEW MODEL A-3 AUTOMATIC FILTER

An endless curtain type oil-bath filter for handling large volumes of heavily dust-laden air at low cost. The efficiency of Staynew Model A-3 is unsurpassed among mechanical self-cleaning filters.

Operation and Features: *Double* filter curtains (1) carried on heavy roller chains driven by sprockets keyed to the shafts of the curtain rollers (2). These rollers float on ball bearings for quiet, frictionless operation. Curtains consist of removable panels (3) made of a single layer of bronze screen cloth to which are attached layers of woven copper mesh.



Model A-3 Automatic Filter (numbered features are referred to in accompanying description)

The first of the curtains is the denser, having about twice the impingement surface of the second or rear curtain. This first curtain acts as the filter and travels through the oil reservoir. The second curtain does not enter the reservoir, but acts only as a safeguard against oil entrainment. This design permits a direction of curtain travel such that cleaned panels (4) are always on the filtered air side. Therefore no dust can be carried across the back or return side of the front curtain to be blown off and carried on by the flow of air.

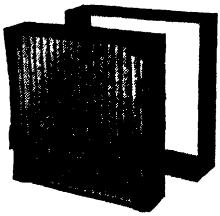
Patented, exclusive Staynew Compressed Air Curtain Cleaners (5) are available for special conditions.

Specifications

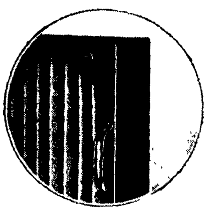
Model A-3 Filters are sectional and may be bolted together to obtain any required capacity. Sections come in two widths, 4 ft 3 in. and 2 ft 9 in. Curtain drive and control mechanism (6) arranged either as an integral part of filter unit or for remote mounting, includes a 1/6 hp motor (7) driving through a reduction gear and a momentary contact time switch (8) for testing and checking curtain travel and compressed air control. All are mounted on a common base plate (9) on clean air side of filter. Shear pin (10) is provided for protection of moving parts from accidental damage.

Drive motor operates for a few seconds at 15 minute intervals, and compressed air curtain cleaners can be arranged to operate simultaneously.

STAYNEW PANEL TYPE FILTERS



Model WKE Panel and Frame

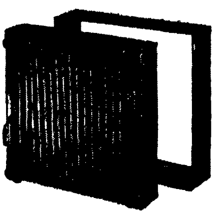


Handles and Latches

Model WKE: Dry-type finned panel filter for use in ventilation and air conditioning systems. Extremely large filtering area in relation to overall size. Adaptable to wide variety of filtering media—in fact, almost any medium obtainable in sheet form that can be crimped. Steel mesh on both sides of medium prevents sagging and makes the WKE *fire-resistant* (models available to meet Class I Fire Underwriters approval), and cleanable without possible damage from vacuum cleaning tool or cleaning nozzle. It may also be washed or dry cleaned when and if necessary. There are no cross bars, spacer bars, or other obstructions to interfere with the cleaning operation. Unaffected by temperature changes.

Filter cells are held in rigid box-type supporting frames of heavy gauge metal by spring-loaded cam-type locking latches. Two lifting handles are provided on each cell. Filtering medium supplied already crimped and cut to size. It may be inexpensively replaced in 2 to 5 minutes right at the filter bank—no special tools required.

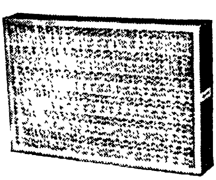
Frames are drilled so that they can be riveted together to form a flat bank, or by the addition of angle uprights into a "V" or staggered arrangement.



Viscous Model DPV Panel and Frame

Viscous Panel (Model DPV): A permanent type panel for air conditioning systems used in heavy duty industrial service. Filtering media consist of a series of layers of crimped galvanized screen cloth and woven mesh. These media when coated with PD-10 Pingene Filter Oil form an unusually efficient filter. Model DPV filters are cleaned easily with live steam or by washing in a suitable solvent. Spring-loaded locking latches and lifting handles are provided as in Model WKE.

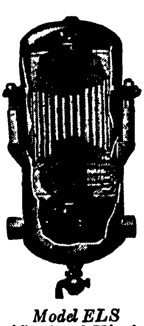
Both Model WKE and DPV cells are furnished in 2 in. and 4 in. depths in various standard sizes.



Model PVR-3 Kitchen Range Filter

Kitchen Range Filter (Model PVR-3): Solves the problem of grease collection in exhaust hoods over ranges, steam tables or cookers. Eliminates fire hazard and odorous, unsanitary conditions. Features exclusive Staynew "RL" steel filtering medium made in 3 in. deep units. High quality in every respect, yet low in price.

STAYNEW LIQUID FILTER



Model ELS (Sectional View)

Model ELS: Widely used for the filtration of cooling water to prevent clogging of spray nozzles. Exclusive, low-cost SLIP-ON INSERT easy to remove, clean, replace. Radial Fin Construction provides all possible filtering area in smallest possible space. Standard models available to handle up to 1000 gpm.

Representatives in Principal Cities

Complete Information from Factory on request

FILTERS FOR INTERNAL COMBUSTION ENGINES, COMPRESSORS, PIPE LINES; ALSO DUST COLLECTORS

Farr Company

Manufacturing Engineers

P.O. Box 10187, Airport Station

Chicago **Los Angeles 45, Calif.** **New York**

Manufactured under license by Control Equipment Co., Ltd., Montreal, Canada

FAR-AIR* FILTERS FOR ALL TYPES OF INDUSTRIAL USE

THE FAR-AIR FILTERING PRINCIPLE . . .

Herringbone-Crimp Media Design

FAR-AIR Filters obtain maximum efficiency through their herringbone-crimp media design. Zinc electroplated wire mesh is formed in alternate layers of flat and herringbone-crimp to make channeled, triangular openings facing the incoming air flow. Design allows greater air flow, turbulence and efficiency. Dirt builds up in the mesh on the entering side which diverts the air flow through the channels to clean mesh where the filtering process continues. The accompanying diagram of progressive dirt loading shows why FAR-AIR filters have higher performance, larger dirt holding capacity, lower pressure loss, easier cleanability and reduced maintenance costs.

Progressive Loading Feature

FAR-AIR CAPACITIES . . .

Recommended face velocities are 519 fpm although velocities up to 700 fpm may be used. Higher air velocities than recommended results in greater pressure loss and a shorter service period. Velocities lower than 350 fpm are not recommended.

The following chart can be used as a guide in estimating FAR-AIR Filter requirements. 4 in. filter thickness has twice the pressure loss of the 2 in. thickness.

NET FACE VEL. F.P.M.	CAP. C.F.M. Per Sq. In.	PRESSURE LOSS CLEAN In. H_2O 2" #44	NOMINAL FILTER SIZE — Standard Stocked Filters — FILTER CAPACITY CUBIC FEET PER MINUTE 16" x 20"	16" x 25"	20" x 20"	20" x 25"	Other Common Sizes 10" x 20"	16" x 16"	20" x 30"	25" x 30"
346	2.40	0.06"	625	795	800	1020	360	485	1235	1575
390	2.70	0.07"	700	895	900	1145	405	550	1390	1775
433	3.00	0.09"	780	995	1000	1275	450	610	1545	1970
476	3.30	0.10"	860	1095	1100	1400	495	670	1700	2170
519	3.60	0.12"	935	1195	1200	1530	540	730	1855	2365
563	3.90	0-14"	1015	1290	1300	1655	585	790	2010	2560
606	4.20	0-16"	1090	1390	1400	1780	630	855	2165	2760
650	4.50	0-19"	1170	1490	1500	1910	680	915	2320	2960
693	4.80	0-21"	1250	1590	1600	2035	725	975	2475	3155

MEDIA DESIGNATION AND APPLICATION

Type 44: 14 mesh zinc-electroplated steel screen. For ventilation, paint, lint, ink, oil.

Type 44F: Coated with vinyl or other special paint. For fume-resistant applications.

Type 44G: Standard filter, unoiled. For grease arrestance.

Type 44MZ: Screen painted with zinc chromate before and after assembly. Frame hot dip galvanized. Marine type—ventilation application.

Type C4C4H: All copper media and filter frame. Corrosion-resistant and water elimination applications.

Type A4A4: All aluminum media and filter frame. Stainless steel and monel filters for special applications can be supplied on special order.

* **Trade Mark Reg.**

Farr Company

Manufacturing Engineers
P.O. Box 10187, Airport Station
Chicago **Los Angles 45, Calif.** **New York**
Manufactured under license by Control Equipment Co., Ltd., Montreal, Canada

HOLDING FRAMES AND DIMENSIONS

FAR-AIR holding frames are made from 16 gage steel and have a baked enamel finish. Felt stripping around the inside flange assures tight filter fit and prevents air by-pass. Frames are interlocking and easily bolt together to form a rigid framework. Calking between frames in a bank is unnecessary. Clips hold filters firmly in place. Bracing is not needed in banks up to 5 filters high by 6 filters wide. Instruction sheets are included in every shipment.

STANDARD STOCKED SIZES (Actual)

2" and 4" Thickness

FILTERS Type 44 and 44H*	Shipping Weight (lbs.)	HOLDING FRAMES	Shipping Weight (lbs.)
15½ × 19½"	9½	16 × 20"	4¾
15½ × 24½"	11½	16 × 25"	5½
19½ × 19½"	11½	20 × 20"	5½
19½ × 24½"	13	20 × 25"	6

* Includes handles

FAR-AIR Filters and Holding Frames can be furnished in practically any size to meet exact specifications. Sizes over 25 in. × 30 in. are not recommended, however, as they are difficult to handle.

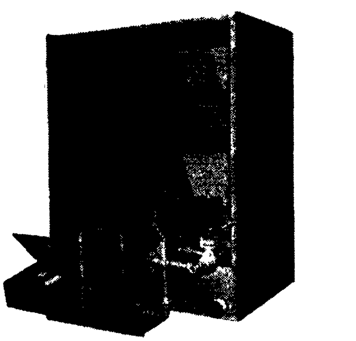
HOW TO SPECIFY STANDARD TYPE 44 FILTERS & FRAMES

Filter shall be permanent, impingement, washable, all metal, panel type. Media shall be zinc electroplated, 14 mesh steel screen, arranged in alternate layers of flat and herringbone-crimped screens, so that at no point is air flow restricted to flow through the screen, 4 layers of each per inch, ⅛-in. rod reinforced, and enclosed in a frame of 16 gage steel with flush mitered corners. Resistance to air flow of a clean filter shall not exceed 0.12 in. w. g. at 3.6 cfm per square inch of net face area.

Holding frames shall be factory built on 16 gage steel "T" section, bonderized and bake enameled, with felt air seal, interlocking edges, and filter locking device.

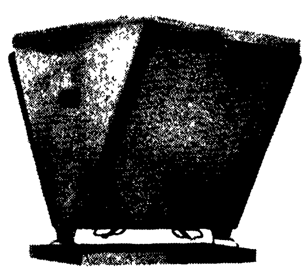
Filters shall be Far-Air Type 44 and frames shall be Far-Air Standard Interlocking as manufactured by Farr Company, Los Angeles 45, California.

SELF WASHING FILTERS



These *completely automatic* units wash and re-oil themselves on any frequency cycle desired. Each is made up from 4 in. thick corrosion-resistant filters in combination to meet any cfm requirement. Washing water and contaminated oil is immediately flushed away, eliminating messy sludge and oil sumps. There is no oil entrainment. Each unit has a safety deluge valve that automatically prevents fire from passing the unit. Installation is simple and inexpensive.

GREASE ELIMINATORS



FAR-AIR Grease Eliminators consist of two standard filters in a special adapter with drip pan. These units halt the deposit of highly combustible grease and lint in the air ducts, reduce maintenance, protect blower and fan, provide better ventilation and sanitation. Available in sizes and types to meet any cfm requirement.

Full catalog and technical information on any FAR-AIR product is available on request . . . or see *Sweet's Architectural File.*

Owens-Corning Fiberglas Corporation

Toledo 1, Ohio

OWENS-CORNING FIBERGLAS

DUST-STOP* AIR FILTERS

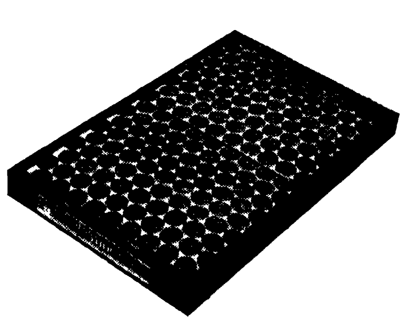
WHAT THEY ARE—Fiberglas* Dust-Stop Air Filters are replaceable, impingement-type filters. Dust-Stops are constructed of glass fibers that are non-absorptive, do not shrink or swell.

WHERE DUST-STOPS ARE USED—Dust-Stop Filters are used in all systems in which there is a mechanical movement of air—central heating, ventilating and air-conditioning systems, and mechanically-circulated warm air furnaces. Composed of numerous glass fibers coated with a viscous adhesive, Dust-Stop Filters catch the dust particles as the air moves through them.

ADVANTAGES OF DUST-STOPS—One of Dust-Stops' greatest plus values is fire safety. The fibers cannot burn and will withstand temperatures of 1,000° F. Being glass, the fibers are inorganic, chemically stable and resistant to corrosive vapors. A new, improved dust-catching adhesive is sprayed on the fibers which accounts for longer life with no loss of efficiency. All Dust-Stop Filters are now listed by *Underwriters' Laboratories*.

No special machinery is necessary to install Dust-Stops. Air filtering costs are further cut by Dust-Stops' high capacity to clean the air of dust, dirt, lint and pollen. Dust-Stops are readily adaptable to practically any system. Their low cost and easy installation make Dust-Stops the engineer's preferred filters.

FACTS ABOUT DUST-STOPS—Available in two standard types: No. 1 (1 inch thick) and No. 2 (2 inches thick). Both types are made in many sizes to fit practically every installation. Each filter is faced with a metal grille and bound on the edges with a fiberboard frame.



WHERE TO BUY DUST-STOPS—Check yellow pages of your phone book for name of the Dust-Stop distributor in your vicinity. If unable to locate a distributor write Dept. 44, Owens-Corning Fiberglas Corporation, Toledo 1, Ohio.

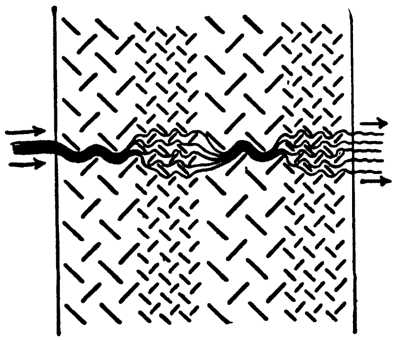
**FIBERGLAS* and *DUST-STOP* are trade-marks (*Reg. U. S. Pat. Off.*) *of Owens-Corning Fiberglas Corporation for products made of or with fibers of glass.*

Research Products Corporation

Madison 10, Wisconsin

AIR FILTERS FOR HEATING AND VENTILATING

Grease Filters for Kitchen Exhaust Systems—Paint Arrestor Pads



RP AIR FILTERS

Controlled turbulence design is provided in every RP filter. Scientifically staggered baffles cause thousands of sharp reversals of air flow in the filter. Media gradation, insures RP *depth loading*, an important longevity feature. Air borne particles are centrifuged against the baffles and are held by adhesive in the air filter and by the viscous qualities of the particles, themselves, in the grease filter and paint arrestor applications.

Constructed of slit and expanded material, RP filters are uniform in construction and performance.

ODOR REMOVAL

Exclusive RP Combination dust and odor removal is available in the D-O Air Filter—a replaceable filter—and with D-O Kote, the dust and odor removing adhesive for application to RP alumaloy washable filters.

"Snap-In" Grid Air Filter Installation Showing Self Seal Edge.

REPLACEMENT

All RP replacement filters employ a fiber media. The oversize RP "Self-Seal" idea was developed due to the ease and safety in handling the exposed media. Soft, resilient, and nonfracturing, the media only is sold in a refill pad for the famous RP Fiber Self-Seal, Air Filter for central systems and units and the RP Snap-In Grid for central systems (illustrated). The filters and pads as well as a conventional cardboard frame filter are available in one and two inch thickness. RP Paint Arrestor Pads, stop and hold overspray in spray booth operation.

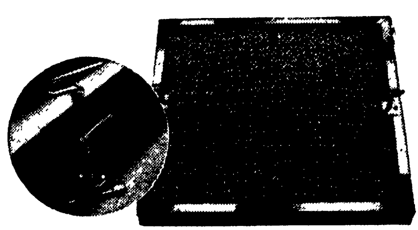
RP Paint Arrestors cut labor costs and maintenance down-time necessary with conventional paint spray booths. RP Paint arrestors have high overspray removal efficiency, versatility in application and low first cost.

WASHABLE

The RP Alumaloy E Z Kleen Air Filter is an economical selection for many applications. Like all RP washable air filters, this filter is extremely easy to clean.

RP Alumaloy Industrial air filters are multi-velocity filters, which have extremely low initial resistance and slow build-up resistance. Holding frames for filter banks, combination lift and lock handles, lift handles and water soluble filter coat adhesive are available.

Efficiency in excess of 99 per cent on grease vapors, gleaming appearance, ease of cleaning and light weight have made RP Alumaloy Grease Filters a volume performance leader in its field. A free comprehensive data booklet on kitchen exhaust systems is available.



RP Alumaloy Air Filter and Inset Showing Handle Lock

H. J. Somers, Inc.

6063 Wabash Ave., **Detroit 8, Mich.**

Agents in All Principal Cities

SOMERS Heavy Duty Industrial Filter

All Welded Vee Type
Patent No's. 2008800, 2130107

Somers Hair Glass Filters provide everything required in an efficient air-cleaning system.

Consider These Features

- High rating for dust, soot and bacteria separation.
- Require no adhesive, coating or impregnation.
- Indestructible in normal service.
- Minimum low-pressure drop.
- Odorless and non-absorptive.
- Fireproof.
- Washable.
- Permanent—Do not rot nor disintegrate.
- All welded zinc-plated 20 ga. steel frame.
- Metal protection strip on apex.
- Glass cloth between hot-dipped hardware cloth.
- Glass ribbon seal so air cannot short circuit.

Somers Hair Glass Filters consist of a 20 gauge hot galvanized frame holding galvanized wire cloth packed with hair-spun glass strands. The glass strands are flexible, do not break up and cannot be drawn into air stream.

Hair Glass being chemically inert, has no facility of absorption; it cannot rust and lasts indefinitely in service. Water either hot or cold may be used to clean it, without impairing its efficiency.

These filters eliminate the necessity, the expense and the inconvenience of periodic replacement.

SOMERS WASHABLE AIR FILTERS All Welded Vee Type Stock Sizes

Frame Size Height and Length	Frame Depth	Filter Surface Square Inches	For Average Dry Filter Installations	Wet Application
8" x 12"	3⅛"	288	288 C.F.M.	144 C.F.M.
12" x 12"	2⅞"	288	288 C.F.M.	144 C.F.M.
12" x 20"	3⅛"	720	720 C.F.M.	360 C.F.M.
15½" x 24½"	3 1/16"	1023	1023 C.F.M.	511 C.F.M.
15½" x 24½"	3⅛"	1674	1674 C.F.M.	837 C.F.M.
15⅝" x 24⅝"	2"	480	480 C.F.M.	240 C.F.M.
15⅝" x 24⅝"	3⅛"	1110	1110 C.F.M.	555 C.F.M.
16" x 20"	2"	384	384 C.F.M.	192 C.F.M.
16" x 21½"	3"	816	816 C.F.M.	408 C.F.M.
16" x 25"	2"	480	480 C.F.M.	240 C.F.M.
16" x 25"	2½"	624	624 C.F.M.	312 C.F.M.
16" x 25"	3"	864	864 C.F.M.	432 C.F.M.
16" x 25"	3⅛"	1344	1344 C.F.M.	672 C.F.M.
16" x 25"	3⅛"	1440	1440 C.F.M.	720 C.F.M.
16" x 25"	3⅛"	1632	1632 C.F.M.	816 C.F.M.
16" x 25"	3⅛"	1056	1056 C.F.M.	528 C.F.M.
18" x 18"	3 1/16"	864	864 C.F.M.	432 C.F.M.
18" x 18"	3⅛"	1134	1134 C.F.M.	567 C.F.M.
18" x 24"	3"	1080	1080 C.F.M.	540 C.F.M.
19½" x 19½"	2"	480	480 C.F.M.	240 C.F.M.
19½" x 19½"	3"	819	819 C.F.M.	409 C.F.M.
19½" x 19½"	3"	936	936 C.F.M.	468 C.F.M.
19½" x 19½"	3"	995	995 C.F.M.	497 C.F.M.
19½" x 19½"	3⅛"	1053	1053 C.F.M.	526 C.F.M.
19½" x 19½"	3⅛"	1170	1170 C.F.M.	585 C.F.M.
19½" x 19½"	3 3/16"	1696	1696 C.F.M.	848 C.F.M.
20" x 20"	2"	480	480 C.F.M.	240 C.F.M.
20" x 20"	2½"	600	600 C.F.M.	300 C.F.M.
20" x 20"	2¾"	780	780 C.F.M.	390 C.F.M.
20" x 20"	2¾"	840	840 C.F.M.	420 C.F.M.
20" x 20"	3"	960	960 C.F.M.	480 C.F.M.
20" x 20"	3 1/16"	1020	1020 C.F.M.	510 C.F.M.
20" x 20"	3⅛"	1200	1200 C.F.M.	600 C.F.M.
20" x 20"	3⅛"	1320	1320 C.F.M.	660 C.F.M.
20" x 20"	3⅛"	1680	1680 C.F.M.	840 C.F.M.
20" x 25"	2"	600	600 C.F.M.	300 C.F.M.
20" x 25"	2⅞"	1020	1020 C.F.M.	510 C.F.M.
20" x 25"	3⅛"	1560	1560 C.F.M.	780 C.F.M.
20" x 25"	3 3/16"	1800	1800 C.F.M.	900 C.F.M.
20" x 30"	3⅛"	1800	1800 C.F.M.	900 C.F.M.
20" x 30½"	3 3/16"	2400x	2400 C.F.M.	1200 C.F.M.
23" x 20"	3⅛"	1656	1656 C.F.M.	828 C.F.M.
23½" x 23½"	3⅛"	1621	1621 C.F.M.	810 C.F.M.
23¾" x 17¾"	3"	1068	1068 C.F.M.	534 C.F.M.
24" x 25½"	3⅛"	1872	1872 C.F.M.	936 C.F.M.
25" x 20"	3⅛"	1800	1800 C.F.M.	900 C.F.M.
26" x 23½"	2½"	936	936 C.F.M.	468 C.F.M.
26" x 23½"	2⅝"	936	936 C.F.M.	468 C.F.M.
26" x 34"	3⅛"	2652	2652 C.F.M.	1326 C.F.M.
28" x 33½"	2½"	1428	1428 C.F.M.	714 C.F.M.
29" x 33½"	3⅛"	3045	3045 C.F.M.	1520 C.F.M.
30" x 15"	3 3/16"	1800	1800 C.F.M.	900 C.F.M.
30" x 20"	3⅛"	1800	1800 C.F.M.	900 C.F.M.
30" x 24"	3"	1800	1800 C.F.M.	900 C.F.M.
31" x 23¼"	3¼"	3162	3162 C.F.M.	1581 C.F.M.

Other sizes also available. Send for complete stock size list.

Frames zinc plated for 100 hour salt water spray test. Refill may be inserted if necessary.

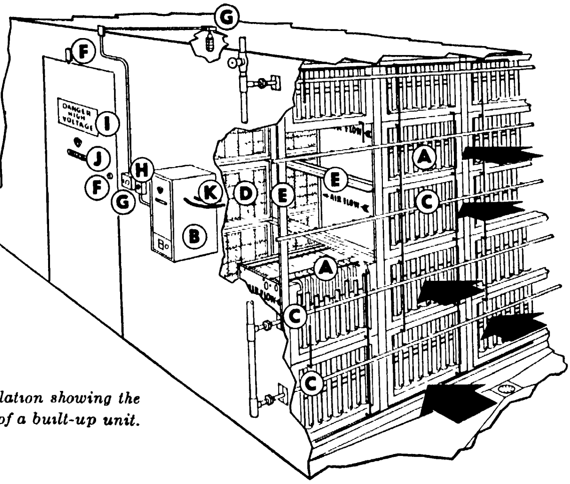
Quotations and further engineering data, including master holding frame drawings will be sent on request.

Just a *few* users of Somers Filters

Chemical Plants: American Viscose Co., American Zinc & Chemical Co., Celanese Corporation, Davison Chemical Corp.
Automotive: Cadillac Motor Car Co., Chevrolet Motor Car Co., Chrysler Corp., Fisher Body Corp.
Refrigeration and Air-Cond.: Frigidaire Corp., Norge Div.—Borg Warner Kelvinator Corp., York Ice Machine Co.
Ships: Amer. Shipbuilding Co., U. S. S. Saratoga, U. S. N. Lake City, Fla., U. S. N. Daytona Beach, Fla., U. S. N. Vero Beach, Fla., U. S. N. Jacksonville, Fla.,
Utilities and Municipalities: City of Kenosha, Michigan Consolidated Gas Co., Detroit Edison Co., New York Edison Co., Westchester Lighting Co.,
Dep't. Stores: S. S. Kresge Co., S. H. Kress & Co.,
Food Processing: Awrey Bakeries, Gilbert Chocolate Co., Kellogg Co.,
Manufacturers: Buffalo Forge Co., Burroughs Adding Machine Co., Clarage Fan Co., Curtiss-Wright Airplane Co., Glenselder Textile Co., Hoover Co., International Heater Co., Kearney & Trecker Corp., Kilian Mfg. Co., National Carbon Co., Inc., Pittsburgh Plate Glass Co., Rockford Machine Tool Co., Sunstrand Machine Tool Co.

TRION, INC.*

1000 Island Avenue, McKees Rocks, Pa.
IN METROPOLITAN PITTSBURGH



Typical Trion installation showing the various components of a built-up unit.

TRION ELECTRIC AIR FILTERS
(Electrostatic Precipitator)

Trion Electric Air Filters remove more than 90 per cent of dust, dirt, smoke, pollen and air-borne bacteria from air streams (*National Bureau of Standards* "blackness" test). Trion manufactures a complete size range of electric air filters for residential, commercial and industrial applications as follows:

BUILT-UP UNITS for commercial and industrial use. Complete unit consists of (A) ionizing-collecting cells for specified cfm at required cleaning efficiency; (B) power pack(s) of proper size and capacity, complete with electronic tubes, indicating instruments and magnetic circuit breaker; (C) complete water-wash cleaning system; (D) built-in bank of water proof, dry type mechanical after-filters; (E) rust resistant steel framework, factory assembled, then matched-marked before shipment; (F) door interlock and time delay screws; (I) "Danger High-Voltage" sign; (J) Trion name plate; (K) all high voltage cable and connectors.

CUSTOM-BUILT "PACKAGED" UNITS are advantageous in smaller installations where space is limited. Models up to 9330 cfm are constructed to specifications as a complete package, including drain pan and access door.

STANDARD "PACKAGED" UNITS for residential and small commercial use are available from stock in 4 sizes up to 4000 cfm at 90 per cent efficiency. Shipped fully assembled, complete with adaptors, interchangeable to accommodate air flow from right or left.

ENGINEERED EQUIPMENT designed and manufactured to meet specific requirements of by-product recovery, nuisance elimination, general purification of air or other gases. Design takes into consideration pressure, temperature, corrosion, dirt-loading, removal of collected dirt.

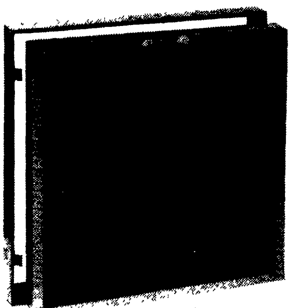
* Designers and manufacturers of equipment for electrostatic cleaning and purifying of air and other gases.

Vortox Company

Claremont, California

Panel Air Filters for all Industrial Applications

Vortox Panel Air Filters are of the cleanable, impingement, viscous-coated type with exceptionally low air flow resistance. They are made for use in air conditioning, ventilation, range canopies, engine intakes, and general air filtration.



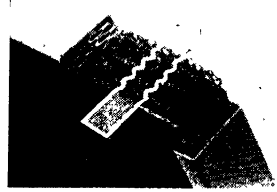
Vortox Panel Filter with Frame

Vortox Type VR Panel Air Filter illustrated above is the type used on engine intakes and in car bodies of diesel locomotives. It is constructed to withstand the severe usage of locomotive service. It has the exceptional characteristics as in all types of Vortox Panel Air Filters.

FILTER ELEMENT. Fabricated of elastic units of fine steel wire positively interlocked to provide a permanent filter, the element offers many advantages: (1) In a 2 in. x 20 in. panel there are from 9300 to over 15,000 feet of fine steel wire, depending upon the type. (2) Even distribution of filaments exposes innumerable viscous-coated surfaces to the air stream, thus obtaining the most effective cleaning for space occupied. (3) Proper spacing of filaments prevents clogging and assures maximum dust removal efficiency. (4) Structural strength and permanent resilience combine to withstand "packing" effect of vibration and pulsation.

EFFICIENCY. Vortox Panel Air Filters are efficient at both high and low air flows. Consequently to reduce costs of the original installation or to save space, fewer Vortox Filters may be used at higher air velocities. To assure a longer service life, more Vortox Filters may be used at lower velocities. In either case Vortox Panel Air Filters provide better cleaning at lower cost.

OPERATION. Numerous changes in direction of dust laden air cause the dust particles to impinge on the adhesive viscous-coated surfaces of the



filter element. Coarser particles collect near the entrance, while finer particles penetrate to a greater depth. As the entrance becomes saturated the cleaning action takes place deeper in the filter and the restriction is increased slightly. Increased restriction in certain sections of the filter diverts the air to cleaner sections, thus distributing the dust load for a higher over all efficiency and a greater total dust-holding capacity. EASY TO CLEAN. All dust and lint are easily and completely flushed from the filter by water or steam sprays.

Nominal Size of Panel Inches*	Actual Outside Dimensions Inches	Capacity Range In CFM**	Outside Dimensions Of Holding Frames Inches
16 × 20 × 2	15⅝ × 19⅝ × 1⅞	640 to 950	16¼ × 20¼ × 2
16 × 25 × 2	15⅝ × 24⅝ × 1⅞	800 to 1200	16¼ × 25¼ × 2
20 × 20 × 2	19⅝ × 19⅝ × 1⅞	800 to 1200	20¼ × 20¼ × 2
20 × 25 × 2	19⅝ × 24⅝ × 1⅞	1000 to 1550	20¼ × 25¼ × 2

*These filters are also made in 4-in. thickness.

**Some filter manufacturers specify very high velocities which are applied to certain internal dimensions of the filter. The range of capacities stated above are computed on the basis of average to high velocities applied to the total filter area using the actual outside dimensions of Vortox Panel Air Filters.

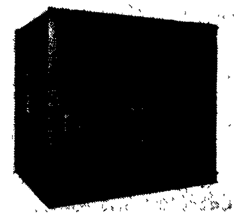
Westinghouse Electric Corporation
Sturtevant Division

Air Conditioning, Heating, Ventilating, Dust Control and Fume Removal Equipment, Electronic Air Cleaners, Compressors. Mechanical Draft Equipment

Hyde Park — Offices in Principal Cities — **Boston 36, Mass.**

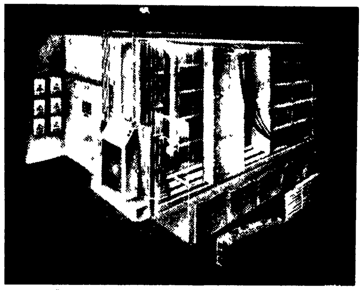
PRECIPITRON AIR CLEANERS

PRECIPITRON electronic air cleaners are applied to remove airborne dust and dirt from ventilating air. Used in commercial establishments, they reduce soilage, redecoration and cleaning costs. Industrially, they reduce contamination of processes and provide the cleaned atmosphere necessary for precision manufacturing.



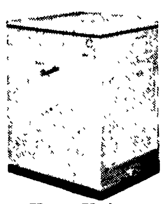
Encased Precipitron Horizontal Airflow

Horizontal Air Flow
Designed for installation in ducts—manual washing. Available in capacities of 1200 cfm and up with efficiencies of 90 or 85 per cent by Blackness Test. Power supply—115 volts, single phase, 60 or 50 cycle.

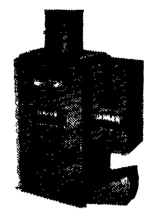


Commercial Precipitron With Semi-Automatic Wash

Semi-automatic Wash
Similar to horizontal air flow but equipped to perform the functions of washing, draining and applying adhesive automatically. Units for 90 per cent efficiency 40,000 cfm up, may have provisions for washing the PRECIPITRON without interrupting the air cleaning. In cases where the cleaner can be shut down for washing, units are available for either 90 or 85 per cent efficiency, in capacities of 24,000 cfm and up.



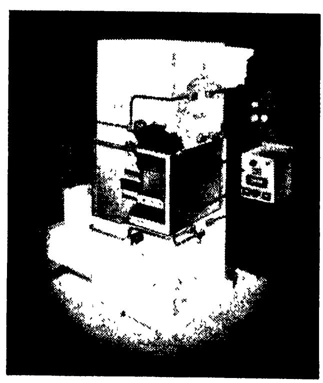
Home Unit



Oil Mist Control Unit

Vertical Air Flow
A PRECIPITRON arrangement for vertical, up or down, air flow—manual washing. Particularly suited to small and medium size installations where floor space is at a premium. Capacities of 1200 to 9600 cfm with efficiencies of 90 or 85 per cent.

Oil Mist Control Units
Designed to collect oil coolant mists generated by high-speed cutting, grinding or machining operations. Available with built-in fan for individual machine application, without fan for group collection system. Capacity—600 cfm.



Commercial Vertical Airflow Unit

Home Unit
Designed for residential use when connected in return air duct of forced warm air heating system. Capacity-1200 to 1500 cfm; efficiency 90 per cent. Operates from house current with total power consumption of 60 watts.

Wilson & Co., Inc.

AIR FILTER DIVISION

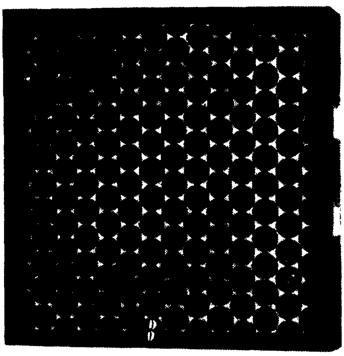
4100 S. Ashland Ave.

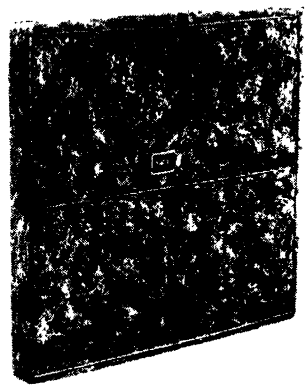
Chicago 9, Illinois

MEMBER NATIONAL WARM AIR HEATING AND AIR CONDITIONING ASSOCIATION INDOOR COMFORT

A Product of WILSON & CO. INC.

WILSON'S HAIR FILTER





The Popular WILSON HONEYCOMB

Here you have the popular HONEY-COMB air filter. Embodying the customary fiber bound edges and cellular metal face, the HONEY-COMB offers a new air cleaning principle to scientific filter design. Laboratory tests prove that no man-made fibre, or other non-absorbent surface, can equal natural hair as a medium for trapping and holding dust—with a minimum resistance to air flow.

The Famous WILSON EDGESEAL

Wilson EDGESEAL filters whip that old problem, marginal leakage. By springing to fit, due to highly processed animal hair, scientifically bound, EDGE-SEAL adds an extra 20 per cent of face filtering area. Even greater is the addition to holding capacity. This filter works perfectly in the home. Filter banks demanding maximum area, a complete seal, and top efficiency should always use EDGESEAL. This advanced filter reaches you ready to install for your greatest satisfaction and protection.

ENGINEERING DATA

Filter Opening Size	Rated Capacity C.F.M. at 300 F.P.M.	WILSON HONEYCOMB ACTUAL DIMENSIONS			WILSON EDGESEAL ACTUAL DIMENSIONS		
		Width	Length	Thickness	Width	Length	Thickness
10″ x 10″ x 2″	200	10″	10″	2″	10⅜″	10⅜″	2″
10″ x 20″ x 2″	400	9⅞″	19⅝″	2″	10⅜″	20½″	2″
15″ x 20″ x 2″	600	14⅞″	19⅝″	2″	15½″	20½″	2″
16″ x 20″ x 2″	640	15⅞″	19⅝″	2″	16½″	20½″	2″
16″ x 25″ x 2″	800	15⅞″	24⅝″	2″	16½″	25½″	2″
20″ x 20″ x 2″	800	19⅝″	19⅝″	2″	20½″	20½″	2″
20″ x 25″ x 2″	1000	19⅝″	24⅝″	2″	20½″	25½″	2″
20″ x 30″ x 2″	1200	19⅝″	29⅝″	2″	20½″	30½″	2″

WILSON HAIR FILTERS are manufactured in all regular and special sizes in 1 in. and 2 in thicknesses (½ in and ¾ in. thickness provided when required.)

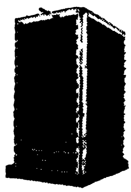
Binks Manufacturing Company

3130–36 Carroll Ave., Chicago 12, Ill.

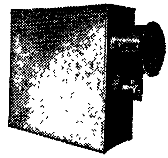
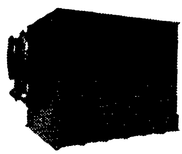
Representatives in all principal cities

Water cooling systems and nozzles . . . a size and type for every purpose

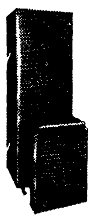
Binks atmospheric spray cooling towers
Small sizes, in a variety of standard units with capacities ranging from 10 to 125 gpm—larger units handle from 600 to 1200 gpm. Special designs furnished in sizes of exceptionally large capacity. Standard tower capacity and temperature performance are based on nozzle pressure of 7 lbs per sq in. Ask for *Bulletin 32*.



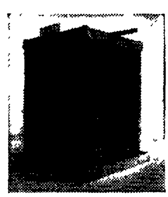

Binks horizontal induced draft cooling towers
The horizontal draft principle of operation results in a tower having relatively low height. Single fan units of the spray filled type have fans from 18 to 30 in. in diameter. Larger twin units have fans from 42 to 48 in. in diameter. Frequently installed in multiples for large capacities. Ask for *Bulletin 34*.

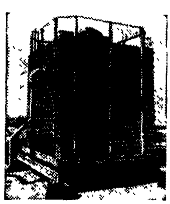
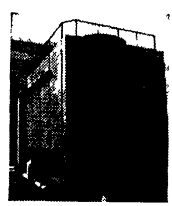
Binks spray filled forced draft towers
Small, compact, quiet, specially suitable for use with packaged air conditioners Nineteen sizes for systems ranging from 3 to 21 tons of refrigeration.
Ask for *Bulletin 35*.



Binks redwood atmospheric cooling towers
In single section units of solid redwood construction, are built in seven standard sizes to handle normal capacities of 20 to 125 gpm of cooling water. Ask for *Bulletin 40*.



Binks steel cased induced draft cooling towers
Towers of this type are of the spray filled or deck filled type—made in 20 standard sizes of sq ft rated area. Larger models are engineered to specification. Air propulsion assemblies for either type can be arranged for V-belt or reduction gear drive, as required.
Ask for *Bulletins 36 and 37*.

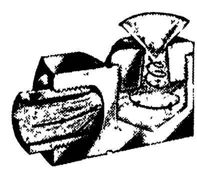



Binks induced draft masonry cooling towers
Engineered for large scale air conditioning systems in towers that harmonize with the architectural features of the building. Spray or deck filled. Ask for *Bulletin 38*.

Binks non-clogging Rotojet spray nozzles

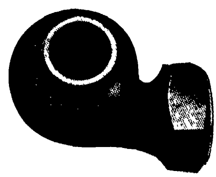
Non-clogging Rotojet nozzles are the heart of every Binks water cooling system. They account largely for the efficiency and satisfactory operation of Binks water cooling installations. In addition to cooling tower applications, Binks Rotojet nozzles have found a wide number of uses in brine-spray and quick-freeze refrigerating systems, air washing equipment, metal cleaning and treating machines, chemical plants, etc. Rotojets produce a uniformly fine fluid breakup in a hollow cone pattern. Standard small and medium Rotojet nozzles are machined from brass bar stock, but can be made on special order from monel, stainless steel, or other machinable metals. Large, heavy-duty Rotojet nozzles for use in large cooling towers and spray pond installations, are cast from high quality brass with precision machined threads and orifices. These nozzles may be cast in other metals for special purposes.

Binks small and medium capacity Rotojet nozzles



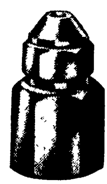
To fit $\frac{1}{8}$ to $\frac{3}{4}$ in. pipe connections. Regularly supplied in brass, with male or female threads, as specified. Discharge orifices are available over a considerable range for each size. Rotojet nozzles of this type are designed on the side inlet whirl chamber principle, which produces a fine fluid breakup and a uniform spray pattern. Full data is contained in *Bulletins Nos. 10 and 11.*

Binks heavy-duty Rotojet nozzles



To fit 1 to $2\frac{1}{2}$ in. pipe connections. Female threads only. Discharge orifices available in various sizes, from $\frac{9}{16}$ in. to $1\frac{13}{16}$ in. The totally unobstructed involute type of whirl chamber produces a uniformly fine water breakup at low pressures (5 to 7 lbs). Ask for *Bulletin No. 12.*

Binks Spra-Rite nozzles

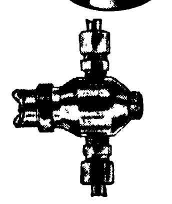


Produce a solid mass cone spray pattern. Small sizes for $\frac{1}{4}$ to $\frac{3}{4}$ in. connections are widely used for air washing, cooling, brine refrigeration, rapid evaporation processes, filtering systems, chemicals, etc. *Bulletin 19.*

Binks large capacity Spra-Rite nozzles

To fit 1 to 3 in. connections meet a variety of heavy-duty uses in blast furnace gas washers, vibrating and revolving screen coal and gravel washers and water cooling. *Bulletin 19.*

Binks pneumatic atomizing nozzles



Series 50 nozzles are designed for use wherever conditions of controlled humidity must be maintained, as in the storage of perishable products, paper storage and printing plants, textile mills, greenhouses, etc. Nozzles of all brass construction deliver round or flat spray and are designed for use with automatic siphon or pressure feed installations. Described in *Bulletin No. 16.*

Engineering service and technical bulletins

Binks engineering service and facilities are available without obligation or cost to architects, heating and ventilating engineers and builders. We welcome the opportunity to be of service in planning and installing cooling systems that will fully meet every requirement of performance. Give us the details of your problem and we will submit our suggestions.

Technical bulletins describing Binks water cooling systems and industrial nozzles are available for all units described on these pages. Write for the ones that will be useful to you. They will be mailed promptly, without obligation.

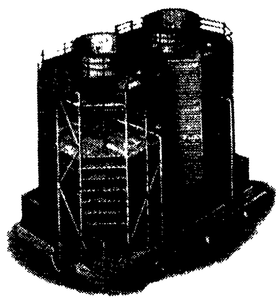
The Fluor Corporation, Ltd.

2500 South Atlantic Blvd.
Los Angeles 22, California

District Offices

New York City · Boston · Pittsburgh · Chicago · Tulsa · Houston · San Francisco

Counterflo Induced Draft Cooling Towers—Aerator Natural Draft Cooling Towers



Thirty years of intensive development in the design, fabrication and erection of cooling towers enable Fluor to offer both natural draft and induced draft cooling towers with comparable first cost based on comparable performance, guaranteed cooling efficiency, and dependable, long-life service.

COUNTERFLO INDUCED DRAFT COOLING TOWERS

DESIGN: Octagonal or rectangular shape with counter-flow design that permits longer air-water contact. Distribution system available in either up-spray or down-spray types. Octagonal shape eliminates "dead" corners and is complementary to modern building or plant design.

PERFORMANCE: Counter-flow principle plus efficient distribution system provides maximum performance far in excess of cross-flow or concurrent types. Distribution system in the upper portion of tower "flashes off" approximately 30 per cent of heat load before water starts down through tower. Performance to customer's specifications is guaranteed in every case.

MAINTENANCE: Decking withstands repeated washing and cleaning and the abuse of removing and replacing. Enclosed distribution system eliminates algae growth. Stainless steel fan blades support a man's weight at their tip. Heavy-duty, precision gear units are designed specifically for the conditions peculiar to cooling tower operation.

MANUFACTURE: Completely prefabricated. Tower structure based on 4 in. x 4 in. members with fan and gear supported on 6 in. x 6 in. beams. Two-inch redwood in fan deck and stack. Internal gusset plate and bolt-type structural joint take both tension and compression loads. Over 250 tower sizes and combinations are furnished from only 258 prefabricated parts to enable fast delivery and erection.

AERATOR NATURAL DRAFT COOLING TOWERS

DESIGN: Fluor Aerator Cooling Towers feature two distinct design advantages: (1) A patented Aerator panel which reduces drift loss to a minimum; (2) a patented bowed deck design for structural stability and greater water distribution.

PERFORMANCE: Trouble-free in operation. Cold water temperatures in the neighborhood of 15 deg. above the wet-bulb temperature present a normal cooling requirement. Performance to customer's temperature requirements is guaranteed.

MAINTENANCE: Designed for extremely low maintenance. Decks withstand severe and repeated washing. Closed distribution system retards algae growth. With proper treatment life expectancy is from 20 to 25 years.

MANUFACTURE: Completely prefabricated. Timber joint structural connectors meet the most rigid building code requirements. Structure is composed of select redwood. Members are 4 in. x 4 in. or greater. Aerator panels are ½ in. x 3 in. battens supported on 2 in. x 4 in. transcords. Hardware according to customer specification.

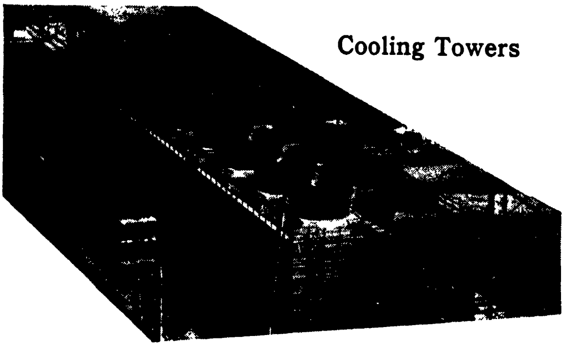
Foster Wheeler Corporation

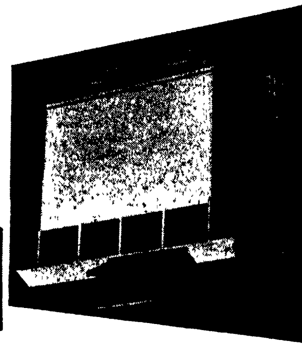
165 Broadway, New York 6, N. Y.

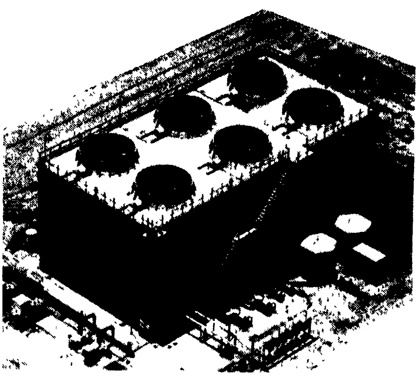
District Offices

Atlanta • Boston • Chicago • Cincinnati • Cleveland • Dallas • Detroit
Houston • Kansas City, Mo. • Los Angeles • Philadelphia • Pittsburgh
San Francisco • Washington, D. C.

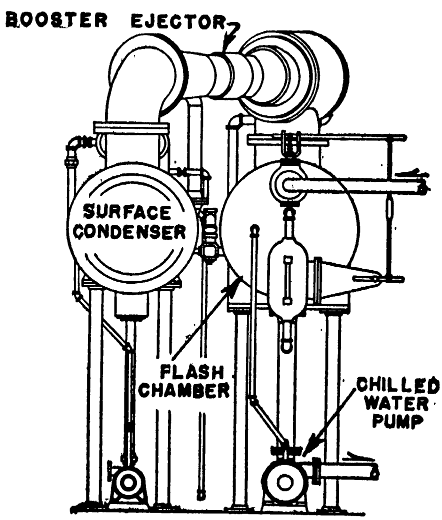
Cooling Towers







Foster Wheeler engineers have had many years of experience in the design and construction of high efficiency cooling towers for service in any geographical location and under all climatic conditions. These cooling towers meet the requirements of a variety of industrial needs such as those encountered in chemical plants, public utilities, textile manufacture, office buildings, department stores, and oil refineries. Foster Wheeler offers cooling towers of all types and capacities employing natural, forced, or induced draft. Recommendations are made only after careful study of the customer's particular requirements.



Vacuum Refrigeration

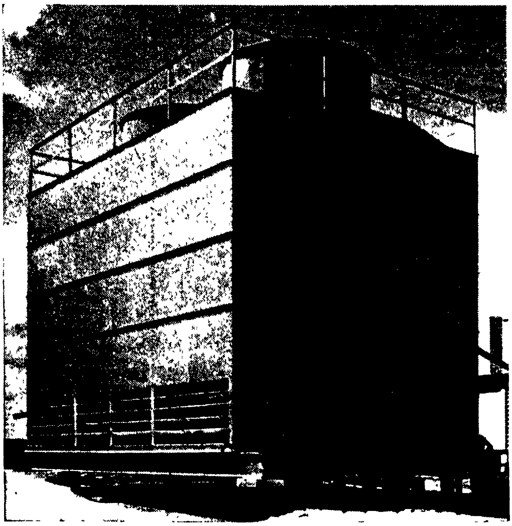
Schematic diagram showing arrangement of a typical vacuum refrigeration system. These systems, which cool water by subjecting it to high vacuum, supply chilled water for air conditioning or refrigeration. When a sufficient quantity of steam is available, vacuum refrigeration systems have several outstanding advantages such as low initial cost, low maintenance cost, absence of toxic and explosive refrigerants and, exclusive of pumps, no moving parts. Entire unit designed and constructed by Foster Wheeler.

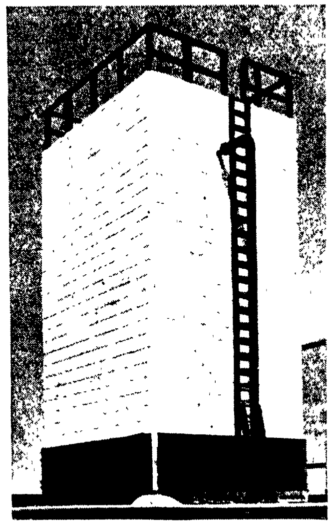
Lilie-Hoffmann Cooling Towers, Inc.

Exclusive Builders of Cooling Towers for 50 Years

4239 Duncan Ave., **St. Louis 10, Mo.**

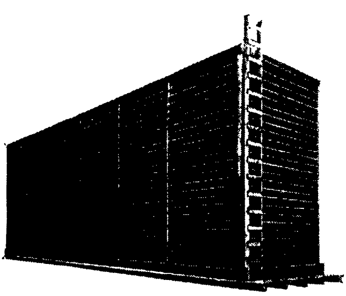
Two Modern Plants—St. Louis, Mo., and Plainview, Texas





INDUCED DRAFT TOWERS

New type filling reduces static pressure. Filling consists of slats, assembled in grids. Slat arrangement insures maximum wetted surface, minimum pressure drop of air flowing through tower packing, and uniform distribution of water and air over the entire effective area within the tower. Gravity distribution system requires only one riser pipe. Teco timber connectors develop 100 per cent working stress of members. Normally constructed of selected California redwood. All fans furnished by Lilie-Hoffmann are *P.F.M.A.* tested and certified.

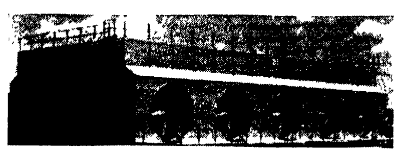


FORCED DRAFT TOWER

Three-tier zigzag pattern spray eliminator cuts drift loss to minimum. Fan openings covered by galvanized screenwire. Towers built in single or multiple cells. Continuous design improvement, based on installation studies, presents record of unfailing operation.

ATMOSPHERIC SPRAY TYPE

Generally offered in capacities up to 3000 gpm. Distribution by galvanized pipe header with smaller lateral arms, equipped with non-clogging spray nozzles. Water atomized as finely as possible. Sides and ends of towers equipped with narrow louvres, which fit into mast slots—no nails required. Recommended where cost is prime factor; and close approach to wet bulb is unnecessary. Rigidly braced and tied together to prevent warping and buckling. Withstand wind pressures up to 100 mph without use of guy lines.

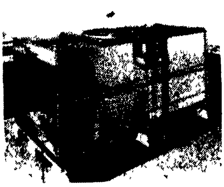


The Marley Company, Inc.

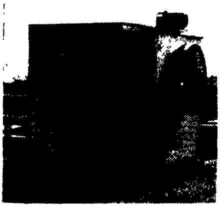
222 W. Gregory, **Kansas City 5, Missouri**

Representatives in All Principal Cities (Consult Classified Phone Directory)

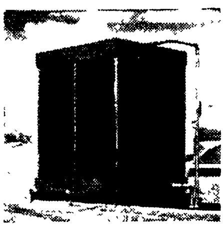
Water Cooling Towers and DriCoolers of All Types and Capacities. Spray Nozzles



MARLEY DOUBLE-FLOW AQUATOWER . . . The new "low silhouette," low pumping head tower for air conditioning and refrigeration jobs of 50 tons or more—a 150-ton model is only 7½ feet high. This entirely new tower provides Double-Flow efficiency combined with Aquatower simplicity. Produced in seven standard sizes of all steel or wood with asbestos cement board casing, it has open distribution, full height louvered sides, and all fittings and mechanical equipment are completely accessible. Write for Bulletin DFA-52.

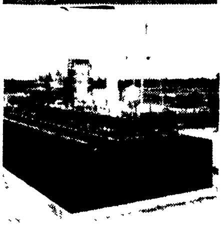


MARLEY Aquatower . . . For "packaged" cool water, the steel Aquatower packs solid performance for even the extra tough jobs. Aquatowers are available in ten sizes ranging from 3 to 60 tons of refrigeration. More than 7,000 Aquatowers are now in service throughout the country. They are carried in stock, shipped completely assembled and require no field erection. Large models may be readily disassembled to facilitate handling. Write for Bulletin AQ-52.

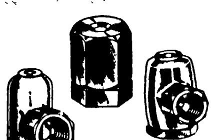


MARLEY Natural Draft . . . These heart quality redwood towers are available in two series. Economical to erect and operate, sturdy and long lasting. Assures top performance. *Series* 100 . . . used primarily for refrigeration and air conditioning small buildings, theaters, locker plants ranging up to 35 tons of refrigeration. *Series* 200 . . . is built in standardized units for larger capacities and is also available with atmospheric sections for indirect cooling of jacket water, oils and gases. Write for Bulletins 100-52 or 200-52.

MARLEY Conventional . . . These structurally excellent COUNTER-FLOW towers assure peak performance and operating economy. Built to high structural standards, they are equipped with Marley mechanical units and other exclusive Marley features. Marley Conventional towers meet every water cooling application and are also adaptable to indirect cooling with atmospheric sections. Available in redwood, steel or asbestos board casing. Medium to large capacity. Write for Bulletin C-52.



MARLEY Double-Flow . . . Has these advanced features in cooling tower design: Horizontal air-flow through full height louver walls, Open distribution system, Low pumping head, Minimum draft loss, Double service from each fan unit, Patented, nail-less filling which retains correct alignment without fasteners, Marley designed and built mechanical equipment, Utmost flexibility, Safe operation and low maintenance cost. Available in redwood or asbestos board casing. Double-Flows are used wherever large gallonage of water must be cooled economically and efficiently. Write for Bulletin DF-52.



MARLEY Nozzles . . . Marley patented, bronze Spray Nozzles achieve the fundamental requirement of every spray installation . . . a maximum breakup at lowest pressure . . . without moving parts. Three styles, one piece, two piece, humidifying. Write for Bulletin SN3-52.

EQUIPMENT DIVISION J. F. Pritchard & Company

Dept. 230, 908 Grand Ave., **Kansas City 6, Missouri**

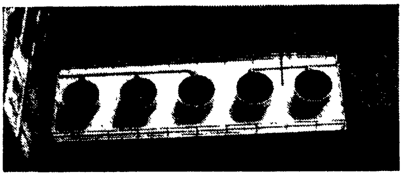
Houston, St. Louis, Chicago, Pittsburgh, Tulsa, New York

Representatives in Other Principal Cities

THREE LEADING LINES—COOLING TOWERS, HEATING EXCHANGERS,

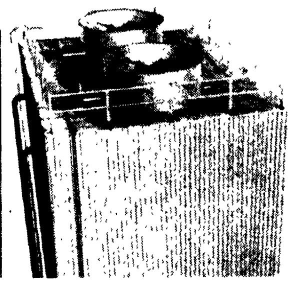
Gas Equipment for

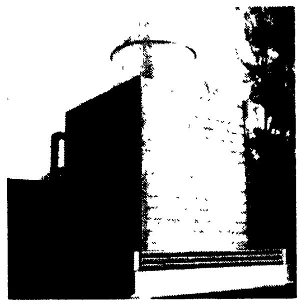
FIVE MAJOR FIELDS—Chemical, Natural Gas, Petroleum, Power and Refrigeration



COOLING TOWERS

Models for every capacity and application. Guaranteed performance ratings assure trouble-free performance and long life. Many patented features not available in any other tower. Specifications, prices and ratings furnished promptly, without obligation.



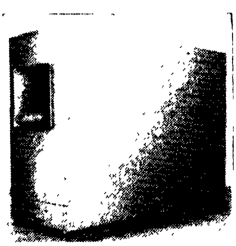


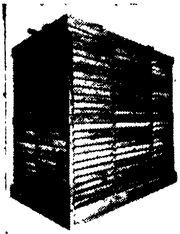
(Left) SERIES H INDUCED DRAFT TOWER. Horizontal flow design. Especially adapted to air conditioning, refrigeration, and industrial heat loads. Steel or Redwood construction. For roof or ground-level installations. Prefabricated for easy assembly. 30 to 100 tons capacity.

(Center) SERIES Q INDUCED DRAFT TOWER. Vertical counterflow model. Especially adapted to air conditioning, refrigeration and industrial heat loads. For roof or ground-level installations. 100 to 1500 tons capacity.

(Right) SERIES G INDUCED DRAFT COUNTERFLOW TOWER. Designed for medium capacity industrial applications. Uniform air distribution for maximum cooling effect. 20 to 100 tons. Prefabricated, Redwood or steel.

(Near Right) SERIES PB "COTOPACK" INDUCED DRAFT COOLING TOWERS. Ideally suited for indoor or outdoor service on small air conditioning or refrigeration installations as well as jacket water and air compressor cooling. Shipped assembled. 6 to 20 tons capacity.





(Above Right) SERIES SR COTOSPRAY* TOWERS. Type "G" is pictured above right. Type "F" also available. For roof or ground-level installation. Designed for air conditioning and refrigeration installations requiring 3 to 35 tons capacity. Simple bolted assembly. Finest Redwood with steel pipe distribution system. **Louvers lift out for easy access to interior.** Series SRE for specific heavy duty industrial applications also available.

Pritchard Accessory Equipment includes: **SEALDFLOW*** Fan Drives, **COTOSPRAY*** Nozzles and **COTOSPRINK*** Distributors, **POWairSAVER*** Fans **MINIMICER*** Louvers.

WRITE FOR BULLETINS

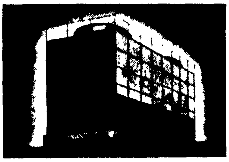
*** Registered trade names for specialized equipment produced by J. F. Pritchard & Co.**

Water Cooling Equipment Co.

New Hampshire Ave. and Weber Rd.
St. Louis 23, Mo.

Fabricating Plants
St. Louis, Mo.
Arcata, Cal.
Houston, Texas

Representatives In Twenty Eight Principal Cities



COOLING TOWER SELECTION is dependent upon many factors and variables including type of service, water pollution, wet bulb air temperature, temperature of water to be cooled, final temperature required, power source, location of proposed installation, adjacent equipment, and many other variables peculiar to each proposed installation. Therefore it is extremely difficult for an engineer to select the proper type and size of water cooling equipment without the assistance of a **WCEC** engineer.
Selection of the type of cooling tower should not be made on the initial cost basis alone, but instead on the cost of providing sufficient water cooling capacity to assure low operating temperature and high performance.
OUTSTANDING DESIGN FEATURES—(1) **Patented Cast-iron Timber Connectors** which develop the full strength of the timber joints, an accomplishment almost impossible to achieve otherwise. (2) **Patented Redwood Drift Eliminators** of the new multiple effect type remove the entrained moisture carried by the air. The eliminator blades are separated by spacers so arranged that the blades are free to expand and contract to avoid warping. No nails or metal fastenings are used. (3) **Patented Non-Clogging Low Pressure Spray Nozzles** provide maximum water break-up at lowest possible pressure. A true paraboloid of revolution is maintained in the whirl chamber thereby eliminating a constricted orifice which is the principal cause of clogging. (4) **Improved Nailless Grid Type Fill** repeatedly interrupts and refilms the water in its passage through the tower. The filling members are arranged to obtain maximum possible retardation of the descending water to allow a long and intimate exposure of the water to the air rising through the tower. The filling guides the air to completely utilize its heat removing capacity with minimum resistance. No nails are used in the filling.
MECHANICAL DRAFT TYPE COOLING TOWERS, either Induced Draft or Forced Draft, are designed for more exacting performance and operate independent of wind velocity. Overall plant economy results from cooler water possible with controlled volume of air. **Three Types of Induced Draft Systems** are built (1) Contraflow, (2) Multi-Stage, and (3) Low Head. Fans located on the top of the tower provide vertical movement of air across the filling, discharging air at high velocity to prevent recirculation. **Contraflow Type** are designed for large cooling systems needed by industry. Water distribution is by gravity splash system. Most extremely large installations are of this type. **Multi-Stage Type** has been designed by **WCEC** to meet the demand for extreme performance and low operation and maintenance costs. They employ low pressure head and have less air resistance, longer air travel, and consequently cooler water. **WLH Low Head Series Towers,** especially suitable for medium duty are compact, low in height, and are shipped completely pre-fabricated, ready for assembly, with minimum field labor. Twelve sizes are manufactured. **Forced Draft Systems,** especially suitable for corrosive or polluted waters, operate on the same principal as Induced Draft Systems except that the fans are located in the air entrance.
WATERFALL SPRAY TYPE ATMOSPHERIC COOLING TOWERS are fabricated in from 5 to 1,000 gpm capacity. Spray nozzles break-up the water with cooling performed by air movement controlled solely by atmospheric conditions. Thirty-three sizes are pre-fabricated for easy assembly. **Deck Type Atmospheric Towers** are manufactured in various sizes for any capacity or cooling requirement. The size is changed by varying the number of bays and cooling decks. All are made of All Heart California Redwood or other suitable material.
ENGINEERING ASSISTANCE is available to help solve the cooling problems in your plant. Our engineering staff is prepared to work with you in the proper selection of a water cooling tower that will meet your requirements with (1) guaranteed maximum efficiency, (2) economy of operation, (3) conservation of water. Write for our Catalog.

American Moistening Company

ESTABLISHED 1888

Providence 1, R. I.

ATLANTA, Ga. BOSTON, MASS. CAMDEN, N. J. CHARLOTTE, N. C.

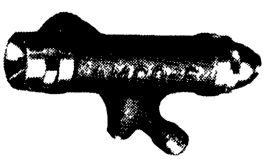
AMCO

Air Conditioning Systems—Humidification, Evaporative Cooling and Central Station

Because Amco installs both ductless and duct systems, you can rely on Amco engineers to give you sound, impartial advice. If you already have a modern, efficient humidification system and desire COOLING, an Amco engineer will probably point out the advantages of an Amco ductless system in which you discard nothing and make only a modest addition to your present humidification system. On the other hand, if your proposed installation calls for a unit duct or a central station air conditioning system, Amco can handle the job. In either case you will get reliable, unbiased advice and expert installation of a system tailored to your needs.

A few of the many AMCO products with a Long Record of Dependable Performance

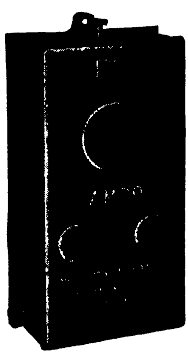
Self-cleaning Atomizers; Humidity Controls; Amtex Humidifiers; Evaporative Cooling Units; Mine Sprays; Fabric and Paper Dampeners; Electro Psychrometers; Sling Psychrometers; Hygrometers; Atometer.



AMCO ATOMIZER, NO. 6

New Automatically Self-Cleaning

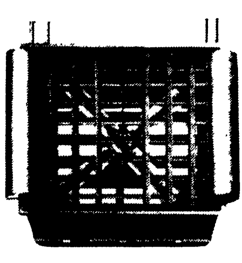
AMCO No. 6 atomizer retains all the time-tested features of the popular No. 5 unit . . . plus additional features which assure improved spray quality. Its better performance and ease of maintenance make this automatically self-cleaning atomizer superior for new installations and replacements.



AMCO HUMIDITY CONTROL

Compressed Air Operated

A unique hygroscopic element, which responds primarily to humidity and not to temperature, forms the heart and brain of the Amco Humidity Control. Its sensitive impulses, amplified by a simple pneumatic circuit, respond automatically to changes in relative humidity of one or two per cent of the desired value, with a minimum of care and supervision. Successful operation in many hundreds of mills throughout the United States attests to its ruggedness and reliability.



AMCO EVAPORATIVE COOLING UNIT

The Amco System of evaporative cooling contributes to smooth production at high speeds in two ways; it maintains the percentage of relative humidity best suited to the fibre and process involved, and at the same time promotes the comfort and efficiency of personnel by obtaining the maximum practical cooling effect from evaporation. It does this by introducing outside air into the room in varying amounts, regulated in accordance with climatic conditions and inside requirements.

A ductless system—very flexible and portable. Can be applied in conjunction with an existing humidifying system.

Monarch Manufacturing Works, Inc.

2509 E. Ontario St., **Philadelphia 34, Pa.**

SPRAY NOZZLES FOR WATER AND OIL

NON-CLOG AIR WASHER NOZZLES

produce an exceptionally efficient, evenly distributed hollow cone spray. Single large tangential inlet to swirling chamber minimizes any possibility of clogging. Also available in ⅜ in. to 1 in. pipe sizes inclusive, and of Brass, Stainless and Monel.

Capacities: Gallons per Hour

Sizes			Lbs. Operating Pressure					
Pipe	Orifice	Lead	10	20	30	40	60	100
¼"	69	69			2.9	3.3	4 0	5.1
	61	61		4.0	5.0	5.5	7.0	8.7
	61	53	4.4	6 2	7.5	8.3	10.3	13 0
	53	53	5.8	8.2	9.8	10.9	14.1	17.3
	49	49	9.1	11.1	13.2	15.0	19.5	24.6
	3/32"	3/32"	11.8	16.6	20.4	23.4	29.0	36 5
	⅛"	3/32"	19.5	27.6	33.3	39.1	49 0	60.0
	⅛"	⅛"	24.3	34.6	42.8	50.0	64.0	78 5
	11/64"	⅛"	31.5	46.1	57.0	64.1	81.9	105
	3/16"	5/32"	52.2	75.0	92.5	109	138	189
	7/32"	3/16"	78.0	112	138	163	205	257
	7/32"	7/32"	82 0	121	152	180	225	300

Fig. 631

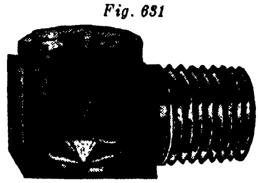


Fig. 629

AIR CONDITIONING AND OIL BURNER NOZZLES

Water Capacity in Gallons per Hour

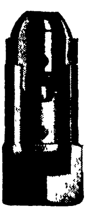


Fig. F-80

Nozzle No.	Lb Operating Pressure				
	25	40	60	80	100
1.35		.57	.69	.83	.92
1.65		.75	.89	.99	1.12
2.00		.94	1.14	1.28	1.40
2.50		1.13	1.45	1.64	1.86
3.00	1.03	1.39	1.62	1.85	1.95
3.50	1.36	1.77	2.11	2.46	2.80
4.00	1.56	2.00	2.42	2.77	3.16
4.50	1.86	2.32	2.77	3.21	3.68
5.00	2.20	2.88	3.57	4.09	4.59
5.50	2.22	2.96	3.75	4.31	4.78
6.00	2.55	3.35	4.01	4.78	5.23
7.00	2.90	3.91	4.60	5.17	6.00

Produce finest breakup possible with direct pressure only. Capacities above are on water. "Nozzle No." is capacity on 34 second Saybolt viscosity oil at 100 lb pressure Larger sizes up to 60.00 gph and smaller sizes down to 0.60 gph.

Furnished of all Brass for Water—Stainless Steel tip and disc for Oil. Standard with ⅛ in. or ¼ in. female pipe Brass adapter and Monel gauze strainer.

SPRAY POND NOZZLES

For recooling condenser water, etc. Operate on pressures from 5 lb upward. Made of Cast Red Brass and in pipe sizes 1 in., 1½ in., 2 in., and 2½ in. Capacities from 4.1 to 88 gpm at 7 lb pressure.

Write for Detailed Catalogs

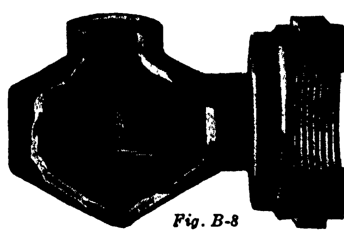


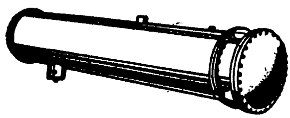
Fig. B-8

ACME INDUSTRIES, INC.

Jackson, Michigan

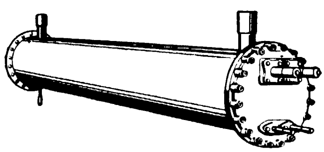
Representatives in Principal Cities

CONTINUOUSLY SERVING THE REFRIGERATION INDUSTRY SINCE 1919



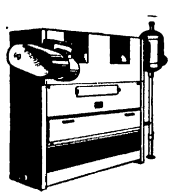
FREON CONDENSERS AMMONIA CONDENSERS

Shell and Tube type for use with Ammonia, Freon or other Refrigerants. Standard or special designs to meet varying water temperatures available and condensing temperatures desired.



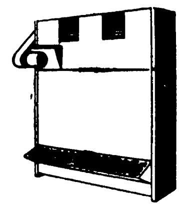
DRY-EX COOLERS

Refrigerant in Tubes, Solution baffled through shell. For cooling water, brine, Glycols or Alcohols by direct expansion of refrigerant.



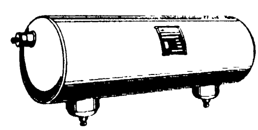
BLO-COLD INDUSTRIAL UNIT COOLERS

Blo-Cold Models are available for either medium temperature or low-temperature applications. *All fabricated steel parts are hot-dip galvanized after fabrication.*



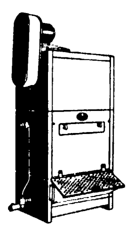
EVAPORATIVE CONDENSERS

All prime surface for Freon or Ammonia Refrigerants—Heavy Gage Sheet Metal Casings, especially processed for Maximum Resistance to Rust and Corrosion. Capacities to 100 tons. *All fabricated steel parts are now hot-dip galvanized after fabrication.*



HEAT INTERCHANGERS

Shell and coil units for small capacities, shell and tube units for large installations. 16 standard models from one ton to 180 tons capacity.



COOLING TOWERS

Twelve sizes of induced draft cooling towers to fit practically any application. Sturdily built, hot dipped galvanized for long life.

Acme Industries, Inc. also manufacture Flooded Water and Brine Coolers, Hi-Peak Water Coolers, Pipe Coils, Flow-Cold Liquid Chillers, Oil Separators, Receivers and Fin Coils.

WRITE FOR CATALOG ON ANY PRODUCT

AEROFIN CORPORATION

410 So. Geddes Street

Syracuse 1, N. Y.

AEROFIN

Standardized Light-weight Heat Exchange Surface

Branch Offices

Boston, Cleveland, Chicago, New York, Philadelphia, Detroit, Dallas, San Francisco and Montreal

Aerofin is the modern Standardized Light-Weight Encased Fan System Heating and Cooling Surface originated by *Fan Engineers* to meet the present and future requirements of this highly specialized field. All Standard AEROFIN Units are furnished as completely encased Units, ready for pipe and duct connections. The patented casings are built of pressed steel and are exceptionally strong and rigid, protecting the Unit from all the strains of pipe connections and expansion or contraction in service. The casings are flanged on both faces, top and bottom, and template punched for bolting together adjacent Units, or for duct connection.

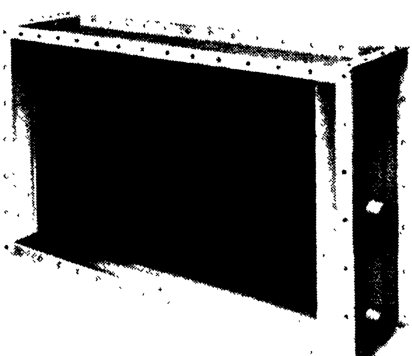


Fig. 1

Aerofin Non-freeze heater (Fig. 1) is non-freeze, non-stratifying spiral fin coil built into casing for air conditioning units or for installing in ducts. May be installed either horizontally or vertically. Used on any two-pipe steam system for preheating or reheating. Modulating control on preheaters.

Available in 13 lengths and 3 widths, from net face area of 2.76 sq ft to 26.28 sq ft.

Fig. 2

Flexitube Aerofin (Fig. 2) is distinguished from all other developments by its off-set tubes, so arranged as to absorb all expansion and contraction strains.

Headers—Steel.

Tubing—5/8 in. O.D. copper, admiralty or aluminum.

Joints—Where admiralty or copper tubes are used together with bronze or steel headers tubes are brazed to headers. Where both aluminum tubes and headers are used tubing is welded to headers.

Casings—Copper, aluminum or galvanized iron.

Design—Constructed with headers on opposite ends making possible installation of units with tubes horizontal or vertical.

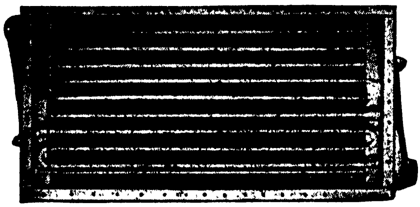


Fig. 3

Universal Aerofin (Fig. 3) is distinguished by its "S" bend construction of tubing, units designed with steel headers on opposite ends, the ends of the "S" bends being connected thereto by compression nuts, the bends taking care of the expansion and contraction of the tubing.

Recommended where close control is desired.

Headers—Pressed steel.

Tubing—1 in. O.D. copper or admiralty.

Casings—Copper, aluminum or galvanized iron.

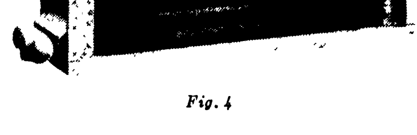


Fig. 4

Aerofin Heavy-Duty Industrial Heating Coil (Fig. 4) for use where extra-rugged coil is needed for close control. Steam pressures from 25 to 450 lb gauge; temperatures to 550 F.

Headers—Pressed steel.

Tubing—1 in. O.D. heavy copper.

Casings—12-gauge galvanized iron.

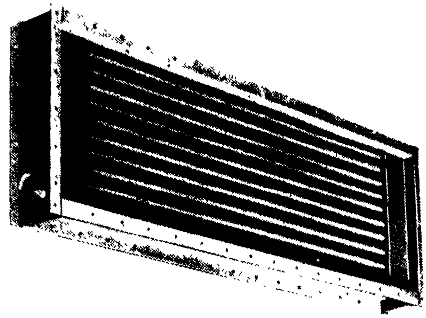


Fig. 5

Booster Aerofin (Fig. 5)—straight tube type, single pass construction for pressures from 1 to 200 lb gauge.

Headers—steel.

Tubings—⅝ in. O.D. copper.

Casings—copper, aluminum or galvanized iron. Recommended where small coils are needed or to raise the air temperatures in branch ducts.

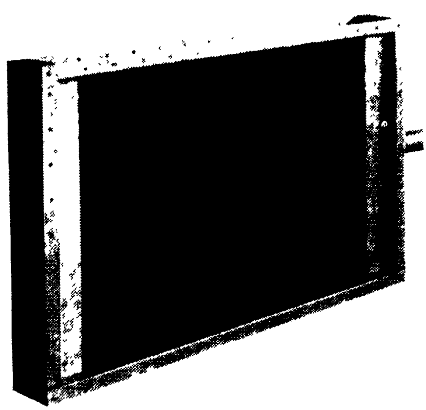


Fig. 6

Narrow Width Aerofin: (Fig. 6) recommended for water cooling or for flooded Freon systems. Made in straight tubes only with headers on opposite ends, joints between headers and tubing being brazed. Construction similar to Flexitube Aerofin.

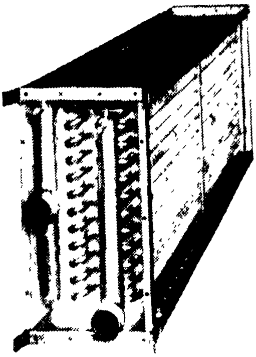


Fig. 7

Aerofin Continuous Tube Water Coils (Fig. 7) are designed for air cooling by circulating cold water through the AEROFIN and air over extended fin surface. Made for either horizontal or vertical air flow.

Tubes and fins are copper, completely tinned with permanent metallic bond between fin and tubes. Headers are made of steel and casings of heavy galvanized iron or copper.

Tested to 100 lb steam, followed by 450 lb air with coil submerged in water.

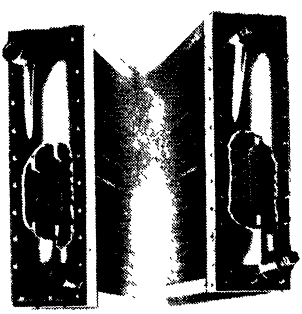


Fig. 8

Aerofin Cleanable Tube Units (Fig. 8) for cooling only made with headers removable to permit cleaning tubes. Recommended for use where sediment or scale forming chemicals are present in the cooling water.

Headers—Fabricated steel.

Tubing—Copper or admiralty.

Casings—Copper or galvanized iron.

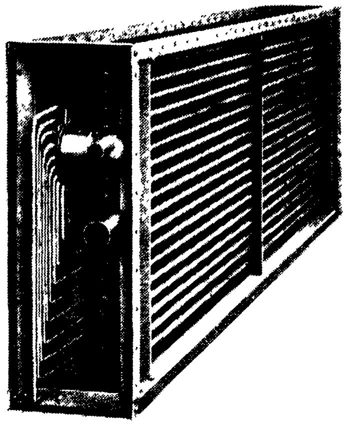


Fig. 9

Aerofin Direct Expansion Units: (Fig. 9) Centrifugal Header Type—For cooling air, using Freon expanded directly into the coil.

AEROFIN Sizes

Flexitube: 13 standard lengths, three widths, one and two rows deep.

Narrow: same as Flexitube.

Universal: 17 standard lengths, two widths, one and two rows deep.

Continuous Tube: 13 standard lengths, three widths, 2-3-4-5 and 6 rows deep.

Cleanable Tube: 17 standard lengths, one width, 2 and 4 rows deep.

Direct Expansion: Centrifugal Header —11 standard lengths, three widths, 2-3-4-5-6 rows deep.

Steel Supporting Legs: 18 in. and 24 in. high. Punched same bolt hole centers as standard casings. Quickly attached. No other foundation required.

Sale: AEROFIN is sold only by manufacturers of nationally advertised Fan System Apparatus. List upon request.

Write Syracuse for Heating Bulletin H-45; Direct Expansion Bulletin DE-48-1 on refrigeration type units; Continuous Tube Bulletin C. T. 39-2 for Water Cooling Coils; or pamphlet on Cleanable Type Aerofin for cooling.

The G & O Manufacturing Company

138 Winchester Avenue **New Haven 8, Connecticut**

G&O

SQUARE FIN TUBING

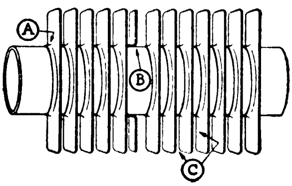
STRAIGHT LENGTHS—U-BENDS—CONTINUOUS COILS

THE use of INDIVIDUAL fins results in high efficiency in heat transfer from primary tube surface to secondary fin surface.

Fins of any size or shape may be obtained giving any desired proportion of primary and secondary surface.

A square fin has about 30 per cent greater surface than a round fin of a diameter equal to one side of the square.

Individual fins permit of any fin spacing; also, of using fins in groups at intervals along tubes.



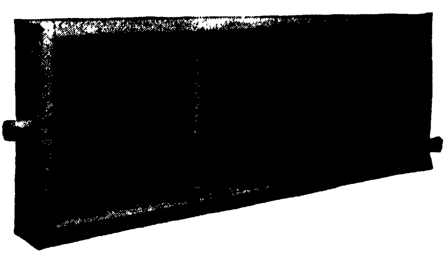
A—Generous Fin Collar provides large contact area between Tube and Fin.
B—Tube expanded against Fin Collar; insures mechanically tight joint, made permanent by bond of high temperature alloy—complete thermal contact.
C—Free air-flow passages; non-clogging.

STANDARD SIZES

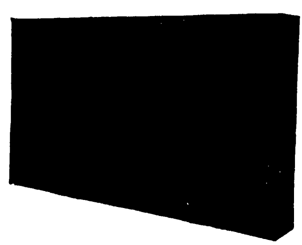
O.D. of Tube	Fin Size	Fin Spacing per Inch	Surface per Linear Foot
3/8"	7/8" sq.	6	0.80 sq. ft.
3/8"	7/8" r'd.	6	0.60 sq. ft
1/2"	1 3/4" X 2" oblong	6	3.65 sq. ft.
5/8"	1 1/8" r'd.	6	0.87 sq. ft.
3/4"	1 1/2" r'd.	6	1.55 sq. ft.
3/4"	1 5/8" sq.	6	2.40 sq. ft.
1"	2 1/8" sq.	6	4.00 sq. ft.
1 3/8"	2 3/8" r'd.	4	2.33 sq. ft.

RADIATING ELEMENTS FOR ALL HEAT TRANSFER PURPOSES

G&O Finned Radiation Coils for industrial applications are available in a wide range of sizes.



Universal Type B



Standard No. 10

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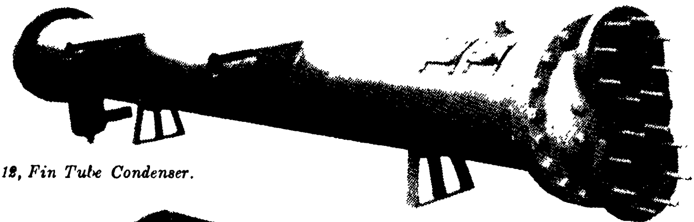
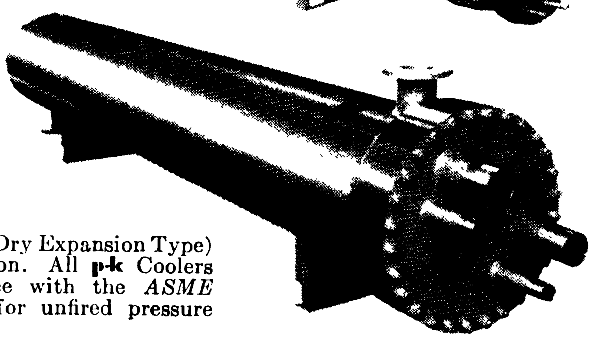
Representatives in Principal Cities

p-k Heat Exchangers for Heating, Air Conditioning and Refrigeration Service. Hot Water Storage Heaters—Instantaneous Heaters—Convertors—Fuel Oil Heaters—Freon Coolers—Condensers—Balance Loaders—Interchangers—Suction Line Exchangers—Slug Eliminators—Coolers for Water, Brine and other Liquids.

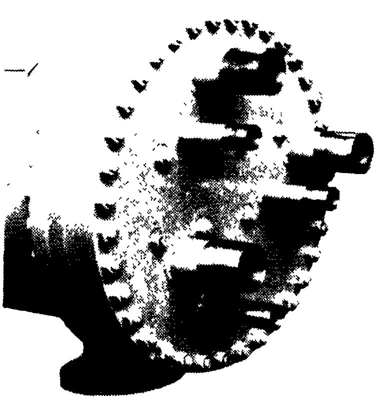
Since 1880, **p-k** has designed, engineered and fabricated heat transfer equipment to meet industries' most exacting requirements. We are prepared to recommend and design exchangers for air conditioning, refrigeration, process and industrial cooling applications.



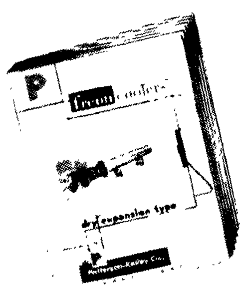
p-k *Freon 12, Fin Tube Condenser.*



p-k Freon Cooler (Dry Expansion Type) 2-circuit construction. All **p-k** Coolers built in accordance with the *ASME* Code, Par. U-69, for unfired pressure vessels.



p-k Freon Cooler (Dry Expansion Type) 3-circuit construction for 200-ton refrigeration capacity.



Catalog number 101 contains tables, charts and complete data which will enable you to select the Cooler which exactly meets your needs.

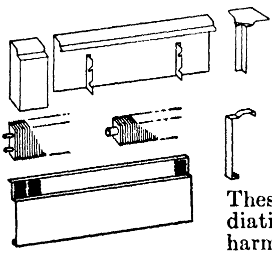
KRITZER RADIANT COILS, INC.

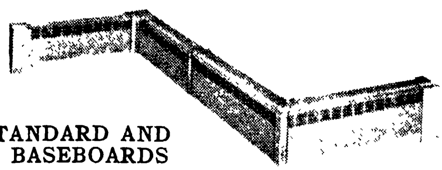
"IF IT'S KRITZER, IT'S RIGHT SIR"

2909A LAWRENCE AVENUE • CHICAGO 25, ILLINOIS

KRITZER "FIN-TUBE" RADIANT COILS

Kritzer "FIN-TUBE" Radiant Coils introduce a new, improved method of radiant heating. They are enclosed in joist or stud spaces, concealed but not imbedded in structural materials. They warm the air in these spaces, and indirectly warm ceilings, floors or walls to produce radiant panels for all areas. No concentration of heat in one spot. Kritzer combines low first cost with lower installation costs. Installation begun and finished when framing is up, and before plastering. Coils are installed crosswise to joists—finned sections lying between joists slightly above plaster; tubing suspended on hangers with metal clad insulating sleeves. Simple construction. A single ⅝ in. copper tube is mechanically bonded to 1½ in. x 3 in. aluminum fins, in sections for highly multiplied heat transfer surface.



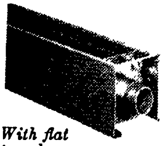


KRITZER STANDARD AND UNIVERSAL BASEBOARDS

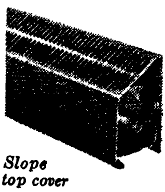
These compact, attractive baseboards supply heat both by radiation and convection—allow maximum use of floor space—harmonize with any interior decoration. Standard Baseboard Coils are made with aluminum fins 2 in. x 5½ in. mechanically bonded to two ⅝ in. copper tubes. Universal Coils are made with 2¾ in. x 3¾ in. fins in steel, aluminum or copper on 1 in. or 1¼ in. diameter steel pipe or copper tubing. Installation requires only three quick, easy steps. (1) Steel back plate is placed against wall on finished floor and fastened. (2) Elements are then snapped into place. (3) Front cover is simply snapped in place and end enclosures installed where required. Damper with control knob is available as optional equipment. Furnished in standard 10 ft, sections easily cut on the job, or cut to length on order.

KRITZER "FIN-PIPE" COMMERCIAL HEATING COILS

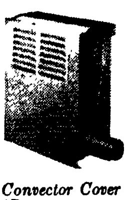
More Btu's for less time, labor and dollars! Engineered for greater efficiency and faster installation. Kritzer "FIN-PIPE" Coil sections easily equal the heat output of equipment costing over twice as much, yet Kritzer Radiation more than cuts in half the time and labor necessary for installation. Fins grip pipe with specially designed deep collars to assure better heat transfer and a permanent, positive mechanical grip. Pipe sections are furnished with chamfered ends, ready for welding, or threaded with modern, improved chaser type die heads that give clean, sharp threads so necessary for fast, leak-proof connections. A variety of covers snap on to wall hangers but do not touch fins. Kritzer "FIN-PIPE" Commercial Radiation is available in a variety of sizes: 2 in.; 1¼ in.; 1 in. steel pipe or copper tube with 4¼ in. square, 3¼ in. square or 2¾ x 3¾ in. fins of steel, aluminum or copper, spaced 24, 32, or 48 per foot.



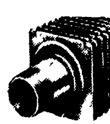
With flat top closure.



Slope top cover



Convector Cover (Damper optional).



Kritzer Commercial Coil

The Rome-Turney Radiator Company

Erie Boulevard. East **Rome, N. Y.**

ROME
TURNEY RADIATOR COMPANY

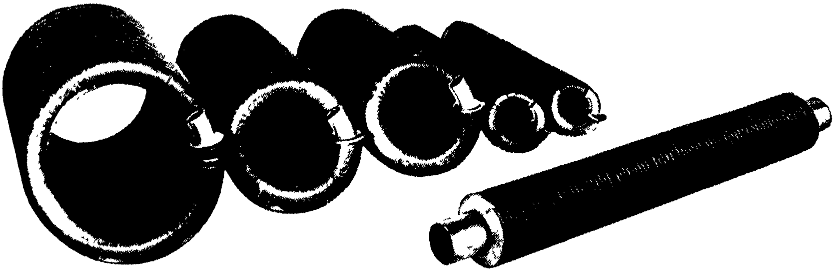
Manufacturers of "Ro-Fin" Tubes for Efficient Heat Transfer

IN PHILADELPHIA SEE W. H. BUNTEN,
1205 HAMILTON ST.

IN ST. LOUIS SEE BRASS AND COPPER SALES CO.,
2817 EUCLID AVE.

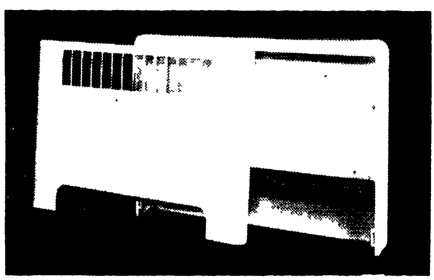
IN BUFFALO SEE J. LANDERS,
170 FRANKLIN ST.

"Ro-Fin" Extended Surface Helical Fin Tubing for Heat Transfer

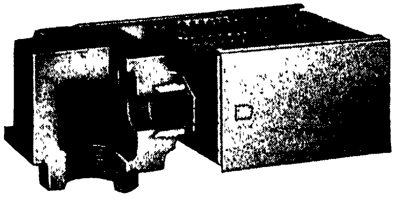


Nearly 100 sizes of "Ro-Fin" Extended Helical Fin Tubing are in production: straight lengths, coils, U-bends, coils-within-coils, with or without threaded end connection, or for standard flare connections. "Ro-Fin" tubes are adaptable for all types of heat transfer work, and are made to the individual specifications of each order.

Complete range of sizes includes tube diameters (outside diameter) from $\frac{1}{4}$ in. to $1\frac{5}{8}$ in., fin widths from $\frac{3}{16}$ in. to $\frac{3}{4}$ in. Number of fins per inch: 3 to 19. "Ro-Fin" tubes can be furnished in continuous lengths, with or without joints. Straight lengths furnished up to 25 ft long.



"Ro-Fin" tubes in a convector radiator for heating homes, offices, churches, schools.



Cutaway section showing replaceable header construction of copper heating element.

Write for information on your heat transfer problems for:

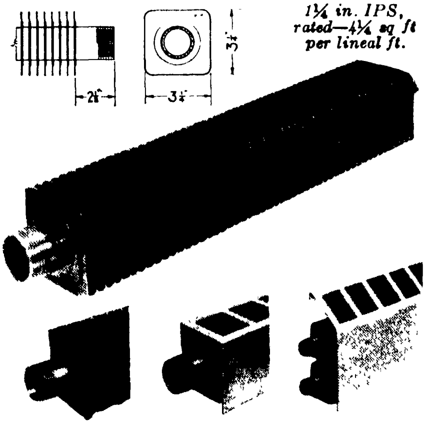
Refrigeration Condensers
Fluid Recovery Condensers
Concealed Radiation Convectors
Steam Condensers
Fan-Type Unit Heaters
Diesel Engine Cooling
Baseboard Radiation

The Vulcan Radiator Company

22 Francis Avenue — Hartford 6, Conn.

Representatives in Principal Cities

Vulcan Radiation is used in railroad cars, ships, hospitals, schools, churches, homes and industrial plants. Available in steel or copper . . . easy to install . . . light in weight . . . requires few fittings and supports . . . tube ends threaded or chamfered for welding. Heat distribution is uniform. Steel radiation comes in two sizes . . . 2 in. IPS, rated—5¼ sq ft per lineal ft at 1 lb steam and 70 deg air . . . for 1¼ in. IPS see illustration—this size also available in copper. Illustrated catalog available.

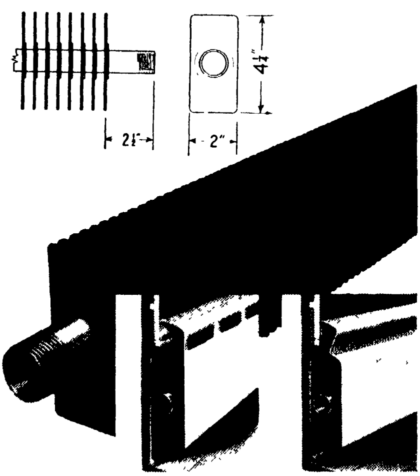


1¼ in. IPS, rated—4¼ sq ft per lineal ft.

Standard Grille Cover — *Linovector Covers Type F* — *Type S*

Vulcan Radiation is fabricated by mechanically imbedding offset fins or plates on seamless steel pressure tube or copper water tube. The patented offset fin construction gives complete rigidity to the entire assembly and extends the heating surface of the tube.

Because of its comparatively light weight and compactness, Vulcan Radiation responds quickly to thermostatic control. Full heat output is obtained almost immediately after steam is supplied. Since most of the heat is given off by convection, the result is EVEN, UNIFORM HEAT from floor to ceiling.



1 in IPS — *Pat. No. 2,484,615 Radi-Vector Covers Type K* — *Type L*

I=B=R ratings for Vulcan Radi-Vector available in Catalog No. 53.

Vulcan Baseboard Radiation . . . fin-on-tube construction with grille covers combines radiant and convection heating. High safe working pressure . . . either hot water or two-pipe steam systems. Light in weight . . . easy to install . . . requires few fittings or supports. Comes in two sizes . . . 1¼ in. IPS . . . steel fins 2¾ in. wide by 3¾ in. high . . . for 1 in. IPS, see illustration. Also available in copper. Illustrated catalog available.

BAKER REFRIGERATION CORPORATION

South Windham, Maine

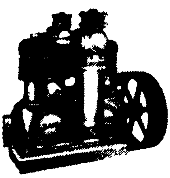
Offices, Warehouses and Parts Depots in Principal Cities

Baker

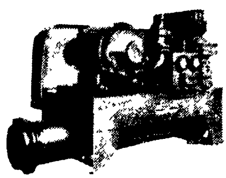
Air Conditioning • Refrigeration

Products: Air conditioning and refrigeration (central plant and unit equipment). **Condensers**—evaporative or shell and tube, wide capacity range. **Cooling Units**—for every requirement, for all refrigerants. **Liquid Receivers**—for ammonia or Freon, with positive liquid seal. **Valves and Fittings**—a complete line.

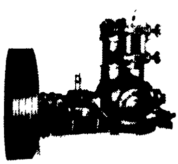
FREON

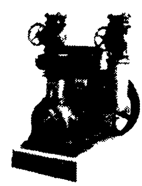


1. **Baker Compressors.** 1½ to 60 hp, feature high efficiency at low operating cost. Timken bearings, forced feed lubrication, gear-type oil pump, positive shaft seal promote long life.
2. **Baker Condensing Units.** ⅓ to 60 hp, are engineered with matched components to assure efficient, trouble free operation for every application. Compressor units, 1½ to 60 hp.



AMMONIA





3. **Baker Ammonia Compressors.** 2 to 125 hp, can be supplied with capacity reduction control down to 33⅓ per cent. Double suction control for operation at two different temperatures is available on special order.
4. **Baker Compressor Units.** 2 to 15 hp, feature automatic motor control with overload and low voltage protection. Complete Condensing Units also available from 2 to 15 ton capacities, ready for installation.
5. **Baker Booster Compressors** for two-stage operation on the low side in low temperature applications. Forced-feed lubrication and balanced fly wheel for "V" belt drive. Compact design requires a minimum of space.

6. **BAKERAIRE**—Delivers maximum cooling per horsepower input. Famous Baker hermetic compressor in 3, 5, 7½ and 10 ton capacities. 4-way air discharge, new Sphericoil condenser, humidity control, spring-mounted compressor.
7. **CENTRAL-AIR**—A complete unit air conditioner in capacities from 5 to 40 tons with built-in evaporative condenser. Factory assembled and tested, delivered ready to operate. Simplified installation keeps first cost low. Fully automatic controls.

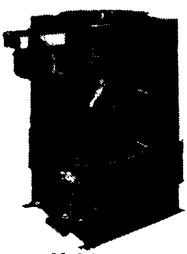
Baltimore Aircoil Company, Inc.

2615 Mathews Street
Baltimore 18, Maryland

Manufacturers of Evaporative Condensers and Cooling Towers

BALTIMORE FAN COMPANY
(A Division of Baltimore Aircoil Company, Inc.)
Manufacturers of Propeller Fan Wheels

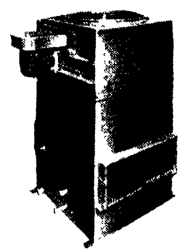
B.A.C. Evaporative Condensers and Cooling Towers are rigidly built throughout for long, trouble-free service. Casings are of galvanized construction, providing full protection against corrosion. Sump pans are hot dipped galvanized after fabrication. Condensing coils are prime surface steel pipe, hot dipped galvanized after fabrication. Non-metallic cooling tower surface is designed for efficient operation.



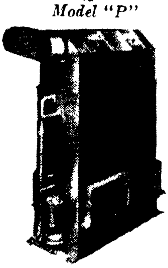
Model "P"

Model "P" Evaporative Condenser: Vertical propeller fan type. . .outdoor installation only. . .Freon or Ammonia. Capacities: 3 TR to 100 TR.

Model "PT" Cooling Tower: Vertical propeller fan type. . .outdoor installation only. Capacities: 3 TR to 100 TR.



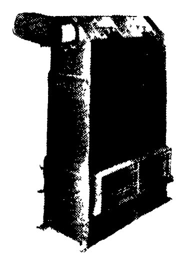
Model "PT"



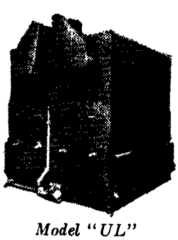
Model "U"

Model "U" Evaporative Condenser: Centrifugal fan type. . .indoor or outdoor installation. . .Freon or Ammonia. Capacities: 10 TR to 100 TR.

Model "UT" Cooling Tower: Centrifugal fan type. . .indoor or outdoor installation. Capacities: 10 to 100 TR.



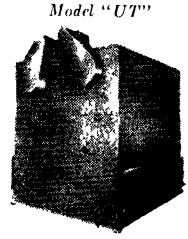
Model "UT"



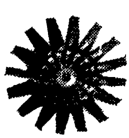
Model "UL"

Model "UL" Evaporative Condenser: Centrifugal fan type. . .indoor or outdoor installation. . .Freon or Ammonia. Capacities: 105 TR to 260 TR.

Model "LT" Cooling Tower: Centrifugal fan type. . .indoor or outdoor installation. Capacities: 105 to 252 TR.



Model "LT"



Baltimore Fan Company Type "IC" Propeller Fan Wheel: Industrial ventilation (free air). 24 in. to 48 in. diameter.

Baltimore Fan Company Type "LS" Propeller Fan Wheel: Medium pressure applications. 24 in. to 96 in. diameter.

Baltimore Fan Company Type "HP" Propeller Fan Wheel (illustrated): Higher pressure applications. 24 in. to 72 in. diameter.

Curtis Refrigerating Machine Division

of Curtis Manufacturing Company

1959 Kienlen Ave., **St. Louis 20, Mo., U. S. A.**

ESTABLISHED 1854

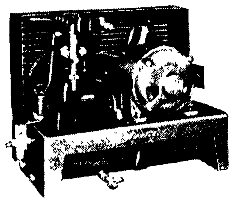
NEW YORK OFFICE
30 VESEY ST.

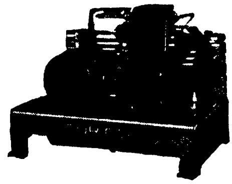
CHICAGO OFFICE
9 S. CLINTON ST

Full Line of Units from ¼ to 40-hp **Unit Coolers and Evaporator Coils**

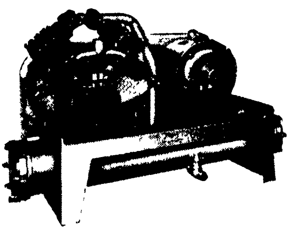
PRODUCTS: Complete Refrigerating Equipment for Dairies, Creameries, Ice Cream Cabinets, Ice Cream Making Plants, Cold Storage Locker Systems, Walk-in Coolers, Drinking Water Systems, Commercial and Low Temperature Cooling, Processing and Air Conditioning Installation, Packed and Remote Types.

Combination air and water cooled Condensing Units. ½ Through 2 hp.



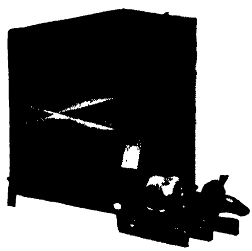


1½ hp Air Cooled Condensing Unit. Other sizes from ¼ to 3 hp.



15 hp Cleanable Shell and Tube Condensing Unit Other sizes from 3 to 40 hp.

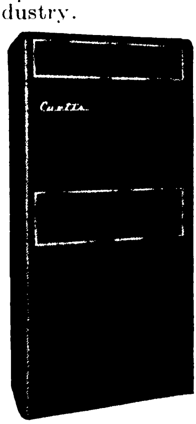
Evaporative Condensers, Cooling Towers, Air Handling Units



Commercial Refrigeration

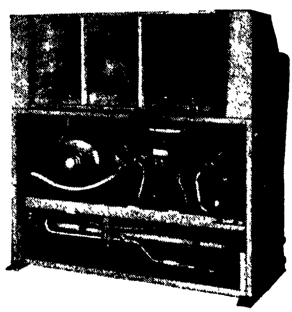
Air cooled condensing units from ¼ to 3 hp, inclusive, and water cooled units from ⅓ to 10 hp, inclusive. All models available for either Freon (F-12) or Methyl Chloride. Mechanical advantages include Timken Bearings. Positive Pressure lubrication.

Special models are available for ice cream, frozen food cabinets and for the dairy industry.



2,3,5, and 7½ ton Packaged Type Air Conditioner.

10–15 ton Remote or Central Type Air Conditioner.



Air Conditioning

For today's Air Conditioning requirements Curtis offers complete packaged, refrigerated air conditioning units, requiring only water and electrical connections to install. Cools, dehumidifies, circulates and filters the air. Eliminates costly installation expense. Adaptable for heating.

ATLANTA
BOSTON
BUFFALO
CHARLOTTE
CHICAGO
CINCINNATI
DALLAS
KANSAS CITY
LOS ANGELES

Frick Company

(Incorporated)

Air Conditioning, Refrigerating, Ice Making and Food Freezing Equipment

Waynesboro, Penna.

Distributors in FRICK Principal Cities

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OKLAHOMA CITY
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PHILADELPHIA
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SEATTLE
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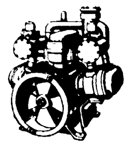
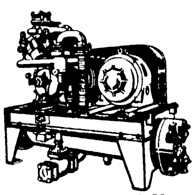
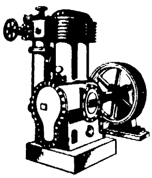


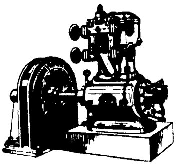
NEW "ECLIPSE" Frick Compressor. Bulletin 100.



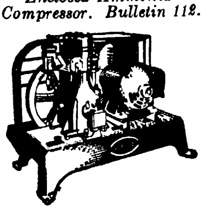
Eclipse Combined Unit with NEW "ECLIPSE" Compressor. Bulletin 100



Enclosed Freon-12 Machine. Bulletin 508.



Enclosed Ammonia Compressor. Bulletin 112.



Low Pressure Refrigerating Unit, Bulletin 97.

AIR CONDITIONING

Complete Frick Systems; also refrigeration for use with equipment supplied by others. Thousands of installations attest the value of Frick air conditioning. Successful experience with exacting commercial and industrial jobs enable us to solve your problems.

See Frick Bulletin 502 on hospitals, 503 on ammonia systems, 504 on typical installations, 505 on engineering details, and 522 on unit conditioners (illustrated).

FREON-12 REFRIGERATION

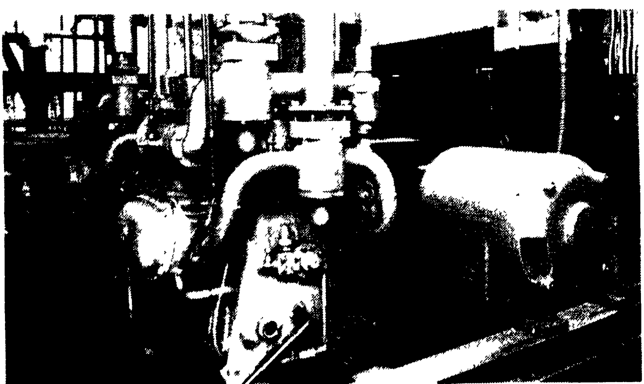
Frick NEW "ECLIPSE" and the larger F-12 compressors provide a complete and efficient line. Coils, coolers, condensers and controls to suit. Patented Flexo-Seal at shaft, pressure lubrication from reversible pump, capacity controls, and other superior features make Frick machines your logical choice.

AMMONIA REFRIGERATION

Combined units and vertical enclosed compressors, with two or four cylinders, in sizes from 2½ tons up. Widely used for air conditioning, with material savings. Ask for Bul. 503 on this subject.

LOW-PRESSURE REFRIGERATION

Commercial and industrial units in sizes from ¼ hp up. Charged with Freon-12. Air and water-cooled condensers. Coils, coolers, and air conditioners. Get in touch with your Frick Distributor; ask for Bul. 97. Our service includes estimates, layouts, manufacture, installation, and maintenance.



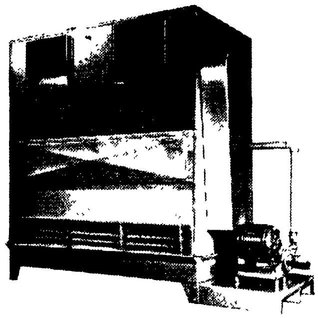
Two Frick NEW "ECLIPSE" Compressors, each with 9 Cylinders, Air Condition Weingarten's Supermarket at Houston.

Marlo Coil Co.

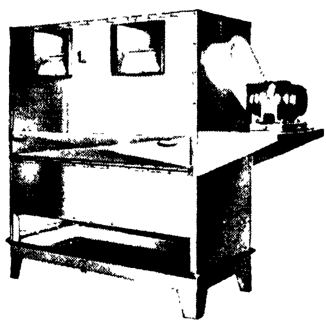
6135 Manchester Ave., St. Louis 10, Mo.

Manufacturers MARLO = HEAT TRANSFER Equipment
Since 1925

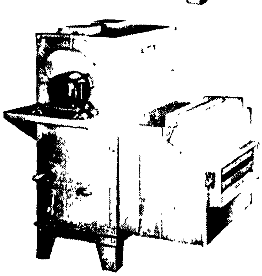
Industrial Coolers—Unit Coolers—Evaporative Condensers—Low Temperature Units—Air Conditioning Units—Heating and Cooling Coils—Cooling Towers—Diesel Engine and Oil Evaporative Coolers.



◀ *Cooling Towers*



Industrial Coolers ◀ *Dry Coil Coolers*



◀ *Floor Type Air Conditioning Units*

COOLING TOWERS. Triple-type; Induced Air—Wetted. Surface—Water Spray. Compact in space, weight and price—3 to 100 tons—Built Sectionally. Write for *Bulletin 406*.

EVAPORATIVE CONDENSERS. 3 to 100 tons—All refrigerants—All prime surface coils—No fins—Quiet—Motor Unidrive—Durable construction. Write for *Bulletin 404*.

INDUSTRIAL COOLERS. Dry Coil and Brine Spray 15 unit sizes—1000 to 24,000 cfm—Floor Type. Galvanized frame and pans—Sectionally built. Variable: (1) Rows of coil and fin spacing (2) Defrost sprays optional (3) All refrigerants. Write for *Bulletin 403*.

AIR CONDITIONING UNITS

Floor and Ceiling Types. Cooling, heating, humidifying, dehumidifying, filtering, circulation—1 to 150 tons—480 to 37,000 cfm. *Bulletin 409*.

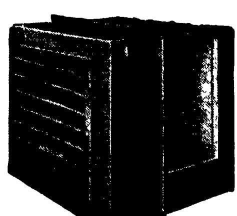
Multi-zone Unit. Complete winter and summer functions—individual zone control—1500 to 17,000 cfm. *Bulletin 409 MZ*.

ELECTRIC DEFROST LT UNITS. Compact ceiling type—High capacity—Low cost. 7 sizes—Ammonia or Freon—$\frac{1}{2}$ to $2\frac{1}{2}$ tons at 12 deg. TD. Defrosted electrically. Quickly installed—Write for *Bulletin 408*. (U.S. Patent 2266373)

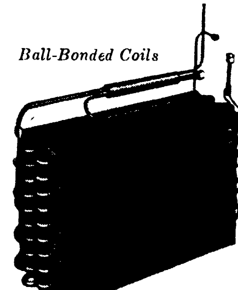
BALL-BONDED COILS. Mechanically expanded tubes to fins—For air conditioning, heating and industrial refrigeration. Any material—All refrigerants. *Bulletins 396 and 495*.

Multi-Zone Air Conditioning Units

Electric Defrost LT Units



Ball-Bonded Coils



Mills *Industries, Incorporated*

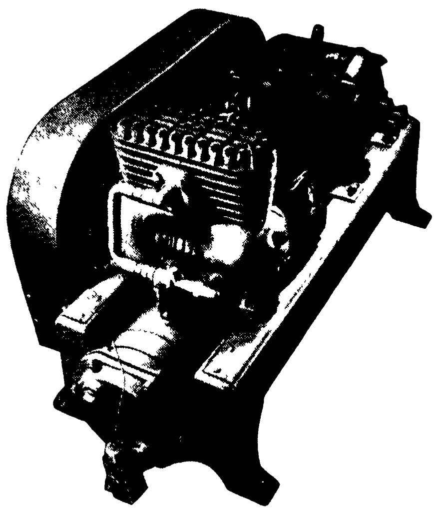
4100 Fullerton Avenue • **Chicago 39, Illinois**

Mills Compression Equipment

for Air Conditioning • Commercial and Industrial Refrigeration

COMPRESSOR

2½ in. x 3 in. four cylinder, vertical, single acting, reciprocating type . . . heavy, one piece alloyed semi-steel cylinder block and crankcase . . . oil sight glass . . . alloy semi-steel pistons, two compression rings, one oil ring . . . hardened piston pins . . . drop forged steel connecting rods . . . bronze piston pin bearing . . . bronze main bearings . . . exclusive Mills design shaft seal.



CONDENSING UNIT

Base of cast iron ends and formed steel sides . . . shell and tube type receiver condenser with removable heads and leak alarm fittings . . . complete with automatic water valve, starting box, low pressure control with high pressure safety cut-out . . . complies with *Underwriters'* specifications . . . dynamically balanced and non-vibrating, light in weight, compact, requiring a minimum of floor space. Mills Condensing Units are economical to operate and maintain. Condensing units are shipped ready to operate. No special foundations are necessary.

APPLICATIONS

Refrigeration in air conditioning applications for comfort, food preservation, and industrial uses for processing and testing. Compactness of condensing units make them ideal for self-contained applications.

ENGINEERING

Mills Industries, Incorporated maintain an engineering test laboratory as a service to their customers, equipped with three hot rooms, calorimeter, indicating and recording, pressure and electrical instruments and all apparatus necessary for a thorough analysis of the use of condensing units to any manufacturer's product or field applications.

CAPACITIES AT 40 F SUCTION GAS

Size	RPM	Displacements CFM	Capacities in BTU	Width	Depth	Height	Net Weight
5 hp	480	16.37	65000	63″	29½″	32″	895 lb
7½ hp	700	23.86	90500	63″	29½″	32″	945 lb
10 hp	890	30.34	115000	63″	30″	34″	1050 lb

Air cooled models from ¼ to 3 hp.
Water cooled models from ⅓ to 10 hp
Combination air and water models
From ½ to 3 hp.

All models are designed for use with Freon-12 or Methyl Chloride refrigerants for low, standard, and high back pressures, for operation throughout the complete range of temperatures.

Compressors are designed for low head temperatures and high volumetric efficiency.

Condensers are matched to the performance ranges of each compressor.

A Member of the Mills Field Organization is Near You and at Your Service

The Ready-Power Co.

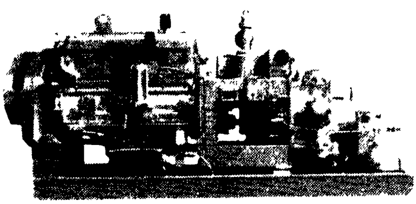
11231 Freud Ave. **Detroit 14, Mich.**

DISTRICT OFFICES OR REPRESENTATIVES LOCATED IN FOLLOWING CITIES

Boston	Philadelphia	Milwaukee	Seattle
Chicago	Pittsburgh	New Orleans	St. Louis
Dallas	Los Angeles	New York	

READY-POWER

AIR CONDITIONING AND REFRIGERATION EQUIPMENT



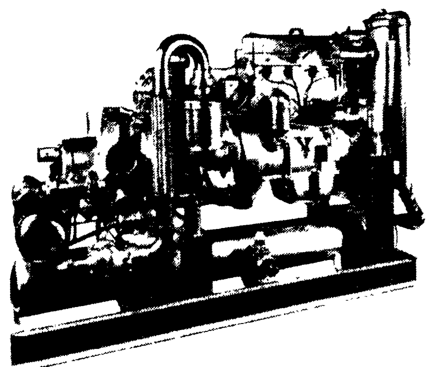
Compressor Type

Ready-Power Engine Driven Compressor and Condensing Units are designed and built to meet the need of (1) Low operating cost air conditioning and refrigeration employing NATURAL GAS fuel and (2) Air Conditioning and refrigeration where electric power is not available or high in cost. It is particularly adaptable to portable equipment for pre-cooling airplanes, perishable shipments by rail or truck, and for installations in refrigerated cars and trucks.

CAPACITY MODULATION

The Ready-Power inherent system of capacity modulation permits operation of the compressor to meet the needs of varying conditions. This assures much better control of temperature and humidity conditions than is possible with the usual "on and off" system generally used with electrically driven equipment.

On units of 15 to 55 ton the capacity modulation is controlled both by varying the engine speed and by unloading the cylinders of the compressor. Since engine speed can be varied from maximum to less than half of maximum this



Condensing Type

more than doubles the usual advantages obtained from unloading.

On the smaller units, the capacity modulation is controlled entirely by engine speed variation.

All capacity modulation is automatically controlled.

3 TO 55 TON RANGE

Unit capacities range from 3 to 55 ton for air conditioning and multiple unit installation of over 100 ton are in successful operation.

Either compressor or condensing units are adaptable for use with evaporative condenser, heat exchanger or radiator cooling of the engine. Water cooled manifolds are standard on some models and available on all models. Engine starting is by either storage battery or AC Motor.

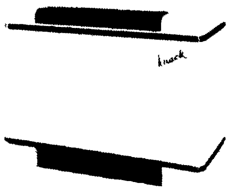
All units are supplied for operation on Natural Gas, Gasoline, Propane or Butane fuels.

Units of 15 ton and larger are also available for operation on Diesel fuel.

Refrigeration Appliances, Inc.

923 W. Lake St. Chicago 7, Ill.

AIR CONDITIONING AND REFRIGERATION EQUIPMENT

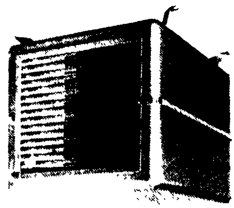


COMF-E-FEX

Remote type ceiling mount comfort cooler. Available with direct expansion or chilled water coil. Heating coil optional. Throway filter in tilting frame at air intake for easy replacement. Sizes: ¾, 1, 2, 3 tons. Specifically designed for professional offices, beauty parlors, barber shops, tourist courts, and small stores—where floor space is at a premium.

CLIMATE MASTER

New console type air conditioner suitable for under window placement. Especially designed for office buildings, hospitals, hotels and motels, in multiple installations using water as a cooling and heating medium. Also available with direct expansion, water, or steam coils, in any combination. Provides individual zone control from a central system while eliminating costly duct work.

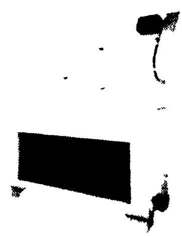


SAN-E-FEX

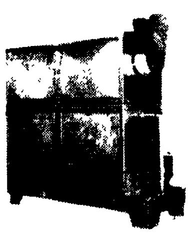
Ceiling mount, central station air conditioner in six sizes from 5 to 25 tons. Available in direct expansion or chilled water coils. Heating coils optional. Permanent or replaceable type filters. Expansion valve furnished with Freon model. Large, low speed centrifugal blowers. Scientifically engineered, rugged construction.

FLOOR TYPE AIR HANDLING UNIT

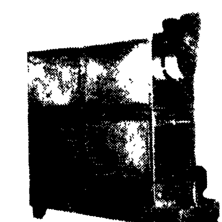
Central station air conditioner in eight sizes from 5–40 tons. Direct expansion or chilled water coils. Heating coils optional. Permanent or replaceable filters. Expansion valve furnished with Freon model. 12 gauge, all steel, channel formed, welded construction. Removable panels for easy access to interior.



EVAPORATIVE CONDENSER AND COOLING TOWER



Floor type evaporative condensers and cooling towers in nine sizes from 5–50 tons. 12 gauge, channel formed, welded steel construction, hot dip galvanized *after* fabrication. Intake eliminators positively contain sump splash. Demountable sections to facilitate handling. Turnover blower section for top, right, or left air discharge. Removable doors for easy access to interior. Separate motors for pump and blower.



Ceiling suspended models in smaller sizes also available.

AIR-E-FEX—Custom-built air conditioning coils for water or direct expansion.

We also manufacture a complete line of unit coolers and gravity coils for commercial or industrial refrigeration and air conditioning applications. Ceiling, wall, or floor mount, for fixture or rooms, for high or low temperatures.

Specification and Data Sheets upon Request

Refrigeration ENGINEERING, INC.

7250 East Slauson Avenue,
Los Angeles 22, California.

RECOLD
TRADE MARK REG.

Manufacturers of Air Conditioning and Commercial Refrigeration Equipment

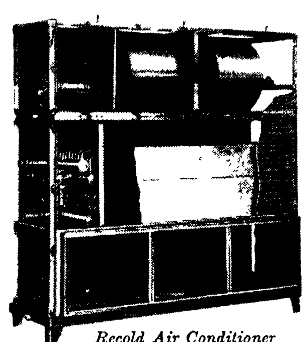
RECOLD HEAT TRANSFER EQUIPMENT

gives performance with economy because each unit is designed for a specific application. Whatever your problem may be—commercial or industrial refrigeration or air conditioning for cooling or heating—you will find that RECOLD equipment will meet the conditions encountered on any job.

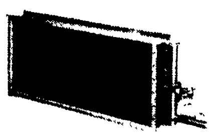
RECOLD AIR CONDITIONERS AND INDUSTRIAL REFRIGERATION UNITS

Outstanding in appearance, performance and versatility, these units have pleasing rounded corners and sparkling blue-grey hammertone finish. Totally enclosed units, with large access doors to motor, expansion valves and coil connections.

Capacities: Air Conditioners up to 50 tons; Industrial Units up to 15 tons
Air Conditioning Units; Multizone Air Conditioners
Commercial and Industrial Coolers; Ammonia and Freon Units



Recold Air Conditioner (Cut-Away View)



COOLING AND HEATING COILS

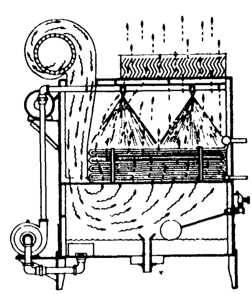
For direct expansion, chilled water, hot water and steam. Complete range of sizes to meet all conditions and capacities.

"DRI-FAN" EVAPORATIVE CONDENSERS AND "DRI-FAN" COOLING TOWERS.

Capacities from 3 to 150 tons.

Diagram Showing How "Dri-Fan" Principle Operates.

Wavy arrows denote warm inlet air stream, other arrows show discharge of moist air.



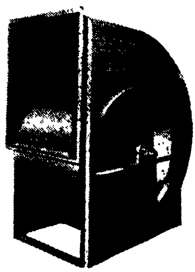
Fan is placed in dry incoming air stream, preventing rust and corrosion. Unique construction provides completely sealed access doors without the use of gaskets of any kind. Galvanized construction.

Write for Complete Catalog and Name of Nearest Distributor

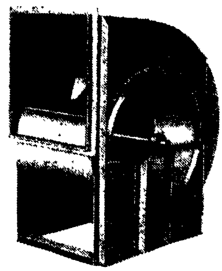
Aladdin Heating Corporation

2272 San Pablo Ave., **Oakland 12, Calif.**

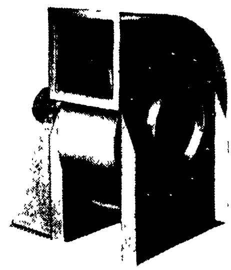
Manufacturers of Centrifugal Blowers,
Heating and Ventilating Equipment.



FC Fan



BB Fan



EX Fan

The Aladdin FC Fan having a forward curved rotor is built in 14 standard sizes, single or double width, of 8 arrangements of drive and 8 directions of discharge. The low tip speed which is characteristic of this fan makes it ideal for general application where quiet operation is essential. Write for Bulletin No. 490.

The BB Fan is a backward curved fan with the non-overloading horsepower characteristic. This fan is built in 12 standard sizes, single or double width of 8 arrangements of drive and 8 directions of discharge. These fans are available in class I, II, III or IV and can be built for special application where required. Write for Bulletin No. 485.

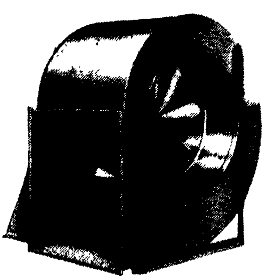
The EX Fan is used chiefly for the conveying of materials, fume exhaust, etc. These fans are reversible and can be furnished in 13 standard sizes of 8 arrangements of drive and 8 directions of discharge. They can be desirable for special applications such as for handling abrasive materials or for acid fumes. Write for Bulletin No. 460.

The RB Fan having a radial curved rotor is used chiefly for kitchen exhaust duty. They are well suited for handling grease and other sticky materials, also for exhausting fumes and vapors from tanks, hoods, etc. This fan is built in 12 sizes, single width only, of 8 arrangements of drive and 8 directions of discharge. Write for Bulletin No. 450.

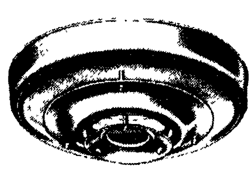
The FC Utility Sets having a forward curved rotor are built in ten standard sizes, of 57 standard speed combinations, single width only. These units are self-contained and can be furnished for any rotation and discharge. All units can be furnished with weather proof hoods when required. Write for Bulletin 525.

Fuseair ceiling outlets are manufactured in a complete range of sizes both in the supply type and the combination supply and return type. These units are fabricated from spun aluminum and all standard units are given an aluminum finish. Write for Bulletin No. 520.

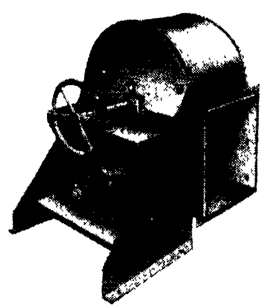
RB Fan



Fuseair



Utility Sets



American Coolair Corporation

3606 Mayflower Street, **Jacksonville 3, Florida**

Exhaust Fans and Related Equipment for Industrial and Home Cooling

Authorized Distributors and Dealers in Most Communities

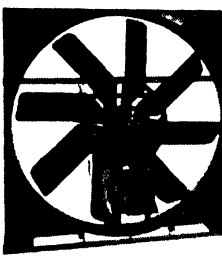
District Representatives:

ATLANTA, GA.—John L. Underwood Co., Inc., 555 Whitehall St., S.W., Atlanta 3, Ga.
BALTIMORE MD.—Herbert C. Hax, 3803 Cedardale Rd., Baltimore 15, Md., Phone Liberty 0781
CINCINNATI, OHIO —Halsey E. Kendrick, R.R. 7, Box 89, Cincinnati 11, Ohio, Phone Humboldt 4573
DALLAS, Texas—J. D. Clower, Whitewright, Texas. LD Phone. Whitewright 919F2.
NEW YORK, N. Y.—E. R. Dexter, 117 N. Middletown Rd., Pearl River, N. Y., LD Phone Pearl River 5-2725
WASHINGTON, D. C.—Robert A. Magee, 7402 Columbia Ave., College Park, Md., Phone Warfield 1480.

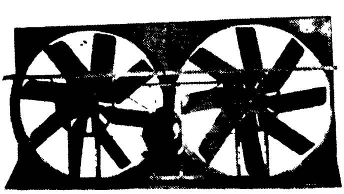
COOLAIR BELT DRIVE FANS are specifically engineered to move large volumes of air *quietly* and at low cost.

CERTIFIED RATINGS, UL LABEL Coolair Fans are rated in accordance with the ASHVE Standard Test Code for Centrifugal and Axial Fans (1938). In addition, fans up to and including the 62 in. sizes are rated by the *A & M College of Texas* and listed under the Re-examination Service of *Underwriters' Laboratories*.

SKF BALL BEARINGS in all models.



TYPES H and C—For commercial and home use. Type H is equipped with sound-absorbing springs, assuring extremely quiet operation. U. S. Patent 2191418. Type C is similar but with rigid frame and heavy duty blades, for commercial and industrial use.



TYPES HT and CT—These twin units often fit perfectly where limited space prevents use of a single fan big enough for the job. U. S. Patents 2109838, 2191418.

TYPE S and SX For large industrial jobs. Heavy-duty double frame construction, pillow-block ball bearings.

Write for special bulletins and catalog sheets on above units, and the following equipment not shown here: Window Fans, Attic Packages, Shutters, Direct Drive Fans.

CONDENSED PERFORMANCE DATA and DIMENSIONS

Coolair V-Belt Drive Exhaust Fans

TYPE H (Ultra-quiet, spring-mounted)

Blade dia (inches)	hp	rpm	cfm	Overall Dim. (inches) Ht.	Width
26	1/6	490	4350	30 5/8	30 5/8
32	1/4, 1/3	420, 475	7730, 8620	36 5/8	36 5/8
38	1/4, 1/3	331, 371	9800, 11050	42 5/8	42 5/8
44	1/3, 1/2	291, 326	13300, 14670	49	49
50	1/2, 3/4	284, 318	17250, 19300	55 1/8	55 1/8
56	1/2, 3/4	242, 272	20000, 22500	61 1/8	61 1/8
62	1/2, 3/4	216, 241	24400, 27200	67 5/8	67 5/8

TYPE HT (Ultra-quiet TWIN UNITS)

26	1/4	468	8400	30 5/8	61 1/4
32	1/3, 1/2	361, 420	13000, 15460	36 5/8	73 1/4
38	1/2	326	19300	42 5/8	85 1/4
44	3/4	317	28800	49	98
50	1	284	34500	55 1/8	110 1/4

TYPE C (Commercial, Rigid Frame).

26	1/6 to 1/2	557 to 780	4900 to 7200	30 5/8	30 5/8
32	1/4 to 3/4	441 to 651	7300 to 10450	36 5/8	36 5/8
38	1/4 to 1	334 to 552	9100 to 14670	42 5/8	42 5/8
44	1/3 to 1 1/2	300 to 494	12040 to 19450	49	49
50	1/2 to 3	284 to 496	17250 to 30000	55 1/8	55 1/8
56	1/2 to 2	242 to 370	20000 to 30000	61 1/8	61 1/8
62	1/2 to 3	216 to 369	24400 to 41500	67 5/8	67 5/8

TYPE CT (Commercial Twin Units)

26	1/4 to 1/2	510 to 612	8900 to 10900	30 5/8	61 1/4
32	1/3 to 3/4	383 to 523	12680 to 17100	36 5/8	73 1/4
38	1/2, 3/4	334, 378	18200, 20600	42 5/8	85 1/4
44	3/4, 1	309, 330	24800, 26500	49	98
50	1 to 2	284 to 351	34500 to 42600	55 1/8	110 1/4

TYPE S (Industrial Units)

72	1 to 5	155 to 270	35000 to 60000	75 1/4	75 1/4
84	2 to 7 1/2	170 to 270	58000 to 85000	87	87

TYPE SX (Industrial Units)

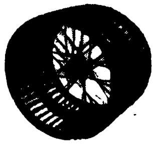
96	3 to 10	150 to 225	80000 to 120000	99	99
108	5 to 15	150 to 215	110000 to 154000	111 1/2	111 1/2

Bayley Blower Company

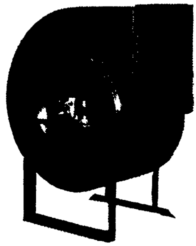
1821 S. Sixty-Sixth Street **Branches in Principal Cities** **Milwaukee 14, Wis.**

Engineers and Manufacturers of Fans, Washers, Heaters and Other Air-Handling Equipment

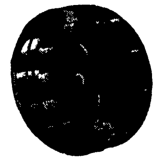
PLEXIFORM



Type "F" Wheel



AEROPLEX

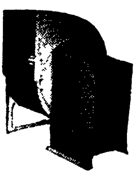


Type "AP" Wheel

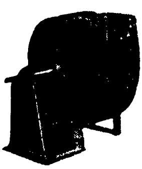
CENTRIFUGAL FANS

Bayley offers a complete selection of types, sizes and arrangements of centrifugal fans for ventilating, conditioning and mechanical draft service, with slow-speed or power-limiting wheels. Made in single and double widths, with capacities up to 300,000 cfm or more. Smaller sizes have reversible housings for convenience and economy.

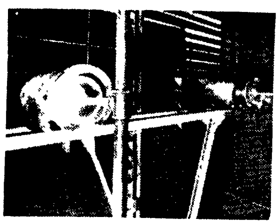
Type "F" Fans are equipped with wheels having forward-curved floats, effecting a slow-speed characteristic. The floats of the wheels supplied with Type "AP" Fans are backwardly inclined, producing a non-overloading characteristic. Choice of fans is determined by the design conditions of the individual installation.



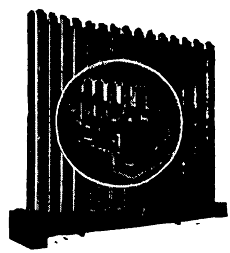
Vent Set



Type "EX" Fan



Turbo Washer



Chinook Heater

VENTILATING SETS—Compact, unitary Ventilating Sets are manufactured with drive complete, ready to use, direct-connected and belted styles with capacities to 15,000 cfm. These handy units serve a wide variety of applications in ventilation and exhaust, for kitchens, lavatories, gymnasiums, restaurants, garages, halls, hospitals, laboratories, industrial plants, and many other locations. May be used for supply, exhaust or both. They can be equipped with weather-protection hood for outdoor or penthouse installation. Carefully engineered and sturdily built to give long-lasting, efficient service.

INDUSTRIAL FANS—Bayley Industrial Fans are available in a wide range of sizes and constructions from which to select the unit best fitted for any application in exhausting, conveying, cleaning, drying, blast, draft, or similar duties. Type "EX" is proportioned primarily for medium pressures and capacities usually encountered in materials handling or exhaust. Type "H" is similar, but proportioned for relative higher pressures and smaller volumes. Designs are adaptable to suit virtually any requirement of temperature, corrosion-resistance or other severe operating conditions.

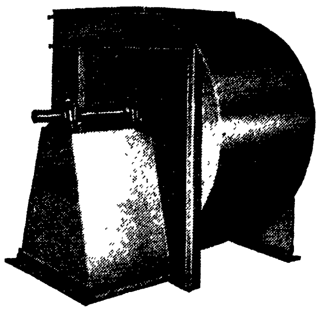
AIR WASHERS—Because Turbo washers atomize by mechanical means, they cannot clog, hence are admirably suited to handling air contaminated with particulate matter. Standard pressure-nozzle washers also available for use where dust content is light.

BLAST HEATERS—The finless, long-lasting iron pipe coils of Chinook Heaters will not clog, therefore maintain full capacity longer. Pipe-within-a-pipe arrangement has single header for supply and return, cannot short-circuit, and absorbs expansion and contraction without damaging stresses. A size and type for any blast heating application.

The Bishop & Babcock Mfg. Co.

Massachusetts Blower Division

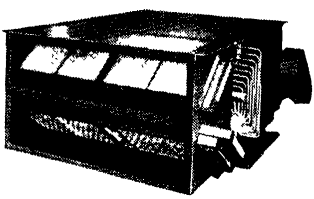
4901 Hamilton Ave. **Cleveland 14, Ohio**



SQUIRREL CAGE AND POWER FIXED FANS

Squirrel Cage Fans, outstanding in performance, slow speed characteristics.

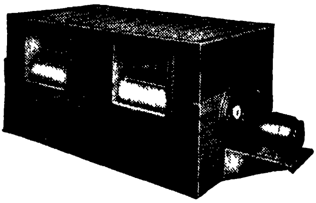
Power Fixed Fans are backward curve blade type, with non-overloading characteristics. Double width, double inlet. Class I or Class II construction. Rating and dimension tables available. Sizes 13½ in. to 86½ in. wheel diameter, single and double width. Write for catalogs.



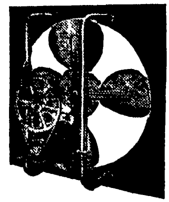
MASSACHUSETTS AIR CONDITIONING UNITS

Designed to combine cooling in summer and heating in winter. Available in seven sizes ranging from 1,000 cfm to 14,000 cfm and may be obtained in either vertical or horizontal design. These units are built in sections to facilitate easy handling and erection. Built with various heating and cooling coil combinations to suit requirements. These units incorporate traditional Massachusetts features of fine workmanship and superior performance. Write for catalog.

The new Design 2 Air Conditioning Furnace Blowers are now available, with a wide variety of stock combinations and discharge arrangements. They can be furnished in special widths or in multiples. Also available are wheel assemblies, housings and housing sides. Sizes 7 in. to 27 in. wheel diameter. Write for catalog.



Unit Heaters. Blower type. Floor and Ceiling type made in 13 standard sizes, with regular or non-freeze coils, filter and damper sections. Ratings from 50,000 Btu up. Propeller Fan type "H," Horizontal made in 16 sizes. Type "V" Vertical projection for ceiling mounting. Write for catalogs for full information.



Propeller Fans available with Belt drive with wheel sizes 24 in. to 48 in. Direct Drive with wheel sizes 12 in. to 30 in. Both types available with single or 2 speed motor. A complete line of Automatic Shutters and Fan Houses are obtainable. Write for catalog.

Air Washers. Four types available. Type A—cooling dominating cleansing. Type B—as much cooling as possible without mechanical refrigeration. Type C— cooling and cleansing with use of mechanical refrigeration only. Type D—Identical with type A except that flooding nozzles have been omitted.

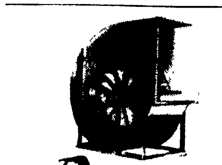
Buffalo Forge Company

450 Broadway, Buffalo, N. Y.

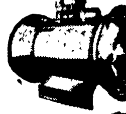
Manufacturers of Unit Heaters, Multiblade Fans, Air Washers, Unit Coolers, Drying Equipment, Mechanical Draft Fans, Air Preheaters, Blowers, Exhausters, Disk Fans, Spray Nozzles. For Complete Information, Write for Bulletins Indicated, or Call Your Trained "Buffalo" Engineering Representative In Nearest City Listed Below.

ENGINEERING REPRESENTATIVES

ALBANY 7, N. Y., Mr. R. B. Taylor, 966 Broadway; ***ATLANTA, GA.***, Mr. J. J. O'Shea, 305 Techwood Dr., N. W ; ***BALTIMORE 1, MD.***, Machinery & Equipment Sales, Inc., Mr. C. A. Conklin III, 1014 Cathedral St.; ***BOSTON, MASS.***, Mr. E. D. Johnson, 507 Main St., Melrose Station; ***KITCHENER, ONTARIO, CANADA***, Mr. A. S. Capwell, Canadian Blower and Forge Co.; ***CHICAGO 6, ILL.***, Emmert & Trumbo., 20 N. Wacker Drive; ***CINCINNATI 25, OHIO***, Mr. S. O. Johnson, 2944 Colerain Ave.; ***CLEVELAND 15, OHIO***, Weager & Sherman, 570 Hanna Bldg.; ***CORPUS CHRISTI, TEXAS***, Langhammer-Rummel Co, 101 N. Alameda St.; ***DALLAS 1, TEXAS***, Mr. T. H. Anspacher, 615 Tower Petroleum Bldg.; ***DENVER, COLORADO***, Hendrie & Bolthoff Co., Box 5110, Terminal Annex D; ***DETROIT 16, MICHIGAN***, Coon-DeVisser Co., 2051 W. Lafayette Blvd.; ***GREENVILLE, S. CAROLINA***, Mr. Roy A. Stipp,P. O. Box 1796, 104 E. Stone Ave.; ***HOUSTON 5, TEXAS***, Mr. D. M. Robinson, 2609 Sunset Boulevard; ***INDIANAPOLIS 4, INDIANA***, S. E. Fenstermaker & Co., 937 Architects and Builders Bldg.; ***DAVENPORT, IOWA***, D. C. Murphy & Co., 305 Security Bldg.; ***DES MOINES 14, IOWA***, D. C. Murphy & Co., 810–5th Avenue; ***KANSAS CITY, KANSAS***, Mr. W. K. Dyer, Box 966, Post Office Bldg., Mission, Kansas; ***KNOXVILLE, TENN***, Mr. C. F. Sexton, P. O. Box 2224, 702 Empire Bldg.; ***LITTLE ROCK, ARKANSAS***, Mr. J. L. Brown, Terminal Warehouse Bldg.; ***LOS ANGELES 13, CAL.***, Halladay & Knauff, 804 Pershing Sq. Bldg.; ***LOUISVILLE 2, KENTUCKY***, H. M. Lutes, 633 South Fifth St.; ***MEMPHIS, TENN.***,Humphrey-Wynne Co., 620 Sterick Bldg.; ***JACKSONVILLE 4, FLORIDA***, H. L. McMurry & Co., 25 Riverside Viaduct, ***MIAMI, FLORIDA***, H. L. McMurry, 46 N. E. 6th St.; ***TAMPA, FLORIDA***, H. L. McMurry & Co., % Peninsular Warehouse & Ter. Co., 411 Hampton St.; ***MINNEAPOLIS 2, MINNESOTA***, E. Floyd Bell, 2102 Foshay Tower; ***NEW ORLEANS 12, LA.***, Devlin Bros., 1003 Maritime Bldg.; ***NEW YORK 7, N. Y.***, Johnson & Norman, 39 Cortlandt St.; ***NEWARK 2, N. J.***, Johnson & Norman, Room 205, 27 Washington St.; ***OMAHA 2, NEBRASKA***, Wain Engineering Company, 2311 Douglas St.; ***PHILADELPHIA 2, PA.***, Davidson & Hunger, 1200 Cunard Bldg.; ***WILKES-BARRE, PA.***, Power Engineering Co., 517 Brooks Bldg.; ***PITTSBURGH 22, PA.***, Mr. H. Lee Moore, 345 Fourth Ave.; ***PORTLAND, ORE.***, Arthur Forsyth Co., 921 S.W. Oak St.; ***ROCHESTER 4, N. Y.***, R. D. Moyer, 846 Sibley Tower Bldg.; ***ST. LOUIS 3, MO.***, Mr. J. W. Cooper, 2118 Pine St.; ***SALT LAKE CITY 1, UTAH***, Pace Turpin & Co., 726 So. Third St ; ***SAN FRANCISCO 3, CALIFORNIA***, Mr. Chas. W. Lockhart, 1214 Central Tower Bldg.; ***SAN ANTONIO 6, TEXAS***, Langhammer Rummel Co., 300 Blum St., P. O. Box 367; ***SEATTLE 1, WASHINGTON***, Arthur Forsyth Co., 3150 Elliott Avenue; ***SPOKANE, WASHINGTON***, Arthur Forsyth Co., 501 Hyde Bldg.; ***TOLEDO 2, OHIO***, Mr. Carl Eyster, 1118 Madison Ave.; ***WASHINGTON 5, D. C.***, Mr. G. S. Frankel, Mgr., 310 Woodward Bldg.; ***WILKES-BARRE, PA.***, Power Engineering Co., 517 Brooks Bldg.;



TYPE "LL" FANS. Used in large ventilation systems in plants, public buildings, and shops because of their quiet operation, non-overloading characteristic and complete selection of sizes from 600 to 500,000 cfm. Backward curved blades, inlet vanes and special housing scroll shape contribute to high efficiencies. BULLETIN 3737.

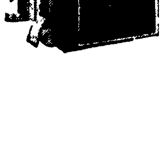


AXIAL FLOW FANS. Like the "LL" Fans above, these are used for ventilation and exhaust. Their "straight-through" delivery, light weight and compact design make them ideal for mounting in straight duct runs like a section of pipe. For light duty service (to about 3 in. static pressure). BULLETIN 3533-C.

AIR WASHERS. "Buffalo" Air Washers have non-clogging, "Buffalo" Spray Nozzles, trouble-free "Buffalo" Pumps—simple eliminator plate and tank design for simple maintenance. May be had in combinations for air spraying, surface cooling, heating or filter cleaning. BULLETIN 3142-D.

BREEZO-FIN HEATERS. Will operate on as low as 2 lbs steam pressure. Heater element is a one-piece, seamless copper tube with square copper fins spaced to give maximum radiation. Its "Buffalo" Breezo Fan throws heat efficiently. BULLETIN 3137-D.

INDUSTRIAL EXHAUSTERS. Available with interchangeable "AW" Air Wheels and "MW" Material Wheels. All-welded steel plate construction gives smooth interior surfaces for minimum friction loss. Models to handle gases up to 750 deg, also rubber-lined models. BULLETIN 3576.



PC CABINETS. Compact central air conditioning units for (1) simple cooling, (2) cooling and de-humidifying, (3) heating and humidifying, (4) continuous air cleaning. Vertical floor types, horizontal suspended types and units with air washer sections like the model shown. Equipped with "Buffalo" fans and pumps. BULLETIN 3703.

Champion Blower & Forge Co.

Manufacturers and Engineers

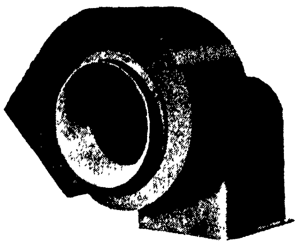
Plant & Offices: **Lancaster, Pa.**

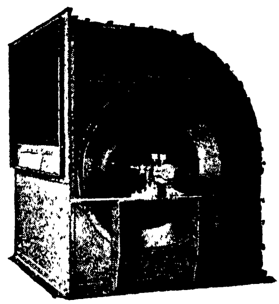
Address Correspondence to Div. 9

Manufacturers of Blowers, Ventilating Fans and Exhaust Fans for Air and Material; and Blast Gates

Representatives in Principal Cities

Type D



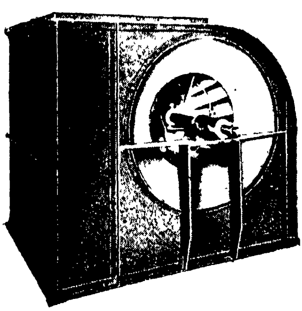


Type S

Type D Backward curve ventilating fans, single width, and electric drive up to 30 in. wheel diameter.

Type S Forward curve ventilating fans, single and double width. Sizes 6 in. to 60 in. wheel dia.

Type BC



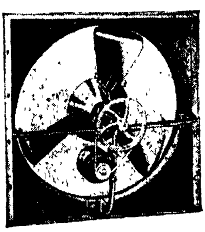
Type CE

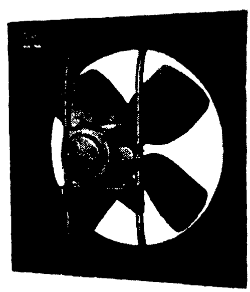
Type BC Backward curve ventilating and exhaust fans, single and double width; belt driven and direct connected electric.

Sizes 12 in. to 60 in. wheel dia.

Type CE Electric cast iron exhaust and forced draft fans. Also volume control dampers.

Type A





Type SV

Type A Industrial Ventilating and domestic attic fans. Built in sizes 30 in., 36 in., 42 in., 48 in.

Type SV Super Ventilating fans, direct motor drive up to 36 in. diameter. Motor belt drive up to 48 in. size.

DeBothezat Fans Division

American Machine and Metals, Inc.

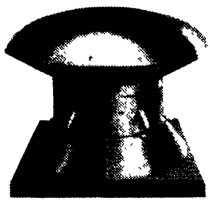
***Main Office and Factory*—East Moline, Illinois**

Foreign Sales Office: WOOLWORTH BUILDING, NEW YORK 7, N. Y.

SALES ENGINEERING OFFICES IN ALL PRINCIPAL CITIES

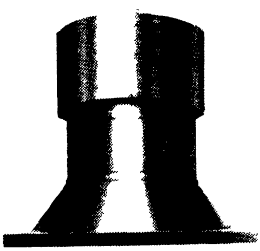
LOOK IN YOUR TELEPHONE DIRECTORY UNDER "FANS" OR "VENTILATING EQUIPMENT"

CONTROLLED VENTILATION



Power-Flow Roof Ventilator

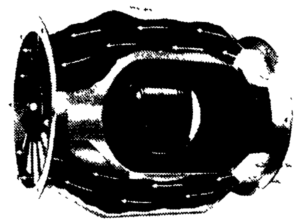
Motor driven fan in weatherproof housing provides positive controlled ventilation at all times, regardless of wind direction or velocity. Streamlined appearance, low height. Operates efficiently with or without duct system. Power-Flow Roof Ventilators are available with fan wheels 12 inches through 48 inches in diameter, and capacities up to 40,900 cfm. Catalog furnished on request.



Vertical Discharge Roof Ventilator

DeBothezat Vertical Discharge Roof Ventilators exhaust straight up at high velocity. Available with fan wheels 36 inches, 42 inches and 48 inches in diameter, with capacities up to 40,900 cfm. Model 7B7 has square base with conical transformation for standard curb openings. Model 7B8 has square base with reinforcing gussets for installations requiring special curb sizes. Bulletin furnished on request.

FUME REMOVAL



Bifurcator (Cut-away view)

For exhausting abnormally hot, corrosive, inflammable or explosive fumes Motor is mounted in separate chamber. Destructive fumes are by-passed (bifurcated) around motor as illustrated above. Bifurcator Fans install directly in the duct, in any position. Available with fan wheels 12 inches through 48 inches in diameter, and capacities up to 45,000 cfm. Catalog furnished on request.

VENT SETS

DeBothezat Axial-Flow Vent Sets are built with fan wheels from 16 inches through 48 inches in diameter. Fan wheels can be volume type (with 4 blades) or pressure type (with 14 blades). Non-overload power characteristic. Certified performance ratings. Catalog on request.

SPOT COOLING

DeBothezat "Hy-V" Air Jets blow a directed stream of cooling air as far as 35 feet without the use of ducts. Available with bracket for wall mounting or equipped with portable wheeled stand. With fan wheels 18 inches through 30 inches in diameter, and capacities up to 12,000 cfm. Catalog on request.

General Blower Company

8600 Ferris Ave., **Morton Grove, Ill.**

Engineering • Manufacturing
Application of Blowers
Fans and Exhausters

FOR BETTER AIR MOVING PERFORMANCE

More than 20 years' experience in building Blowers, Fans, and Exhausters to meet the exacting specifications of leading Architects Engineers and Contractors.

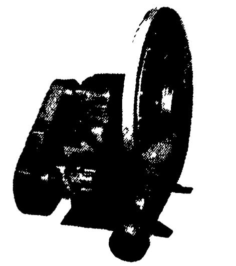
Engineering skill and integrity in building and supplying all types of Centrifugal Fans, Turbo Blowers, and Gas Boosters enables us to supply the type of equipment best suited for the particular need, whether standard or custombilt.

Sales Engineering offices maintained in all large cities to bring the engineering knowledge and skills, and the resources of General Blower Company to your organization.

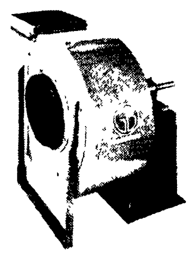
We engineer and build many special types of fans and blowing equipment. Consult us on your particular problems.

Write for our illustrated Products Bulletin covering the complete line of

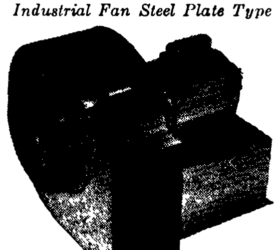
"LUNGS FOR INDUSTRY"



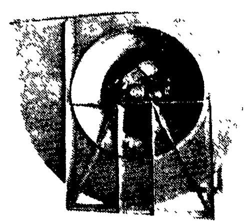
Turbo Blower With Oil Pump Assembly



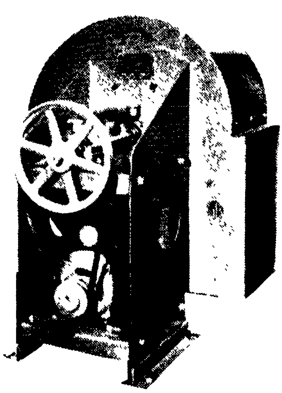
Industrial Fan Steel Plate Type



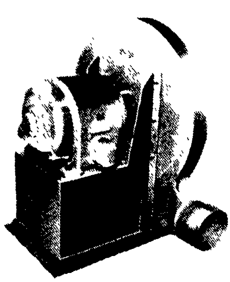
Direct Driven Multi-Vent Set



Centrifugal Fans. Backward and Forward Curve Types in all Standard Arrangements



Belt Driven Multi-Vent set weather cover removed

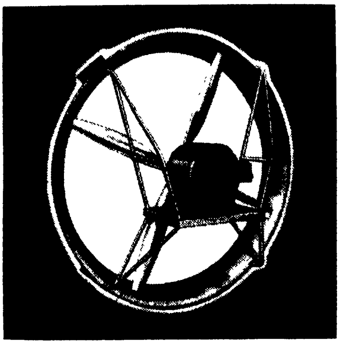


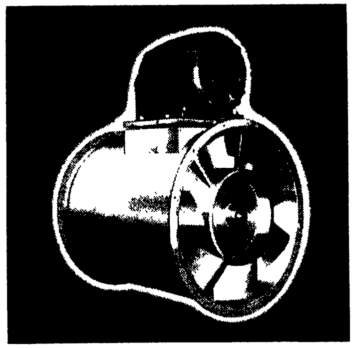
Turbo Blower

Hartzell Propeller Fan Co.

DIV. OF CASTLE HILLS CORP.

Piqua, Ohio





PROPELLER TYPE FANS

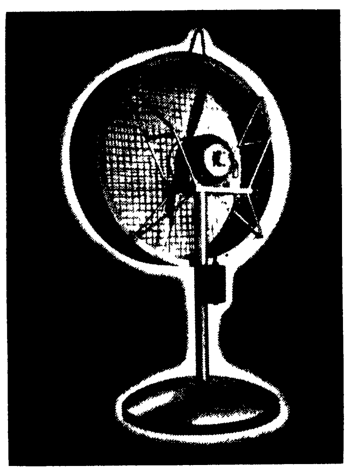
Single-Propeller, Two-Propeller and Multiblade. 12 in. to 60 in. Cast aluminum alloy propellers . . . not sheet metal stampings. Standard make motors. Curved orifice air-seal ring provides extra air delivery.

VANEAXIAL BLOWERS

Compact, easy-to-install blowers for moving air against pressures up to 8 in. water gauge. 12 in. to 48 in. diameter. Belt-drive and direct-drive models. Wheel is a single, precision casting of aluminum alloy. Standard *NEMA* motors. Occupy no floor space.

COOL BLAST FANS

22 in. to 60 in. Stationary and portable models. Built to withstand heavy industrial use. Equipped with totally enclosed, ball-bearing motors, conservatively rated for continuous duty. Utility fan, not shown, 14 in. to 36 in. is shorter, lighter, more easily portable; fan can be revolved 360 degrees vertically.

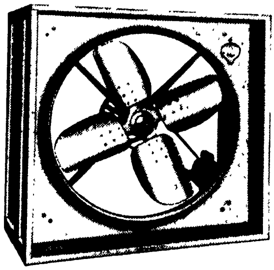


There is a Hartzell fan or blower to meet any air-moving need—Lo-Noise, Extension-Shaft, Pulley-Drive, Duct, Belt-Drive, Reversible, Cooling Tower and Mine Fans. Also makers of Unit Heaters, Fan-Powered Roof Ventilators, Penthouses, Intake Air Units, and Farm Crop Driers. Engineering representatives in principal cities.

Hunter Fan and Ventilating Company

Exclusive Fan Makers Since 1886

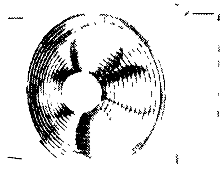
400 S. Front Street, **Memphis, Tenn.**



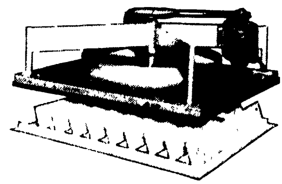
BELT-DRIVEN FANS

Capacities from 5100 to 22500 cfm; sizes 24 in. to 48 in. Certified air delivery ratings. Fans tested by *A & M College of Texas* according to Standard Test Code as adopted by ASHVE and *PFMA* and air deliveries are in accordance with the standard test code for centrifugal and axial fans and conform with U. S. Department of Commerce Standard No. CS 178-51. Heavy die-formed blades, balanced for quietness. Ball bearings throughout—sealed against dirt and grease leakage. Thrust type bearings permit installation in any position. Rubber-mounted ball-bearing motors, with built-in thermal overload protectors. *Underwriters' Laboratory* Label, with re-inspection service. Fan guaranteed five years; motor one year.

18 AND 22 INCH WINDOW FANS



Two speed reversible capacitor type motor. *Cat. No. B2500:* 18 in. blade, 2500 cfm—cabinet 24 in. high, 27 in. wide adjustable to 35 in. wide. *Cat. No. B3400:* 22 in. blade, 3400 cfm—cabinet 27⅝ in. high, 29¾ in. wide adjustable to 39 in. Certified air delivery ratings.



PACKAGE ATTIC FANS

Complete with ceiling shutter. Heavy duty motor, rubber mounted for quietness. Precision balanced blades. Built-in fuse link. Simple, inexpensive installation. Ball bearings throughout.

Underwriters' Laboratory approved. Fan guaranteed 5 years; motor and shutter, 1 year. 4750 and 6800 cfm models are complete with fan, motor and automatic ceiling shutter. Resilient rubber cushion on fan frame forms vibrationless air seal. Shutter with integral metal trim is finished in ivory and is designed to operate quietly and to eliminate drafts when closed. 7700 and 9700 cfm models are complete with fan, motor, manual shutter, built-in switch and ceiling trim. Pull-chain located at one corner of shutter controls fan and locks shutter open when fan is operating. No accessories are required.

30 INCH WINDOW FAN

Cat. No. W-3051. Certified air delivery: 7000 cfm. Electrically reversible. 30 in. blades. Belt-driven for quietness. Cools 4 to 6 rooms. Modern safety grille.

HUNTER ENGINEERING SERVICE

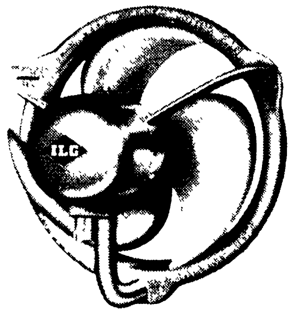
Hunter's Engineering Department will assist you with your cooling and ventilating design problems. See Hunter Section in Sweet's Catalog. Write for new Hunter manual "How to Cool for Comfort" giving methods and installation details.

ILG Electric Ventilating Co.

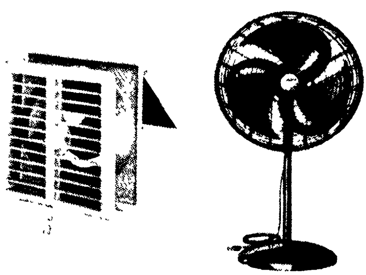
2880 North Crawford Ave., **Chicago 41, Ill.**

Offices in more than 40 Principal Cities

Propeller Fans, Centrifugal Fans, Unit Heaters, Kitchen Ventilators, Night Cooling Fans

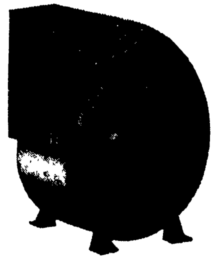


ILG Direct-Connected Self-Cooled Motor Propeller Fans
Used for exhaust of stale air, fumes, heat, dust, odors, etc. Self-cooled motor combines protection of enclosed motor with low operating cost of open motor—constantly cooled by fresh, clean air, circulated internally—never "gums-up" from contact with foul air—saves 5 to 10 per cent on power costs. Rugged, heavy-duty framework. Dynamically-balanced fan wheel, direct-connected to motor. Smooth, quiet, effortless operation—economical, long lived. "ONE-NAME-PLATE" Guarantee. Certified ratings. Sizes 8 in. to 72 in.

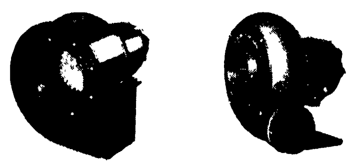


Kitchen Ventilators and Night Cooling Fans
Kitchen Ventilators (left)—wide range of sizes, types for mounting in wall, ceiling, or window. Full capacity. Extra-quiet. Rigid construction. Headlined by new LC10 Built-in Ilgette shown left above.

Night Cooling Fans (right above)—portable model for use at attic or downstairs window. For permanent installation in attic, use ILG Self-Cooled Motor Propeller Fans. (top of page)

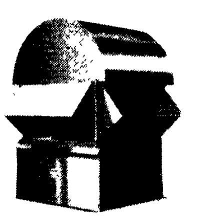


ILG Direct-Connected Centrifugal Fans
Type "BC"—Load-limiting type with backward curved blades. Motor load remains constant over wide range of air volume and change in static pressure. Wheel mounted directly on motor shaft with motor partially recessed in side of casing. No motor base required. Unobstructed inlet. 10 sizes. Also available for belt-drive in 12 sizes.



Volume Blowers

Type "B" (left)—small volume, low pressure, quiet running. Multi-blade wheel direct-connected to motor shaft. Cast iron base. Universal discharge. 12 capacities.
Type "P" (right) for exhausting dust, fumes, removal of steam, vapors. Four discharge positions to avoid friction in short bends. 7 capacities.

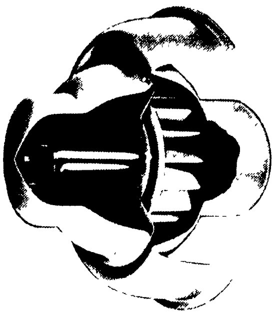


Type "PRV" Power Roof Ventilators
Centrifugal fan type, for exhaust from vertical flues or duct systems. Direct-connected, self-cooled motor. Non-overloading, backward curved wheel. 10 sizes. Up to 1½ in. SP.

Jenn-Air Products Company

Architects & Builders Building **Indianapolis 4, Indiana**

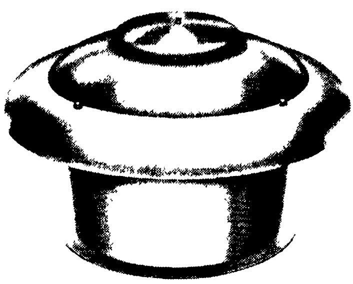
REPRESENTATIVES IN PRINCIPAL CITIES



Wall Exhauster

WALL EXHAUSTERS

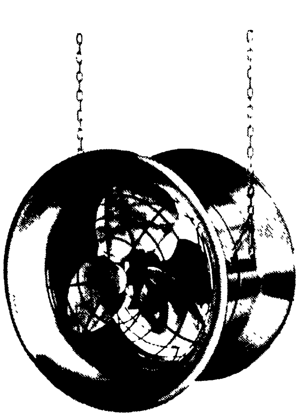
Jenn Air wall exhauster has proven its position in the commercial and industrial ventilating field. It is a complete package unit designed for mounting on the wall exterior. The aluminum housing provides enclosure for the motor which is mounted out of the air stream. The centrifugal wheels are non-overloading and the air is discharged radially. In many cases such an exhauster can be installed with but a fraction of the duct work required for venting through the roof.



Roof Exhauster

ROOF EXHAUSTERS AXIAL AND CENTRIFUGAL TYPES

These direct connected power units are available in axial as well as centrifugal types. Their low contour, special motor enclosure and all aluminum construction indicate the quality built into this product. The power unit is floated on live neoprene isolators to eliminate vibration. The axial roof type is recommended for lower static pressure applications while the centrifugal units operate well against higher resistances. Both have non-overloading characteristics.



Suspended Air Circulator

AIR CIRCULATOR

Jenn Air offers a positive way to overcome the heat problem. This industrial air circulator is suspended from above and thereby eliminates the possibility of accidental body contact. They take up no valuable floor space and have an adjustable angle of suspension which allows their use for short or long distance projection. The venturi design has the effect of increasing the efficiency of the fan without sacrificing quiet operation.

All of the above equipment is available in a wide range of sizes and also in explosion proof types. Write Department H for information.

JOY MANUFACTURING CO.

General Offices: Henry W. Oliver Building, Pittsburgh 22, Pa.

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AND MORE THAN 500 DISTRIBUTORS THROUGHOUT THE WORLD

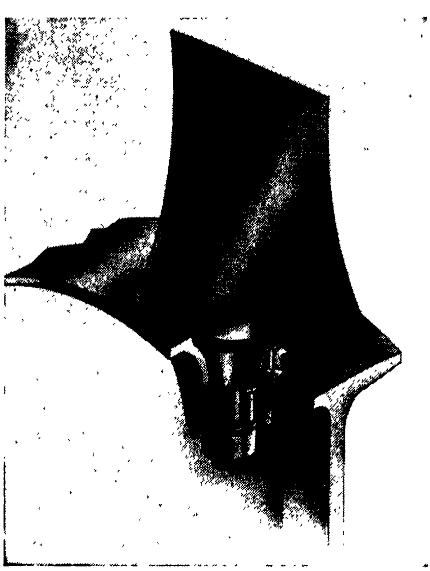
SERIES 1000 AXIVANE* INDUSTRIAL AND COMMERCIAL FANS

Joy Series 1000 adjustable blade **AXIVANE*** Industrial fans are available in 124 sizes ranging in volume capacity up to 100,000 cfm with pressures up to 9.6 in. W. G. Housing diameters range from 18 in. to 60 in. For complete specifications, construction details, and selector charts giving pressure-volume range for each fan, write for bulletin number J-605.

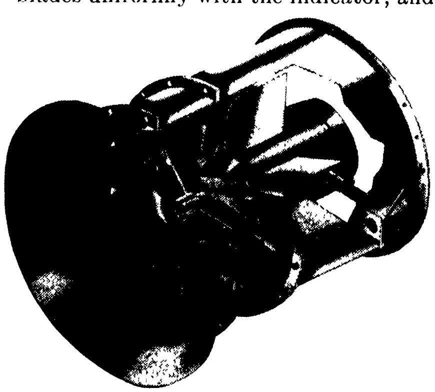
Joy **AXIVANE*** Series 1000 fans are efficient, quiet, compact, flexible, and easy to install.

ADJUSTABLE BLADES

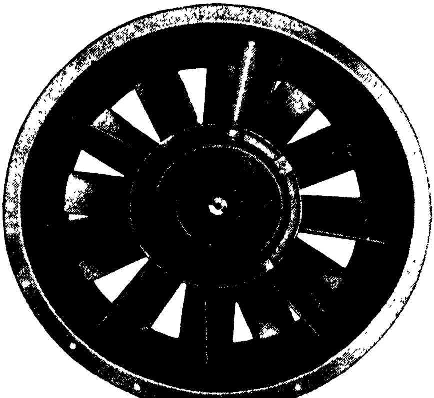
Joy **AXIVANE*** Industrial Fans have the extra performance flexibility of adjustable blades. Adjustable blades are standard equipment on all Series 1000 fans. The factory blade setting can be quickly changed to provide either a wide pressure range for any particular volume or a change in a volume simply by loosening a lock nut with a wrench, setting the blades uniformly with the indicator, and



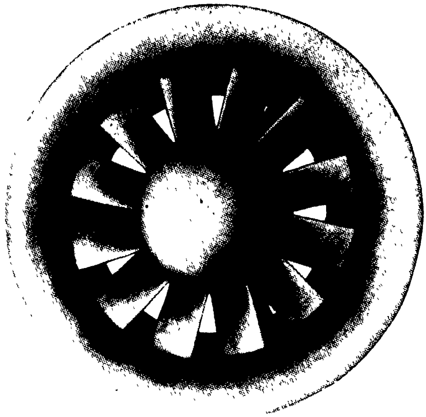
Blades are adjustable on the job by loosening one lock nut



Cutout drawing showing location of motor and compact construction



Rear view, showing vanes and motor



Front view, showing simplicity of construction

retightening the lock nut. A permanent stop prevents setting blades in a position likely to overload the motor. Minimum blade settings are limited by the fan housing.

Adjustable blades permit on-the-job correction for unpredictable duct resistance or for poorly installed duct work.

MORE EFFICIENT

Stationary straightener vanes, located immediately behind the rotor, partially recover the rotative energy imparted to the air by the rotor, and re-establish axial flow to the air leaving the vanes. This eliminates excess turbulence at the point where the air enters the duct system and increases efficiency by decreasing pressure loss.

The Joy **AXIVANE*** fan utilizes an aerodynamically efficient blade and stationary vane design.

QUIETER OPERATION

For equal weight and space the Joy **AXIVANE*** fan is quieter than a centrifugal type fan of equal volume and pressure. The streamlined airflow from an **AXIVANE*** fan makes sound insulation a simple and inexpensive operation when required for the ventilation and air conditioning of quiet spaces such as hospitals, auditoriums, radio stations, etc., where insulation against system noise must be used.

MORE COMPACT

Joy **AXIVANE*** fans are built around the motor, the fan housing becoming an actual part of the duct system. This produces a more compact design than is possible with a centrifugal fan. An **AXIVANE*** fan, installed on an in-line connection with ventilation ducts, parallel to and close by an overhead structure, may require 70 per cent less space than a conventional belt-driven centrifugal fan. The compactness of a Joy **AXIVANE*** fan assures a maximum of net operating or rentable area. Fan rooms are virtually eliminated.

EASIER TO INSTALL

The Joy **AXIVANE*** Series 1000 fan develops a greater volume and pressure per pound of fan and motor because of its compact, in-line construction. This light weight permits a simplicity of installation that minimizes installation costs and total weight by eliminating heavy foundations, complex duct offsets and elbows, drives, and guards. **AXIVANE*** fans can be installed quickly and easily, even by relatively inexperienced or un-skilled labor.

MATCHED ACCESSORIES

Inlet bells, screens, and fan supports are accessories designed to fit all **AXIVANE*** Fan housings. In ordering, it is only necessary to state the model number with or without the accessories as desired. If required with accessories, these will be furnished to fit the fan model ordered without special number.

No matter how carefully a duct system is planned, an incorrectly selected inlet bell will reduce fan efficiency by increasing intake turbulence. This excess turbulence will also increase the noise level of the fan. When a fan takes its air directly from the weather, a plenum, a fan room, or from a duct system larger in circumference than the fan housing, a bell should be used.

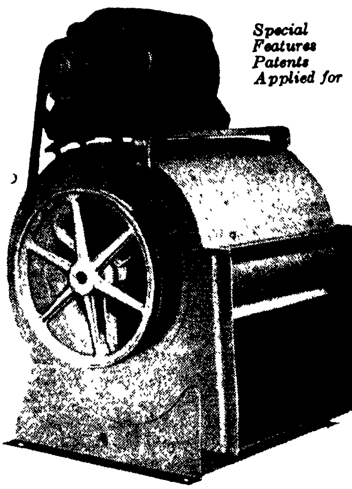
***Reg. U S. Pat. Off**

The Lau Blower Company

2007 Home Avenue, Dept. H, Dayton 7, Ohio

Engineers and fabricators of general Air Handling Equipment
Blower Assemblies • Blower Wheels • Propeller Fans • Accessories

NEW Series "A" Blower Assemblies

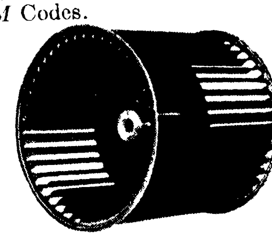


Special Features Patents Applied for

The Lau Series "A" Blower Assembly—result of years of exhaustive tests of all types of blowers—years of research and design evolution—is the all-time, outstanding achievement in the blower field. Greater mechanical strength. Greater efficiency. A more compact unit (overall size considerably smaller than formerly). Will fit more jobs. Embodies many new and revolutionary features exclusive with Lau. Includes new 3-point suspension type bearing bracket—an integral part of the shroud—identical for various angles of discharge. New frictionless, self-aligning bearing—completely encased in Neoprene. New center suspension wheel (see below). New discharge outlet design and construction. Cut-off cannot set crooked on outlet. No wavy edges. Faster installations. New 1-piece motor mounting easily convertible, rear to top or vice versa, by simple use of two sheet metal screws. Many other features. Complete range of sizes Every size tested and rated for performance in accordance with A.S.H.V.E. and *NAFM* Codes.

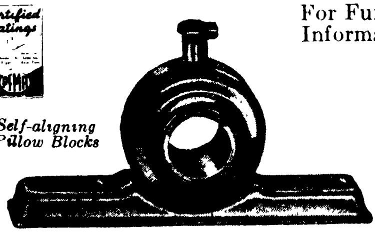
NEW Series "A" Blower Wheels

New center-suspension wheel *tested and proved* by us to have greater mechanical strength, truer concentricity and far more efficient performance than ordinary types of wheel. Complete details supplied. →



Write Dept. H.
For Further
Information

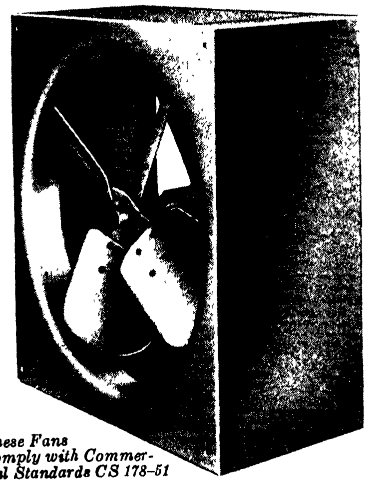
Certified Ratings



Self-aligning Pillow Blocks

Propeller-type "Niteair" Fans

For a wide variety of applications where it is necessary or advantageous to exhaust undesirable air and provide fresh air from the outside. Equally applicable for industrial, commercial, residential and farm building installations. Efficient and economical method for correcting innumerable air-control problems—removing dust-laden, foul, contaminated, or excessively hot air, fumes, gases, smoke. Venturi-type entrance housing reduces air "drag" and turbulence—eliminates most common cause of "air noise." 5 sizes—24 in. to 48 in.

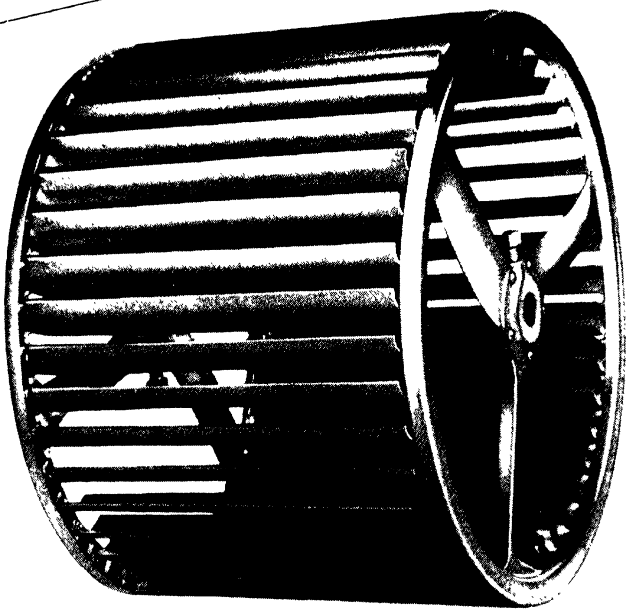


These Fans Comply with Commercial Standards CS 178-51

Morrison Products, Inc.

A 25 Year Firm

East 168 Street and Waterloo Road, **Cleveland 10, Ohio**



MORRISON BLOWER WHEELS—*Made Exclusively for original equipment manufacturers* in heating, ventilating and air conditioning.

Morrison Blower Wheels are double width—double inlet in standard diameters from 10 in. to 16 in., and in width from 6 in. to 16 in.

One-Piece Blade Construction. Three-Piece Balanced Assembly—one-piece blade and two pressed rings with integral hubs welded together.

Equalized Weight Distribution with end mounting. No shaft whip—reduced deflection.

Available with Morrison Blower Wheels is Complete Engineering Service. Included are templates, shop drawings, tables, data, cost analysis, graphs, charts, sources of component parts. Housing Squares and Scroll Sides available for Low Cost Assemblies.

Catalogs: Morrison Blower Wheels—Copies Mailed Upon Request.

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FANS ● BLOWERS ● UNIT HEATERS ● MAKE-UP AIR UNITS
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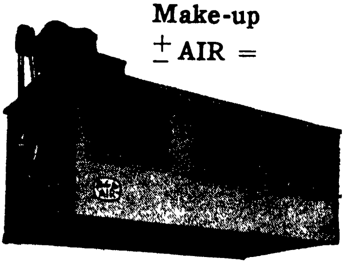
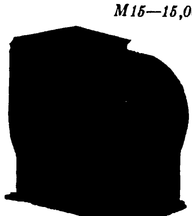


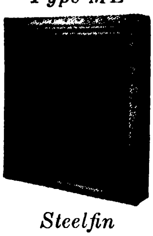
M15—15,000 cfm

Comet Exhaustair

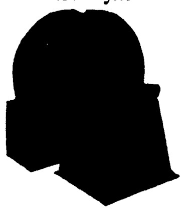
Comet Unit Heater



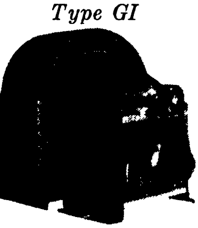
Type ME



Steelfin



Type GI



General Purpose Fan

M15—A unit that delivers, warmed, filtered, outside air to industrial spaces to replace exhausted air and balance minus pressure. Corrects drafty conditions and uncontrolled infiltration. Made in 4 sizes from 5,000 cfm to 20,000 cfm. Described in *Bulletin 502.*

Type ME Centrifugal Fans—Capacities up to 101,000 cfm. Slow speed wheels offered from 7½ to 73 in. Capacities **up** to 123,000 cfm. Quiet operating (PL) medium speed wheels with non-overloading horse-power characteristics for heating, ventilating and air conditioning or industrial applications. Wheel diameters from 18 in. to 73 in., with any speed or discharge required. Class I, II, III or IV construction. *Write for Bulletin 493.*

Steelfin Hot Blast Heating Surface—Extra heavy duty, fin-and-oval tube, all-steel, welded construction. A hot dip metallic coating over all, including headers, affords perfect bonding and conductivity. Suitable for continuous heating service on steam pressures up to 150 lb. *Bulletin 492.*

Comet Exhaustair—Delivers large volumes of air at low resistance and low current consumption. All wheels are machine balanced for smooth, vibrationless operation. Made in two types and eight basic sizes. Wheel diameters from 12 in. to 48 in. Direct or belted drive. Capacities from 400 cfm to 23,500 cfm. *Ask for Bulletin 511.*

Comet Unit Heaters—Heavy duty, welded steel, fin-and-tube heating element. Suitable for continuous heating service on steam pressures up to 150 lb or more. 10 sizes with capacities from 30 Mbh to 490 Mbh. *Bulletin 513.*

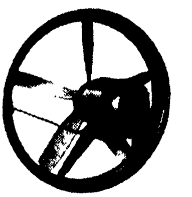
Type GI Industrial and Heat Fans—For dust and gas removal, conveying of materials and handling hot gases. Housings, drives, and discharge arrangements to meet any requirement. Wheel diameters from 10 in. to 66 in. Capacities from 450 cfm to 60,500 cfm. *Details and engineering data in Bulletin 482.*

General Purpose Fans—Portable, self-contained units for Class I industrial and ventilating applications. Recommended for ease of installation, low maintenance and space saving features. Made in three types and eight basic sizes. Capacities 400 cfm to 18,000 cfm. *Bulletin 512.*

PROPELLAIR Div.

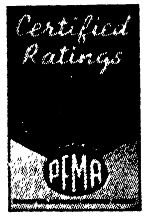
ROBBINS & MYERS, INC.
1947 Clark Boulevard
SPRINGFIELD, OHIO

VENTILATING SPECIALISTS IN ALL PRINCIPAL CITIES



Propellair Direct-Connected Fans

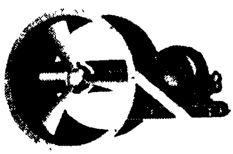
For Ducts, Walls, Windows, Hoods, Roof Ventilators



For use wherever motors may operate within the air stream, from free air to medium and relatively high resistance. A compact, durable design with fan having two to six blades. Sizes: 12 in. to 60 in. Capacities: 800-85,000 cfm. Type "C.D."

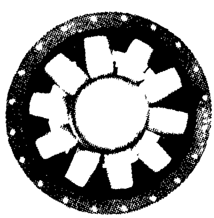
Belt-Driven Propellair Tube-Axial Type

For Heat, Acids, Alkalies -Fumes, Gases, Dust



A complete fan unit in short duct section ready for installation in lines from 20 in. to 48 in. diameter. Type "CS" may be used for severe acid or alkaline conditions, explosive fumes and gases. Type "CSV," for excessive temperatures, circulates outside air through belt and fan shaft tubes to keep drive and bearings cool. Capacities: 4,100 to 43,000 cfm.

Propellair Vaneaxial Fans

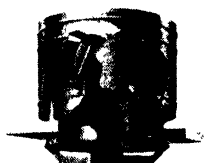


A compact, highly efficient pressure fan using standard steel drum sections incorporating standard *NEMA* frame motors in direct drive models. Also available in belt-driven ratings with motor outside the air stream. Cast aluminum airfoil propeller and guide vanes for maximum efficiency and durability. Available in 20, 24 and 30 in. diameters, ranging from 4,000 to 15,000 cfm.

For Heat, Moisture, Fumes, Dust, and Gases

Propellair Extended-Shaft Fans

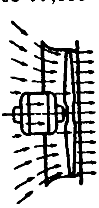
This design locates motor outside air stream when fan is installed in duct at right angle turn, elbow, "Y," or offset. Simple installation usually can be supported by duct without auxiliary bracing. Drive shaft is enclosed and sealed within steel tube. Sizes: 12 in. to 60 in. Capacities: 2,000 to 68,000 cfm. Type "CE."



Propellair Sky Blast

Power Roof Ventilator

Dependable and economical power roof ventilators. Butterfly dampers open wide the instant fan is started, close automatically as fan coasts to a stop, offer virtually zero resistance as heat, fumes, moisture, dust shoot high into air. Rain is prevented from entering by fan when operating. Drainage gutter prevents leakage when dampers are closed. Sizes: 20 in. to 60 in. Capacities: 3700 to 77,000 cfm.



Airfoil-Section Blades

Airfoil Principle Entrance Ring

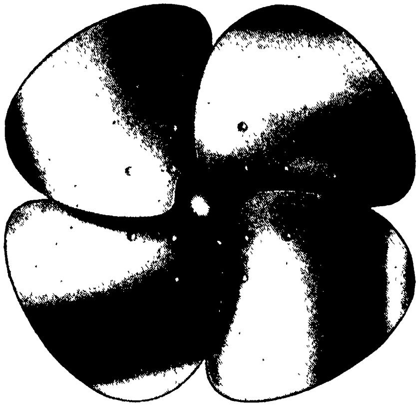
Propellair fans have airfoil-section blades with variations of pitch, curvature, and thickness to compensate for different lineal speeds of points at various radii. Air movement is uniform over whole fan area. The Propellair curved entrance ring eliminates eddy currents; helps efficient Propellair blades deliver highest pressure and volume.

The Torrington Manufacturing Co.

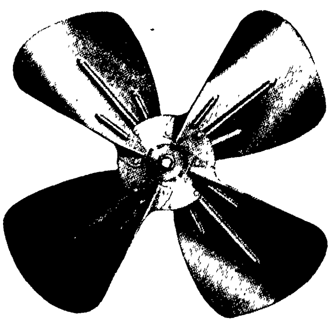
50 Franklin Street, Torrington, Conn.

Manufacturers of Blower Wheels and Propeller Type Fan Blades.

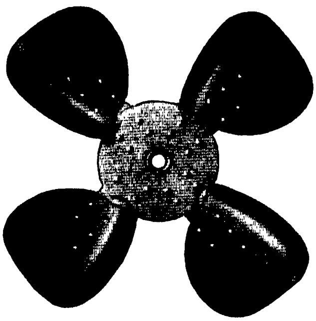
AIRISTOCRAT *Quiet Propeller Fan Blades* AIROTOR *Blower Wheels*



4-Blade Airistocrat Fan "E" Series



4-Blade Airistocrat Attic Fan "M" Series



4-Blade Airistocrat Attic Fan "B" Series

"E" Series AIRISTOCRAT Fan Blades—Outstandingly high efficiency is the chief characteristic of this truly *new* fan blade. It delivers more air for any given horsepower. Size for size it looks bigger, more powerful.

In impartial tests, ten competitive fan blades were recently compared with "E" Series blades of the proper diameter and pitch. In every case, air delivery was sharply increased. Within the same space limitations and with the same power, the "E" blade delivered as much as 28 per cent more air.

This high efficiency is the result of four years of research and development which from the beginning was devoted to bringing out a superior fan blade.

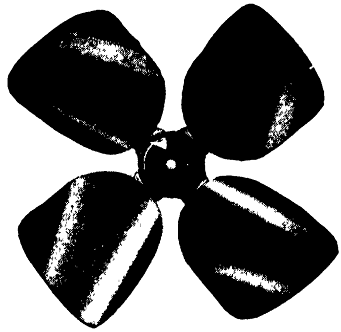
Convincing proof of the superior performance of this new fan may be found in the *NEMA* and *NAFM* tables prepared as a guide to selection. The catalog containing these tables and specifications will be mailed upon request.

Specifications: Three-blade models in 10 in., 12, 14, 16, 18 and 20 in. diameters; four-blade models 8 in., 10, 12, 14, 16, 18, 20, 22 and 24 in. diameters. Five pitches in most sizes. Aluminum blades, steel spider and hub. Standard finishes.

AIRISTOCRAT "M" Series Attic Fan Blades—Three outstanding features of this new design are: (1) Extremely high efficiency, which gives maximum cfm per horsepower; (2) knockdown construction which drastically lowers shipping costs; (3) quiet operation—a point of major interest to the consumer.

This all steel four-blade fan is manufactured for attic use exclusively, in 24, 30, 36, 42 and 48 in. diameters, in 40 deg pitch only.

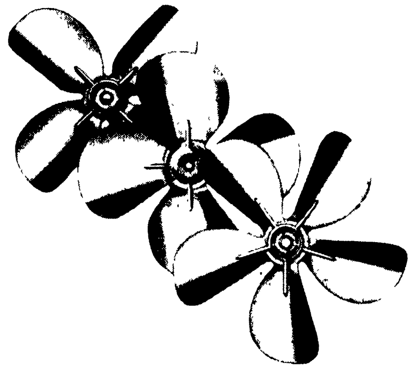
AIRISTOCRAT "B" Series Attic Fan Blades have the same proportions, proved aerodynamically correct, in all diameters. New larger center disc and heavier spider arms increase strength and a new blade shape adds to the appearance of this carefully designed product. Available in 3, 4 or 5 blades in standard diameters 24, 30, 36, 42 and 48 in. All steel construction. Available in the following finishes: 1. Plain. 2. All one color lacquer.



4-Blade Airistocrat Pressure Fan "U" Series

Pressure "U" Series—Four blade models of steel designed for pressure operation. Sizes 20 in., 22, 24, 26, 28 and 30 in. diameters.

"One-Piece" Airistocrat Fan Blades—Exceptionally rigid models blanked from one piece of metal. Made in both steel and aluminum. Sizes 3 in., 4, 4½, 5, 5½, 6, 6½, 8, 9, 10, 12 and 16 in. diameters, all four blades; also 5½, 7, 8, 9, 10 in. 5-blade. Available in the following finishes: 1. Plain. 2. Lacquered. 3. Zinc plated (steel only).

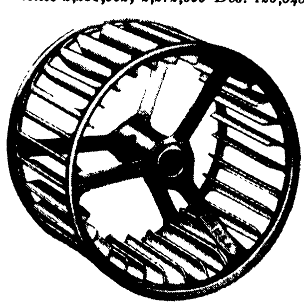


"One Piece" Airistocrat Fan

Torrington Airotor Blower Wheels are light, sturdy and inexpensive—incorporate new principles of design and construction, which insure rigidity and concentricity. **Single Width—Single Inlet** wheel is of simple four-piece construction. **No rivets or welds are used;** concentric rib serving as backing for blade strip is formed at same time as hub socket, insuring trueness of wheel. Rigid radial ribs prevent deflection by thrust. Three thicknesses of metal in rims make for maximum strength. Excellent for many heating and ventilating uses. Manufactured in both aluminum and steel in 1½ in., 2, 3, 3⅝, 4, 4½, 5, 6, 7, 7½, 8½, 9 and 10½ in. diameters. Clockwise or counterclockwise rotation. Same sizes available in DA type double width, double inlet wheels.

Airotor Blower Wheel—Single Width—Single Inlet
Patents 2,231,062; 2,272,695 Des. 126,043

Torrington Airotor Blower Wheel—Double Inlet—Spider End Plates. Has blades punched and formed in a single strip, rigidly held by flanged single piece end rings. Hubs are rigidly mounted by peening. Wheels of 2½ in., 3⅝ in., 7½ and 10½ in. diameter are available at present. Additional sizes now being developed.



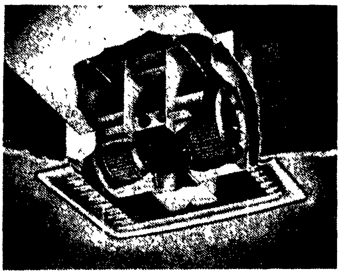
Airotor Blower Wheel Double Inlet—Spider End Plates

Trade-Wind Motorfans, Inc.

5725 So. Main St.
Los Angeles 37, Calif.

Representatives In All Principal Cities. Carried In Stock By Many Electrical Jobbers

TRADE-WIND CLIPPER CEILING VENTILATORS

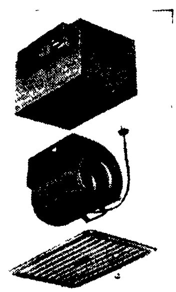


Cutaway view of Model 2501 shows double blowers used in this unit.

THE TRADE-WIND CLIPPER BLOWER is a small capacity centrifugal blower, primarily used for exhaust application. While used extensively for home kitchen ventilation, it is adaptable for other applications, such as in school toilets, therapy and treatment rooms in hospitals, dental laboratories, ticket booths, offices and x-ray and photographic dark rooms.

This is a "unit package" assembly, complete with ceiling grille and necessary electrical connections. It is *Underwriters*, approved and listed. It carries a (pro-rata) five year guarantee. The installation requires only the application of a discharge duct of the same dimension as the outlet collar and the optional addition of an automatic shutter for the end of the duct.

The assembly is inherently quiet and is installed rigidly in the structure without need for resilient mountings or flexible duct connections. The blower is normally quiet at its maximum speed. The motor and wheel unit is removed easily without tools through the ceiling inlet opening. The patented construction of the Clipper entirely isolates the motor from the air stream. This special feature keeps the motor free from contaminated air, adding to its service life.



LEFT: Blower unit and grille in all Trade-Wind models are easily removed without tools. Unit shown is Model 1501.

RIGHT: Model 3501 installs in a cabinet over the stove. Inlets, equipped with washable filters, are provided in base of unit and at ceiling and fold-under hood is optional.

Selection Chart and Specifications
TRADE-WIND CLIPPER VENTILATORS—115 Volt, 60 Cycles, A.C.

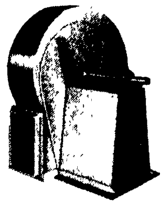
Cat No.	Description	Type Blower	Net Air CFM	Recomm. Max. Room Size Cu Ft	Duct Size	Motor Watts
1201 Clipper	Horizontal discharge complete with grille.	Single Wheel	100	Bathrooms Only	4″ Round	60
1501 Clipper	Interchangeable horizontal or vertical discharge, complete with grille.	Two Wheels	300	1000	10″x3½″	90
2501 Clipper	Interchangeable horizontal or vertical discharge complete with grille, 2-speed motor & switch.	Two Wheels	425	2000	10″x3½″	145
3501 Super Clipper	Cabinet Installation Includes housing, 2 filters, 2-speed motor and switch; stainless steel hood optional.	Two Wheels	550	3000	13″x4″	185

Western Blower Company

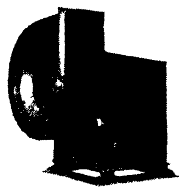
Main Office and Plant: 1800 Airport Way, **Seattle 4, Washington**

Sales Offices in the Principal Cities West of Rocky Mountains

Turbine Multiblade Fans



Mill Exhausters



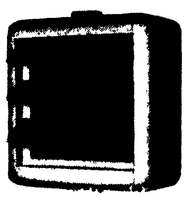
Centrifugal Exhaust Fans

Turbine Multiblade Fans—forward curved blade, Type TR, or backward curved blade, Type "S," fans for heating, ventilating, mechanical draft, etc. Bulletins No. 30 and 31.

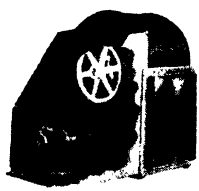
Mill Exhausters—single or double for material conveying exhaust systems, direct connected or V-belt driven. Bulletin No. 32-3.

Pulley Driven Utility Sets—V-belt driven, slow speed, quiet operating, for general utility duct ventilating systems. Bulletin No. 31.

RB Volume And Pressure Fans



Western Unit Heaters



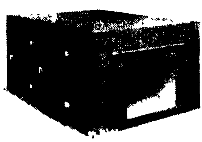
Pulley Driven Utility Sets

RB Volume And Pressure Fans—radial blade type either direct connected or V-belt driven for ventilating and conveying applications. Bulletin No. 39.

Western Unit Heaters—vertical or horizontal, for general heating and drying applications. Bulletin No. 53.

Centrifugal Exhaust Fans Series 59—for general exhaust systems. Complete packaged units. Bulletin No. 59.

Air Washers



Volume Heaters

Spirovane Propeller Fans

Air Washers—for cleaning, cooling, humidifying and dehumidifying. Bulletin No. 30.

Volume Heaters—with one or more centrifugal fans for heating, ventilating and air conditioning. Available in vertical or horizontal cabinet units. Bulletin No. 54.

Spirovane Propeller Fans—furnished either direct connected or V-belt driven for commercial or industrial ventilation. Bulletin No. 50.

Olympic Heat Exchangers—Converters, Side Arm Heaters, Immersion Heaters, Oil Heaters, and Condensate Coolers.

Bulletins as listed above furnished upon request.

Westinghouse Electric Corporation
Sturtevant Division

Air Conditioning, Heating, Ventilating, Dust Control and Fume Removal Equipment, Electronic Air Cleaners, Compressors, Mechanical Draft Equipment

Hyde Park | Offices in Principal Cities | Boston 36, Mass.

CENTRIFUGAL FANS
Silventvane (R) centrifugal fans are highly effective in both commercial and industrial service. Available in capacities ranging from 500 to 480,000 cfm volume —total pressures up to 6¾ in.

AXIFLO FANS
Especially designed to operate against resistance. Either vertical or horizontal air flow. Widely used in industrial heating, ventilating and fume removal.

VENTILATING SETS
Rexvane (R) ventilating sets are direct-connected general purpose blowers or exhausters made in 7 sizes up to 4000 cfm. V-Belt ventilating sets are self-contained units consisting of a centrifugal fan and motor with V-belt drive. Enclosing cover available. Air deliveries from 500 to 14,000 cfm.

HEAT TRANSFER SURFACE
Sturtevant cooling and heating coils are available for Freon, chilled or hot water, standard steam and steam distributing types.

AIR BLENDERS
For both winter and summer conditioning of individual rooms-mixes conditioned air from a central system with recirculated air.

AIR HANDLING UNITS
For year-round air conditioning when connected with cooling media in summer and source of steam or hot water in winter. Units available for heating and ventilating, or for ventilating only. Available in horizontal or vertical type, with or without coil sections. Sizes from 1200 cfm to 18,000 cfm.

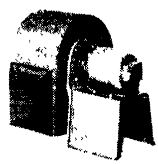
INDUSTRIAL HEATERS
Designed for large area, severe heating jobs using steam, or hot water. Capacities range from 77,700 to 1,270,000 Btu/hr and 1420 to 17,600 cfm.

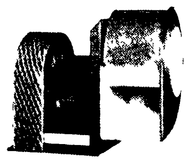
SURFACE DEHUMIDIFIERS
For central plant air conditioning, these sprayed coil units are available with chilled water or direct expansion coils. Capacities range from 5 to 115 tons of refrigeration with 1600 to 45,000 cfm.

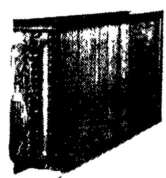
AIR WASHERS
For evaporative cooling, humidifying, dehumidifying and cleaning—four types available in capacities from 4500 to 110,000 cfm.

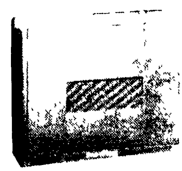

Silentvane Centrifugal Fan Design 10


Elbow Axiflo Fan Design 2AE

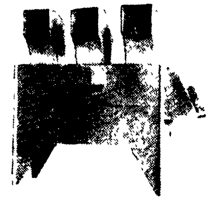

Rexvane Ventilating Set

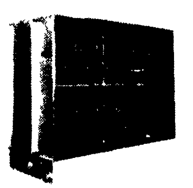

V-Belt Ventilating Set

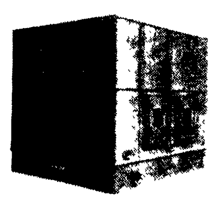

Evaporator Coil Direct Expansion Type EA


Air Blender


Air Handling Unit Type AH


Industrial Heater


Surface Dehumidifier

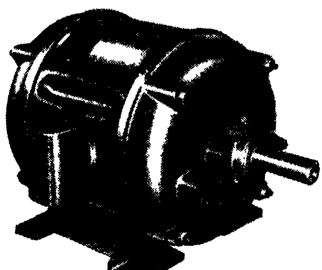

Air Washer, Type C

Wagner Electric Corporation
EST. 1891

6464 Plymouth Avenue
St. Louis 14, Mo., U.S.A.

Sales Offices in 32 Principal Cities

WAGNER MOTORS for Heating, Ventilating and Air Conditioning Equipment



SINGLE-PHASE MOTORS

Wagner open-type 40 C general purpose single-phase motors are built in standard models with sleeve or ball bearings and rigid bases. Resilient bases are available on fractional hp and small integral-horsepower ratings, and face and flange type endplates can be furnished to meet specific requirements. Thermal protectors can be supplied on ratings through $1\frac{1}{2}$ hp.

Type RA—Repulsion-Start Induction

High starting torque, low starting current. $\frac{1}{2}$ to 15 hp, all standard voltages and frequencies.

Type RK—Capacitor-Start Induction

High starting torque, normal starting current. $\frac{1}{6}$ to 3 hp, all standard voltages and frequencies.

FAN DUTY MOTORS

Type TM—Shaded-pole, low starting torque, for shaft mounted propellor fans which draw air over the motor. Totally enclosed non-ventilated, no temperature rating, with sleeve bearings, round frame or rigid or resilient base. Ratings 1/125, 1/80, 1/40, 1/30 and 1/20 hp, 115 or 230 volts, 60 or 50 cycles. Three-speed reactor controllers can be supplied if specified. A companion line of direct current fan-duty motors is also available.

Type TZ—Permanent Split-Capacitor

Totally enclosed non-ventilated, 55 C, with sleeve bearings and resilient base or rubber rings for mounting. Ratings 1/20 to 3/4 hp, constant speed, two speed or adjustable speed, standard voltages and frequencies.

Split-phase, squirrel-cage and direct current 55 C, totally enclosed non-ventilated fan-duty motors are also available.

POLYPHASE SQUIRREL-CAGE MOTORS

Wagner open-type 40 C single-speed polyphase squirrel-cage motors are built with sleeve or ball bearings and rigid bases. Multispeed motors, vertical and flange mounted motors, and splash-proof and totally enclosed motors are also available.

Type RP-1—Normal Torque Normal Slip

Ratings 1/6 to 400 hp, 3 or 2 phase, all standard voltages and frequencies.

Type RP-5—High Torque High Slip

Ratings $1\frac{1}{2}$ to 200 hp, 3 or 2 phase, all standard voltages and frequencies.

Part-Winding Increment Type Motor-and-Starter Combination

A low cost combination of a squirrel-cage motor wound with two parallel star circuits and a magnetically or manually operated two step increment type starter, which limits the inrush of current at starting. Available with normal or high torque open or enclosed motors.

Write For These Bulletins

MU-185—Describes and illustrates all types of Wagner motors.

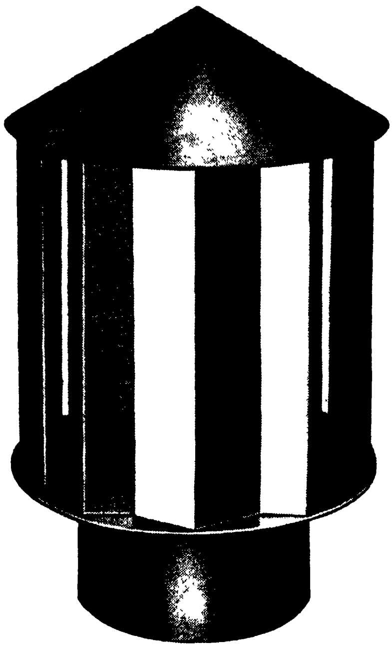
MU-40—Lists part numbers and prices of Wagner Motor repair parts.

G. C. Breidert Co.

3129 San Fernando Road, **Los Angeles 65, Calif.**

Representatives Located in Principal Cities of the U. S.

BREIDERT AIR-X-HAUSTERS FOR ROOF VENTILATING, VENT FLUES & CHIMNEY TOPS



Patent No. 2269428

HOW YOU CAN GET BETTER RESULTS WITH LESS MONEY

WHEN YOU CONSIDER VENTILATORS, ASK FOR PROOF OF PERFORMANCE . . .

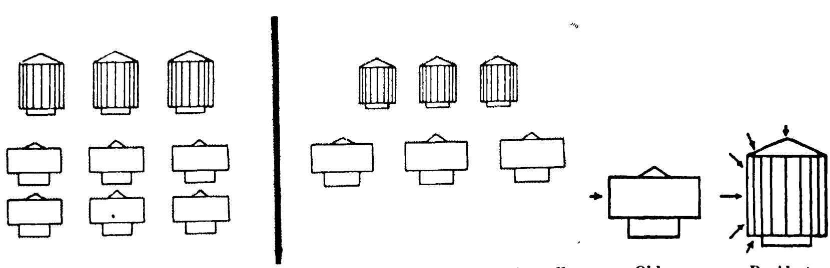
Ventilators are usually selected on the basis of their capacity ratings. That makes sense, but do you know that:

1. Capacity ratings of most ventilators are not certified.
2. Capacity ratings of most ventilators are based on horizontal wind tests only.

Breidert Air-X-Hausters provide safe, sure ventilation no matter which way the wind blows!

and . . . Breidert Air-X-Hausters pioneered with published certified capacity ratings based on tests* made with wind blowing in all directions as shown below. Only such tests can guarantee the capacities a ventilator will deliver under actual operating conditions. No matter which way the wind blows, barring interior negative pressures, the Breidert provides safe, sure ventilation . . . on roofs, vent flues, chimney tops. Stationary, no moving parts, nothing to jam or get out of order.

Briedert Air-X-Hausters are ideal for Kitchen Ventilation, for Vent Flue Caps, for Chimney Tops. They also are widely used on all types of factories, commercial buildings, and residences.



It takes fewer Breiderts to do the job as same size conventional ventilators. **(or)** *Use the same number of smaller size Breiderts to do the same job as conventional ventilators.*

Old Method ***Breidert Method***

Write for Free Engineering Data Book . . . contains specifications and installation data, certified capacity ratings, etc. Address Dept. HV.

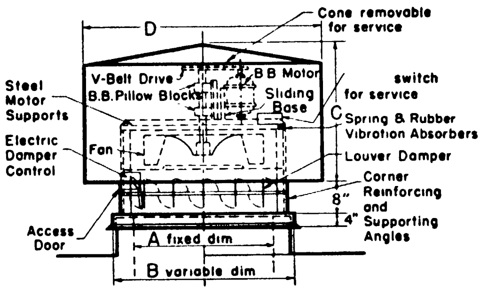
* **By Pittsburgh Testing Laboratories**

Hirschman-Pohle Co., Inc.

200 Lent Ave. . . . **LeRoy, N. Y.**

Sales Representatives in Principal Cities

The STATICK Power ventilator has been especially designed as pressure exhauster for roof mounting, and its use conserves much valuable space within the building in addition to reducing installation costs without sacrifice of efficiency. It uses the conventional type of backward curved blade fan wheel that will not overload motor at any static pressure, and it provides a factor of safety for possible later changes in duct system.



The low speeds possible with this type of fan, plus solid construction of the entire unit and the vibration dampeners used, permit the quiet operation so desirable for many types of installations. Because of the flexibility of belt drives, each size unit has a wide range of capacities at static pressures within the limits of the motor horsepower. Standard capacity tables or suggestions to meet your requirements available on request.

It permits room for dampers as an integral part of the ventilator with means of access to the dampers and their operators, also permits of fitting to other than a standard square curb where building construction makes such odd shapes desirable or necessary.

The entire fan and motor assembly is mounted on rugged welded steel angle frame by means of spring and rubber vibration absorbers. Motors used are of standard manufacture designed for vertical operation, ball-bearing, continuous duty. Our standard design is such that the motor compartment is permitted to receive free ventilation for motor cooling purposes, but this compartment can be isolated entirely from the exhausted air where injurious fumes are being exhausted. The exhaust cowl itself has been designed to permit unrestricted air outlet and to present a pleasing appearance. Either the top cone or the entire cowl are removable for servicing.

These units can be furnished of galvanized steel, aluminum, copper or other available metals, with or without dampers, as desired. Dampers included can be of the self-acting type or for chain, electric or pneumatic control.

The Hirschman Type "F" Electric Ventilator (not shown) uses a highly efficient propeller type of fan designed to exhaust free air or at the lower static pressures, also permitting high gravity exhaust during periods when fan operation is considered unnecessary.

It can be furnished with any type or size of base connection desired, with or without any type of damper for any type of control. For extreme heat conditions or where injurious fumes are to be exhausted, the motor can be isolated from the path of the exhausted air by our Isolated motor section, either single or double shell.

Motors used are of standard manufacture designed for vertical operation, fully enclosed, ball-bearing, continuous duty.

Data for any capacity will be gladly furnished on application.

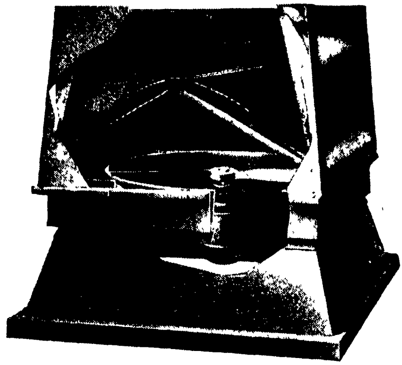
We also manufacture a complete line of rotary and stationary gravity ventilators for any type of application.

Iron Lung Ventilator Company

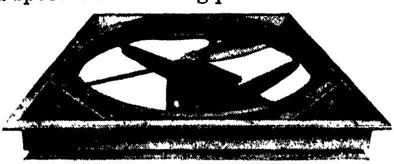
4013 Prospect Avenue
Cleveland 3, Ohio

Manufacturers of IRON LUNG Roof Ventilators and Air Intake Units

Hi Velocity Discharge exhausts polluted air so high above the roof that impurities cannot return to building through open doors and windows.
All IRON LUNG roof ventilators are equipped with Airfoil Propeller Axial Flow Fans having air delivery ratings certified in accordance with the Standard Test Code for Axial Fans endorsed by the *PFMA* and the ASHVE.
Without cost or obligation our experienced engineers will cooperate in planning efficient ventilation for any type or size of industrial building. Write for Catalog listing all sizes with complete specifications. Data sheet will be included to make it convenient to explain a specific ventilating problem.



"Cut-a-way View of IRON LUNG"



"IRON LUNG Chassis, Assembled"

Left: IRON LUNG chassis, a sturdy all-welded unit that supports the fan, fan ring, and motor—one of the really great improvements to electrically operated roof ventilators. Adds strength to the entire unit—absorbs vibrations.

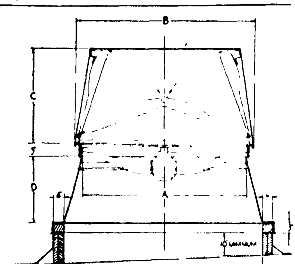
IRON LUNG RATINGS

Cat. No.	Size	RATING CFM	HP	RPM	Approx. Net Wt.	Approx. Shpg. Wt.
			110/220 Volt, 1 Phase, 60 Cycle			
300	20"	3720	¼	1725	175 lbs.	250 lbs.
340	24"	5200	¼	1725	209 lbs.	316 lbs.
			220/440 (or 550) Volt, 3 Phase, 60 Cycle			
341	24"	6510	½	1725	233 lbs.	356 lbs
348	30"	8880	½	1725	288 lbs.	455 lbs.
351	30"	10900	1	1725	300 lbs.	465 lbs.
358	36"	13500	1	1725	390 lbs.	565 lbs.
370	42"	21150	2	1140	613 lbs.	808 lbs.
376	48"	28800	3	1140	773 lbs.	983 lbs.
383	60"	40000	3	1140	875 lbs.	1175 lbs.

IRON LUNG DIMENSIONS

All Dimensions are for Units of 5 HP or Less

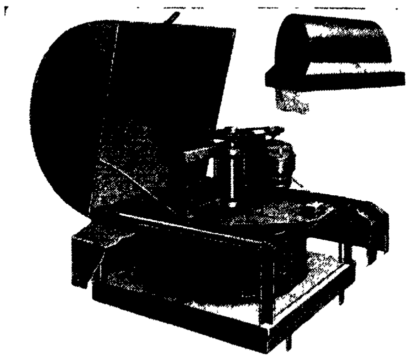
Size of Iron Lung "A"	20"	24"	30"	36"	42"	48"	60"
B	27"	31"	37"	43"	49"	55"	67"
C	14¾"	17"	20⅛"	23½	26⅞	30"	36⅝"
D	15"	15"	15"	19"	19"	22"	22"
E	30"	34"	40	48"	54"	62"	74"
Overall Curb	37"	41"	47"	55"	61"	69"	81"



The Swartwout Company

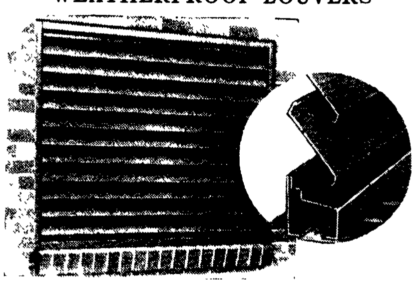
18511 Euclid Avenue, **Representatives in Principal Cities** **Cleveland 12, Ohio**

DUCT EXHAUST VENTILATOR



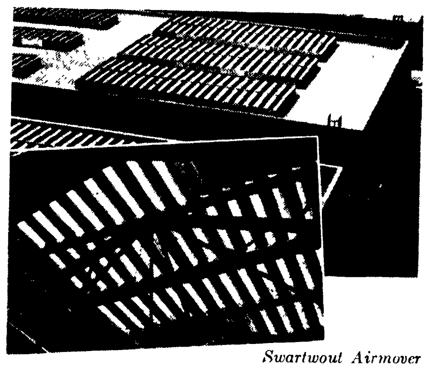
The Airlift is a centrifugal fan unit for duct exhaust against static pressure. Operates at very low noise levels. Fan has backwardly curved blades completely non-overloading; streamlined inlet close-coupled to duct throat increases efficiency, avoids turbulence; 14 sizes, 49 capacity variations.

WEATHERPROOF LOUVERS



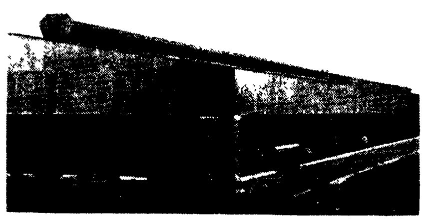
Swartwout Air louvers are fabricated to order in a wide selection of sizes up to 74¾ in. high by 48 in wide per unit. Provide adjustable or fixed weatherproof louver equipment to fit wall opening desired. Large openings fitted with bank of units of equal size for best appearance. Optional operating methods; optional anchoring methods.

GRAVITY ROOF VENTILATORS



Swartwout Airmover

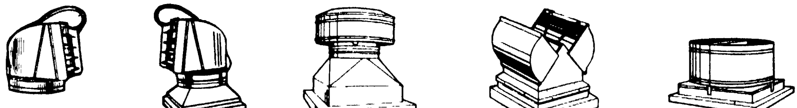
Compact multiple opening type featuring very short air travel, built in units 10 ft x 7 ft 6 in. x 32 in. high; 30 sq ft of opening per unit. Used as single units, or single or multiple width runs. Unusual large scale ventilation results possible by roof coverage such as illustrated. Can be adapted to any type of roof. Insert shows "open roof" effect.



Swartwout-Dexter Heat Valves

Continuous opening natural draft ventilator particularly effective for ridge of peaked roof, saw-tooth construction or skylights; adaptable to flat or slant roofs. Made in throat opening sizes 4, 6, 9, 12, 15, 18, 24, 30, 36 and 42 in. Ten foot units. Adjustable damper. Weatherproof.

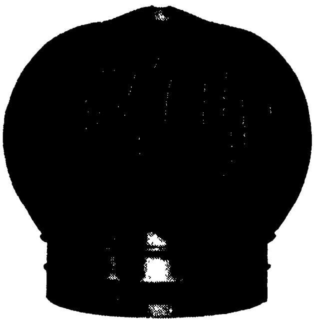
OTHER SWARTWOUT ROOF VENTILATOR TYPES



ROTARY, gravity *AIRJECTOR, powered* *VALVENT, gravity and powered* *JECT-O-VALVE, powered* *WHIRLOUT, powered*

See ***Sweets Architectural or Engineering File*** **or send for Swartwout general catalog**

HIGH EFFICIENCY ROOF VENTILATORS



Western Rotary

Efficiency: Extremely high exhaust capacity due to "backward curved blower wheel" type mechanism. Momentum operates the Western Rotary Ventilator between wind peaks when any stationary ventilator is inoperative.

Capacities certified by a foremost university laboratory. (See results in our catalog.)

CERTIFIED CAPACITY RATINGS

(See our complete catalog)

Bulk, height, weight and cost are below other types yielding comparable capacities. Weight is important when designing roof loading.

Appearance: Modern functional design adds beauty. No guy wires needed.

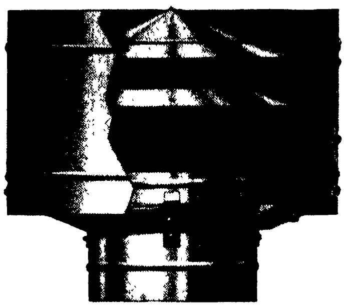
Guarantee: Bearings are guaranteed for ventilator life

Sizes: Available from stock in sizes 6 in. through 48 in.

Westernaire Curb-Mounted Fan

May be used with any ventilator of same throat size to increase exhaust volume. High speed-High velocity or Slow Speed-QUIET Operation.

For fumes, smokes, dust or excessive heat.



Forbes Syphonaire

Efficiency: Exhausts high volume. Will not backdraft. Tests of a foremost university laboratory show the outstanding capabilities. Openings exposed to wind will lower efficiency in a stationary ventilator. The Forbes Syphonaire has a solid 360 degree wind-band to prevent this possibility.

CERTIFIED CAPACITY RATINGS

(See our complete catalog in Sweet's)

Bulk: Low design, light in weight.

Appearance: Modern, attractive, no guy wires needed.

Projects: Used for many Government projects.

Sizes: Fabricated from galvanized iron. Also available in aluminum and copper or stainless steel. Sizes from 6 in. through 48 in.

SEE OUR COMPLETE CATALOG IN SWEET'S ARCHITECTURAL FILE and ARCHITECTS AND ENGINEERS CATALOG FILE or write to above address for additional catalog and list of Representatives.

Air Devices, Inc.

Air Diffusers ● Exhausters ● Air Filters
Filter Holding Frames ● Hot Water Generators

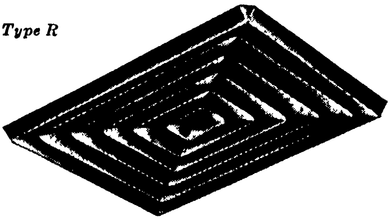
17 East 42nd St.
New York 17, N. Y.

AGITAIR PRODUCTS

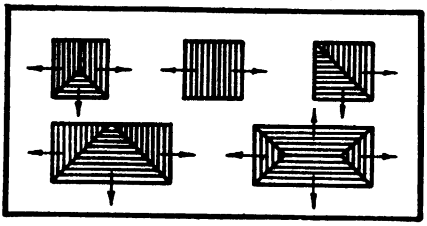
Agents in All Principal Cities

AGITAIR DIFFUSERS

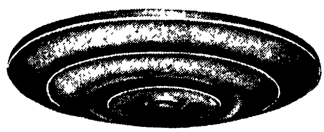
Type R



Square or Rectangular in Shape

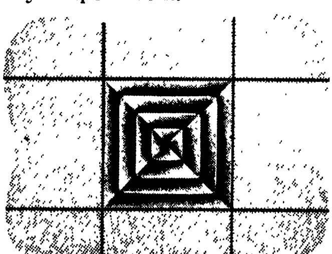


Type A



Type CSF

Type R AGITAIR air diffusers, are square or rectangular in shape with patented built-in diffusing vanes which can be assembled in a variety of arrangements to divide and discharge the air in proportion to the area to be served, in one—two—three or four directions without the use of baffles or blank-offs. Each side delivers a quantity of air proportional to the areas served. These AGITAIR diffusers can be installed in any location in the ceiling or sidewall and will perform efficiently, diffuse the air quietly, draftlessly, and with rapid temperature equalization throughout any shaped room.

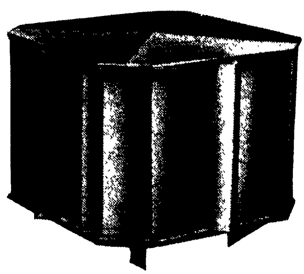


ACOUSTICAL CEILINGS

The AGITAIR **Type RTC** square or rectangular in shape is especially designed for installation in acoustical ceilings. Made in sizes to conform to standard tile dimensions.

Circular AGITAIRS combine beautiful design with finest operating features to give rapid temperature equalization and draftless diffusion of air. Model CSF is quiet, easy to install. Pressure losses at minimum. Type A diffusers are smaller, weigh less, are easy to install. Type CM is for marine use. In all sizes for all types of mounting and with lighting combinations. Dampers are provided where needed.

The AGITAIR Diffuser Data Book, available to architects and engineers, will help you design and install air distribution systems. Consult our engineers.



AGITAIR WIND-ACTUATED EXHAUSTERS

Provide proper ventilation regardless of wind direction, and with positive elimination of down-draft. Functions at peak efficiency at average low wind velocities. Will not restrict the flow of air or gases when there is no movement of outdoor air across the head.

Fan-equipped units also for **higher ratings.**

Air Control Products, Inc.

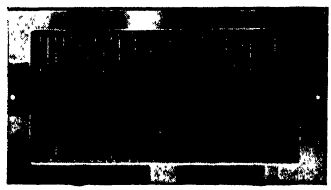
Coopersville, Michigan

Air Conditioning Registers, Grilles, Ceiling Diffusers and Leigh Building Products.

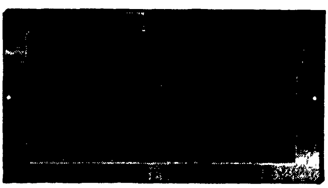
NO. 10 SERIES REGISTERS AND GRILLES. A truly high value line that gives you top performance and appearance. No. 10 Series Registers provide adjustable, dual control of the air stream. These Registers are made for side-wall or base-board installation. They are available in a wide variety of standard sizes.

NO. 10 SERIES REGISTERS are equipped with the famous Push Button operator that opens or closes at a touch of the finger. Air flow valve may be locked to any up or down deflection.

A complete line of matching Grilles are styled to go with the No. 10 Registers.

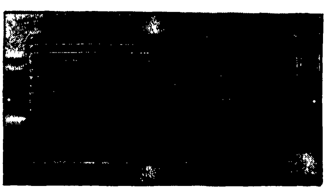


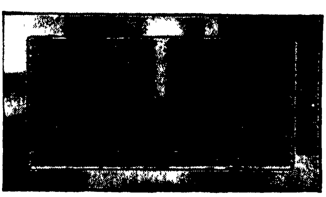
NO. 110 SERIES REGISTER



NO. 110 SERIES REGISTERS have vertical adjustable fins for use where vertical air control is not needed. Single shutter type damper. Operator sets close to face yet holds damper in a position.
New Adjusto-Stop permits use of damper as a balancing damper.

NO. 210 REGISTERS GRILLES



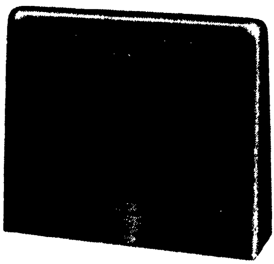


NO. 210 SERIES REGISTERS have adjustable horizontal fins. Horizontal fins are used where a vertical control of the air stream is needed. A single shutter damper is used to open or close the Register. Grilles are designed to match Registers. Like the No. 10 and No. 110 Series, they are available in sidewall and baseboard design. All of the above Registers and Grilles are painted in beautiful beige prime coat.

Installation frames to fit these Registers are available in a complete range of sizes.

No. 61, 61 and 62 return air intakes are built with the same type of adjustable fins.

Write for New Air Control 4 Color 52 a.c. Catalog.

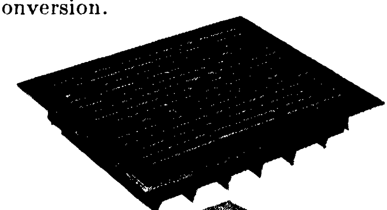


NO. 50 SERIES BASEBOARD REGISTERS. A beautiful gravity type Register that blends with any room interior. Removable face makes streak proof installation easy. Outer shell stamped from heavy gage metal. No welds to break loose. Balanced damper holds damper in any position.

NO. 80 SIDE WALL GRAVITY REGISTERS are styled to match with baseboard registers. Adaptable for side wall installation above the base board. Equipped with balanced dampers. Fins on gravity Registers can be adjusted for forced air conversion.

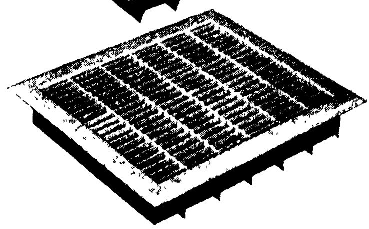
NO. 30 FLOOR REGISTERS—Unit-Grid Construction. A strong rugged low priced Register. Face is stamped from one piece of heavy gage metal. Dial operator easily opens or closes the valve with a touch of the foot.

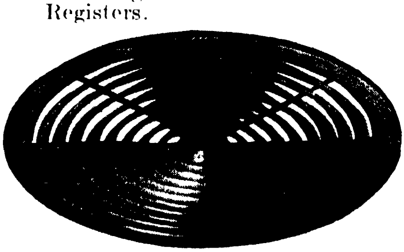
No. 31 Return Air Face is the same Unit-Grid Construction as No. 31 Registers.



NO. 40 FLOOR REGISTERS feature famous **Rigid-Lock** type construction. Each fret is locked to each crossing fret and to the margin. Dial-operator valves run the short way of the face. Medium mesh ($7/16$ in.) between frets.

No. 41 Floor Return Air Faces are same Rigid-Lock construction as above Registers.





AIR CONTROL DIFFUSERS

Built in two styles—Flush Type or S.D. stepped down type. Air flow ring design gives ideal air distribution. Both styles are made in seven sizes—6 in. to 22 in. A low cost unit that does the job right.

INSTALLATION RINGS

Installation Rings provide a firm base for fastening diffuser in place. Diffuser screws to ears on side of ring. Made in a complete range of sizes. **DROP RINGS**—Use where it is desirable to drop diffuser down from ceiling. 2½ in. deep. Rubber gasket on top edge for tight seal. Sizes from 6 in. to 22 in.

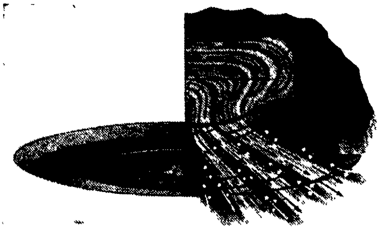
CEILING DIFFUSERS DAMPERS. A low cost easy method of regulating the air issuing from the diffuser. Use either in the end of a round pipe or with a horizontal duct. Damper operates with chain that comes through the center of the diffuser.

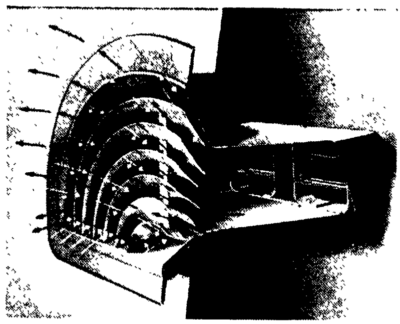
Write for New 4 Color 52 a.c. Catalog—complete engineering data. Also information on Vision-Proof Grilles.

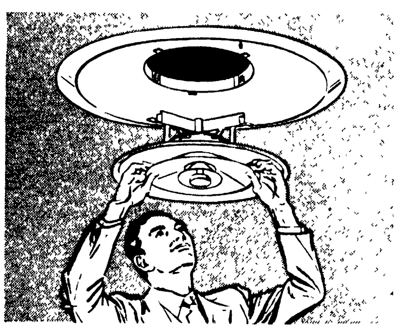
Anemostat Corporation of America

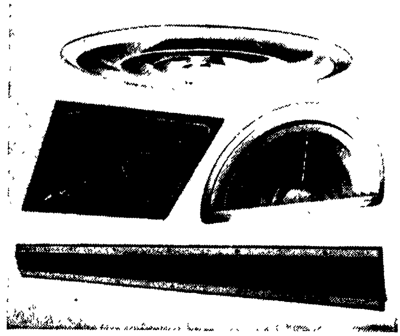
10 East 39th Street **ANEMOSTAT** Representatives in

New York 16, N. Y. **DRAFTLESS** *Aspirating* **AIR DIFFUSERS** Principal Cities









PRINCIPLE OF ASPIRATION

The Anemostat Air Diffuser splits the supply air into multiple, separate air streams and at the same time creates low pressure areas in certain parts of diverging passageways. This causes room air to be siphoned into the device where it is thoroughly mixed with supply air. The mixture is discharged in a series of expanding turbulent air layers and, therefore, readily entrains a large amount of additional room air.

As a result, temperature and humidity are equalized throughout the room and a continuous air motion in the occupancy zone within low acceptable velocity limits is established. Stagnant air pockets are also avoided.

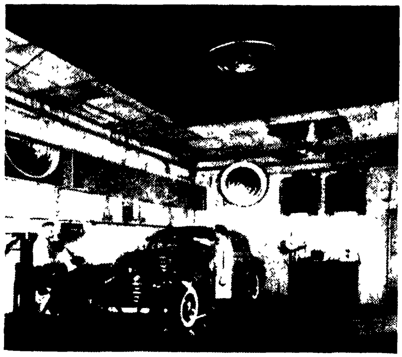
The quantity of room air drawn into Anemostat Air Diffusers depends on the specific design, size and type of the various units and is equal to as much as 35 per cent of the supply air. This effective aspiration distinguishes Anemostat Air Diffusers from all other air outlets.

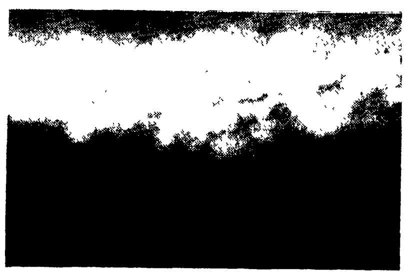
SMALLER DUCTS

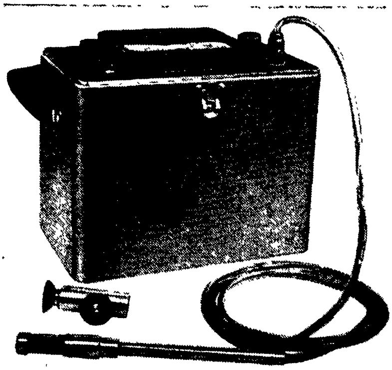
Aspiration permits the use of greater temperature differentials and higher supply air velocities than customary, which results in savings in the initial cost of fans and ducts and in the operating cost of the system. Duct layouts may be simplified because Anemostat Air Diffusers distribute air evenly in spite of unusual room plans, columns or other obstructions.

THE COMPLETE LINE

Anemostat Corporation of America has developed air distribution devices and related equipment for every purpose. Circular, semi-circular, square and straightline Anemostat Air Diffusers of various types and sizes for ceiling and wall distribution are available for comfort conditioning and industrial heating, ventilating and air conditioning installations. Units are also available for high pressure systems, handling velocities up to **5000 fpm and 6 inches of static pressure.**







QUALITY

Anemostat Air Diffusers are scientifically designed according to modern fluid flow theory and manufactured according to modern production standards.

ANEMOSTAT PATENTS

The Anemostat Corporation of America holds over fifty patents covering the use and design of diffusers in the field of air distribution. The superior functioning of the Anemostat is protected by these patents which are the result of inventive spirit, diligent research and constructive engineering.

RESEARCH AND ENGINEERING

Anemostat Corporation of America maintains a large, well equipped laboratory manned by experienced scientific and technical personnel for testing purposes and the development of new products. Engineers from all over the world have visited the Anemostat laboratory to witness demonstrations of Anemostat products and their applications to any conceivable problem pertaining to air diffusion. Anemostat has spent over a million dollars in research and engineering and will continue to make substantial contributions to the progress of the science of air distribution.

EXPERIENCE

Over a million Anemostat Air Diffusers are now doing an excellent job in thousands of comfort conditioning and industrial heating, ventilating and air conditioning installations. In addition to applications in commercial buildings, industrial plants, hotels, stores, hospitals, theatres, restaurants and homes, Anemostat Air Diffusers are used in military and commercial aircraft, railroad cars, buses and ships.

SPECIAL INSTRUMENTATION

Anemostat engineers have developed various instruments for the accurate and convenient testing of the performance of air outlets. The Anemotherm Air Meter, a self-contained, compact, portable instrument, facilitates the balancing and checking of air distribution systems by engineers and contractors. It is now commercially available and is being used with great success by engineers and contractors throughout the country.

The Auer Register Co.

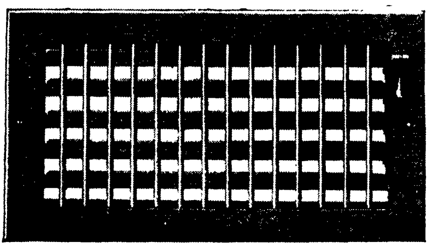
6600 Clement Avenue, **Cleveland 5, Ohio**

Manufacturers of Registers and Grilles for Gravity and Air Conditioning Systems; Metal Grilles for Radiator Enclosure, Ventilation, Concealment

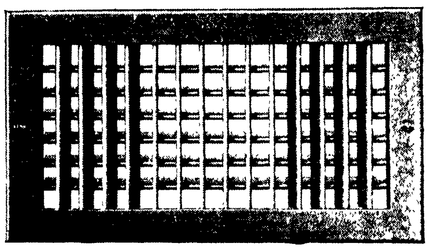
AIR CONDITIONING REGISTERS AND GRILLES

Auer has a complete line for warm air or air conditioning, a wide choice of styles for every purpose. For gravity systems, the Heat-Rite is a quality model, adjustable for up-or-down flow. DuraBilt floor registers and intakes provide extra strength with interlocked cross-bar construction. "Streamliner" registers and grilles for high velocity outlets are made in 8 styles, with single bank of adjustable bars, vertical or horizontal, also with double bank, front vertical and rear horizontal (or the reverse), also all these types with the addition of multi-louvre valves. Auer Register Book showing entire line sent on request.

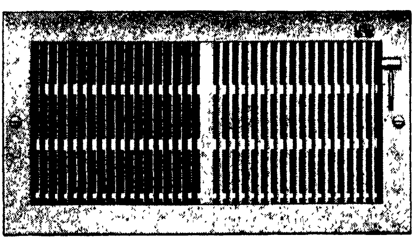
Auer perforated grilles are made in many designs and sizes, in steel (or stainless), aluminum, brass or bronze. Finished or plated as desired. Grille Catalog. "G" gives full scale details, tables of openings and free areas.



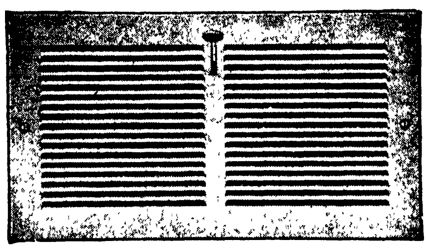
Streamliner Register No. 1005V-HML. Adjustable bars, multi-louvre valve.



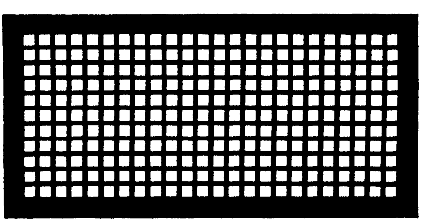
Streamliner No. 1205VH. Double deflection grille, adjustable bars.



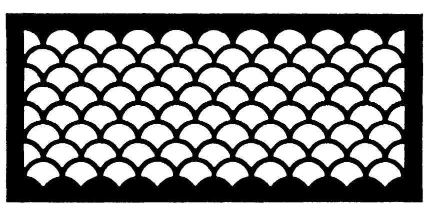
Airo-Flex No. 4432 Register—Multi-louvres adjustable up, straight or down. Grille bars adjustable for right or left flow. Grille to match.



Airo-Flex No. 7032 Register—Grille bars set to direct air downward at 22½ deg, but adjustable for other angles. Single louvre. Grille to match.



5A—Square Lattice Grille



32A—Shell Grille

Barber-Colman Company

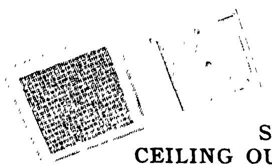
ENGINEERED AIR DISTRIBUTION OUTLETS

BARBER COLMAN

AUTOMATIC TEMPERATURE CONTROL SYSTEMS

Rockford, Illinois

uni-flo



SQUARE CEILING OUTLETS

Completely new air distribution principle provides flexibility in wide range of air pattern adjustment. Units are sized to match acoustical tile and are inconspicuous in design to blend with any type of ceiling. Ruggedness and simplicity facilitate installation.

line-o-flo

LINE TYPE CEILING OUTLETS

A new idea in air distribution, matching modern architectural concepts and continuity of design. Linear panel units can be used singly, end to end, or in patterns. Also furnished for combination with M21118 Day-Bright fluorescent lighting unit.

DOUBLE DEFLECTION GRILLES

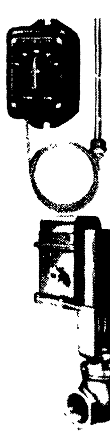
Greater versatility from simple units is obtained by adjustable vertical fins and removable core. High aspiration efficiency, rapid diffusion, maximum temperature differential, minimum pressure drop, and *guaranteed performance.*

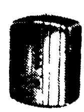
venturi-flo

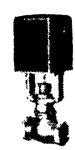
CEILING OUTLETS

Provide uniform peripheral air distribution with quiet operation and high diffusion efficiency. Neat in appearance and made in several types and a full range of sizes and capacities. Adaptable to all kinds of ceiling systems.

AUTOMATIC CONTROLS







Automatic Temperature Control Systems are available for all types and sizes of heating, ventilating, and air conditioning installations. These systems are composed of selected, skillfully-engineered combinations of various types of thermostats, Motor-Operated Valves, and Control Motors. Specialized Barber-Colman control units include the Micro System for non-hunting proportioning control, the Econostat for indoor-outdoor zone control, a line of self-contained Temperature Regulators, a series of Dual Bulb Thermostats, an Adjustable Ratio Control, a Thermostatic Adjuster, and others. All units of Barber-Colman design and manufacture are ruggedly built for accurate operation, long life, maximum performance, and minimum maintenance.

W. B. CONNOR ENGINEERING CORP.

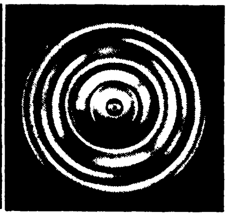
Danbury, Conn. — **Representatives in All Principal Cities**

In Canada: Douglas Engineering Co., Ltd., Montreal, P. Q.

KNO-DRAFT *Adjustable* AIR DIFFUSERS

CONTROLLED AIR DIFFUSION

Type KDA for supply air. Aluminum, sizes 4 to 36 in. neck dia. Capacities 50 to 15,000 cfm per unit.



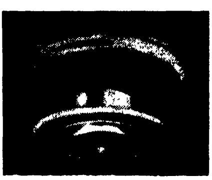
Type SRD for combination supply and return air. Sizes 6 to 24 in. supply neck dia. Capacities 50 to 2,500 cfm per unit, return neck area 75% of supply.

Kno-Draft Adjustable Air Diffusers are designed to give accurate control of air distribution, plus installation and operation economies. With air direction and air volume adjustments on each diffuser, "custom-made" air patterns can be created which will insure draftless diffusion and equalized temperatures for comfort conditioning or specific patterns for industrial processes.

Installation, balancing and inspection are fast because of features like the Type HD quick-opening set-lock assembly, the self-contained inner unit and the sleeve-type damper.

System design problems are eased because Kno-Draft Diffusers are adjustable *after installation*. The often difficult and hazardous job of figuring everything about the air movement in advance is eliminated. And the air pattern in an area can be changed with the seasons or when processes, people or partitions are relocated. Kno-Draft Diffusers are geometrically proportional, size for size, insuring like resistance at like neck velocity for any size—a considerable advantage when selecting various size diffusers for a common system.

Designed for high or low ceilings or attachments to exposed ducts, these diffusers will effectively distribute large volumes of air and pre-mix room and supply air. They permit the use of high duct velocities—resulting in smaller ducts and lower costs. Duct designs are simplified.

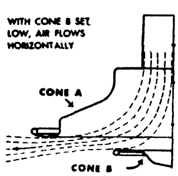


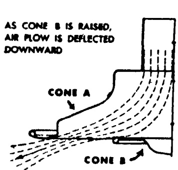
Diffuser and Cone separated

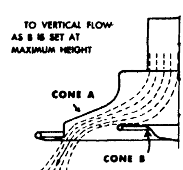
Diffuser and Cone assembled

The simple, attractive design of the Kno-Draft Diffusers enables them to blend with either period or modern interiors. In their original aluminum, they create an interesting and unobtrusive decorative accent. Painted to match the ceiling, they become self-effacing.

Anti-Smudge Cone: Where exceptionally sooty or dusty air conditions are expected or where rough-textured ceilings are employed, the use of this accessory cone is recommended. It furnishes the additional control needed to provide the precise air separation which inhibits smudging.



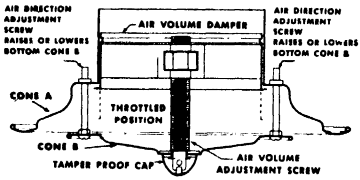




Kno-Draft Diffusers are covered by U.S. Patents Nos. 2,365,367; 2,369,119; 2,432,289 and others pending; Canadian Patents Nos. 429,206; 443,236.

Any angle of air discharge needed to suit ceiling heights and heating, ventilating or cooling air patterns can be obtained by raising or lowering bottom Cone B.

TypeD Air Volume Control operates independently of the air directional adjustment. It varies only the quantity, not the characteristic of the air distribution. It consists of a cylindrical, sliding, sleeve-type damper connected by a specially designed spider



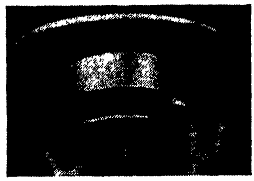


Fig. 1 Self-Contained Inner Unit

Fig. 2 Type HD Set-Lock Assembly

to a centrally operated screw. The shank of the screw extends through the lower cone of the diffuser and is concealed by a tamper-proof cap. With this damper on each unit, a series of diffusers can be quickly balanced.

Kno-Draft Features Speed Installation

The development of the self-contained, removable inner assembly alone (see Fig. 1) reduced installation time as much as 50 per cent. The addition of the Type HD set-lock assembly (see Fig. 2) reduced the time for that part of the installation to a matter of minutes. No tools are required. The B cone or inner element of the diffuser is secured to the combined suspension and adjustment screws by a springloaded catch

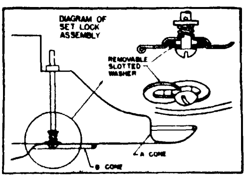


Fig. 3 Now Standard Equipment

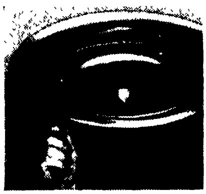


Fig. 4 Air Direction Adjustment

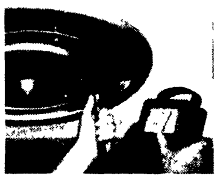


Fig. 5 Balancing

Fig. 6 Air Volume Adjustment

which is kept in compression by a slotted washer. The holes in B cone pass over the bolt heads. All that is necessary is to press up on B cone and insert or remove the slotted washers. (See Fig. 3.) Even where ceilings already exist, the outer cone is easily attached to a duct or collar.

The air direction adjustment is also accomplished quickly. All that is needed is a screwdriver to adjust the suspension screws for any angle of air discharge from horizontal to vertical. (See Fig. 4.)

System balancing is fast and simple. The single annular air stream permits immediate and accurate velometer reading. (See Fig. 5.) Desired air supply ratios may be rapidly obtained by adjusting the volume control dampers. A twist of the wrist regulates the volume instantly. (See Fig. 6.)

Nation-wide Sales and Engineering Service

The W. B. Connor Engineering Corp. maintains a research laboratory with a staff of trained specialists and district representatives in leading cities. Their services are at the disposal of consulting engineers, architects, air conditioning dealers and plant engineers. They can assist you in getting the best possible performance from your air conditioning system by creating custom-made air patterns which will thoroughly mix room and supply air, eliminate drafts and maintain uniform temperature throughout an area.

Free Handbook on Air Diffusion

It contains the latest engineering data on air diffusion and is profusely illustrated with charts, photographs, sketches and dimension prints that simplify the selection, application, location, assembly, erection, testing and adjusting of Kno-Draft Adjustable Air Diffusers. It is designed to help you get top efficiency from an air conditioning system by creating "custom-made" air distribution patterns.

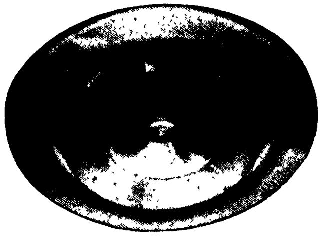
For your FREE copy, please write Dept. Y-29.

See pages 1188, 1189, 1190 and 1191 for data on Dorex Air Recovery and Air Purification Equipment.)

Charles Demuth & Sons, Inc.

Mineola, N. Y.

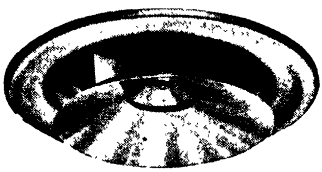
Demuth Draftless Air Distributors



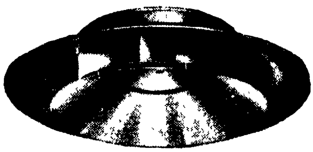
Supply & Return

Designed and Patented by Demuth

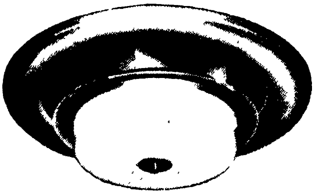
Illustration showing curved vane principle



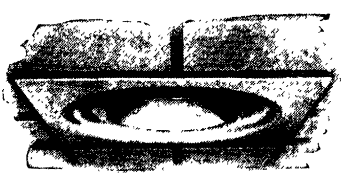
Type (S)



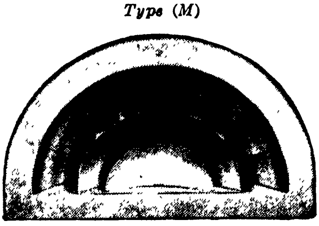
Type (I)



Type (SL)



Type (M)



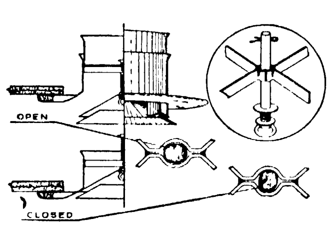
Type (SW)

The DEMUTH DRAFTLESS AIR DISTRIBUTOR consists of a series of curved vanes, mounted on a deflecting cone, to insure 360 deg distribution without the aid of secondary equalizing devices. A second hollow deflecting cone is arranged at a tangent to the primary cone, forming an injection nozzle.

The curved vanes discharge the supply air in turbulent streams, mixing room air with conditioned air, thereby providing constant air motion, uniform temperature and draftless distribution throughout the room.

Secondary air currents, which are created by the turbulence of the discharge air, are drawn into the diffuser by the bottom or secondary cone, mixed with the supply air, resulting in positive aspiration.

Lighting Fixtures as illustrated by Type SL are standard accessories and are available in Type S, Type F and Type I. For pendant or other special lighting applications,please consult your local representative or write direct to the factory. Volume Controls can, upon request, be incorporated into the diffuser at the time of manufacture. This device is intricate in design and simple to install—two very desirable features. It adjusts the volume of air supplied without decreasing the efficiency of the DISTRIBUTOR.

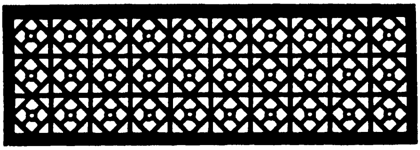


Diamond Manufacturing Company

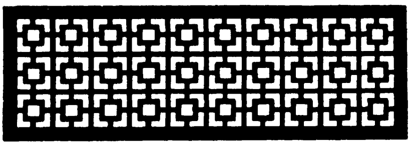
Main Office & Works
Wyoming, Penna.

Sales Representatives in All Principal Cities

DIAMOND PERFORATED-METAL GRILLES: AIR-CONDITIONING GRILLES AND REGISTERS; PERFORATED-METAL PLATES, SHEETS AND PARTS FOR ALL ARCHITECTURAL AND INDUSTRIAL REQUIREMENTS.

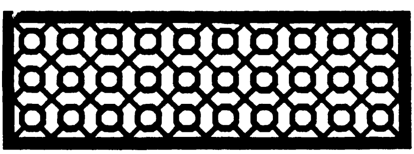


"Jack Special" Grille Pattern, 66 per cent open area

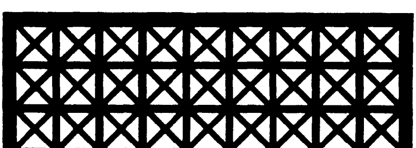


"Greek" Grille Pattern, 58 per cent open area

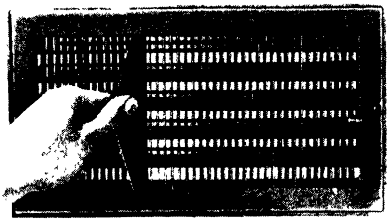
Diamond Architectural Grilles have been specified by leading architects and engineers for more than a third of a century and are now installed in many of America's finest buildings. Superior workmanship and finish in heavy-gage metal assures lasting beauty—standing up through the years against rough treatment which no cast-metal grille could survive. More than a hundred modern grille designs are illustrated in our catalog No. 36—most of which are available in any metal and in any thickness up to ¼ in. Catalog also gives complete working data.



"Egyptian" Grille Pattern, 70 per cent open area



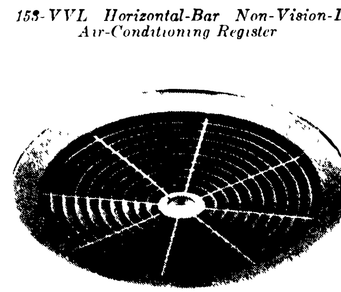
"Maltese Cross" Grille Pattern, 35 per cent open area



No. 256 Multiple-Valve Flex-Bar Air-Conditioning Register with 4-Way directional flow.

Diamond Air-Conditioning Registers & Grilles are available in a wide range of standardized types and sizes to meet any probable requirement. Catalog No. 36 illustrates many popular types and gives complete working data. It also illustrates and describes Diamond Fabricated Grilles, Diamond All-Steel Registers and other modern equipment for heating, ventilating and air-conditioning.

No. 153-VVL Horizontal-Bar Non-Vision-Design Air-Conditioning Register



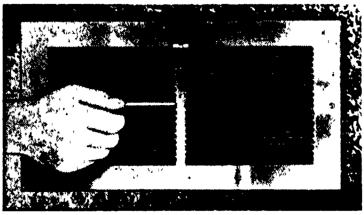
No. 500 Round Ceiling Outlet. Available in five standard sizes, with or without installation frames

Other Diamond Products include a complete line of Ornamental Cane and other perforated metal for Radiator Covers, Metal Furniture and scores of other architectural and industrial applications. Write for illustrated catalogs.

Hart & Cooley Manufacturing Co., Holland, Mich.

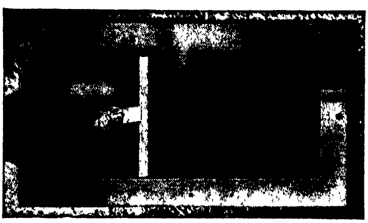
Air Conditioning Registers and Grilles—Warm Air Registers
Damper Regulators—Furnace Regulators—Pulleys—Chain

NO. 74 DESIGN—Economy Type Air Conditioning Register. Flexible-fin face



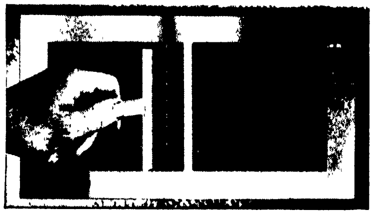
permits any sideway deflection of airflow. Positive, single-shutter valve with stop screw beneath valve handle for accurate volume control at the register face. Sponge rubber gasket prevents streaking. Finest quality construction. Available as Sidewall and Baseboard Registers in sizes 8x4 through 14x8—Grilles 8x4 through 30x8.

NO. 75 DESIGN—For Air Conditioning at Its Best. Flexible-fin face provides



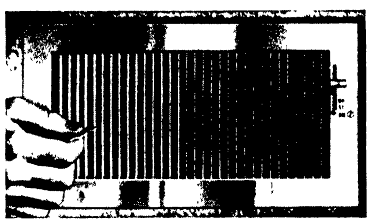
for any deflection of airflow sidewise. Turning-Blade Valve, an exclusive H&C feature, smoothly turns airflow up, straight or down with 30 per cent less resistance than other valves. Assures thorough distribution to all parts of room. Instantly adjustable. Sponge rubber gasket prevents streaking. Available as Sidewall and Baseboard Registers in sizes 6x4 through 14x8. Grilles 6x4 through 30x8.

NO. 76 DESIGN—For Shallow or Horizontal Ducts. An ideal air conditioning



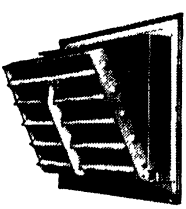
register for shallow or horizontal ducts which will not accommodate the turning blade valve of our No. 75 Design. Multi-shutter valve (depth 1⅛ in. from wall) and flexible-fin face provide for all deflections desired. Any desired up or down deflection can be maintained with adjusting screw beneath valve handle. Sealed to prevent streaking. Sidewall and Baseboard type Registers, in sizes 6x4 through 30x8.

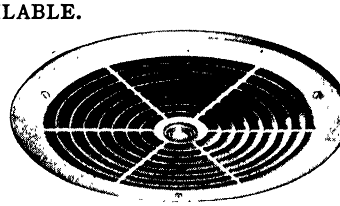
NO. 88 DESIGN—Shallow Ducts, Large Installations. Similar to our No. 76



except that face bars are pivoted and adjustable in 2½ in. sections (one moves all in section). A deluxe sidewall register and particularly advantageous for large installations. Largest size, 30x24, has only two valve handles. Removable handle available to avoid tampering. Sidewall Registers or Grilles 6x4 through 30x24.

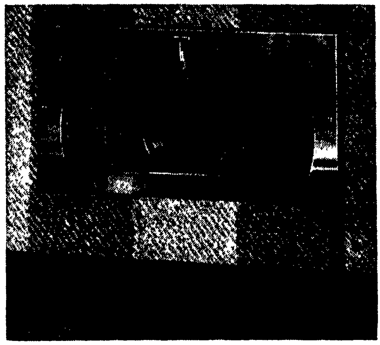
INSTALLATION FRAMES TO ACCOMMODATE ALL THE ABOVE ITEMS ARE AVAILABLE.





CEILING DIFFUSER—Primarily designed for use with residential package heating, where its low resistance eliminates the necessity of changing the blower. Will also do an excellent job on any residential or small commercial heating or cooling installation. 5 popular sizes. Gasket prevents streaking.

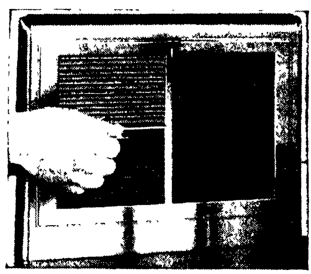
NO. 401 DESIGN—For Perimeter Heating. The No 401 "Diffusaire" Sidewall Register, installed above the baseboard on outside walls, provides an excellent means of blanketing cold wall and window areas with a curtain of warm air, thus counteracting downdrafts of cold air from these areas. With this register the air is dispersed in a full 180 deg spread upward in addition to permitting a portion of the airstream to be directed downward over the floor. An adjustable stop on the handle permits balancing the system at the register face. Available in sizes 10x6, 12x6, and 14x6. Also made in Baseboard type—No. 404.

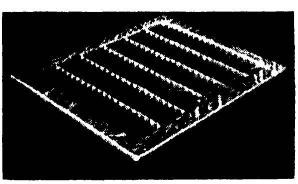


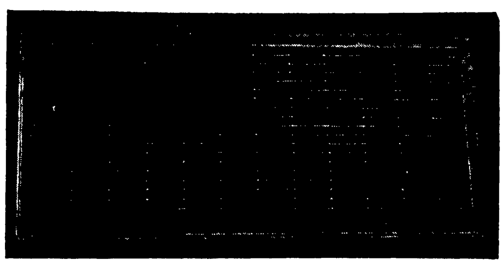
NO. 410 DESIGN—For Perimeter Heating. No. 410 "Diffusaire" Floor Register is specifically designed for perimeter heating to counteract downdrafts from windows and cold walls. Opposed louvre valve mechanism with set screw adjustment provides accurate volume control for easy balancing of the system. Design of diffuser blades permits even spread of air in fan shaped pattern with minimum amount of resistance. Available in sizes $2\frac{1}{4}$ in. x 14 in., 4x10, 4x12, 4x14, 6x10, 6x12, 6x14.

NO. 130 DESIGN—For Gravity or Conversion. This Baseboard Register has a removable face, flexible fins which may be easily adjusted to desired upward or downward deflection of airflow. valve that holds securely under all conditions, and approximately 80 per cent free area. It's tops for gravity installations and as a replacement register where an existing gravity job is converted to forced air. Furnished in beautiful METALUSTRE finish in sizes 10x8 through 13x11.

NO. 330—Sidewall Register. Companion piece to the No. 130 Baseboard Register. Furnished in sizes 10x8 and 12x8.







NO. 210 "NO-FLEX" Floor Register. Grid-type, very sturdy with heel-proof mesh ($\frac{7}{16}$ in. x $1\frac{15}{16}$ in.). Free Area is over 75 per cent. All steel body with smoothly operating valve running the short dimension. Furnished in all standard finishes including Oak. Sizes: 4x6 through 30x30.

NO. 265 RETURN AIR FACE—Matches No. 210 Register.

FOR COMPLETE DESCRIPTIONS AND ENGINEERING DATA OF THESE AND OTHER ITEMS WRITE FOR CATALOG.

Hendrick Manufacturing Company

48 Dundaff Street, **Carbondale, Pa.**

Sales offices in principal cities—consult telephone directories

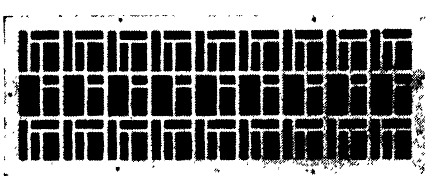
Hendrick Bulators; Hendrick Perforated Metal Grilles; Hendrick Mitco Open Steel Flooring, Armorgrids, Shur-Site Treads

HENDRICK BULATOR

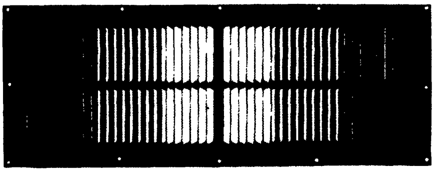
the *dual-unit* combination of a deflecting vane grille and an ornamental grille

Now you can secure in a single installation all the advantages of an adjustable vane grille—*to direct air flow*—and an ornamental grille—*to harmonize with the decorative scheme*—by specifying the new dual-unit Hendrick BULATOR*. It is a practicable combination that meets both the engineer's specifications for air throw and spread, and the decorative requirements of the architect.

In this dual-unit, the deflecting vanes are not noticeable, although they are mounted just behind the ornamental grille. The vanes are adjustable so that the air flow can be deflected to right or left, up or down, or in a combination of both directions.



Photograph taken with deflecting vanes less than an inch behind grille, shows that vanes are not noticeable.



Vertical deflecting vanes, showing how the vanes may be set to produce any desired air stream pattern.

For the imposed ornamental grille section of the dual-unit Bulator, Hendrick offers a wide selection of attractive designs, with the essential open area.

Tested at Case Institute

To determine the efficiency of the dual-unit, tests were made at the Case Institute of Technology, Cleveland, under the direction of Professor G. L. Tuve, as a result of which, "it was found that at a given air volume, the presence or absence of the 'Mosaic' (design) grille made very little difference on either the air stream pattern or the throw."

A copy of the detailed report on these tests will be mailed on request.

How to specify or order Hendrick Bulators

In specifying or requesting quotations on the Hendrick dual-unit Bulator, the following information is required:

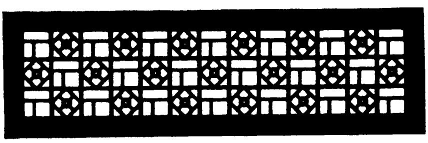
1. The name or description of the desired ornamental grille, as given in the Hendrick Grilles catalog.
2. The metal, gauge and finish of the ornamental grille, that is required.
3. The dimensions of the air duct opening.
4. The type of deflection desired in the deflecting vane section; whether right and left, up and down, or a combination of both.

* Beauty + Ventilator

HENDRICK PERFORATED METAL GRILLES

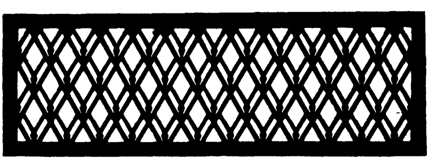
Hendrick decorative grilles are furnished in over a hundred patterns, and in a wide variety of overall dimensions, bar sizes, and number and size of perforations.

Many exclusive Hendrick designs, originally produced to meet an architect's specifications for some particular project, are now available as standard numbers, and facilities for making special designs to specifications make the Hendrick service even more complete. The wide range of patterns permits the choice of a grille that will harmonize with any style of architectural design or period construction.

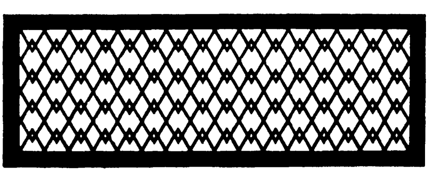


Arglin—63 per cent Open Area

Grilles are fabricated in heavy-gauge aluminum, bronze, copper, Monel, steel, stainless steel, and other commercially rolled metals. With ample open areas and accurate sizes, Hendrick grilles are characterized by clean-cut perforations, fine finish, and freedom from burrs and other imperfections.

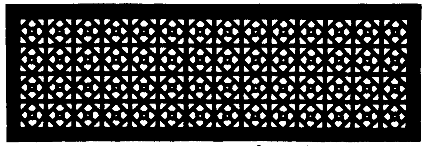


M-No. 2—57 per cent Open Area

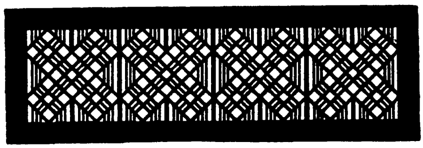


M-No. 9—67 per cent Open Area

They are easy to install, and always lie flat because of a special flattening operation in their manufacture.



Argive—60 per cent Open Area

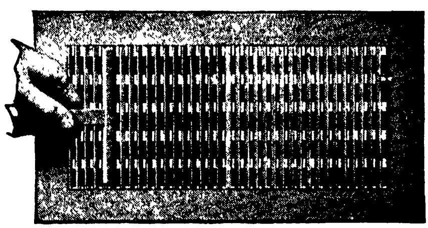


La Crosse—55 per cent Open Area

AIR CONDITIONING GRILLES AND REGISTERS

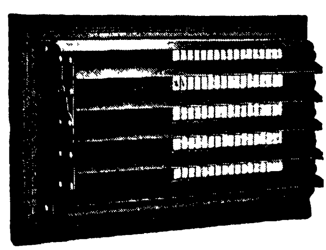
Hendrick air conditioning grilles and registers for directed air flow are made in various meshes, the standard having ⅜ in. opening between face bars. All types are furnished with either horizontal or vertical directional bars.

The standard mesh can either be furnished with grille bars permanently set at the factory for straight or directional air flow, or furnished so that the grille bars may be individually adjusted on the job to direct air flow to any desired degree.



Front View

Rear View

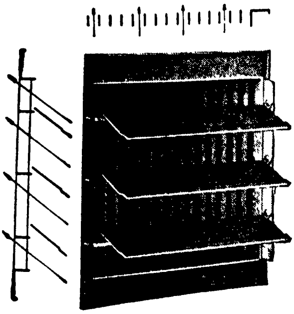


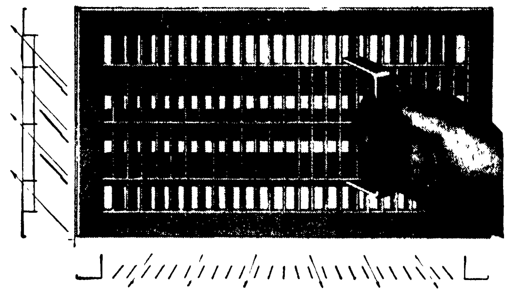
The Independent Register Co.

Established 1898

3747 East 93rd Street, **Cleveland 5, Ohio**

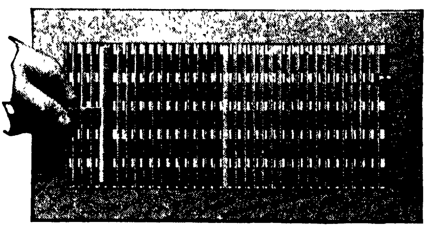
AIR CONDITIONING REGISTERS AND GRILLES



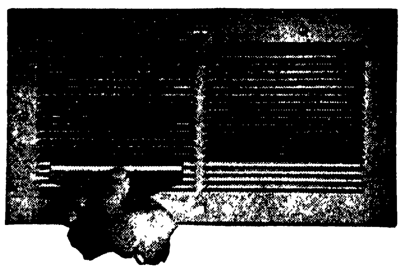


Rear View
Showing Adjustable Deflecting Vanes

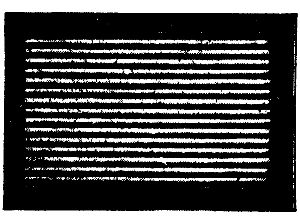
No. 321A Grille with Deflecting Vanes—With vertical grille bars and horizontal deflecting vanes. The grille bars may be individually adjusted to direct air flows to right or left; and the vanes are made individually adjustable to deflect air flows up or down.



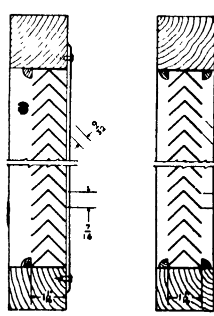
No. 238 Wrought Steel—4-way adjustable direction of air flow. Flexible vertical grille bars, multiple valves.



No. 139 Wrought Steel—Flexible horizontal grille bars, bendable for up, down or straight air flow. Single valves.



Independent No-Vision Grilles—No. 1312 for Doors, Walls and Partitions The grille bars are "V" shaped; it is impossible to see through the grille from any viewpoint.



R C

No. 1312R—With overlapping rim ⅝ in. wide, on all four sides.
No. 1312C—With grille core only, installed with moulding.

The Pyle-National Company

Multi-Vent Division
1363–78 N. Kostner Ave.
Chicago 51, Illinois

Sales Engineers and Agents
in Principal Cities of
U. S. and Canada

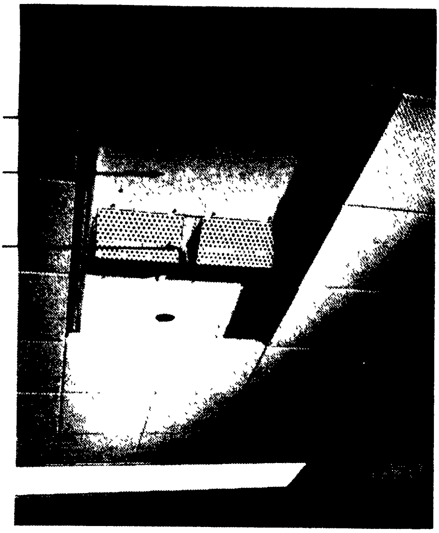
MULTI-VENT* *LOW VELOCITY* AIR DIFFUSION PANELS

Multi-Vent installations are simple, quick to balance and easy to clean. They have been applied with remarkable results to almost every type of building, new or old, and are particularly well adapted to the *lower ceilings and movable partitions* in modern architecture.

Panel Frame . . . installed in the bottom of air supply duct or plenum.

Control Plate . . . supporting one or more valves per panel may be easily lowered to provide ready access to duct above for cleaning.

Pressure Displacement Air Valve . . . single adjusting screw raises, and lowers a valve plate above opening in control plate to regulate volume of air flow from duct into dual V shaped primary distribution sections, the design of which insures even distribution of air over the entire perforated area below panel.



Concealed Multi-Vent Panel exposed by removal of six squares of metal acoustical ceiling.

11 Outstanding Advantages

1. Radiant Panel Heating and Cooling Effect Adds to the Comfort Factor—The large areas of the ceiling which function as distribution plates for the Multi-Vent panels are heated or cooled to the temperature of the supply air.

2. Complete Absence of Strong Air Streams or Blow Eliminates All Air Direction Adjustments by Diffusing Vanes or Baffles: With Multi-Vent duct velocities are so radically reduced (within the diffuser itself) . . . diffusion is so rapid, thorough and wide-spread . . . that no air movement in excess of ASHVE comfort zone requirements exists more than six inches away from the perforated distribution plate.

3. No Deflection Problems to Restrict Location or Capacity of Outlet Panel: With Multi-Vent the location and the capacity of the diffuser can be determined solely by load considerations assuring maximum effectiveness and efficiency. The proximity of seating locations or the relative positions of partitions and lighting fixtures—which must be a major consideration in locating high velocity diffusers to avoid drafts—need not be considered with Multi-Vent regardless of ceiling heights.

4. Complete Freedom of Partition Movement With No Panel Alteration or Adjustment Necessary: In resizing office or store space Multi-Vent panels can even be bisected by partitions with no possibility of draft hazards or other undesirable air diffusion problems.

5. No Change in Air Diffusion Patterns or Cold Drop When Desired Volume of Air Delivered is Varied: Multi-Vent's adjustable pressure displacement valve can be easily set for delivery of various amounts of air without disturbing the

**This application of low velocity, pressure displacement air diffusion is fully protected by patents. Only with Multi-Vent can you enjoy its benefits. Multi-Vent is the registered trade mark of the Pyle-National Co., Chicago, air distribution systems and parts thereof.*

THE PYLE-NATIONAL COMPANY

Multi-Vent Division

balance of the overall system. Neither single panel adjustments to suit occupants special requirements nor substantial reduction or increase of air capacity at source to meet seasonal demands will in any way affect the desired air flow pattern.

6. 40 Per Cent Higher DTD Will Meet Comfort Zone Requirements: Multi-Vent will permit raising the usual 15 deg Diffusion Temperature Differential to as high as 25 deg (with an 8 ft ceiling for example). Thus 40 percent less air need be used to handle a given load making possible substantial economies in ducts, fans, filters and coils.

7. No Protruding Outlets to Interfere with Style and Location of Lighting Fixtures or with Interior Ceiling Design—Multi-Vent panels are completely concealed in metal acoustical ceilings. Installed flush in any other type ceiling, Multi-Vent is less conspicuous than diffusers of any other make.

8. Exceptional Uniformity and Control of Room Temperature: Multi-Vent can achieve a temperature differential of as little as 1 deg within the comfort zone in all seasons . . . and 2 deg is guaranteed. This insures *true* air conditioning comfort and will meet the most exacting air conditioning requirements for scientific research and industrial processing.

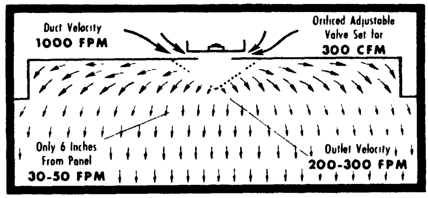
9. More Room Air Changes Per Hour With Less Air Motion: Multi-Vent is designed to handle the greatest amount of air in proportion to room size and therefore is particularly well suited to locations having high load or high ventilating requirements.

10. Elimination of Dirt Impingement on Ceiling and Wall Surfaces Reduces Costly Redecorating—Dirt Particles are not driven into paint and plaster by horizontal high velocity primary air streams, but are deposited gently downward on easy-to-clean furniture and floor surfaces.

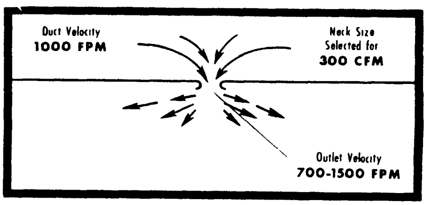
11. Complete absence of the sound of rushing air.

* See foot note on page 1280.

MULTI-VENT*—LOW VELOCITY DISTRIBUTION *BY DISPLACEMENT*



OTHER DIFFUSERS—DISTRIBUTION *BY HIGH VELOCITY INJECTION*



Panel Type	Panel Frame Sizes—Inches	No of Air Valves	Maximum CFM Capacity
MVAR-122-1	12 x 24	1	75†
MVAR-123-2	12 x 36	2	150†
MVAR-124-2	12 x 48	2	150†
MVAR-125-3	12 x 60	3	225†
MVAR-126-3	12 x 72	3	225†
MVAR-244-1	24 x 48	1	300†
MVAR-245-1	24 x 60	1	300†
MVAR-246-1	24 x 72	1	300†
MVAR-364-1	36 x 48	1	300†
MVAR-365-1	36 x 60	1	300†
MVAR-365-2	36 x 60	2	600†
MVAR-366-1	36 x 72	1	300†
MVAR-366-2	36 x 72	2	600†

Special sizes to accommodate lighting or ceiling decoration are available

† Allowable cfm per sq ft of panel area must be determined by DTD and ceiling height. Write for selection data.

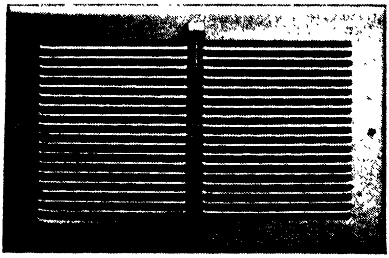
Standard types *with perforated ceiling plate* for plaster or fibre ceiling are listed above. Panels are also supplied *less perforated plate* for use in metal acoustical ceilings—designated MVAMC.

Application data will be furnished on request.

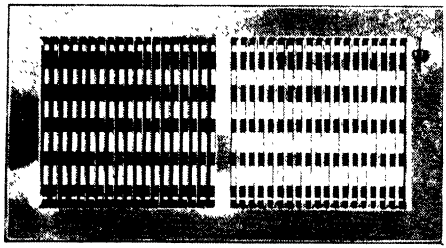
Standard Stamping & Perforating Co.

3111 W. 49th Place, **Chicago 32, Illinois**

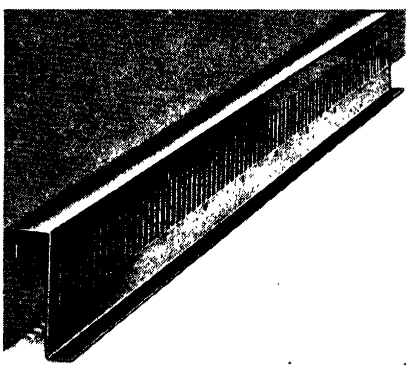
Air Conditioning Registers and Grilles—Cold Air Faces
Perforated Metals for all Purposes



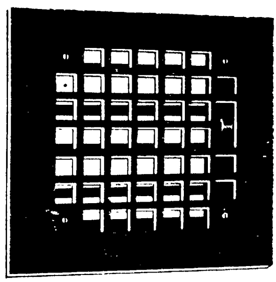
No. 41H Sidewall Register
Equipped with a single damper and a 3/16 inch turned down edge for flush side wall installations.



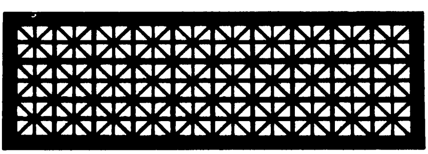
No. 331 Sidewall Register
Horizontal multiple valve louvres attached to vertical "bend-ezy" faces, adjustable for four way deflection.



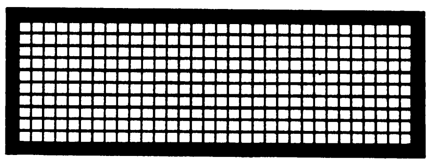
C-F4 Perimeter Baseboard Register
For use in homes, churches, and commercial installations. Fabricated of 20 gage steel with prime coat finish only. Simplifies balancing of heating systems.



No. 100-LO Multiple Louvre, Lever operated register.
Also available as: 100-KL, a Key Operated register; 100-RK, a register with a Removable Key; 100-PO, a Pulley Operated register (especially suited for inaccessible installations). Register faces are attached to louvre boxes with screws so that the boxes are detachable.



Union Jack Design



Plain Lattice Design

All Standforated Grilles and ornamental designs are available in steel from 16 gage to ¼ in. thicknesses. They can also be furnished in non-ferrous metals, such as aluminum, brass, bronze and stainless steel, and in varying thicknesses according to the physical properties of each metal.

Complete specifications will be furnished upon request

Stewart Manufacturing Co., Inc.

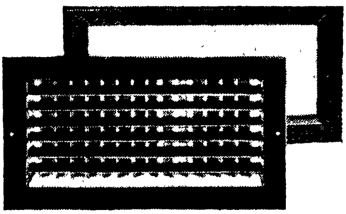
Cedar Grove, N.J.

Technical and price information available through representatives in all principal cities, or at factory headquarters.

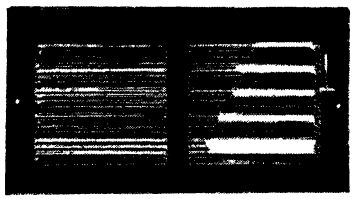
In Canada: Douglas Engineering Co. Ltd., 101 Murray St, Montreal 3, P.Q.

A complete line of registers, grilles and scoops for all types of industrial and residential air conditioning applications.

Stewart presents 54 styles in 2140 sizes for engineer or contractor with special as well as standard specification requirements. Engineering data available, conveniently arranged for estimating large or small installations. Standard material is cold rolled steel, but stainless steel, aluminum and brass are also available for most products.



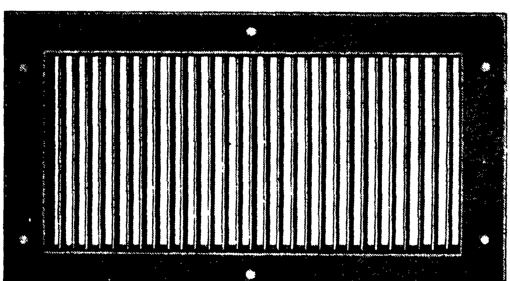
Style DDH *Plaster Frame*



Style 94

DEFLECTAIRE Style DDH, ALL-WAY SELECTIVE AIRTHROW, has two banks of individually adjustable fins—front bank horizontal, rear bank vertical. Fins are stream-lined, ¾ in. deep, spaced on ¾ in. centers. Bank of lever- or key-operated 1 in. multiple valves can be furnished as addition or substitution. Valve blades overlap when closed and open through 110 deg arc. Overall size of face 2 in. greater than duct opening. PLASTER FRAME allows for removal of grille when desired without damage to surrounding painted areas. Usable with composition duct and becomes the base to which register is attached.

Style 94, MULTIPLE VALVE WALL REGISTER, is a unit of the Stewart residential line. Horizontal face bars ¼ in. center to center and ¼ in. deep, deflect air downward. Multiple valve damper is lever operating. 1 in. valves overlap when closed and open on 110 deg arc. Face bars can be bent with removable key furnished. All grilles and registers have prime coat finish and sturdy rubber gasket is cemented on registers at factory.



DV Close

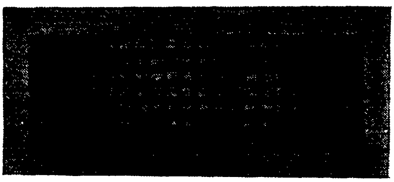
The DEFLECTAIRE Line is also available with face fins on CLOSE centers. This applies to all DEFLECTAIRE products, first bank only. Spacing of fins on CLOSE-type is ½ in. center to center. Fins are ¾ in. deep, streamlined, sturdy, lightweight. Lever or key-operated multiple valves available with CLOSE-Type outlets and returns.

Airfoil
AIR CONDITIONING OUTLETS

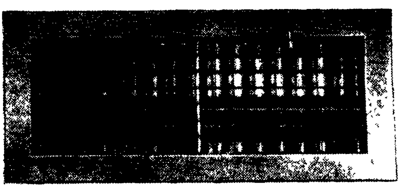
Titus Mfg. Corp.

WATERLOO, IOWA

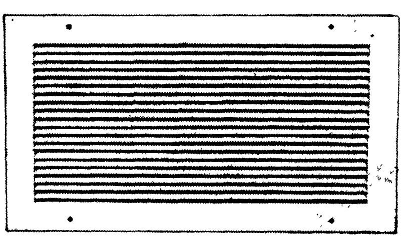
Titus Mfg. Corp. are designers and manufacturers of Airfoil Grilles featuring the Airfoil *Louver*—patterned after the airfoil section of an airplane. The following distinctive features identify Titus Airfoil Louvers. (1) Smooth as glass streamlined surface (2) Solid construction (3) Noiseless performance (4) Minimum turbulence.



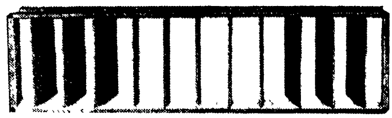
270 **Airfoil Grille** Gives 4-way directional control. Louvers set on $\frac{3}{4}$ in. centers. Individually adjustable to create any air pattern desired.



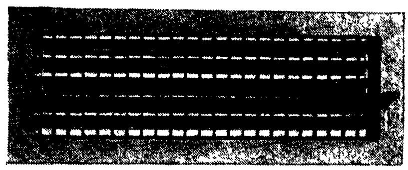
274 **Multi-Shutter Register** Airfoil Louvers are featured in front. Individually adjustable. Rear multi-shutter damper blades.



RL-21 Return Air Grille The RL-21 features the blades on $\frac{3}{8}$ in. centers. They present a smooth grille surface facilitating easy cleaning.

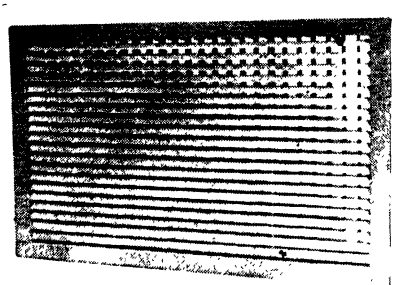


AG-25 Volume Controllers provide positive control of air volume. Blades individually adjustable. Sponge rubber gasket holds unit firmly in duct.

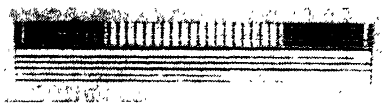


276 **4-Way Multi-Shutter Register** Consists of the #270 4-way directional grille combined with a multi-shutter damper. Damper blades inter-locked when closed to provide complete shutoff.

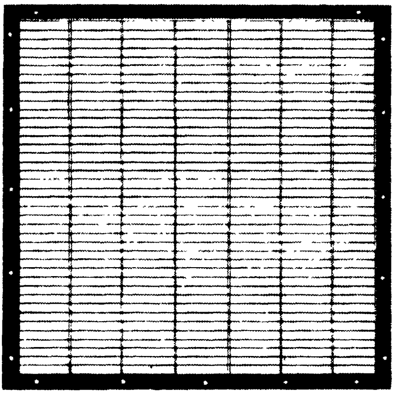
230 **Return Air Grille**—blades on $\frac{3}{4}$ in. centers. Parallel to long dimension. Deep fins. Standard $1\frac{1}{4}$ in. beveled border. Any size grille can be furnished.



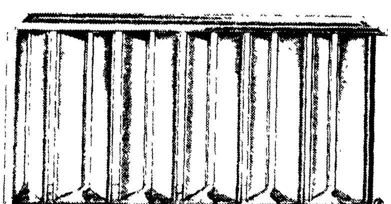
240 **Return Air Register**—Features rear multi-shutter louvers—quick shut off. Rear blades are deep-spaced on 1 in. centers.



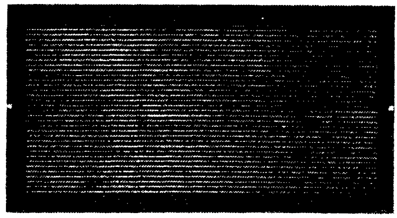
L-8—features horizontal front louvers, vertical rear louvers. Blades are individually adjustable. Front louvers are spaced on ½ in. size.



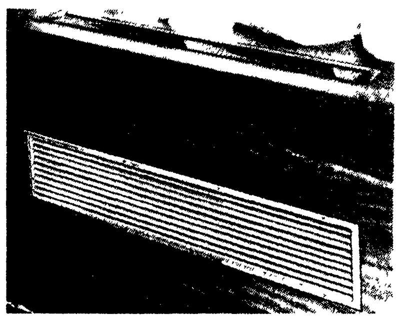
HEAVY DUTY INDUSTRIAL GRILLE—features 14 gage steel blades. Vertical support bars placed on 6 in. centers. Standard grade primer coat finish. Uses 16 gage steel, extra wide border for easy mounting. Made in two sections . . . Grille Face and Volume Controller. Especially designed to take lower wall use and abuse.



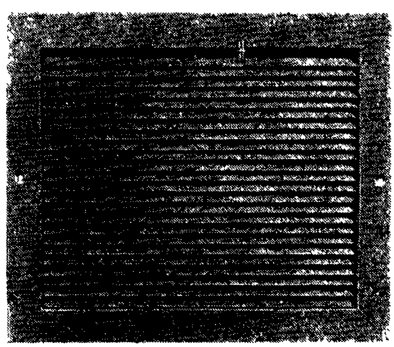
AG-35 VOLUME CONTROL DAMPER . . . features opposed acting blades which operate simultaneously toward or away from each other in pairs. Sets back of Grille Face—controls distribution of air to grille outlet. Furnished with key for quick, easy adjustment.



T-700-B—No Vision door and partition grilles. Made with flange frame or channel type frame. All steel—V-shaped louvers on ½ in. centers. Widely used for exhaust and return air grilles.



CONVECTOR GRILLES—Louvers closely spaced. Heavy-duty support bars on 6 in. centers. Over 70 per cent free area. Can be installed in any special surface, such as marble or tile. Extra wide blades. Each "hemmed" for extra strength and safety. Available with damper and knob control.



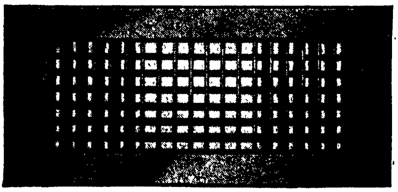
RL-22—Return air grille with multi-shutter damper blades in rear. Front blades on ¾ in. spacing. Rear blades operated by lever on face of grille. Available with removable lever lock.

TUTTLE & BAILEY

Engineered Products for Residential, Commercial and Institutional Air Conditioning, Heating, Ventilating

NEW BRITAIN, CONNECTICUT

Tri-Flex GRILLES AND REGISTERS



FOR SUPPLY

Available as grilles, double deflection grilles, multi-shutter registers, double deflection multi-shutter registers . . . with individually adjustable horizontal or vertical bars. Designed to meet every requirement of supply air delivery with maximum control of air direction, throw, drop and volume. Stocked in 26 standard sizes for quick specifying and economy. Where required, non-standard sizes can be furnished.

SANTROLS

A flexible device designed for installation with Tri-Flex grilles to provide positive control of air volume . . . uniform distribution over the entire supply outlet. Stocked in the same 26 standard sizes as listed for Tri-Flex. Also available in non-standard sizes.

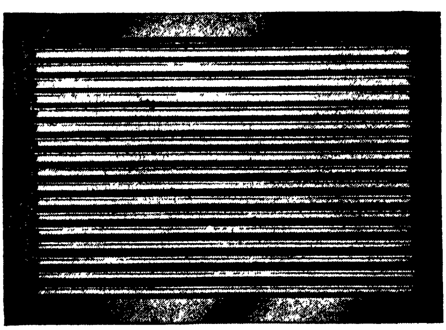
26 STANDARD SIZES

8 x 4	14 x 4	20 x 6	30 x 6
10 x 4	14 x 5	20 x 8	30 x 8
10 x 6	14 x 6	24 x 5	30 x 10
12 x 4	16 x 5	24 x 6	30 x 12
12 x 5	16 x 6	24 x 8	36 x 8
12 x 6	20 x 5	24 x 10	36 x 10
		24 x 12	36 x 12

AEROVANE GRILLES AND REGISTERS

FOR RETURN

Aerovane grilles and registers are available with horizontal or vertical bars, styled to match the Tri-Flex line. Stocked in 20 standard sizes for quick specifying and economy. Where required, non-standard sizes can be furnished.



20 STANDARD SIZES

10 x 6	24 x 24
10 x 8	30 x 12
12 x 6	30 x 18
12 x 8	30 x 24
12 x 12	36 x 18
18 x 6	36 x 24
18 x 12	36 x 30
18 x 18	48 x 24
24 x 12	48 x 30
24 x 18	48 x 36

Aerofuse DIFFUSERS

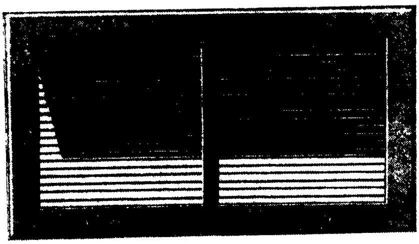
TYPE	LISTED SIZES		FEATURES
	FOR INSTALLATION ON CEILING		
 D	12″ x 12″ 16″ x 16″ 20″ x 20″ 24″ x 24″ 30″ x 30″		Delivers supply air in 360 deg. pattern. Type DF for flush mounting in standard acoustical ceilings. . . . Type DE for installation on plaster ceilings.
 S	4½″ 6″ 8″ 10″ 12″	15″ 18″ 21″ 27″ 33″ 38″	Flush type, extends only ⅝ in. below ceiling. Temporary deflection of air stream issuing from outer passage minimizes streakage.
 LF	15″ 18″ 21″	27″ 33″ 38″	Combines flush type diffuser and attractively styled light fixture. Six standard sizes . . . for use with 100 to 300 watt bulbs.
 H	8″ 10″ 12″ 15″	18″ 21″ 27″ 33″ 38″	Half round diffuser for installation on ceiling where it is desirable to distribute air from a point adjacent to a side wall.
	FOR INSTALLATION ON CEILING OR EXPOSED DUCT		
 EAC	6″ 8″ 10″ 12″	15″ 18″ 22″ 27″ 33″	Manually operated auxiliary effective area control ring provides complete on-the-job adjustment of air delivery.
 ES	6″ 8″ 10″ 12″	15″ 18″ 22″ 27″ 33″	High capacity diffuser with rings offset and stepped down maximum distance. For given effective area, neck diameter is reduced to minimum.
 R	8″ 10″ 12″ 15″	18″ 21″ 27″ 33″ 38″	Combination supply and return (or exhaust) unit. Designed for installation where simplification of duct layout is essential.
	FOR INSTALLATION ON EXPOSED DUCT		
 E	4½ 6″ 8″ 10″ 12″ 15″ 18″ 21″ 27″ 33″ 38″		E 1 Flush type, extends only ⅝ in. below duct. Supply air is directed in horizontal plane on leaving diffuser. E 2 Rings graduated downward medium distance effecting an average increase of approximately 45 per cent in capacity over that of Type E1. E 3 Rings stepped down maximum distance effecting an average increase of approximately 90 per cent in capacity over that of Type E1.

United States Register Company

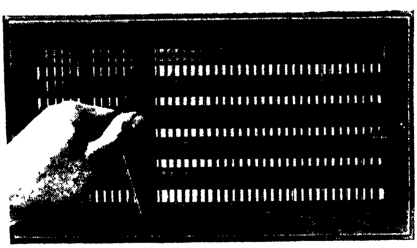
General Offices: **Battle Creek, Mich., U.S.A.**

Branches: MINNEAPOLIS, MINN., KANSAS CITY, MO., ALBANY, N. Y.

Air Conditioning Registers, Vents and Grilles



No. 249—Multiple-Valve Air Conditioning Register. Gives complete Air Control. Vertical Front Bars—Key-pin adjusted to provide 45 deg Right and Left or Two-way Side Flow. Lever operated Horizontal Backvalves give from Full Closed to any degree of Upflow and to 45 deg Downflow. FULL FACE COVERAGE. Can be supplied with any style of Setting Frame. Fits all Stack Heads of Standard Size Dimensions.

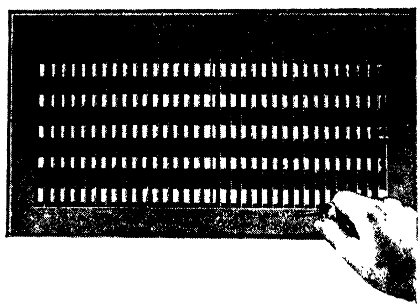


All of above Styles can be supplied with either Lever or Individually adjusted Multiple Valves or Louvers. *i. e.* 177VVI—Vertical Valves Individually adjusted. 145VVL—Lever operated Vertical Valves.

Grilles and Vents in Matching designs are available.

For Complete Information Write for Catalog No. 51 or Pocket Manual No. 51.

No. 153—Single-Valve Air Conditioning Register-Bars ¼ in. deep—Spaced 4 openings to the inch affords Non-Vision. Can be supplied in Directional Flow in either Horizontal or Vertical Bar Styles. Can be furnished with all styles of Setting Frames.

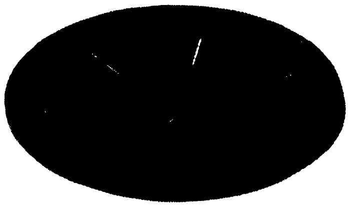


No. 256—Multiple-Valve Flex-bar-Air Conditioning Register. Vertical Front Bars set 22 deg Right and Left. Side Flow Deflection attained by setting of Grille Bars with bending wrench to accommodate room condition. Backvalves give same Up and Down control of air flow as No. 249 above. FULL FACE COVERAGE. Can be supplied with any style of Setting Frame. Fits all Stack Heads of Standard Size Dimensions.

No. 153-VVL Horizontal Bar Non-Vision Design-Vertical lever operated rear valves.

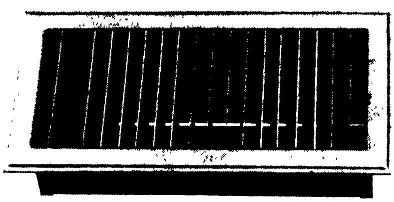
Complete Gravity and Air Conditioning Register, as well as Fitting Catalogs furnished on request.

No. 500 U.S. Ceiling Outlet



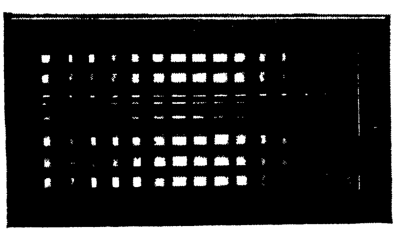
No. 500 U. S. Round Ceiling Outlet. Made in 6 in., 8 in., 10 in., 12 in. and 14 in. sizes. Furnished with or without No. 900 frames. However, frames are recommended.

No. 410 U. S. Perimeter Floor Register.



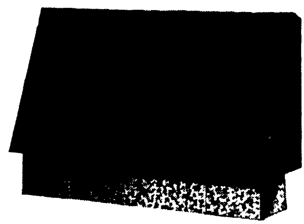
No. 191 U. S. Multi-Flex Register Streamlined Double-Edged Grille bars. Front bank of bars -Horizontal Adjustable. Second bank of bars—Vertical Adjustable. Rear Valves—Horizontal Lever operated. May be furnished Key operated.

No. 191 U. S. Multi-Flex Register



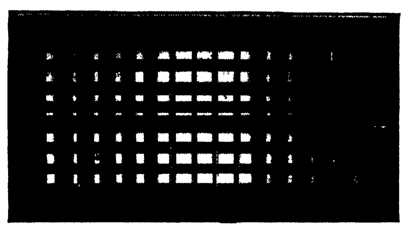
No. 165-3¾ U. S. (Out-of-Wall) Air Conditioning Register is made in 10 x 6, 12 x 6, 14 x 6 in. sizes—with heads (No. 165-3¾)—without heads (0165—3¾). Intakes to match No. 175-3¾, made in 5 sizes 10–12–14–24 & 30 x 6 in. "Dodge" old house troubles. Avoids cutting sills, joists, walls, carpets, rugs and floor. Ideal for perimeter jobs.

No. 165-3¾ U. S. Out-of-Wall Air Conditioning Register



Valves of No. 410 run the long way to give "Away-from-the-Wall" Deflection.

No. 190 U.S. Multi-Flex Register



No. 190 U. S. Multi-Flex Register Stream-lined, Double-edged Grille bars. Front bank of bars—Vertical, Adjustable. Second bank of bars—Horizontal, Adjustable. Rear Valves, Horizontal Lever operated. May be furnished key operated.

No. 192 U. S. Multi-Flex Register—same as No. 190—but with Vertical Bars only.

No. 193 U. S. Multi-Flex Grille—same as No. 190—but less Rear Valves.

No. 194 U. S. Multi-Flex Grille—same as No. 191—but less Rear Valves.

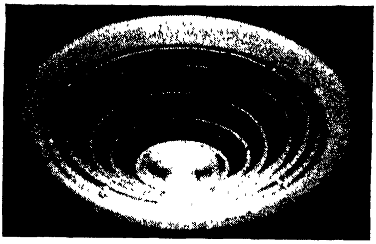
No. 195 U. S. Multi-Flex Grille—same as No. 193—but Vertical Bars only.

No. 196 U. S. Multi-Flex Grille—same as No. 194—but Horizontal Bars only.

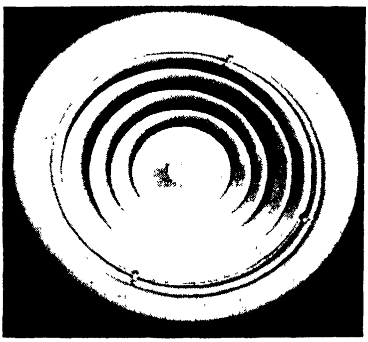
Universal Diffuser Corp.

890 Whittier St., **New York 59, N. Y.**

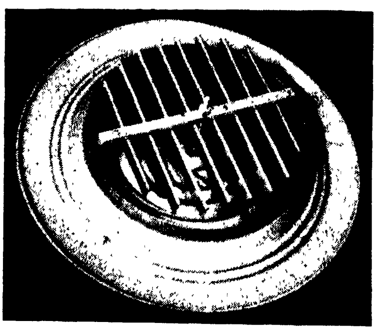
Manufacturers of FLEXIFLO Adjustable Diffusers



Flexiflo Fully Open



Flexiflo Closed



Individually movable blades of equalizing deflector

VARIABLE EFFECTIVE AREA

The effective area of **FLEXIFLO** is infinitely variable; all the blades move simultaneously when the control knob is moved up or down. The air delivery can thus be varied from near zero to full output without changing the characteristic air pattern, which is a flow parallel to the ceiling. Since the throw is inversely proportional to the effective area, precise control of air flow is obtained by simply changing the opening of the **FLEXIFLO** to meet your specific requirements.

OPERATING PRINCIPLE

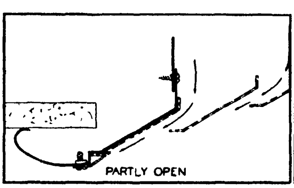
It is essentially a conical spiral with flanged edges, flexibly tied together by means of a centre rod which can slide up and down. In action the **FLEXIFLO** simulates an infinite number of concentric jets. The air, on leaving the blades, travels parallel to the ceiling; the low pressure area created by its temporary high speed, creates *a high aspirating effect* resulting in a large entrainment of room air which is rapidly mixed with the cold air from the duct. This high rate of mixture dissipates the energy of the supply air so that objectionable drafts are eliminated. Moreover this rapid mixture of cold and warm air prevents stratification and quickly equalizes the temperature.

UNIFORM DISTRIBUTION

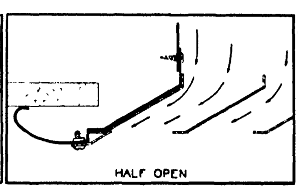
Every **FLEXIFLO** diffuser is equipped with an equalizing deflector attached directly to the main cross bar. The blades are individually movable and will remain fixed in any position.

Besides acting as instruments to direct the flow of air and equalize its flow, they can be used to restrict the volume in case a large reduction in neck velocity should become necessary *after* installation.

Write for Catalog



Partly Open



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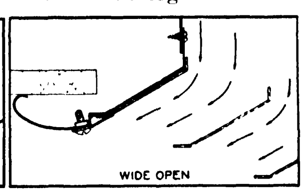


Wide Open

Young Regulator Company

5209 Euclid Avenue, **Cleveland 3, Ohio**

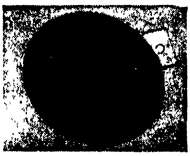
DAMPER REGULATORS; REMOTE CONTROLS SYSTEMS

Sales Representatives in Principal Cities

Young Regulators meet practically every condition where damper regulators are required for controlling air volume.

No. 1—For installation on finished wall. May be locked in any position.
No. 700—For remote control of one or more dampers at distances up to 250 ft.
No. 704—Corner pulley eliminates friction where there are many turns.
No. 301 and 201—For imbedding in plaster. Cover plate flush with finished wall.
No. 605 and 602—Bearing set used in place of CRS rod when regulator is mounted on side of duct.
No. 656—End bearing provides a bearing for the damper rod at side away from regulator and has a rubber gasket which prevents air leakage.
No. 401—Valcalox regulator for mounting on duct. May be locked in any position.
No. 4—Adapter for connecting different sizes and shapes of damper rods to various models of regulators and to adjust length of rod.
No. 403—Valcalox for mounting on duct, lever adjustment. May be locked in any position.
No. 910—Convector regulator for operating damper hinged on back of an enclosure.
No. 900—Air split regulator for operating a splitter damper.
No. 914—Concealed air split regulator with mitre gears used when the regulator is operated from a suspended ceiling
No. 1015—Volume control grille gives equal distribution of air over entire grille and directional flow.
No. 807—Register with fusible link for operation by remote control.
No. 805—Damper with fusible link for operation by remote control.
No. 815—Relief damper with motor and remote bulb thermostat controls temperature of individual rooms.


No. 1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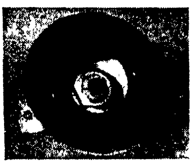
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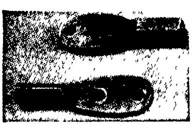
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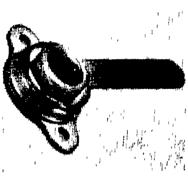
No. 401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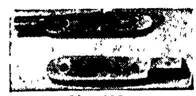
No. 700


No. 704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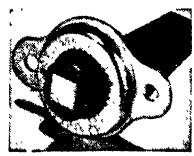
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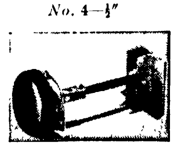
No. 4—½"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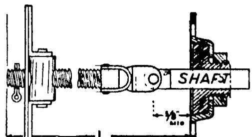

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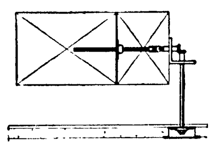
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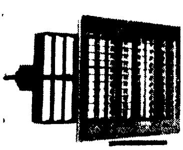
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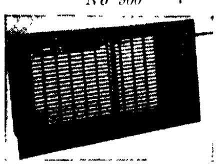
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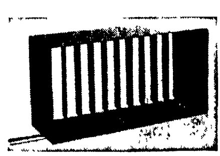
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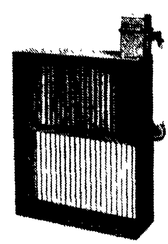

No. 815

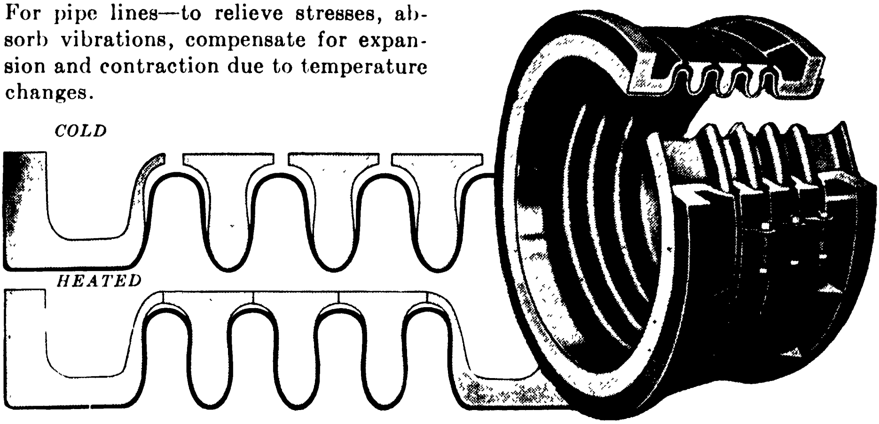
Badger Manufacturing Company

230 Bent St., Cambridge 41, Mass. • 60 East 42nd St., New York

Representatives in principal cities

Badger "PACKLESS" Corrugated Expansion Joint

For pipe lines—to relieve stresses, absorb vibrations, compensate for expansion and contraction due to temperature changes.



EXCLUSIVE *all-curve* corrugations equalize stresses

Directed Flexing Self-Equalizing Rings on Badger Joints assure even distribution of stresses whether caused by *temperature, pressure* or *movement* . . . no excessive strains.

INSTALLATION, OPERATING, MAINTENANCE ECONOMIES

"Packless"—Made from a *single tube*. . . no packing . . . no servicing so no manholes or tunnels are required.

Flexible—Fast response to movement reduces to minimum the thrust on adjacent equipment.

Wide range of traverses—By varying the number of corrugations, traverses from a fraction of an inch up to any practical limit are possible.

Wide range of pressures—Standard joints for normal pressures . . . special joints for higher pressures.

Compact, easy to install—Outside diameter is about the same as regular pipe flange—nothing clumsy or hard to install.

Directed flexing, all-curve corrugations plus all-curve self-equalizing rings—Exclusive features on Badger Joints prevent stresses from *localizing*—assure long life, greater dependability. Rings *limit* and *progressively control* flexing movement.

Heat treatment—Scientific heat-treatment during manufacture removes forming stresses . . . lengthens life of the joint.

Available in copper, stainless steel and other metals—To overcome temperature, pressure and corrosion condition.

Flexonics Corporation

EXPANSION JOINT DIVISION

(Formerly Chicago Metal Hose Corp.)

Maywood, Illinois

District Offices

Atlanta Boston Cincinnati Cleveland Detroit Ft. Worth
Los Angeles New York Philadelphia St. Louis San Francisco

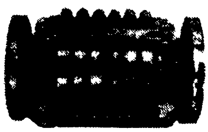
In Canada: Flexonics Corporation of Canada, Ltd., Brampton, Ontario

CMH CONTROLLED-FLEXING EXPANSION JOINTS

For pressures up to 300 psi—temperatures to 1600 F

The control rings of CMH Controlled-Flexing Expansion Joints provide guided flexing of the joint and prevent any permanent deformation of corrugations. Control rings are firmly anchored into corrugations to close working dimension tolerances. Expansion travel up to 7½ in. may be secured with a single CMH Expansion Joint.

Standard sizes from 3 in. to 48 in. I.D. in either stainless steel or copper. Available with or without stainless steel internal sleeves.



CMH Controlled-Flexing Expansion Joint with flanged ends.

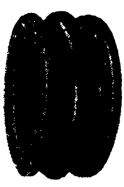
CMH Controlled-Flexing Expansion Joint with Welding Ends.

CMH FREE-FLEXING EXPANSION JOINTS

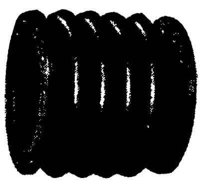
For pressures up to 30 psi . . . temperatures to 1600 F

CMH Free-Flexing Expansion Joints are made with single or multiple corrugations. Designed for expansion travel up to 1 in. per unit. Additionally, misalignment correction or offset motion is calculated at $\frac{1}{16}$ in. per corrugation when two or more corrugations are used.

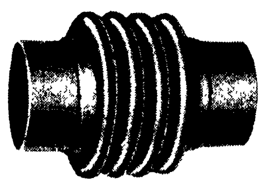
Standard sizes from 3 in. to 48 in. I.D. in either stainless steel or copper. Available with or without stainless steel internal sleeves.



Single Corrugation CMH Free-Flexing Expansion Joint with flanged Ends.



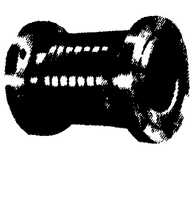
CMH Free-Flexing Expansion Joint with multiple corrugations and flanged ends.



CMH Free-Flexing Expansion Joint with welding ends.

CMH FLEXONIFLEX EXPANSION JOINTS

For pressures to 1500 psi . . . temperatures to 1600°F



CMH FLEXONIFLEX units make it possible to utilize the expansion joint's advantages of compactness and simplicity of installation at pressures far beyond those which had previously been considered safe. While units have been designed for pressures of 5500 psi, pressures in excess of this may be handled by units of special design.

Standard FLEXONIFLEX units have stainless steel pressure carriers of the bellows type formed within integral control rings and end sections. They are available in single or multiple ply, lined or unlined. Sizes range from ⅜-in. through the normal range of high pressure pipe sizes.

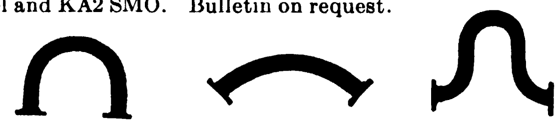
Arthur Harris & Co.

210–218 N. Aberdeen Street **Chicago 7, Ill.**

ENGINEERS AND BRONZE FOUNDERS—FABRICATORS OF NON-FERROUS METALS AND STAINLESS STEEL

Metals Fabricated—Aluminum, Block Tin, Brass, Bronze, Copper, Everdur, Monel, Nickel, Inconel, Stainless Steel and KA2 SMO. Bulletin on request.

Bends



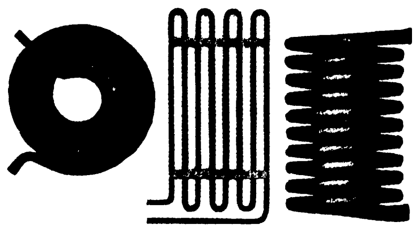
We make bends in every shape from all sizes of copper tube, pipe and tubing in copper, brass, aluminum, stainless steel, monel, tin and nickel. Standard or special connections. U-bends for storage water heaters.

Also special pipe work for industrial installations, plumbing, heating and brewing.

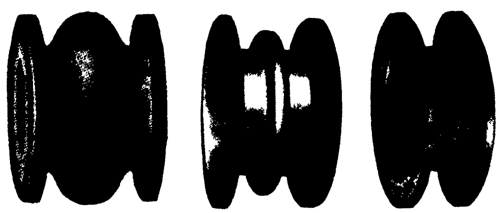
Non-Ferrous Castings—"Dairywhite" nickel silver for Process Industries Equipment. Suitable for milk and food products machinery. Castings also of 88-10-2. 80-10-10, 85-5-5-5, silicon bronze and manganese bronze, and special mixtures.

Coils

For heating, cooling and condensing. All shapes made from any size pipe or tube—standard or special connections, of copper, brass, aluminum, stainless steel, KA2 SMO, monel, inconel, nickel, block tin, and Everdur.



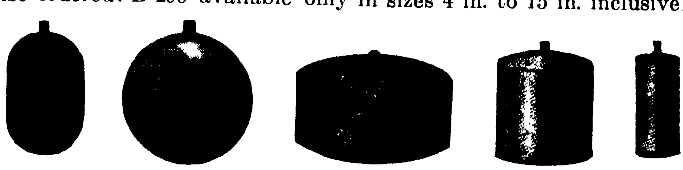
Copper Expansion Joints



B-280 Convex *B-290 Convex* *B-281 Concave*

For low pressure and vacuum. Made in two styles—convex and concave. Sizes 4 in. to 60 in. diameter. Cast iron or steel flanges. Flanges drilled to American standard unless otherwise ordered: B-290 available only in sizes 4 in. to 15 in. inclusive.

Metal Floats



Column *Ball* *Flat Cylindrical* *Cylindrical* *Cylindrical*

Made of copper, plain steel, copper plated steel, stainless steel, KA2 SMO, aluminum, brass, Monel, pure nickel, Admiralty and Everdur, for open tank and all pressures. Seamless copper ball floats carried in stock in diameters of 3 in., 4 in., 5 in., 6 in., 7 in., 8 in., 10 in., 12 in. for open tank and pressures of 25, 50, 100 and 150 lb. Floats in special sizes and pressures—made to order. Stainless steel ball floats 2½ in. to 12 in. for high pressure and corrosion carried in stock—special stainless steel floats made to order—stainless steel ball floats larger than 12 in. diameter can be made up specially. Float catalog sent on request.

Tube Turns, Inc.

TRADE MARK

General Offices and Factory: Louisville 1, Ky.

TRADE MARK

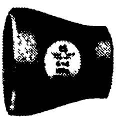
DISTRICT OFFICES:

NEW YORK, 150 Broadway............Rector 2-7844
PHILADELPHIA, Broad Street Station Bldg. Rittenhouse 6-0722
PITTSBURGH, 3001 Grant Bldg.......Atlantic 1-8848
CHICAGO, Suite 904, 600 S. Michigan Ave. Harrison 7-8526
TULSA, 420 Wright Bldg...............Phone 2-9193
HOUSTON, 1709–11 Commerce Bldg.....Charter 1668
LOS ANGELES, 2417 E. 24th Street . Jefferson 8257
SAN FRANCISCO, 2611 Russ Bldg. ... Garfield 1-2594

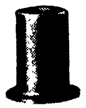
In Canada: Tube Turns of Canada, Limited, Chatham, Ontario

DISTRIBUTORS IN PRINCIPAL CITIES

In addition to carbon steel welding fittings listed here, the complete TUBE-TURN line embraces many alloys—types 304, 347 and 316 stainless, carbon moly and chrome moly steels, copper, aluminum, brass, Monel, Inconel, nickel and wrought iron. Dimensions and engineering data are included in TUBE-TURN catalog and Engineering Data Book No. 211, sent on request.

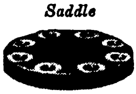

Concentric Reducer


Eccentric Reducer


Lap Joint Stub End

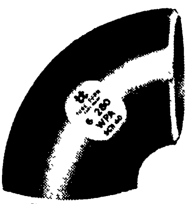

Cap

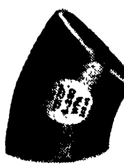

Saddle

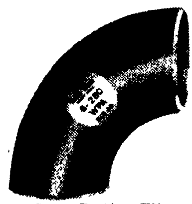

Blind Flange

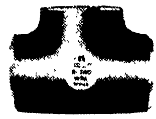

Welding Ring

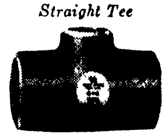

Lapped Flange


90° Short Radius Elbow

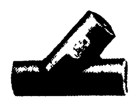

45° Long Radius Elbow


90° Long Radius Elbow


Straight Tee


Reducing Outlet Tee

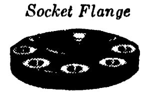

180° Long radius Return


Straight Lateral


Straight Cross


Welding Neck Flange


Socket Flange


Threaded Flange

TUBE-TURN SEAMLESS WELDING FITTINGS—RANGE OF SIZES

DESCRIPTION	STANDARD WEIGHT	EXTRA STRONG	SCHEDULE 160	DOUBLE EXTRA STRONG	LIGHT GAUGE NOMINAL PIPE SIZE	LIGHT GAUGE IRON PIPE SIZE
ELBOWS—90° Long Radius	½"-30"	½"-30"	1"-12"	¾"-8"	4"-24"	¾"-12"
ELBOWS—90° Short Radius	1"-30"	1½" 30"				
ELBOWS—45° Long Radius	½" 30"	½"-30"	1" 12"	¾"-8"	4" 24"	¾"-12"
RETURNS—180° Long Radius	½"-30"	½"-30"	1" 12"	2"-8"	4" 24"	¾"-12"
RETURNS—180° Short Radius	1" 30"	1½"-30"				
RETURNS—180° Extra Long Radius	1"-2½"	1"-2½"				
TEES—Straight	½"-24"	½" 24"	½"-12"	½"-8"		
TEES—Reducing Outlet	½"-24"	½"-24"	¾" 12"	¾"-8"		
REDUCERS—Concentric and Eccentric	¾" x ⅜" 24" x 16"	¾" x ⅜" 24" x 16"	¾" x ⅜" 12" x 5"	¾" x ½" 8" x 3½"		
CAPS	¾"-24"	¾"-24"	1"-12"	1"-8"		
STUB ENDS—Lap Joint	½"-24"	½"-24"				
SADDLES	2"-24"**					
LATERALS—Straight	1"-24"	1"-24"				
LATERALS—Reducing-on-run	1"-24"	1"-24"				
CROSSES—Straight and Reducing	¾"-24"	¾"-24"				
RINGS—Welding, Groove Type	¾"-24"	1"-24"		1¼"-8"		
RINGS—Welding, Ridge Type	¾"-12"	¾"-12"				
SLEEVES—Welding	2"-24"**					

** Since saddles and sleeves are for external reinforcement only, they do not conform to iron pipe size thicknesses.

TUBE-TURN FORGED STEEL FLANGES—RANGE OF SIZES

DESCRIPTION	150 LB.	300 LB.	400 LB.	600 LB.	900 LB.	1500 LB.	2500 LB.
WELDING NECK	½"-24"	½"-24"	½"-24"†	½"-24"	½"-24"*	½"-24"	½"-12"
SLIP-ON	½"-24"	½"-24"	½" 24"†	½"-24"	½"-24"*	½"-24"	½"-12"
LAP JOINT	½"-24"	½"-24"	½"-24"†	½"-24"	½"-24"*	½"-24"	½"-12"
THREADED	½"-24"	½"-24"	½"-24"†	½"-24"	½" 24"*	½"-12"	½"-12"
BLIND	½"-24"	½"-24"	½"-24"†	½"-24"	½"-24"*	½"-24"	½"-12"
SOCKET TYPE	¼"-24"	¼"-4"		¼"-3½"			
REDUCING—Threaded or Slip-On	¾"-24"	¾"-24"	¾"-24"†	¾"-24"	¾"-24"*	¾"-24"	¾"-12"
ORIFICE—Threaded		1"-24"	4"-12"	1"-12"	3"-12"	1"-12"	
ORIFICE—Slip-On		1"-24"					
ORIFICE—Welding Neck		1"-24"	4"-12"	1"-12"	3"-12"	1"-12"	

† Dimensions on sizes thru 3½" same as for 600 lb. flanges.
* Dimensions on sizes thru 2½" same as for 1500 lb. flanges.

"tt" and "TUBE-TURN" are trade marks of Tube Turns, Inc.

Alco Valve Company

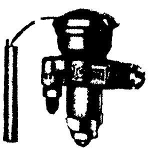
ENGINEERED REFRIGERANT CONTROLS

851 Kingsland Avenue, St. Louis 5, Mo.

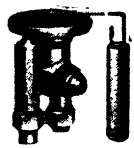
New York Office: 55 West 42nd Street   Chicago Office: 4534 North Broadway

THE COMPLETE LINE OF REFRIGERANT CONTROLS

ALCO THERMO EXPANSION VALVES: for automatic control of liquid refrigerant on all types of refrigeration and air conditioning systems. Capacities—from fractional tonnage to 100 tons Methyl Chloride, 50 tons "Freon-12." Low temperature valves for −40 F to −100 F.

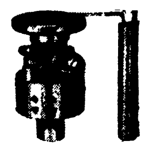


Type 402 with pressure limiting feature



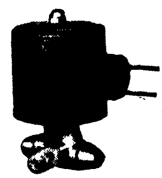
Type TK "3 valves in 1"

Type TCL

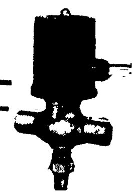


Type TR—Multi-Outlet

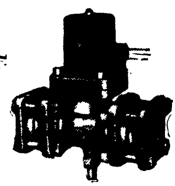
ALCO SOLENOID VALVES: for all types of service. For Liquid: "Freon"—up to 75 tons. Methyl Chloride—up to 150 tons. For Suction: "Freon"—up to 8.8 tons. Methyl Chloride—up to 17 tons. For brine, water, gas, air and steam.



Type S1

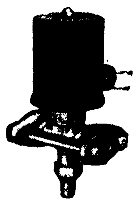


Type M3

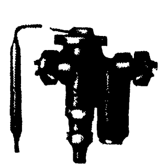


Type R2

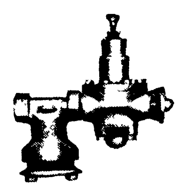
ALCO AMMONIA CONTROLS: Solenoid Liquid Valves—up to 172 tons. Solenoid Suction Valves—up to 28 tons. Thermo Expansion Valves—from fractional tonnage to 125 tons. Automatic Expansion Valves—from fractional tonnage to 60 tons.



Type M9F

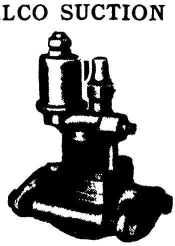


Type UG

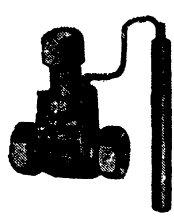


Type E with Strainer

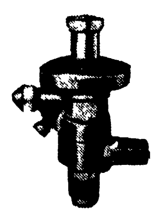
ALCO SUCTION LINE CONTROLS:



Type EPR—for all refrigerants, with connection sizes up to 6 in.



Type 732 SNAP-ACTION SUCTION VALVE—Temperature operated—½ ton, "Freon-12"—1 ton, Methyl Chloride



Type 760 "EVAPOTROL"—Pressure Regulator—½ ton, "Freon-12"—¾ ton, Methyl Chloride

ALCO ALSO MAKES: Constant Pressure Expansion Valves—Liquid and Suction Line Strainers.

Cam-Stat, Incorporated

Division of The Paul Henry Company

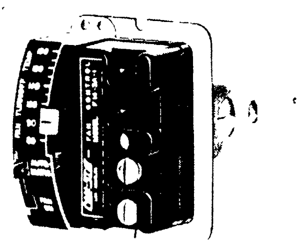
11831 W. Olympic Blvd. **Los Angeles 64, California**

District Offices in All Principal Cities

AUTOMATIC TEMPERATURE CONTROLS

These compact warm air furnace controls are available, as shown below, for integral mounting on the furnace, and also in conduit housings to meet *U.L.* requirements for plenum or duct mounting. Any one of the controls below is available in combination with any one of the other controls, complete in a conduit housing. Extreme sensitivity is obtained through the use of a low mass bimetal actuator. This insures uniformly accurate control results even with high rates of temperature change.

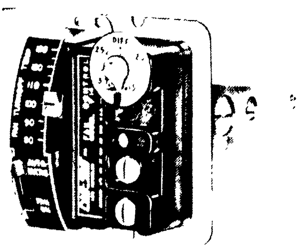
FAN CONTROL



Model No. F14-3A

Specification: Range—80 to 130 F Differential—15 F. (Fixed). Rating—½hp at 120 or 240 volts A.C. Equipped with manual summer fan switch.

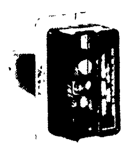
FAN CONTROL



Model No. F14-4A

Specifications: Range—80 to 130 F. Differential—15 to 30 F. (Adjustable). Rating—½ hp at 120–240 volts A.C. Equipped with manual summer fan switch.

FIXED LIMIT CONTROL



Model L22-7A

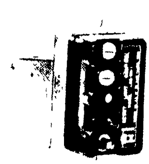
Specifications: Fixed cut-out and differential to manufacturer's requirements. Rating—90 volt-amps at 30 volts A.C.

ADJUSTABLE LIMIT CONTROL

Model L22-8A

Specifications: Range—170 to 200 F, or to manufacturer's requirements. Differential—Fixed anywhere between 15 and 50 F. Rating—90 volt-amps at 30 volts A.C.

RESET LIMIT CONTROL



Model No. L22-9A

Specifications: Fixed cut-out to manufacturer's requirements. Equipped with push button to reset after cut-out. Rating—90 volt-amps at 30 volts A.C.

COMBUSTION CONTROL CORPORATION

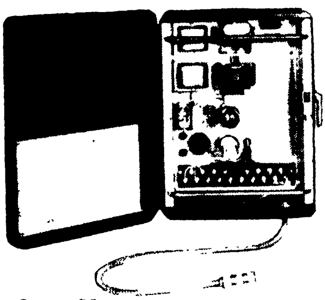
Flame Failure Safeguards (fireye) ***For Oil and Gas Flames***

77 Broadway, **Cambridge 42, Mass.**

New York—Philadelphia—Chicago—San Francisco—Washington, D. C.

District Offices in all Principal Cities

FIREYE FIRETRON FLAME FAILURE SAFEGUARD AND CONTROL FOR GAS AND OIL BURNERS—SERIES FP



Type 26SJ5 *Type 45PT1*

FIRETRON is a photo-conductive cell which "sees" all types of gas and oil flames, even relatively transparent flames. Fireye offers Systems FP-2 and FP-4, using the FIRETRON cell.

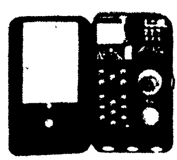
System FP-2 provides operating and starting protection for automatic gas, oil, and combination fuel burners. System FP-4 provides operating and starting protection for semi-automatically and manually ignited gas, oil, and combination fuel burners. Since the FIRETRON scanner supervises both pilot and main flames, a single scanner provides complete supervision.

Fireye equipment is designed with completely fail-safe circuits, whereby component failure results in immediate safety shutdown.

FIREYE PHOTOELECTRIC FLAME FAILURE SAFEGUARD AND PROGRAMMING CONTROL FOR OIL BURNERS—SERIES FF

Complete operating and starting protection for industrial and commercial oil burners with Flame Rod protection of gas pilot. Type 24PJ8 automatically starts burner and programs sequence of gas pilot, ignition, burner motor, oil valves, providing scavenging period, fuel valve delay, post ignition time. Flame Rod, Type 45JP1, monitors gas pilot flame, preventing opening of oil valve unless gas pilot is established. Scanner Type 45PH5 takes over monitoring of oil flame after pilot is established. Failure of either gas flame during ignition or main oil flame during normal operation results in immediate shutdown of burner system.

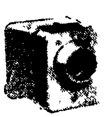
Type 45PH5



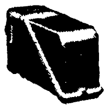
Type 24PJ8

Type 45JP1

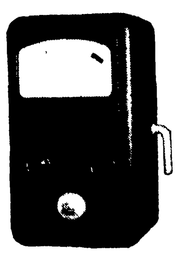
FIREYE SMOKE INDICATORS AND DETECTORS



Type 47FU1



Type 44DD4



Type 27LH6

Photoelectric Smoke Detection Systems available for indicating smoke density passing through stacks, and also for detecting smoke in air conditioning duct systems.

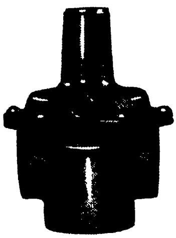
Photoelectric equipment for duct systems detects the presence of even small amounts of smoke. At the first sign of smoke, automatically turns off blowers, closes automatic louvres, and signals the maintenance department. Recommended for theatres, stores, hotels, and other locations where smoke is a hazard to property or a possible cause of panic.

Detroit Regulator Co.

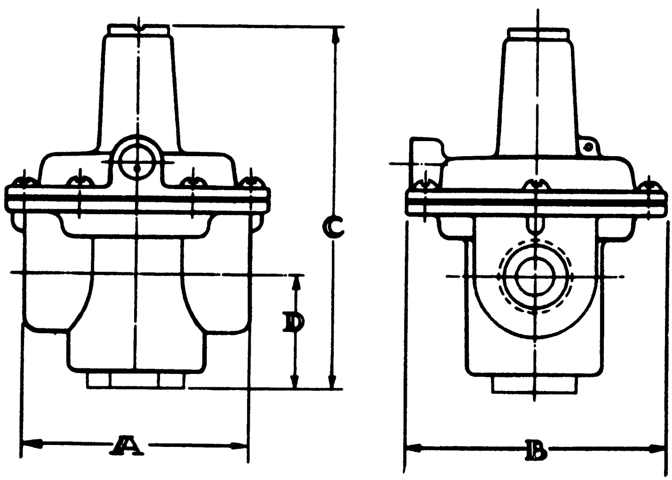
1742 Rivard Street **Detroit 7, Michigan**

Maxitrol LOW PRESSURE GAS REGULATORS

Pacific Coast Distributor; PACIFIC SCIENTIFIC CO.
San Francisco, Los Angeles, Seattle, Portland



Maxitrol gas regulators feature the patented "Straight-Thru-Flow" design principle. This construction sharply reduces pressure loss commonly encountered in conventional type gas pressure regulators. The new feature permits greater flexibility in design of manifolds for domestic equipment, and frequently allows a reduction in pipe size of gas manifolds. **Capacity charts at greater differential pressures for industrial use at inlet pressures up to 5 psi available on request.**



CAPACITY RATING (*A. G. A.* Listed)

Model No.	Pipe Size	Cu Ft/ Hr at 0.3 Pressure Drop 0.6 Sp. Gr. Gas	Btu/Hr Gases 800 Btu/ Cu Ft or More (Natural Gas)	Btu/Hr Gases Less Than 800 Btu/ Cu Ft (Mfd. Gas)
RV40	½	146	108,000	73,000
RV50	½	270	200,000	135,000
RV50	¾	270	200,000	135,000
RV51	¾	450	333,000	225,000
RV51	1	460	340,400	230,000
RV60	1	676	500,200	338,000
RV60	1¼	705	521,700	352,500
RV80	1¼	1,250	925,000	625,000
RV80	1½	1,260	932,400	630,000
RV90	2	2,030	1,502,000	1,015,000
RV90	2½	2,030	1,502,000	1,015,000
RV110	2½	4,200	3,108,000	2,100,000
RV110	3	4,900	3,626,000	2,450,000

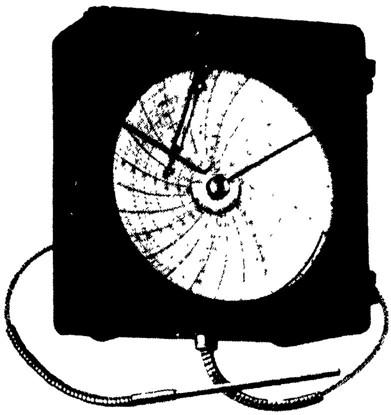
DIMENSIONS

Model No.	A	B	C	D	Shipping Weight Each
RV40	2½	2⅞	4¼	1⅜	⅝ #
RV50	3	3 11/16	5 7/16	1⅞	1 #
RV50	3	3 11/16	5 7/16	1⅞	1 #
RV51	3⅝	4 9/16	6 7/16	2 1/16	1¾ #
RV51	3⅝	4 9/16	6 7/16	2 1/16	1¾ #
RV60	4⅜	5 7/16	7⅛	2 5/16	2½ #
RV60	4⅜	5 7/16	7⅛	2 5/16	2½ #
RV80	6	7	8 5/16	2⅝	5 #
RV80	6	7	8 5/16	2⅝	5 #
RV90	7⅛	9⅛	9 11/16	2⅜	8¼ #
RV90	7⅛	9⅛	9 11/16	2⅜	9 #
RV110	9	12¾	14¼	3⅜	20 #
RV110	9	12¾	14¼	3⅜	20 #

The Electric Auto-Lite Company

INSTRUMENT AND GAUGE DIVISION
Department HV
TOLEDO 1, OHIO
NEW YORK • CHICAGO • SARNIA, ONTARIO

TEMPERATURE INDICATING & RECORDING THERMOMETERS



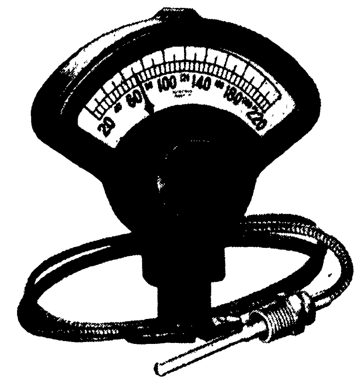
Model 500 Recording Thermometer

Temperature cycles are permanently charted by the Auto-Lite Model 500 Recorder. Precision-engineered for accuracy, it has legible 6 in. chart. Uniformly spaced subdivisions insure accurate readings. The movement is liquid-filled and responsive to changes throughout the temperature range. Head is compensated for temperature changes.

All recorders are enclosed in dustproof and moistureproof aluminum cast cases. Movements and all actuating parts made of non-ferrous metals. With double braided flexible armored capillary tubing of bronze composition or $\frac{3}{16}$ in. semi-rigid copper tubing where capillary is to be immersed in liquids. Standard chart ranges between minus 40°F and plus 550°F.

Choice of 24-hour or 7-day mechanical chart rotation. Complete with 100 charts, bottle of recorder ink, ink dropper. Wide choice of temperature ranges.

Model 500 is made in 3 standard types: WALL MOUNTING, with brackets for mounting; bottom connection. PORTABLE with spool-wound capillary, and strap handle. PORTABLE, SELF CONTAINED, with strap handle.



Model F-1 Indicating Thermometer

This thermometer is designed to facilitate systematic temperature observation for air conditioning, refrigeration or heating applications. It has large, easily read dial, evenly calibrated and fully compensated for temperature changes at the indicating head. Choice of temperature ranges between minus 40°F to plus 750°F.

Equipped with flexible capillary tubing for distance reading, or with rigid stem for direct mounting.

Auto-Lite's perfected one-to-one liquid filled movement eliminates delicate parts. Due to its strong construction, these instruments are particularly suited for installation on equipment where vibration is a factor.

Available with adjustable, electric alarm contact at small added cost.

Auto-Lite Indicating Thermometer, Model F-1, may be mounted in 3 positions by simple screw adjustment.

Send for illustrated Catalog describing styles and types of Auto-Lite Thermometers, including detailed information on dial and chart ranges available.

Fulton Sylphon Division

ROBERTSHAW-FULTON CONTROLS CO.

Knoxville 4, Tenn.

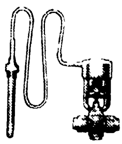
Sales Representatives in Principal Cities

Manufacturers of

Sylphon Automatic Temperature Controlling Instruments and Packless Expansion Joints

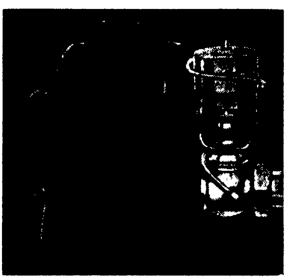
HOT WATER SUPPLY

No. 999



Temperature Regulator controls temperature of liquids. Particularly suited for storage water heaters and for all industrial processes requiring accurate temperature control. Stainless steel frame for minimum heat conduction. Large size Sylphon Bellows provides added power. Self-operating. Valve sizes from ½ in. to 4 in. Temperature ranges start at 40 F., up to 420 F. Bulletin HVG-A.

No. 902 Sylphon Thermostatic Water Mixers



Utilize hot water from any storage tank or instantaneous heater, and effectively regulate the amount of cold water required to temper it to the desired degree, actually mixing the hot and cold water together before delivery. Temperature remains constant in spite of fluctuations in supply water temperatures or pressures. Four sizes with capacities ranging from 5 to 131 gpm. Bulletin HVG-A

SPACE HEATING CONTROL

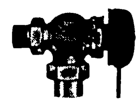
No. 885

Automatic Radiator Valve—

For exposed radiation. Small, neat, attractively finished, adjustable to room temperature desired. Simply replace ordinary radiator valves with these Sylphon Automatic Valves—no wiring, piping or auxiliary equipment is required. These valves answer the demand for an inexpensive means of providing accurate, dependable space temperature control in rooms, sections or throughout large buildings, new or old. Similar type valves for concealed radiation—get Bulletin HVG-80.

Sylphon No. 890

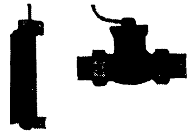
Radiator Control Valve



For either exposed or concealed radiation. Similar in appearance and action to other Sylphon Automatic Valves, but operated by an electric wall thermostat. The closing of the thermostat circuit energizes a low voltage electric heater coil surrounding a bulk containing a volatile liquid. This liquid expansion causes pressure on a bellows in the valve head operating the valve. This provides radiator valve control from a remote location, permits regulation of several radiators from a single thermostat, enables a time switch to be installed, if desired, offers effective zone control of large areas at a fraction of the cost of conventional motor-operated valve systems. Bulletin HVG-80.

Sylphon No. 7

Temperature Control



A self-contained, self-powered regulator for controlling unit heaters, wall or ceiling type radiators, heating coils in duct-type heating systems, etc. Quickly installed, holds temperatures within close limits. Valve placed in steam line to one or a battery of heaters, thermostat mounted on wall or column. For use on regular heating pressures up to 15 lb. Similar regulators, Nos. 7-2 and 7-3 for 50 and 75 lb. Pressure and temperatures up to 170 F. Bulletin HVG-50.

REFRIGERATION CONTROLS

No. 945-Z Regulator

Adaptable wherever brine is used as the refrigerant. Latest development is a "freeze-proof" valve (illustrated) on the popular Sylphon No. 945-Z Regulator. Bulletin HVG-A.

PACKLESS EXPANSION VALVES

No. 110-M Sylphon Expansion Joint

The Sylphon Packless Expansion Joint eliminates useless building height, expensive construction, non-revenue producing space. No leaks or repairs, no repacking, always tight; heating system operates at full efficiency. Bulletin HVG-65.

HEATING AND AIR CONDITIONING CONTROL

Almost any type of heating, ventilating or air conditioning system can be advantageously controlled wholly or in part by Sylphon Regulators. Basic advantages of Sylphon Controls are:

Modulating—Maintains ideal conditions—not continually correcting too hot, too cold, too humid or too dry conditions

Compensating—Many Sylphon Regulators offer compensating control, automatically raising their low limit setting at a predetermined rate as outside temperatures fall.

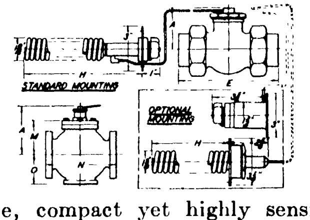
Sensitive—Close operating temperature differentials. Quick response.

Simple—in design.

Rugged Construction—To give years of satisfactory service.

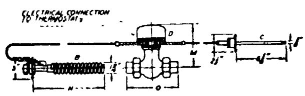
Adaptable—Any one of many combinations of Sylphon Instruments can be arranged to control any air conditioning system and to provide exactly the conditions desired. *Write for Bulletin SAC-850*

The No. 928-C Regulator



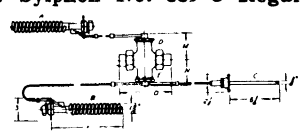
Simple, compact yet highly sensitive. Suitable for modulating control of air temperatures in ducts. Bulb is constructed of numerous coils of copper tubing giving sensitivity to the slightest temperature variation. Packless valve eliminates service problem and makes this regulator ideal for installation in inaccessible locations. Suitable for steam pressures up to 15 lb; other types available for pressures up to 75 lb.

The No. 928-ECC Sylphon Regulator



Room control and low-limit control in a single valve regulator for modulating control of ventilating systems. Main control from an electric room thermostat operating through the electric head "D" on the valve. Low-limit control by Bulb "B" located in discharge duct from the heater. Bulb "C", located in inlet side of the duct to the heater, compensates Bulb "B." Compensating thermostat can be furnished to raise low-limit setting at predetermined rate with falling outside temperature. Suitable for steam pressures up to 15 lb.

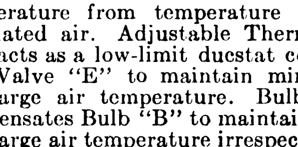
The Sylphon No. 889-C Regulator



A modulating, dual-function regulator for control of duct heating and ventilating systems—two independent valves in a single body.

Adjustable Thermostat "A" governing Valve "D" functions to maintain room temperature from temperature of recirculated air. Adjustable Thermostat "B" acts as a low-limit ducstat controlling Valve "E" to maintain minimum discharge air temperature. Bulb "C" compensates Bulb "B" to maintain even discharge air temperature irrespective of demand. Compensated Thermostat "B" can also be furnished to raise its setting at a predetermined rate with falling fresh air temperatures if desired. Suitable for steam pressures up to 15 lb.

The Sylphon No. 889-C7 Regulator



The No. 889-C7 regulator has a wall type adjustable thermostat that is placed in the room or space to be controlled. This thermostat "A" operates upper half of the regulating valve. Otherwise, same as No. 889-C regulator described above.

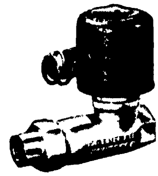
GENERAL CONTROLS

801 ALLEN AVENUE **GLENDALE 1, CALIF.**

Manufacturers of Automatic Pressure, Temperature, Level & Flow Controls

FACTORY BRANCHES: Baltimore 5, Birmingham 3, Boston 16, Buffalo 3, Chicago 5, Cleveland 15, Columbus 15, Dallas 2, Denver 4, Detroit 21, El Paso, Glendale 1, Houston 6, Indianapolis 4, Kansas City 2, Milwaukee 3, Minneapolis 2, Newark 6, New Orleans, New York 17, Omaha 2, Philadelphia 23, Pittsburgh 22, Salt Lake City 4, St. Louis 3, San Francisco 7, Seattle 1, Tulsa 6, Washington 6, D.C. DISTRIBUTORS IN PRINCIPAL CITIES.

(A) Type T-70



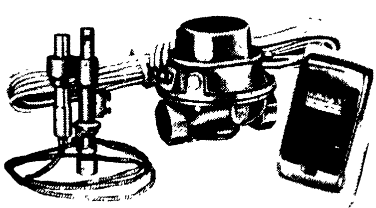
(B) Type K-10

(A) THERMOSTATS

Compact, snap-action, T-70 Thermostat. Functional beauty for accurate, remote control of desired temperature. Extends only ⅞ in. from wall. Streamlined stainless cover, sensitive to slightest temperature change, ivory plastic base.

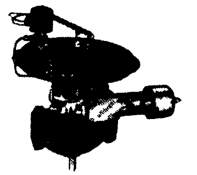
(B) MAGNETIC VALVES

Provides six times more power than ordinary solenoid valves. Controls air, gas, water, light, heavy oils, steam. Positive opening, complete shut-off, packless, hum-free. Available for any voltage, a.c. or d.c., in sizes up to 1¼ in. I.P.S., port sizes up to ⅝ in.



(C) BX-69 GAS ACTUATED PACKAGE SETS

No outside current required. Operates on all types of gases. Safe, quiet, dependable. For all gas-heating appliances. Set consists of a PG-9 500 millivolt pilot generator, a B-60 gas control valve, a T-70 snap-action thermostat, thermostatic cable and vent tubing. Everything needed in convenient package for remote gas control.



(D) Type B-55-G

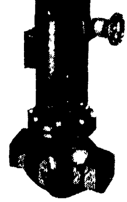
(E) Type K-3-B

(D) SLOW OPENING GAS VALVES

New combustion control afforded by these diaphragm-controlled gas valves. Governor regulates fuel supply to burner in direct ratio to steam pressure, eliminating hunting aspect. Available 1 in. to 6 in. I.P.S.

(E) MAGNETIC GAS VALVES

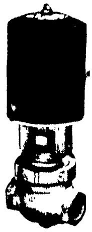
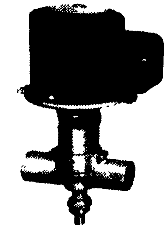
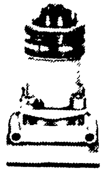
Versatile, two-wire, straight magnetic current-failure valve. Packless. Insures tight shut-off indefinitely. Humless. Size range, ⅜ in. to 6 in. I.P.S. Operating pressures up to 10 lb. Voltages and frequencies a.c. or d.c. Quiet, positive, troublefree. Available in explosion-proof housing.



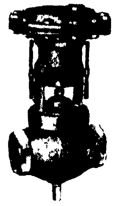
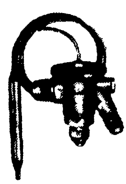
(F) Type V-110

(F) MANUAL RESET VALVES

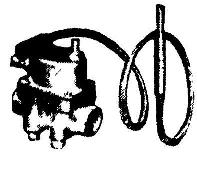
Equipped with manually-reset electromagnetically-held valve operator. Current flowing to operator permits manual opening by turning valve wheel at side. Current failure releases operator allowing valve to close. Trip-free mechanism cannot be opened under unsafe conditions. Once closed, valve must be reopened manually.

(I) Type PV (AV)-3

(K) Type V-25-6 (L) Type V-200



(M) Type MR-2

(G) Type G-1-7 (H) Type K-15

(G) HYDROMOTOR VALVES

Simplify valve control installations. Two-wire, current failure, electric-hydraulic operation. Ample motor-driven power, slow opening and closing movement. G-1 Series, designed for low-pressure steam, hot and chilled water circulating systems. G-2 and G-3 Series for high pressure applications.

(H) REFRIGERANT CONTROLS

Magnetic piloted, two-wire current failure, high pressure, packless. Handle large capacities with minimum pressure drop and loss. Tight shut-off. Operates on air, steam, water, and refrigerants.

(I)*hi-g MAGNETIC VALVES

Designed for positive operation on aircraft, trucks, tractors, tanks, graders, ships, and other moving equipment. Handle all fluids, vapors and gases on anything that rolls, floats or flies at pressures up to 3000 lb or more. Packless, two-wire, current-failure type, available normally open, normally closed for intermittent or continuous duty.

(J) Type RS-100

(J) RELAYS AND TRANSFORMERS

Type RS-100 handles single phase motor loads up to 1 hp or heating loads up to 1.1 kw. Combines double-break relay and integral transformer. Normally open; large double-break contacts. Two-wire control circuit; maximum holding current 0.4 amps. Furnished with ½ in. conduit connections and low voltage outlet. a.c. only.

(K) GAS FUEL GOVERNORS

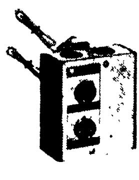
Throttle gas lines according to boiler pressure applied to diaphragm. Ball bearing thrust adjustment, ground and polished non-corrosive stems, low friction packing gland seal, multiple calibrated springs, high lift for maximum capacity. Suitable for butane, natural or manufactured gas. Available ⅜ in. to 3 in., I.P.S.

(L) THERMAL EXPANSION VALVES

Type V-200 with new selective capacity cartridge provides instant sizing adjustment. Only one valve required for full capacity range in each body size at all back pressure or suction temperature ranges. For Freon, Methyl Chloride or Sulphur Dioxide.

(M) THERMOPILOT (Valve Model)

Manually-reset, electromagnetically-held-open valve with current generated by single couple subject to heat of pilot flame. Available ⅜ in. to 1½ in. I.P.S.




(N) Type L-54 *(O) Type V-300*

(N) FAN AND LIMIT CONTROL

Combination. Fan and Safety Limit Control incorporates two separate switch units, one acting as a fan control; the other for safety limit operation. Each switch makes and breaks its own circuit independently. External adjusting knob on fan switch may be turned to "fan summer" position for manual control of fan for summer ventilation.

(O) LOW PRESSURE GAS REGULATORS

New V-300 Series are reliable, trouble-free valves with high capacity, close regulation, yet small and compact. Regulator size range from ⅜ to 6 in. I.P.S. Internal parts, corrosion-resistant.

* Trade-mark—"hi-g" indicates positive ability to function in any position, regardless of vibration, change of motion or acceleration.

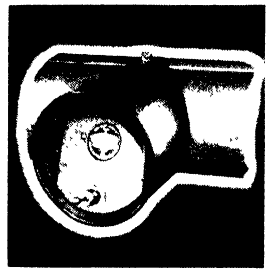
Field Control Division

of H. D. Conkey & Company, Mendota, Ill.

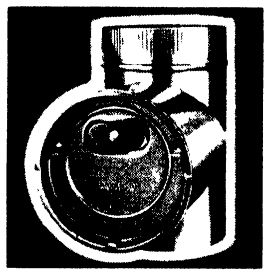
Manufacturers of **FIELD DRAFT CONTROLS**

FIELD BAROMETRIC DRAFT CONTROLS

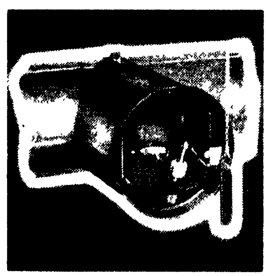
FIELD TYPE M: For automatic heating equipment, designed to assure finer performance, greater fuel economy, through highly accurate control of drafts. Available in sizes from 6 in. through 24 in. for pipe diameters of 6 in. through 25 in. Features "Rocking Chair" gate action, off-center gate mounting, sidewings, extended housing. Widely used in the heating industry.



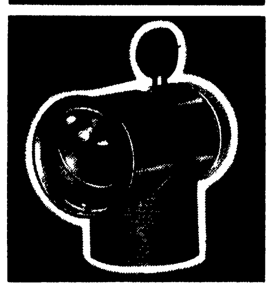
FIELD SCOTTY: Available in pre-set or adjustable models, for use on space heaters and ranges. Adjustable model is adjustable for high, medium or low draft. Pre-set model is pre-set at factory to manufacturers specifications. 6 in. and 6–7 in. sizes for space heaters and ranges with 6 in. or 6–7 in. outlets. For horizontal or vertical installation. 26 gauge Tee, and stub, 24 gauge ring. The choice of leading manufacturers.



FIELD BAROCHEK: Combination barometric draft control and check damper for use on hand-fired furnaces. Will provide fully automatic control of stack drafts, or can be manually checked in open position. Also ideal as part of a damper motor set. Available in 7 in. through 24 in. sizes for 7 in. through 25 in. pipe diameters. Reduces fire hazard, cuts fuel consumption, reduces furnace tending and furnace wear.



FIELD SCOTTY-W: For hot water heaters. 6 in. Tee for 6 in. outlets. 26 gauge steel throughout. Pre-set at factory to manufacturer's specifications. 2 in. diameter lighting opening with cover on Tee. Collar, from heater to tee, $3\frac{3}{8}$ in. Tee length, 11 in. Control Tee to collar, 3 in. A precision made Field Control.

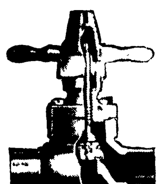


Henry Valve Company

MELROSE PARK, ILLINOIS
(A Chicago Suburb)

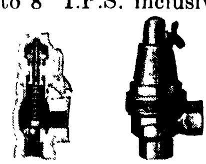
HENRY PRODUCTS FOR REFRIGERATION, AIR CONDITIONING, AND INDUSTRIAL APPLICATIONS: CONTROL DEVICES, VALVES, STRAINERS DRIERS, FITTINGS, AND ACCESSORIES.

Balanced-Action Diaphragm Packless Valves
STANDARD TYPE Two-way, branch shut off, and angle types—flare or solder connections. Hand expansion, purge and charging types also available. Forged brass body and bonnet, ports-in-line, non-directional. *Back seat and ball check permit diaphragm inspection and replacement under pressure.* Stock sizes ¼″ thru ⅝″ S.A.E.; ¼″ thru 1⅝″ O.D. solder; ¼″ thru 1″ F.P.T.
BLUE BANTAM TYPE—Two-way line shut off valves, flare or solder connections. Contain same field proven features as STANDARD line except that diaphragms cannot be inspected or replaced while valves are under pressure. Stock sizes ¼″ thru ⅝″ S.A.E. and solder.



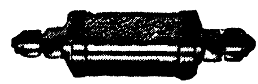
WING CAP PACKED VALVES Bronze with solder connections in globe and angle types, ⅞″ thru 5⅛″ O.D.; semi-steel with F.P.T. connections in globe and angle types, sizes ½″ thru 2″; semi-steel with bolted bonnets and square companion flanges with brass tailpieces for O.D.S., 1⅝″ thru 5⅛″. Also available with steel tailpieces for welding to pipe 1½″ to 8″ I.P.S. inclusive.

Ferrous Type



Diaphragm Type

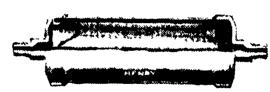
RELIEF VALVES
FERROUS TYPE—For ammonia. Approved under safety codes.
DIAPHRAGM TYPE—For low-pressure refrigerants. Approved under safety codes.
DRIERS Filled with silica gel—other dehydrants on special order.



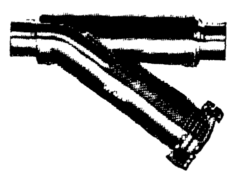
TYPE 705—Has two inches more dehydrant capacity than established practice. Brass shell, forged brass end caps with integral fittings. Dehydrant capacity 8 to 32 cu in. Sizes, ¼″ thru ½″ flare.

TYPES 748, 756 and 757 cartridge driers with side outlet, dispersion tube, safety cylinder, cartridge retaining spring and distortion-proof access flange. Dehydrant capacity 12 to 500 cu in. Sizes ⅜″ thru 2⅛″ O.D. solder.

STRAINERS



TYPES 891 and 892. Screen area 11 and 25.5 sq in. respectively. Sizes ¼″ thru ⅝″ flare and ⅜″ thru ⅝″ O.D. solder.



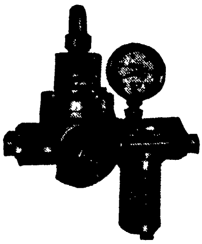
TYPE 895 steel "Y" strainer, forged brass end caps. Distortion-proof access flange. Screen area 23 to 150 sq in. Sizes ⅝″ thru 4⅛″ O.D. solder; 1″ to 3″ FPT.

Sold by refrigeration jobbers. Write for Free Catalog.

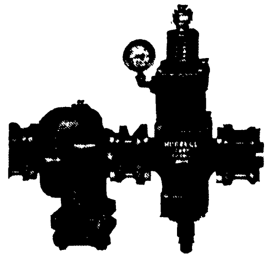
Hubbell Corporation

P. O. Box 700, Hawley Road
Mundelein, Ill.

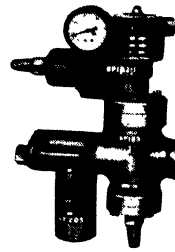
Designers and Manufacturers of Automatic Control Valves For All Refrigerants



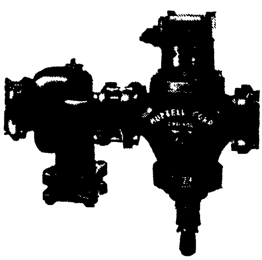
Type SA-5, SF-5



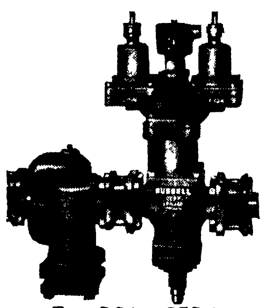
Type SA-6, SF-6



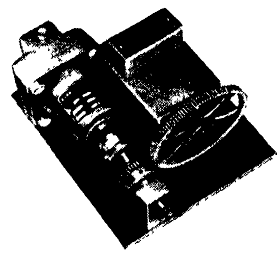
Type SA-7, SF-7



Type "T"



Type DSA-9, SFD-9



Type SAC-6, SFC-6

Type SA-5, SA-6, SF-5 and SF-6 Back Pressure Regulating Valves are of the conventional type used to maintain a constant evaporator pressure.

Type SA-7, SA-8, SF-7 and SF-8 Combination Back Pressure Regulator and Stop Valves. This regulator is of the conventional type used to maintain a constant evaporator pressure and the addition of a small electric pilot valve built into the head, makes it a suction stop valve.

The DSA-9 and DSF-9 is a dual regulator which will control evaporators with two load conditions requiring different refrigerant temperatures. The diaphragms in the dual head may be set for any two evaporator pressures and will automatically change from one to the other by the opening or closing of the electric pilot valve.

The SAC-6 and SFC-6 valves are of the compensating type and are used where a constant temperature is desired in the medium being cooled. These valves will increase or decrease the evaporator pressure to compensate for the increase or decrease of the cooling load. Operated with air or electricity.

The Type "T" suction stop valve is used where automatic suction line control is required. It is operated by high pressure gas and its construction makes a tight closing valve and its dependability far surpasses the conventional magnetic stop valve.

All valves have built-in opening stems, which eliminates the by-passes usually used for manual operation. With either screwed, welding type flanges or copper tube connections, in sizes from ¾ in. to 8 in. inclusive.

Solenoid Valves from ⅜ in. to 2 in. inclusive for liquids and gases with composition seat discs readily renewable, coils for any electrical characteristics, built-in lifting stem standard, with either screwed, welding flanges or copper tube connections.

Strainers in all sizes for liquid and gas with very large screen areas and arranged to bolt directly to valve or with screwed, welding flange or copper tube connections for installation wherever strainer is necessary.

Write for complete information on these and our many additional Refrigeration Controls and Accessories.

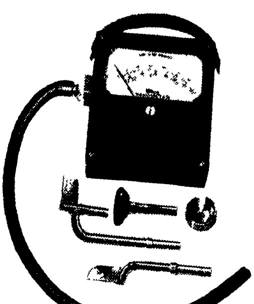
Illinois Testing Laboratories, Inc.

Room 516, 420 N. La Salle St., Chicago 10, Ill.

Alnor

Precision Instruments for *Every* Industry

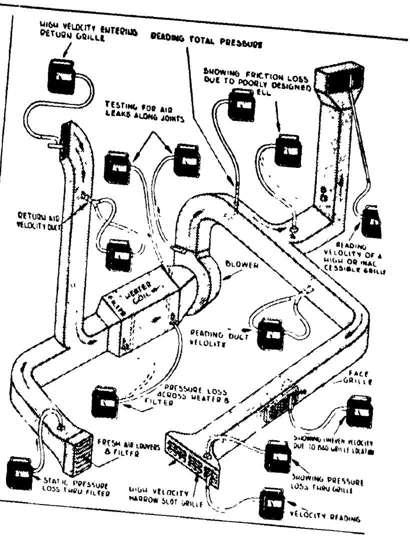
The direct-reading ALNOR VELOMETER

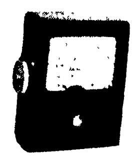


TYPE 4-F Standard minimum set for heating and air conditioning air velocity measurements.

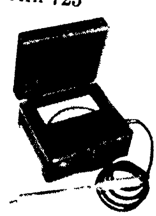
The Alnor Velometer is an instantaneous, direct-reading air velocity meter designed for convenient, rapid determination of air velocities in air conditioning, heating and ventilating, and exhaust systems. It gives instantaneous direct readings in feet per minute, without timing, calculations, or reference to tables or charts. Accurate information on performance of equipment, duct systems, etc., can be obtained with a few moments inspection with the VELOMETER. It can be effectively used to locate drafts and leaks around windows and doors, or in duct systems.

The Alnor Velometer is built in several standard ranges from 20 fpm to 6000 fpm, and up to 3 in. static or total pressure. Special ranges available as low as 10 fpm and up to 25,000 fpm velocity and 20 in. pressure.





Alnor Velometer, Jr. A miniature, direct reading Velometer—4 in. high, 3 in. wide, 1-½ in. deep. Weight, 8 oz. Accurate, strong. Available in single and double scale ranges: 0-200 to 0-2500 fpm. Bulletin 725



Alnor Thermo-Anemometer. For accurate measurement of low air velocity. Compact, direct-reading, battery operated, self-contained, portable. Scale 6 in. Meter ranges: 0-600 fpm and double range 0-300/100-2000 fpm. Accurate readings as low as 5 fpm. Temporary Bulletin 913-A.

Johnson Service Company

AUTOMATIC TEMPERATURE AND AIR CONDITIONING CONTROL

General Office and Factory **Milwaukee, Wis.**

Direct Branch Offices in Principal Cities

Johnson Temperature Regulating Co. of Canada, Ltd., 3615 Danforth Ave., Toronto 13, Ont.
Halifax, N. S. Montreal, Que. Winnipeg, Man. Calgary, Alta. Vancouver, B. C.

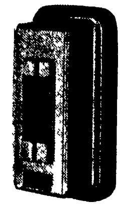
PRODUCTS AND SERVICES

Manufacturers, Engineers and Contractors for automatic temperature and humidity control systems applied to all types of heating, cooling, ventilating, air conditioning and industrial processing installations.

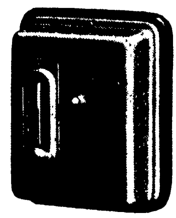
Space Control—Automatic control of room temperatures and humidities, applied to convectors, radiators, radiant heating, unit ventilators, unit heaters and heat delivery ducts. Also, Johnson "Duo-Stats" to maintain proper relationship between outdoor and heating system temperatures for groups of radiators or "heating zones." A complete line of controllers for air conditioning systems, heating, cooling, humidifying, dehumidifying.

Process Control—Automatic temperature and humidity control for every range required in manufacturing and industrial processing. Thermostats, valves and dampers applied to tanks, dryers, vats, kettles, curing rooms, coolers, kilns, etc., in textile, rubber, pulp and paper, petroleum refining, meat packing, dairying, baking, sugar refining, brewing and distilling, tanning, candy making and other industries.

Nation-wide Service—Johnson sales engineers, and trained installation men available at all branches listed above. None is an agent, jobber, or part-time representative. All are salaried employees, devoting their efforts to the interests of the Johnson Service Company and its customers. **Send for Bulletins.**



Single Room Thermostat T-400



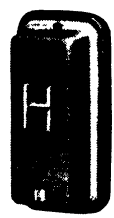
"Dual" Room Thermostat T-460

JOHNSON THERMOSTATS

Room Thermostats—Proportional (gradual) or two-position (positive) action, maintaining temperatures within one degree above or below point of setting. Various covers allow wide selection of adjusting features, guards and mounting method. Red-reading thermometers with magnifying tube attached to covers.

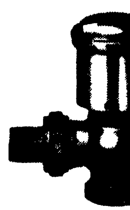
Insertion and Immersion Thermostats—Rigid stem or capillary. Liquid-filled capillary systems for temperatures which are measured at point remote from location of operating mechanism. Various types of bulbs. Standard connecting tubing 8 ft long; 15, 25, 35 or 50 ft on special order.

Thermometers—High grade insertion or immersion thermometers to measure temperatures in ducts, tanks, etc., with red-reading mercury column in heavy lens glass tube and 9-in. scale. Insertion thermometers have patented adjustable tilting feature. Dial thermometers with liquid-filled capillary elements.

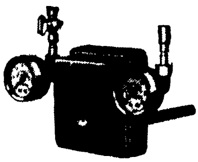


Room Humidostat H-107

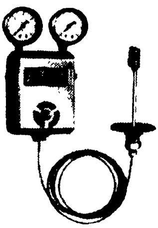
Special Controllers—For applications in industrial processes. "Record-O-Stat," combination capillary temperature controller and recorder. 12-in. chart and liquid-filled capillary systems. Single or duplex type, the latter controlling and recording wet and dry bulb temperatures. **Pressure Regulators**—Pressure ranges 30 in. of vacuum to 250 psi. pressure. Types and sizes for required pressure range and for medium to be controlled: Air, water, steam, or freon. **Liquid Level Regulators** (Float type)—Control within extremely close limits. Mounted through wall of containing vessel by stem with 1 in. pipe thread. Floats of copper, stainless steel or special alloys. **Static Pressure Regulator**—Measures variations in pressure from .009 in. to 3 in. of water. Also used as differential regulator, measuring difference in pressure between two chambers.



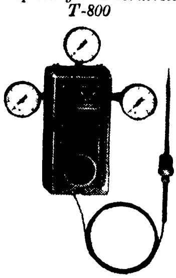
Piston Valve for Convectors V-160



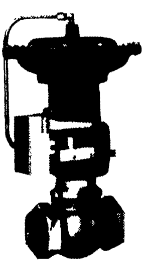
Rigid Steam Insertion Thermostat T-802



Capillary Thermostat T-800



Sub-Master Capillary Thermostat T-901



V-103 Globe Valve with Pilot Positioner

Sub-Master Thermostats—An important development for industrial applications and air conditioning. Available in various types of controllers where readjustment must be made from a remote point.

Johnson Sensitivity Adjustment—A distinctive feature affording convenient means of adjusting the sensitivity of thermostats and humidostats, on the job, balancing "time-lag" with respect to capacity of conditioning apparatus. "Hunting" and temperature fluctuations prevented. Available on Johnson proportional action insertion and immersion thermostats, insertion humidostats, capillary thermostats, pressure regulators and certain room thermostats and humidostats.

JOHNSON HUMIDITY CONTROL

Johnson Humidostats—Automatically control supply of moisture delivered to air by a humidifier or other means, maintaining constant relative humidity. Available in room and insertion patterns with various elements, the most sensitive controlling within 1 per cent at relative humidities as high as 95 per cent at 100 F. Humidostatic elements are wood cylinder, by-wood strip, bow-wood, horn, hair or animal membrane.

Johnson Humidifiers—"Steam grid" type -perforated pipe supplied with low pressure steam) or pan type with copper evaporating pan, brass heating coils and float control.

JOHNSON VALVES

Johnson Diaphragm Valves—Simple, rugged. Diaphragms of special molded rubber, resistant to age and oxidation, operate valve stems against pressure of dependable springs. Available also with Sylphon seamless metal bellows. In standard sizes and patterns including compact valves for convectors and unit conditioners. Normally open (direct acting) or normally closed (reverse acting). Three-way mixing and by-pass valves, for steam, water, brine and other gases and liquids.

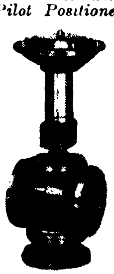
Johnson "Streamline" Diaphragm Valves—With modulating discs and special internal construction. Superior proportional control. Where maximum power is required for repositioning at slightest demand of controlling instruments, the larger molded rubber diaphragm valves are fitted with pilot positioner, independent of friction and pressure variations.

JOHNSON DAMPER AND SWITCHES

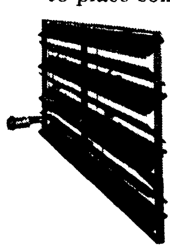
Standard Johnson Dampers—Galvanized blades in flat steel frames with adequate bracing to form rigid assembly. Black lacquer or special corrosion-resisting finishes. Angle iron frames optional. **Special Dampers**—Galvanized iron frames and monel metal, aluminum, copper or rust-resisting steel blades on order. Frames of same material as blades, if desired. Brass pins in steel bearings or ball bearings.

Johnson Damper Operators—Similar in principle to valves. Seamless metal bellows or specially molded rubber diaphragm operates damper through suitable linkage. Johnson "Piston" damper operators afford long travel at full power. With or without pilot mechanism, as described for "Valves."

Johnson Pneumatic Switches—For operation of dampers and to place controllers in and out of service, from remote points. "Masonite" is standard. Ebony, asbestos, steel, polished oak, and genuine or imitation marble on order. Various apparatus is mounted on special switchboards, including gradual, lever type and multiple-step switches, clocks, air pressure gauges, recording gauges, etc.



3-Way Mixing Valve) (Rubber Diaphragm V-108



Proportioning Louver Damper D-226

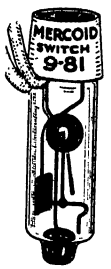
Piston Damper Operator D-251

The Mercoid Corporation

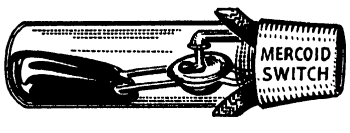
Main Office and Factory, 4201 Belmont Avenue, Chicago 41, Illinois
New York Office, 205 East 42nd St. Philadelphia Office, 3137 N. Broad St.

AUTOMATIC CONTROLS FOR HEATING, AIR CONDITIONING, REFRIGERATION AND VARIOUS INDUSTRIAL APPLICATIONS

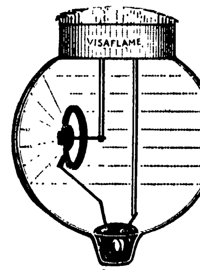
MERCURY SWITCHES BY MERCOID



Magnetic Type



Tilting Type

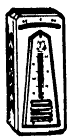


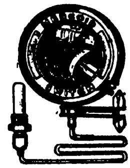
Light Actuated Type

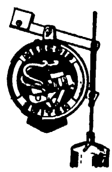
There are three primary operating types of Mercoid Mercury Switches:—1. MAGNETIC TYPE, applied in a stationary position. Circuit is opened or closed with magnetic attraction by means of a permanent magnet or a small milli-ampere electromagnet. 2. TILTING TYPE, a variety of sizes, electrical capacities and arrangements available. 3. LIGHT ACTUATED TYPE. These switches are immune to dust, dirt or corrosion.

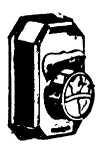
THE 100% MERCURY SWITCH EQUIPPED CONTROLS

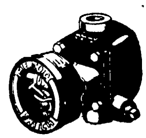

No. 1


No. 2


No. 3


No. 4


No. 5


No. 6


No. 7


No. 8

No. 1. Pressure Controls Noted for their accuracy and dependable performance. The outside double adjustment feature and visible dial eliminate all guesswork when setting the operating range. Various ranges and circuits available.

No. 2. Low Voltage Thermostats Mercoid Sensatherms operate on a total differential of 1 degree F. Type H is the popular room thermostat. Type DNH is a hand wound day and night thermostat. Type HBH is a two-stage thermostat for control of high-low gas or oil burners.

No. 3. Temperature Controls Available with or without remote connection for use with liquids or gases such as air, oil, water or distillate vapors. Equipped with a Bourdon tube and outside double adjustments.

No. 4. Lever Arm And Float Controls The lever arm type is used where it is desired to mechanically open and close electric circuits. The float type is used to maintain fluid levels in tanks, or for control of sump pumps, etc.

No. 5. Line Voltage Thermostats No. 855 thermostats will directly handle the full motor load without the use of a relay. Available with "on-off" manual switch for unit heater applications.

No. 6. Low Water Controls Available as a combination pressure and low water control or as a low water control only. May be furnished with quick-hook up fittings designed in accordance with the *ASME* code.

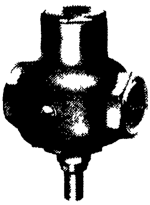
No. 7. Transformer-Relays Type V is a reliable low voltage mercury contact relay which also acts as a transformer inducing low voltage (24 volts) on the pilot circuit. Available in various voltages, cycles and circuits.

No. 8. Visaflame Control For domestic and industrial oil burners. Operates direct from the light of the flame instead of from the heat in the stack. It may be built into the burner unit.

Milwaukee Gas Specialty Company

730 North Jackson St., **Milwaukee 2, Wisconsin**

BASO* THERMOELECTRIC SAFETY PILOTS

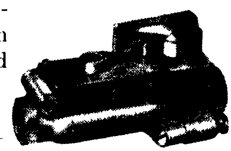


A 814

Straight through valves with one pilot tapping each side. Cast, heat treated aluminum alloy bodies with replaceable electromagnetic hood assemblies and reset assemblies. 100 per cent shutoff safe lighting.

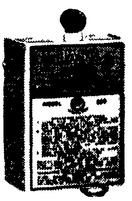
Model	Inlet	Outlet	Pilot Tapping	Capacity Btu/hr.	Thermocouple Type
A814-1	½" F.P.T.	½" F.P.T.	⅛" F.P.T.	80,000	88D
A814-2	⅜" F.P.T.	⅜" F.P.T.	⅛" F.P.T.	101,000	88D
A814-3	¾" F.P.T.	¾" F.P.T.	⅛" F.P.T.	130,000	88D
A505-1	1" F.P.T.	1" F.P.T.	⅛" F.P.T.	280,000	58D
A506-1	1¼" F.P.T.	1¼" F.P.T.	⅛" F.P.T.	329,000	58D

Straight through low height valves with built in plug cock for main burner and pilot burner control. 100 per cent shut-off and safe lighting. With or without pilot adjustment on either side. Model 841 has special plug head for single rod control of floor furnaces. Uses 88D thermocouple lead.



840

Model	Inlet	Outlet	Pilot Tapping	Capacity
840-1	½" F.P.T.	½" F.P.T.	¼" C.C.	85,000
841-1	½" F.P.T.	½" F.P.T.	¼" C.C.	85,000
A812	¾" F.P.T.	¾" F.P.T.	¼" C.C.	165,000

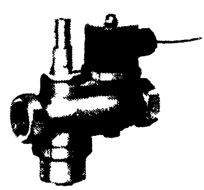


850 Switch

Electrical Rating:
.3 Amps at 250 A.C.
.06 Amps at 230 Volts D.C.
Uses 88D thermocouple lead

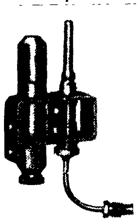
Switch Type Baso is wired in series with automatic gas valve and other controls. Opens circuit to main valve in case of pilot failure. Main burner cannot be lighted until pilot is operating and switch held in closed position.

Basoid valves include in one cast brass body a solenoid valve for automatic main gas control and a 100 per cent shutoff safety pilot unit similar to Models A814 or A505. Available in many current types. Two pilot tappings. Series C Basoid in ¾ in. size only.



Basoid

Model	Inlet	Outlet	Pilot Tapping	Current Type	Capacity	Couple Lead Type
B4411	¾" F.P.T.	¾" F.P.T.	⅛" F.P.T.	115V, AC	159,000	58D
B4511	1" F.P.T.	1" F.P.T.	⅛" F.P.T.	115V, AC	189,000	58D
B4451-B1	¾" F.P.T.	¾" F.P.T.	⅛" F.P.T.	20V, AC	159,000	58D
B4551-B1	1" F.P.T.	1" F.P.T.	⅛" F.P.T.	20V, AC	189,000	58D
C4411	¾" F.P.T.	¾" F.P.T.	⅛" F.P.T.	115V, AC	94,000	88D



ANNULAIR PILOT BURNERS are available in a variety of tips and dual orifice inlet fittings for all kinds of fuel gas and pilot line connections. They are made in four types, "B" for low consumption appliances, "C" for medium consumption, and "D" and "F" for larger capacity appliances.
Thermocouple leads of the 88D and 58D types are available in standard lengths from 12 in. to 72 in. and in special lengths to 240 in.

* Reg. U. S. Pat. Off.

Write for complete catalog SV-300-3.

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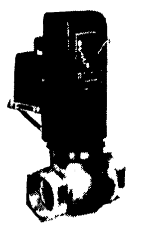
Each Honeywell branch office maintains a staff of experienced factory-trained engineers who are qualified to give unbiased advice on control applications and to install and service all types of control equipment. They are prepared to assist in the writing of specifications and to furnish control layouts and cost estimates without charge.



Modutrol Valve

ELECTRIC CONTROL

Honeywell electric controls are noted for variety, versatility, dependability and precision operation. The trade mark "Modutrol" is used to designate Honeywell electric control systems designed for air conditioning or heating applications (other than domestic). It is your guarantee of Honeywell quality. A wide variety of both modulating and two-position motors, controllers and valves provide a flexible selection of control equipment.

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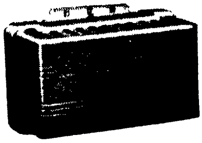
Now, because of the many special features of Honeywell *Electronic* Control, it is easy to achieve results which previously would have been extremely difficult or impossible. *Electronic* controls are super-sensitive and accurate. They are simple in operation and very flexible in application. Honeywell *electronic* thermostats have no moving parts. A single one can be used to control heating, ventilating and cooling. The control settings can be maintained constantly or may be reset automatically. An almost unlimited number of averaging and compensating controls can be used to obtain practically any desired result.



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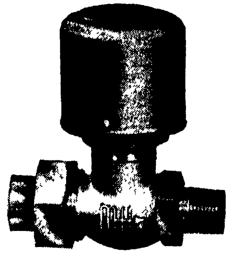
PNEUMATIC CONTROL

The "Gradutrol System" designation is applied to any combination of Minneapolis-Honeywell automatic pneumatic controls used to govern the operation of air conditioning or heating systems. This equipment may be used to obtain either two-position or modulating control in any desired sequence. Such features as the Gradutrol Relay, a Honeywell development, which eliminates friction loss and allows accurate graduation of valves and damper motors, make the Gradutrol System a truly remarkable advance in pneumatic control.



Pneumatic Radiator Valve

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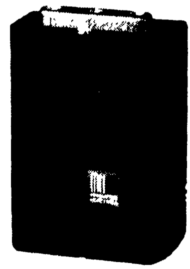
The outstanding advantages of the Honeywell pneumatic Gradutrol System, the electric Modutrol System, and the Electronic Control System may be combined in a single installation. Thus, maximum flexibility, low-cost installation and dependable control results can be obtained. Honeywell can furnish controls for any particular type of system or for any combination of systems. This is your guarantee of fair and unprejudiced engineering advice as to the type of control best suited to your needs.



Electric-Pneumatic Relay

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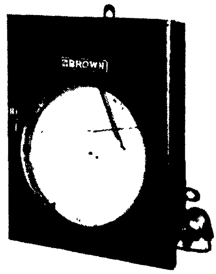
To obtain best results from modern heating and air conditioning equipment, it is necessary that the engineer in charge have a visual picture of actual conditions at all times. Brown instruments are designed to supply this information and to help operate the system at its highest efficiency. A complete line of industrial instruments is available for measuring, controlling, and recording temperature, humidity, flow, pressure, rpm, liquid level, pH, and fire safety.



Recording Thermometer

FREE HONEYWELL LITERATURE

Listed are but a few of the many pieces of literature available, giving full information on various Honeywell control devices and systems. Any inquiries you may have pertaining to Honeywell equipment or some specific control problem will be given prompt, personal attention—whether you contact the main office in Minneapolis, or any of the Honeywell subsidiaries or branch offices located in principal cities throughout the United States, Canada, and other countries.

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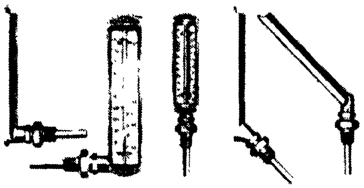

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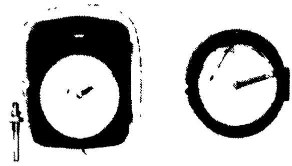

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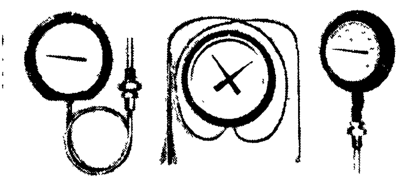

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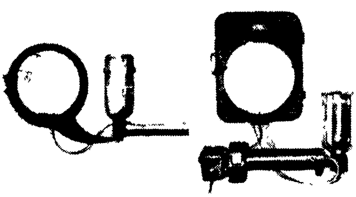
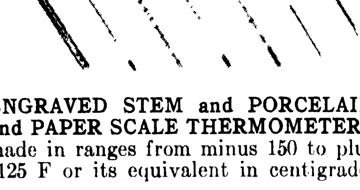


RECORDING PSYCHROMETERS, Mercury Actuated. Round and Rectangular Case, 12 hr, 24 hr or 7 day charts, self contained or remote reading, with or without motor driven fan.



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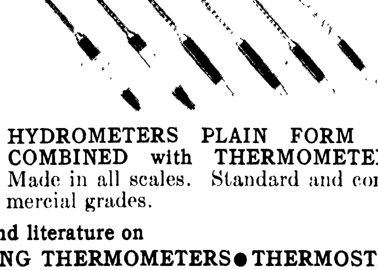

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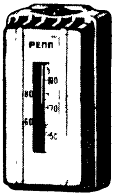
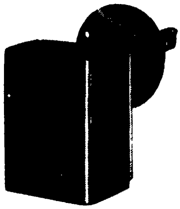
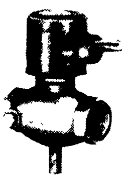


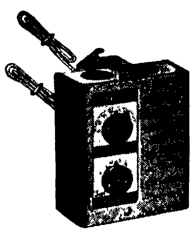
Thermostats



Oil Burner Stack Switches



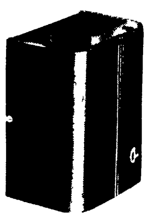
Type 3B-Magnetic Gas Valves



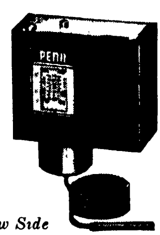
Warm Air Fan and Limit Controls

HEATING CONTROLS

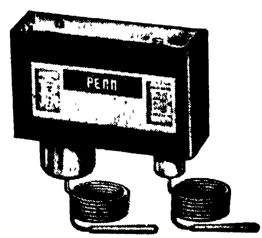
A wide selection of controls is available for automatic heating service on steam, vapor, hot water, or warm air systems . . . gas, oil or coal-fired. Typical controls in Penn's complete line are: Low and Line Voltage Room Thermostats—Stoker Timers — Oil Burner Stack Switches—Magnetic and Motorized Gas Valves—Hot Water and Warm Air Temperature Controls (including a new line of Liquid Expansion Fan and Limit Controls)—Vapor and Steam Pressure Controls — Relays and Relay-Transformers—Humidistats—Day-Nite Tem-Clocks—Pilot Burners and Pilot Generators — Thermopilot Relays — Low Pressure Gas Regulators — Damper Motor Controls—and Solenoid Valves. For complete descriptions and specifications write for free catalogs.



Stoker Timers



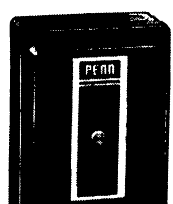
Series 270 Low Side Pressure Control



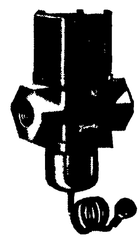
Series 271 Dual Pressure Control

REFRIGERATION CONTROLS

A complete line of automatic commercial refrigeration controls . . . in a wide choice of pressure and temperature models to fit every need. Series 270 Heavy Duty Refrigeration Control features two-pole, load-carrying contact structure and direct-reading calibrated scales. Other controls in the refrigeration line include: Cooling Room Thermostats — Humidistats — Relays — Solenoid Valves—Line Starters and Motor Contactors in Size 0, 1, and 1½—and Water Regulating Valves in sizes up to 2½ in. I.P.T. Write for free catalogs which give complete descriptions and specifications.



Motor Starters and Contactors



Water Regulators

PERFEX CORPORATION

Milwaukee 7, Wis. **Perfex Controls Ltd., Toronto 1, Ont.**

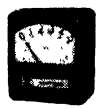
REPRESENTATIVES IN—Boston, Buffalo, Chicago, Cleveland, Indianapolis, Milwaukee, New York, Philadelphia, San Francisco.

DISTRIBUTORS IN—Atlanta; Baltimore; Bangor, Me.; Boston; Butte; Cedar Rapids; Chicago Cincinnati; Cleveland; Denver; Des Moines; Detroit; Fort Wayne; Grand Rapids; Greensboro; Hartford; Indianapolis; Kansas City; Knoxville; Los Angeles; Louisville; Milwaukee; Minneapolis; Nashville; New Orleans; New York; Omaha; Philadelphia; Pittsburgh; Portland, Oregon; Providence; Richmond; Rochester, N. Y.; Salt Lake City; San Francisco; Seattle; St. Louis; St. Paul; Sioux Falls; Syracuse.

INDUSTRIAL CONTROLS AND INSTRUMENTS

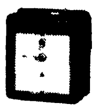
Increased boiler efficiency, lower fuel costs, elimination of smoke, saving of operator's time—these are some of the benefits of installing dependable Perfex combustion controls and instruments in the boiler room. Overfire draft control systems and modulating systems controlling fuel and air input for oil, gas, or stoker-fired boilers, draft gages, pressure gages, flue gas temperature indicators, combustion controls, draft controls, program controls and actuators are included in this line of cost-cutting instruments.

Flue Gas Temperature Indicator

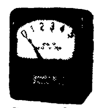


Pressure Gage

Actuator

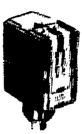


Draft Control



Draft Gage

AUTOMATIC HEATING CONTROLS

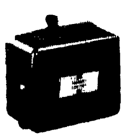


Pressure Controls

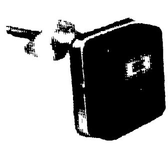
Dependability and accuracy characterize the complete Perfex line of automatic controls for heating systems, unit heaters, electric heating, etc. The line includes thermostats (low and line voltage), limit and operating controls (for steam, hot water and warm air), primary controls (for gas, oil, stoker or hand firing) time switches, relays, and barometric draft regulators. Feature of Perfex controls is the "Twin Contact" Switch—which gives double action, positive contact, magnetic snap-action, immunity to vibration, doesn't require leveling and has no flexible leads to impede action or impair calibration.

Hot Water Controls

Line Voltage Thermostats



Stoker Primary Controls



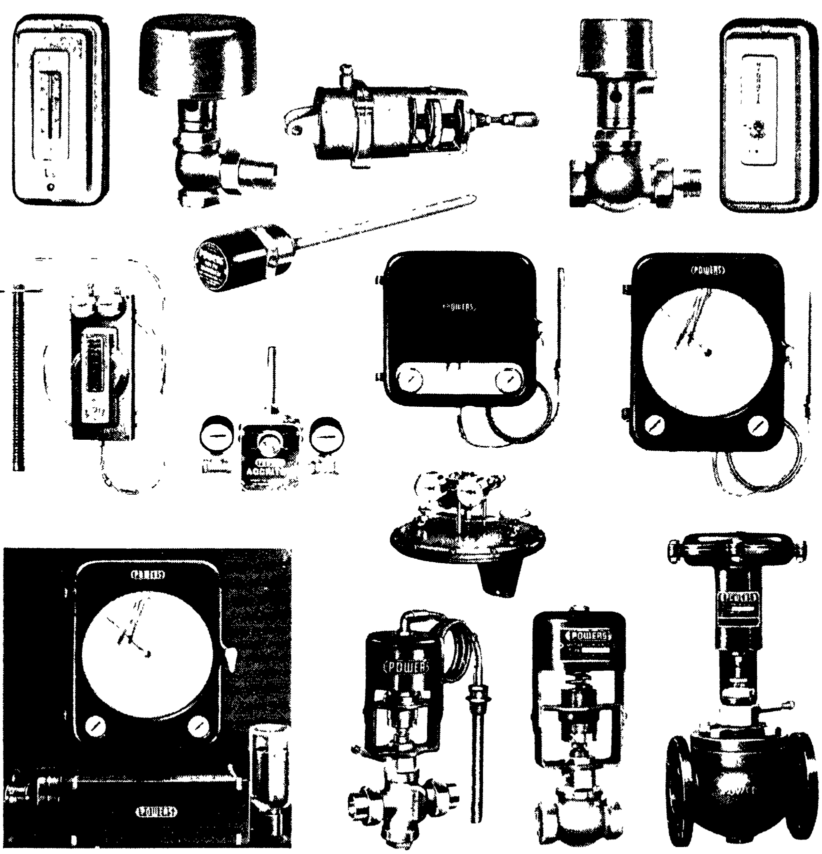
Oil Burner Primary Controls

Solenoid Gas Valves

THE POWERS REGULATOR CO.

60 Years of Temperature and Humidity Control—Offices in over 50 Cities

GENERAL OFFICE AND FACTORY: Skokie, Ill. NEW YORK 17, N. Y., 231 East 46th St.—Chicago 13, 3819 N. Ashland Ave.—LOS ANGELES 5, 1808 W. 8th St.—BOSTON 15, 125 St. Botolph St.—DETROIT 1, 320 McKerchey Bldg.—PHILADELPHIA 32, 2240 N. Broad St.—CLEVELAND 3, 1670 E. 40th St.—ST. LOUIS 3, 2118 Pine St.—NEW ORLEANS 12, 208-209 Vincent Bldg.—KANSAS CITY 8, 1510 Main St.—SEATTLE 1, 3150 Elliot Ave.—ATLANTA 3, 142 Spring St. N.W.—DALLAS 1, 2415 N. Pearl St.—GREENSBORO, 733 Jefferson Standard Bldg.—TORONTO, ONT., 195 Spading Ave.—MEXICO, D.F. Apartado 63 Bis.



A very complete line of Temperature, Humidity, Pressure—indicating, controlling, and recording regulators—for heating and air conditioning systems, industrial processes and all types of hot water heaters.

Sixty years of experience in furnishing and installing temperature and humidity control for every conceivable purpose in all types of buildings have given us a wealth of experience from which you can draw in selecting the proper type of control for any purpose. Catalogs and Bulletins describing our products furnished upon request. Phone or write our nearest office. See your phone directory.

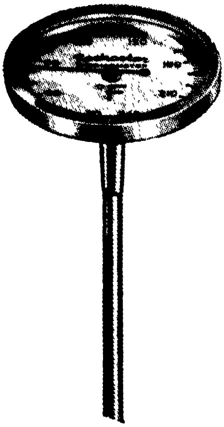
Rochester Manufacturing Co., Inc.

80 Rockwood St., **Rochester 10, N. Y.**

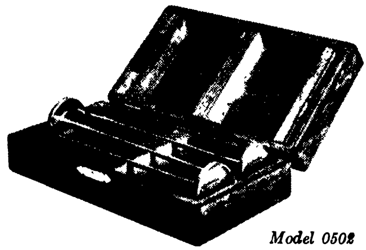
274 Madison Ave., NEW YORK 16 · 9443 S. Ashland Ave., CHICAGO 20 · 2949 Harriet Ave., S., MINNEAPOLIS · 1355 Market St., SAN FRANCISCO 3 · 1011 Greenmount Ave., BALTIMORE 2 · 6270 Souder St., PHILADELPHIA 24 · 4 Manor Rd., East, TORONTO, ONTARIO, CANADA.

Rochester Gauges—Dependable Accuracy

Liquid level, pressure and temperature gauges in a wide range of types



CIRCULATION AIR CONTROL THERMOMETER KIT

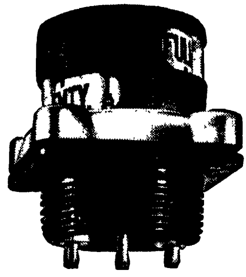


Model 0502

Packed three thermometers to a Kit, these thermometers permit accurate balancing of air temperature in any type of forced air heating. No further equipment needed. To be inserted in warm air duct, return air duct and near blower switch all at one time. Stainless steel with 2 in. easy to read dial 30–240 F and 9 in. stem with tapered bushing to fit snugly in awl or nail pierced hole. Sturdily and accurately made.

LEAK-PROOF, PRESSURE-TIGHT OIL TANK GAUGES

These highly dependable, easy-to-read fuel level gauges have been the standard of the industry for 25 years. They feature a magnetic gauge action which makes them leak proof and pressure tight. The pointer is actuated by a permanent non-electric magnet in solid gauge head. *Underwriters'* listed. Mounts on top of tank at center or either end in pipe thread fitting—1½ in., or 2 in. for all basement tank depths—22, 24, 27, 42, 44 or 47 in.



Model 3175

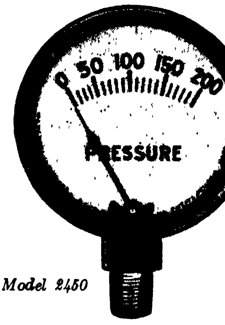
PRESSURE AND VACUUM GAUGES

Designed for indicating and testing. Pressure gauges, 0–200 lb-vacuum 0-30 in., 2 in. dial, ⅛ in. bottom connection.

Kit 0500 One 200 lb pressure gauge, one 30 in. vacuum gauge.

Kit 0501 As above plus one stack thermometer—2 in. dial 100–1000 F.

All gauges *individually calibrated* for accuracy.



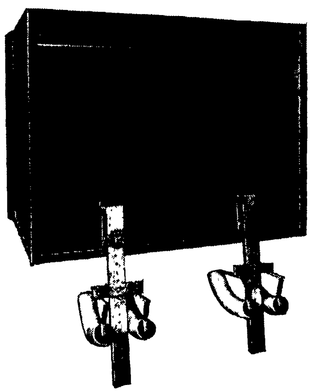
Model 2450

Simplex Manufacturing Company

198–206 North Main Street

Fond du Lac, Wis.

SIM-TROL Barometric Draft Controls—SIMPLEX Roof Vent Flashings



Type "A'

Breechings and stacks must provide sufficient draft under adverse atmospheric conditions. During normal and high barometric periods, cold months and windy days they generate excessive intensities with resulting inefficiency and fuel waste.

ADVANTAGES

SIM-TROLS automatically maintain a minimum draft required for good combustion, fuel economy, increased heat transfer by reduction of gas velocity through the boiler, and minimize cold air infiltration and inrush causing sudden shrinkage of boiler parts. They eliminate local hot spots, floating and pulsating flame and sucking out of pilots. They improve feed water regulation, super-heater results, boiler life and boiler room ventilation.

SIM-TROLS aid materially in smoke abatement and boiler cleanliness by reduction of unburned combustibles and air dilution of stack gases. Lowered stack temperatures prolong liner life and reduce fire hazards.

CONSTRUCTION

Properly sized SIM-TROLS adjust easily to any desired draft intensity. The races adjust laterally, the race assembly vertically, and the arm angle may be varied in relation to the plane of the gate.

Gates of Type A and Type C SIM-TROLS rotate on cold rolled arbors and dust proof ball bearings. Curved tubular counterbalances contain metal balls which constantly change position to compensate for varying rotation angles of the gate, maintaining over-fire draft within 0.01 in. water, plus or minus.

SIM-TROLS are protected by hard, heat resisting boiler room enamels. All counterbalance members are outside, free from encrustment and corrosion by products of combustion.

DESIGN FACTORS

Sizing is important. The input opening should equal the stack area, plus 10 per cent for each 35 ft of stack height over 65 ft thereof. Preference should be given to control width, rather than height for ease of adjustment and smooth operation.

Horizontal clearance from installation point approximates gate height plus 30 in. Over-all height approximates gate height plus 37 in. Over-all width approximates gate width plus 8 in. Designs for limited space may be made if plans are submitted.

Types A, C and F are individually designed to plant specifications presented, including breeching and stack dimensions, clearances, fuel, method of firing and number of boilers. Detailed sketches are submitted for approval without obligation.

GAS MODEL SIM TROLS

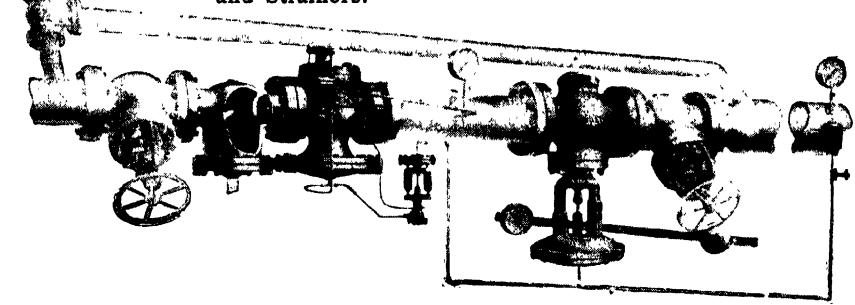
Types A, F and H are available to provide full back pressure relief and draft control. If specified, Types A and F are designed for use with coal, gas or oil by minor adjustment by the operator.

SPENCE

Spence Engineering Company, Inc.

28 Grant Street, **Walden, N. Y.**

Regulators for the accurate control of Pressure, Temperature, Differential, Back Pressure and Liquid Level; also Pump Governors, Electrically Controlled Regulators, Desuperheaters and Strainers.

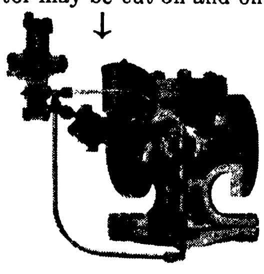


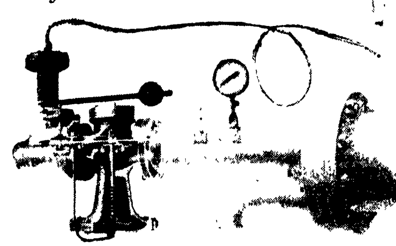
The SPENCE Type ED-W27 Two-Stage Pressure Reducing Station is designed to serve a heating system dependably, at low cost and with utmost safety. The primary is a pilot operated, Type ED Pressure Regulator. SECO Metal trim affords guaranteed resistance to wiredrawing. The secondary is a direct operated, Type W27 Valve, simply constructed, yet engineered for long trouble-free, heating duty.

Note that the Diaphragm of the SPENCE Safety Pilot is connected to the low side. Should the secondary valve fail, this Pilot will assume control of the primary; thereby effecting a one-stage reduction from primary to final delivery pressure.

A Spence Type E2T100 Temperature Regulator reduces the steam pressure in addition to accurately modulating the flow as required to control the temperature. No separate reucing valve is necessary.

By adding a **SPENCE** Solenoid Pilot any **SPENCE** Regulator may be cut on and off electrically





The Spence design provides a self-operated, pilot controlled regulator entirely packless in construction and guaranteed to shut tight. SECO Metal seats and discs will not be cut by the wire-drawing action of steam. Spence thermal elements are not injured by overheating.

Pilot Type T100 is weight-loaded for heaters requiring not more than 10 psi pressure. Pilot Type T150 is spring-set for pressures up to 50 psi. Either may be used on a storage or instantaneous heater in combination with a main valve selected to suit the initial steam conditions.

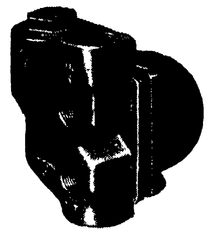
Sterling, Inc.

3738 North Holton Street — Milwaukee 12, Wisconsin

Heating and Temperature Control Products

Distributed through leading Heating and Plumbing Wholesalers

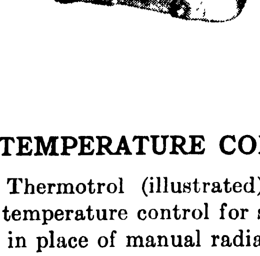
Sales Representatives in Principal Cities



HEATING SPECIALTIES

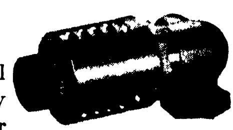
Thermostatic Traps: sizes ½ in., ¾ in., 1 in.; max. pressure 15, 65, 100, 125 psi; Angle, Straightway, Corner, Vertical Body Styles. Float and Thermostatic Traps: sizes ¾ in. to 2 in.; max. pressures 15, 100 psi. Strainers: max. pressure 125 psi; Cast Iron Sizes ½ in. to 2 in.; Brass Sizes ⅜ in. to 1 in. Also Radiator Valves, Boiler Return Traps.
Illustrated at right is the compact, sturdy ¾ in. 69B F & T Trap, which like all other Sterlco traps has a bellows type thermostat and replaceable seats and valve members.

CONDENSATION and VACUUM PUMPS



4100 and 4200 Series (illustrated): neat, convenient unit with either steel or cast iron tank. Dual voltage capacitor type motor and carbon type rotary seal—for all jobs up to 14,000 sq ft EDR, 20 psi. 3500 Series: heavy duty units with pedestal mounted steel tank for a wide range of jobs—2000 to 65,000 sq ft, up to 150 psi. 3700 Series: with heavy cast iron tank for installation underground or in wet locations. 2000 to 20,000 sq ft, 20 or 30 psi. Vacuum Pumps: Type V, 2500 and 5000 sq ft, 20 psi. Type S, 10,000 to 40,000 sq ft, 20 to 40 psi.

TEMPERATURE CONTROL VALVES



No. 120 Thermotrol (illustrated): self-contained individual radiator temperature control for steam or hot water. Easily installed in place of manual radiator valve without wiring or other external connections. Tank Temperature Controls: self-powered modulating control valves for fluid heaters and process equipment. Sizes: ⅜ in., ½ in., ¾ in., 1 in., 1¼ in., 1½ in., 2 in.

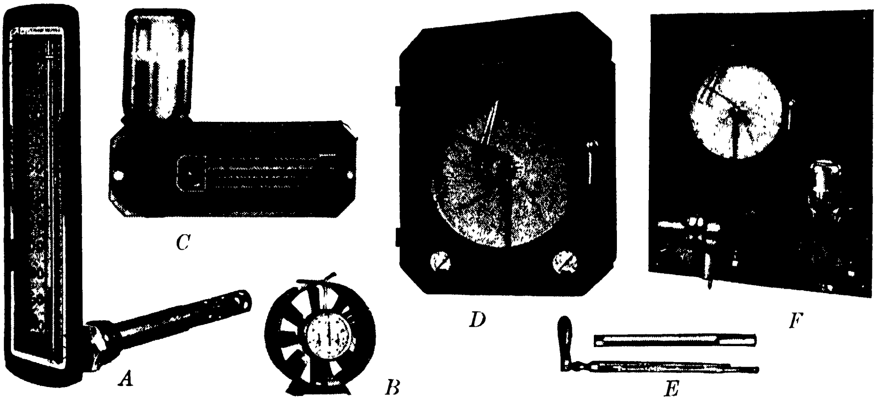
Taylor Instrument Companies

Rochester 1, N. Y., U. S. A.

IN CANADA—TAYLOR INSTRUMENT COMPANIES OF CANADA LTD., TORONTO

NEW YORK	LOS ANGELES	ST. LOUIS	CINCINNATI	ATLANTA
CHICAGO	PITTSBURGH	SAN FRANCISCO	TULSA	MINNEAPOLIS
BOSTON	CLEVELAND	HOUSTON	BALTIMORE	WILMINGTON
PHILADELPHIA	*Manufacturing Subsidiary in Great Britain,*			SCHENECTADY
BUFFALO	*Short & Mason, Ltd. London*			SCOTIA

Taylor Instruments for Indicating, Recording and Controlling Temperature, Pressure, Humidity, Flow and Liquid Level



(A) Taylor Industrial Thermometers – with BINOC* Tubing Includes many styles and scale ranges with bulbs for every application. These thermometers use the well known Taylor BINOC Tubing—a designed and optically correct glass tubing which assures ease of reading generally lacking in industrial thermometers. BINOC tubing more than doubles the angle of vision within which readings can be made. Because of the patented triple-lens construction, its broad mercury column can be read easily and accurately with both eyes. Bore reflection is absent.

(B) Taylor Biram's Anemometer For measuring air velocities with fan revolutions indicated on dial. Various models for a wide range of air speeds and registration limits.

(C) Taylor 10BG Hygrometers For air conditioning supply and return ducts, dryers, and other closed compartments where temperature and humidity readings are desired. Combines the accuracy of an etched-stem thermometer and ruggedness of an industrial thermometer. Available with bottle or constant automatic water supply.

(D) FULSCOPE* Recording Controller An air-operated controller that gives practically any character of process control regardless of time lag in apparatus. Available for controlling temperature, pressure, humidity, rate of flow, liquid level. Where extreme load changes or badly balanced operating conditions exist, precision control can be maintained by the automatic reset feature. For applications where a record is not essential, Taylor supplies an Indicating FULSCOPE Controller.

* Trade Mark

(E) Sling Psychrometer Two accurate etched-stem thermometers mounted on die-cast frame, with the bulb of one covered with a wick to be moistened. Whirling bulbs subject this hygrometer to complete air contact to produce extreme accuracy of temperature and humidity measurement.

(F) Recording Hygrometer Records both wet and dry bulb temperatures on the same chart in different colored inks, making comparison very easy.

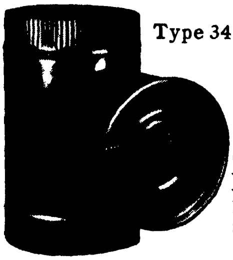
Type shown has motor-driven fan for conditioned rooms or passages where circulation is poor. Furnished without fan for installations where circulation across bulb is good.

Taylor also offers a complete line of the famous Taylor Recording and Dial Thermometers; Ratio, Pneumatic Set, Self-Acting and Type "P" Controllers; Indicating Hygrometers and many types of Humidiguides.

WALKER *Automatic* DRAFT REGULATORS

Walker Mfg. & Sales Corp.

Sales Office & Factory
1701–10 Penn St.
St. Joseph, Missouri



Type 34 B

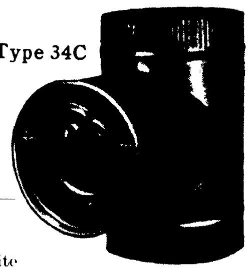
These controls for Space and Water Heaters, Floor and Utility Furnaces, Ranges and Stoves. Ring and Damper pressed from steel; fittings are brass. Larger sizes are interchangeable for economy with Domestic Controls. These controls feature the Walker patd., non-clogging, closed box hinge.

SIZES, SMOKE PIPE	3″	4″	5″	6″	7″	8″	9″	10″
with Tee Joint	x	x	x	x	x	—	—	—
with Collar	x	x	x	x	x	x	x	x
Damper Section only	x	x	x	x	x	x	x	x
Standard Finishes	Blued	Blued	Blued	Blued	Blued	Cad.	Cad.	Cad.

x denotes availability

These controls made in selected sizes are similar to Type 34B except that adjustment feature and balance assembly are refined to provide ease of setting and proper draft control. Prices are slightly higher, but these controls are extensively used by manufacturers who demand the best.

Type 34C



SIZES, SMOKE PIPE	4″	6″	8″
with Tee Joint	x	x	x
with Collar	x	x	x
Damper Section only	x	x	x
Standard Finishes	Blued	Blued	Cad. or Zinc Iridite

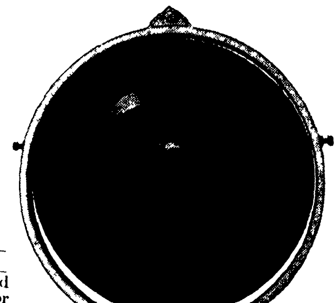
Type 34
DOMESTIC CONTROLS

These controls are for central heating plants in homes, Apt. Bldgs., Churches, Stores, Schools, etc. for economy and peak performance. All materials are carefully selected for resistance to heat, corrosion, moisture and soot. Aluminum ring assures positively rigid construction. Aluminized steel features the Damper Plate construction. Pivot pins are cadmium plated; adjustment and balance assemblies are brass.

SIZES, SMOKE PIPE	6″	7″	8″	9″	10″	12″	14″	16″	18″	20″
with Collar	x	x	x	x	x	x	x	x	x	x
Damper Section only	x	x	x	x	x	x	x	x	x	x

Standard Finishes — ALL SIZES—Aluminized Steel Damper Plate or Vane Cast Aluminum Frame or Ring Brass Fittings

TYPE BB—Industrial—These controls are ruggedly constructed for industrial boilers. Pivots are ball bearing. Close-grained cast iron used in frame for rigid, proper alignment. Cast iron and brass fittings, in conjunction with temperature-resistant finish, assure long life. No installation collars furnished. For details of installation or other recommendations, write for further information.

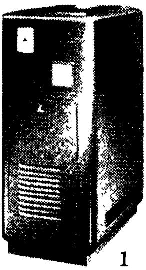


SIZES, Equiv. to Rd. Pipe	16″	18″	20″	22″	24″	26″	30″	36″
Draft Regulator only	x	x	x	x	x	x	x	x

Standard Finishes — ALL SIZES—Cast Iron Frame and Painted Steel Damper Plate or Vane

American Radiator & Standard Sanitary CORPORATION

P. O. BOX 1226, PITTSBURGH 30, PENNA.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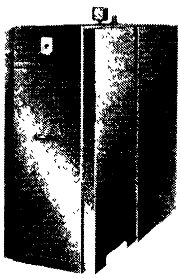
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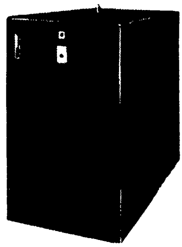
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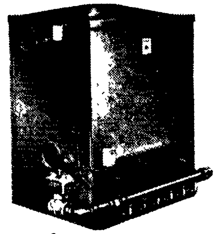
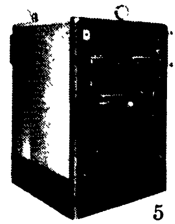

4

1. EMPIRE BOILER For all types of gas. Made in a complete range of sizes for smaller homes and buildings with or without basement; also for domestic hot water supply. Approved by *American Gas Association*. *A.G.A. Ratings:* Steam—255 to 425 sq ft, Water—960 to 25,600 sq ft.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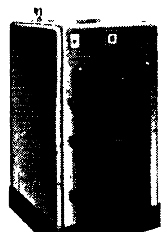
2. ARCOLINER WET BASE BOILER For oil. Compact boiler, with or without Arcoflame Burner, for small homes. Wet base construction ideal for homes with or without basements. Ratings: Steam—260 to 520 sq ft, Water—485 to 940 sq ft, installed radiation.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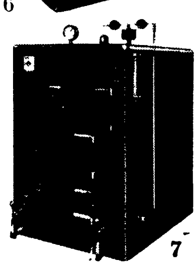
3. OAKMONT BOILER For oil. Exclusively for oil firing. Also supplied as complete oil heating unit with Arcoflame Burner. Ratings: Steam—390 to 810 sq ft, Water—715 to 1440 sq ft, installed radiation.

4. STANDARD BOILER For all types of gas. Designed for larger homes and buildings. Approved by *American Gas Association*. *A.G.A. Ratings:* Steam—600 to 16,000 sq ft, Water—960 to 25,600 sq ft.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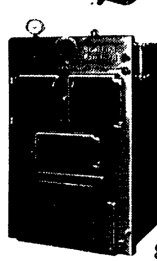
5. EXBROOK BOILER For oil or stoker. Boiler in sizes adapted for larger homes and buildings. Available with Arcoflame Burner as oil heating unit. Ratings: Steam—775 to 1825 sq ft, Water—1380 to 3095 sq ft, installed radiation.

6. SEVERN BOILER For all fuels. Efficient boiler with advanced features for convenience and economy. Available with Arcoflame Burner as oil heating unit. Ratings: Steam—350 to 480 sq ft, Water—560 to 1390 sq ft, installed radiation.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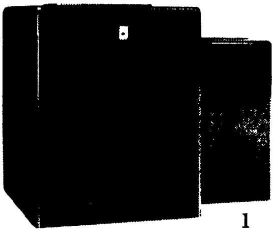
7. REDFLASH BOILER For all fuels. Economical heat for any size or kind of building. Attractive jacket, fully insulated. Ratings: Steam—770 to 9900 sq ft, Water—1230 to 15,840 sq ft, installed radiation.

8. WATER TUBE BOILERS For oil or stoker. For medium to large buildings. Efficient and economical. Ratings: Steam—930 to 4600 sq ft, Water—1640 to 7360 sq ft, installed radiation.



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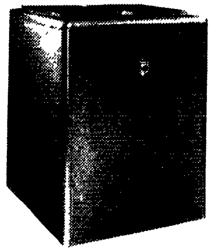

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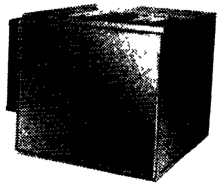

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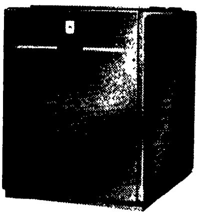

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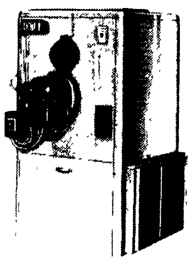

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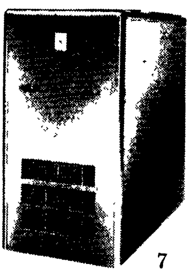

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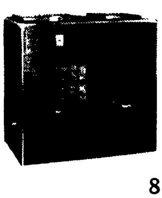

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1. **WESTMORELAND WINTER AIR CONDITIONER** For oil. Steel unit expertly engineered for automatic oil fired heating, with or without Arcoflame Oil Burner. Five sizes, from 150,000 to 330,000 Btu capacity at register.

2. **WYANDOTTE WINTER AIR CONDITIONER** For gas. Steel utility type unit for small homes with or without basements. Factory assembled, pre-wired, cleanable heating element. Five sizes with *A.G.A.* input ratings from 55,000 to 125,000 Btu. Side or bottom return air inlet.

3. **MAYFAIR SUMMER AIR CONDITIONER** For summer cooling. The Mayfair summer air conditioner, latest addition to the American-Standard line, converts forced warm air heating system to year 'round conditioning. It uses same duct system. . .mechanically cools and dehumidifies the air.

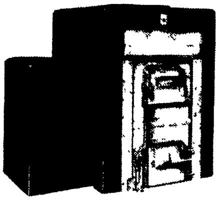
4. **MAGNE-FILTER AIR CLEANER** For clean air year 'round. The electronic Magne-Filter air cleaner, installed in the return duct of *any* winter or summer air conditioning system, traps even the smallest dirt particles, removes pollen, air-borne bacteria, dust and smoke.

5. **MOHAWK WINTER AIR CONDITIONER** For gas. Deluxe cast iron winter air conditioner for large or small homes. Preheated air offers fuel economy, extra efficiency. Ribbon burner for any type of gas. Eight sizes, with *A.G.A.* input ratings of 80,000 to 300,000 Btu.

6. **WINTERGLO WINTER AIR CONDITIONER** For oil. Steel utility type unit for small homes and individual apartments. Factory assembled and pre-wired; two sizes, 85,000 and 105,000 Btu output at bonnet. Side or bottom return air inlet. *Underwriters'* listed with flange type Arcoflame Burner for less than standard clearances.

7. **WINTERWAY WINTER AIR CONDITIONER** For oil. Steel basement type unit with right or left side flue connection, solid base pan with leveling screws for quick installation. *Underwriters'* listed with flange type Arcoflame Burner. Three sizes, with 100,000 to 150,000 Btu capacity at register.

8. **SENECA WINTER AIR CONDITIONER** For gas. Low cost steel gas fired basement unit for small and medium size homes. Cleanable heating element. Four sizes with *A.G.A.* inputs of 85,000 to 150,000 Btu.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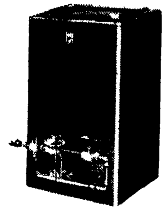
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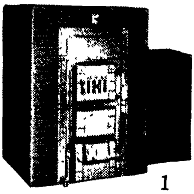
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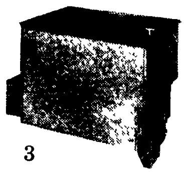
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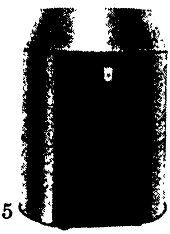
1. ALLERTON WINTER AIR CONDITIONER For all fuels. Moderately priced steel unit. Four sizes, 20 to 27 in. shell diameters, from 86,100 to 127,900 Btu capacity at register.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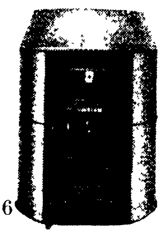
2. CLIFFDALE WINTER AIR CONDITIONER For all fuels. Rugged steel unit. Two sizes, 30 and 34 in. shell diameters with 162,500 and 177,000 Btu capacity at register.

3. NAVAHO FLOOR FURNACE For gas. Shallow steel unit for homes with or without basements. Factory assembled. Available with floor grille or dual wall register, automatic controls. Three sizes, with *A.G.A.* inputs of 25,000 to 50,000 Btu.

4. SHAWNEE WARM AIR FURNACE For gas. Compact steel furnace for economical heating. Cleanable heating element. Two factory assembled sizes, with *A.G.A.* inputs of 65,000 and 80,000 Btu; three larger sizes, with inputs of 105,000 to 140,000 Btu.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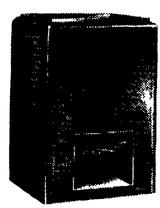
5. KENWOOD WARM AIR FURNACE For all fuels. Cast iron furnace for hand fired coal. Heating element of durable cast iron. Dependable and economical. Four sizes, from 18 to 24 in. fire pot diameters—53,000 to 81,200 Btu capacity at register.

6. ARLINGTON WARM AIR FURNACE For all fuels. Modern steel furnace, in pipe and pipeless models, for coal (hand fired or stoker) or oil. Readily adaptable to gas. Heavy steel heating element. Four sizes, from 20 to 27 in. shell diameters—67,800 to 100,800 Btu capacity at register.


7

7. CLIFTON WARM AIR FURNACE For all fuels. Quality steel furnace for larger installations. Burns all fuels. Readily adaptable to gas. Two sizes, 30 and 34 in. shell diameters—128,200 and 139,300 Btu capacity at register.

8. BLOWER-FILTER UNIT For converting existing gravity installations to forced air. Replaceable filters remove dust, dirt and pollen. Quiet, dependable, rubber mounted motor, built-in overload protection. Six sizes, from 9 to 21 in.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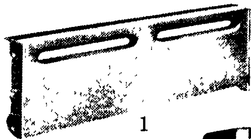
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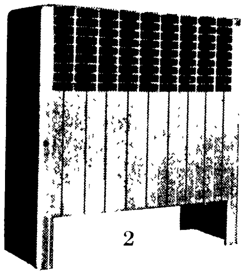
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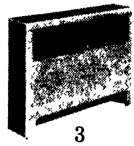
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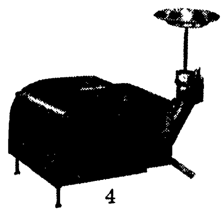
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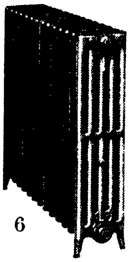
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1. **RADIANTRIM PANELS** Replace ordinary baseboards. Provide all the advantages of both convected and radiant heat. Sheet metal accessories also available for complete installation.

2. **SUNRAD RADIATORS** Recessed or free-standing. Need no enclosure. Two sizes: 5 in. deep x 20 in. high and 7½ in. x 23 in. Inlet grilles if desired.

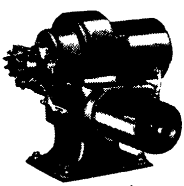
3. **CONVECTORS** With cast iron (Arco) or non-ferrous (New Multifin) heating elements. Styles and sizes for every need. Special designs for hospitals, institutions, etc. New Multifin Type K (shown) available in 63 packaged stock sizes.

4. **STANFLAME CONVERSION BURNER** Gas fired. Vertical, upshot type burner for boiler, furnace, or winter air conditioner. Fits round or square combustion chamber. Runner pilot igniter. Burns all gases. Three models, with *A.G.A.* inputs of 60,000 to 335,000 Btu.

5. **BUDGET WATER HEATER**—Approved by *A.G.A.* Heats water automatically, stores it for instant use. White enameled jacket, black trim. Thrifty and dependable. Three sizes—20, 30 (shown) and 40 gal.

6. **ARCO RADIATORS**—Modern, slim tube radiators that occupy less space and give more heat. Available in four widths—3, 4, 5 and 6 tubes—and four heights—19, 22, 25 and 32 in.

7. **ARCOFLAME OIL BURNER**—Listed by *Underwriters' Laboratories.* Complies with Commercial Standard CS-75. Special flange types for oil heating units. Pedestal types for conversion. *Three models: Capacities up to 7 gal per hour.*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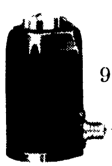
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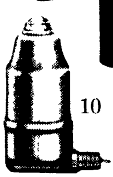
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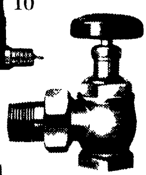
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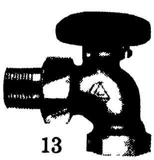

13

"DETROIT" HEATING CONTROLS AND ACCESSORIES—A complete line of Heating Controls and Accessories for all types of systems.

8. No. 861 Hurivent Vent Valve (for mains)

9. No. 300 Multiport Adjustable Air Valve

10. No. 500 Airid Air Valve

11. No. 999 Packless Radiator Valve

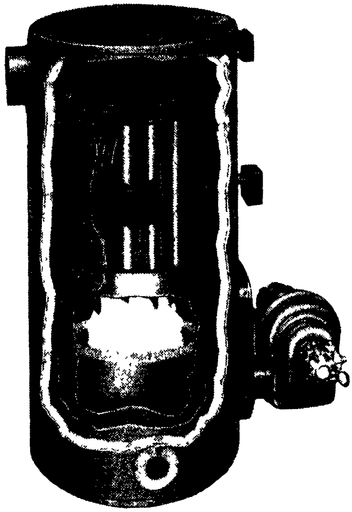
12. No. 5000 Variport Airid Adjustable Air Valve

13. No. 116 Radiator Valve. Complete line of elbows and fittings for all types of hot water systems.

Aldrich Company

121 E. Williams St., **Wyoming, Illinois**

Boiler-Burner Units, Domestic and Commercial Oil Burners



Aldrich Boiler-Burner Units—available in 7 sizes for Hot Water Heating, Steam Heating, and Hot Water Supply—designed for home, apartment, garage, club, restaurant, hotel, and factory installations. Gages, covers, and trim furnished to suit model and unit type ordered. Heavy-duty, double-spiral hot water coils factory-installed and tested in SC and WC models. Matched Aldrich Oil Burner with each unit. Standard equipment includes full set of basic automatic controls. Series BG Boilers are gas-fired with *A.G.A.*-approved burners for first five sizes. Oil and gas burners are readily interchangeable.

Welded steel fire box lined with cast refractory having high resistance to deterioration. Large burner - mounting plate provides access to fire box for replacement of combustion chamber. Extra thick wool insulation blanket provides high insulating value and minimizes heat loss through radiation.
Universal tappings on all models except "Bantam."

BOILER-BURNER UNITS

Sizes—Specifications—Dimensions

Size of Boiler	*Bantam*	118	160	225	315	514	808
Rating Sq. Ft. Hot Water EDR	660	750	1000	1500	2100	3300	5000
Rating Sq. Ft. Hot Water Standing Radiation	440	500	650	1000	1400	2200	3333
Rating Sq. Ft. Steam EDR	425	500	630	935	1275	2025	3000
Rating Sq. Ft. Steam Standing Radiation	285	333	420	620	850	1350	2000
Rating BTU Per Hr. (Max.)	100,000	118,000	160,000	225,000	315,000	514,000	808,000
Water Heater Delivery GPH @ 100° Rise	93	125	190	280	450	610	850
Storage Capacity Gallons	20	28	38.5	72	99	120	170
Firing Rate—GPH Maximum	1.00	1.25	1.65	2.75	3.65	5.7	9.00
Firing Rate—GPH Minimum	.65	.75	1.25	1.75	2.4	3.7	6.00
Firing Rate—GPH Best	.75	1.00	1.35	2.00	2.5	4.5	7.5
Model Burner Furnished	CX1	SAX1	SAX2	SAX3	SAX3	BX	BX
Sq. Ft. of Heating Surface	17	21	27	44	67	93	136
Dia Main Shell (Inside)	16½″	19″	21″	24¼″	28″	32″	38″
Height of Main Shell	42½″	46½″	50½″	58½″	66½″	66½″	70½″
Dia. of Fire Box (Inside)	12″	14¼″	16″	19″	22½″	27″	32½″
Height of Fire Box	21	24″	24″	26″	26″	26″	26″
Number of Tubes	16	16	20	30	42	60	94
Length of Tubes	15⅝″	15⅝″	19¼″	23⅜″	30⅝″	29⅝″	31⅝
Output Hot Water Coil @ 100° Rise WC—GPM	3	1.75	2.00	2.66	5.25	6.50	7.25
Output Hot Water Coil @ 100° Rise SC—GPM	3	2.00	2.75	3.67	6.75	8.00	10.00
Shipping Weight Lbs.—Boiler Only	440	697	843	1170	1760	1975	2370
Shipping Weight Lbs.—Burner Only	77	90	90	90	90	100	100
Shipping Weight Lbs.—Total	517	787	933	1260	1850	2075	2470

MODEL DESIGNATIONS

Without Coil	
Steam Heating	S
Hot Water Heating	W
Hot Water Supply	HGS*
With Coil	
Steam Heating	SC
Hot Water Heating	WC
Hot Water Supply	—

*** Galvanized, with dress jacket.**

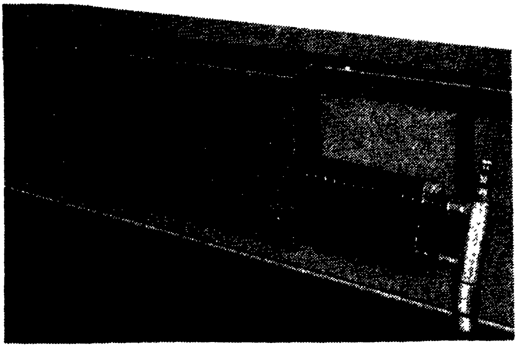
ALDRICH OIL BURNERS—Capacities from 0.75 gph to 19 gph. Models SAX-CX-BX-JU. 6 sizes, 4 models for domestic and commercial warm air, steam, or hot water systems. UL listed. SAX (illustrated) 3 firing ranges—0.75 to 1.35; 1.35 to 2; 2 to 4.5. Model CX 0.75 to 2. BX 4 to 8.5. JU 7 to 19.00 gph.

Brown Products Company

97-12 Metropolitan Ave., Forest Hills, New York

BROWN

BAYCE-HEET

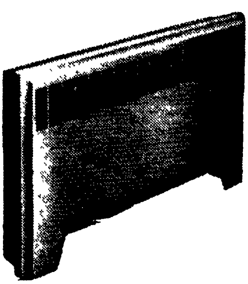


Brown Bayce-Heetis a "high-output" baseboard designed for residential and commercial use. This smart looking new unit has a rating of 1100 Btu at 215°F. (4.6 sq ft EDR per lineal ft) made possible by the remarkable TILT-FIN design wherein two diagonally opposed fins are folded downward and the entire heating element (fins and tubes) is tilted to allow more efficient air circulation. For use on forced hot water or two pipe steam system. Heating element: ½ in. copper tubes expanded into rectangular beaded aluminum fins for greater strength and heat transfer area.

Brown HEET

Automatic Oil-Fired Boiler Burner Units
Domestic Water Heaters

Available in 6 sizes for hot water radiation 68,000–472,000 Btu/hr. For steam radiation 5 sizes available from 93,000–472,000 Btu/hr (388–1965 sq ft EDR). Boilers come with built-in combustion chamber and flange mounted gun-type oil-burner with controls. Available with 3, 4, or 5 gal instantaneous coil for hot water. Publication 100-2.



Brown HEET

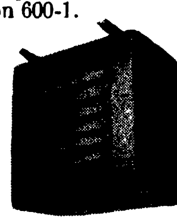
CONVECTOR UNITS

Brown Heet Convector Units in the distinctive heavy gage cabinet design can be used with forced circulating hot water, one or two pipe, steam vapor or vacuum systems. Heat regulation is by simple chain control damper. Free standing or recessed style 20 in. or 24 in. high, 6 in. depth, 20 to 64 in. length, ratings from 17 to 68.6 sq ft EDR (4075 Btu/hr to 16,464 Btu/hr). ½ in. copper tube—expanded into aluminum fins. Publication 600-1.

Brown HEET

Unit Heaters Horizontal Suspended Type

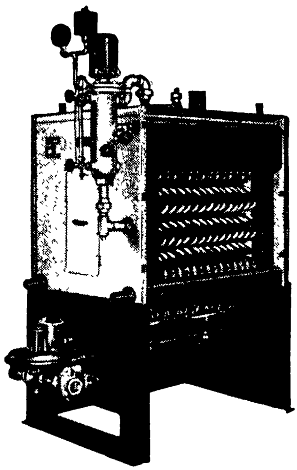
Brown Heet Unit Heaters are available in capacities from 2,400 to 240,000 Btu/hr (100–1000 sq ft EDR) operating at 2 lbs steam pressure 60 F air entering. Hot water use is optional. Heating element consists of ½ in. OD copper tubes expanded into aluminum fins providing positive permanent mechanical bond. Publication 500-1.



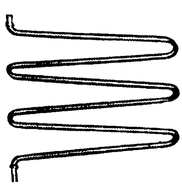
Bryan Steam Corporation

Chili Pike, Peru, Indiana

Hot Water and Steam Boilers designed exclusively for oil or gas firing. Products include domestic hot water or vapor steam boilers, commercial low pressure heating boilers for hot water or steam, non-explosive high pressure boilers up to 50 HP



Bryan Copper Tube Boilers are engineered expressly for oil or gas firing, where the flame is either on at full intensity or entirely off. Such combustion characteristics call for a boiler that is able to capture heat units with utmost rapidity—and able to withstand sudden expansion and contraction. Average stack temperature of the Bryan is 412 degrees.



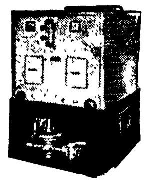
BRYAN COPPER TUBES

The Bryan starts with copper tubes. Copper has a coefficient of heat transfer approximately 6 times that of iron or steel. Heat of a Bryan tube is therefore transferred to the water inside 6 times as fast.

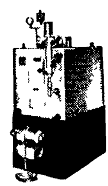
The design of the Bryan tube is such that water circulates rapidly through all parts of the boiler. There is a veritable nest of tubes directly over the flame where heat is most intense. These tubes break up the path of heat travel, reducing "surface film" to a minimum.

A Bryan Copper Tube Boiler never explodes. The worst that can happen is a split tube which may be replaced in a few minutes' time by an inexperienced person.

Domestic Boilers



Commercial Heating Boilers



High Pressure Boilers

DOMESTIC BOILERS

The Bryan Domestic Boiler is available in 7 sizes for hot water radiation. 70,000 Btu's to 560,000 Btu's.

For steam radiation 5 sizes are available, 390 to 1560 sq ft of radiation.

All models come complete with built-in combustion chamber and flange mounted oil burners. In gas fired models the burner is built in.

COMMERCIAL HEATING BOILERS

Any of the domestic Bryans may be used for small offices or buildings; but they are supplemented with three additional larger capacity boilers. These boilers are rated 2250 to 4400 sq ft of steam radiation or 3600 to 7050 sq ft of hot water radiation. They are used also for supplying hot water or low pressure steam in industrial applications.

HIGH PRESSURE BOILERS

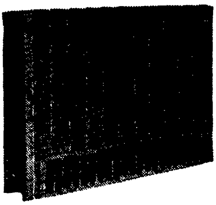
Bryan High Pressure Boilers are made in 5-10-20-35- and 50 hp ratings. Every one carries the *A.S.M.E.* stamp. They are made for high efficiency on long, hard pulls but are exceptionally useful in operations requiring fast, safe steam on short notice. Hospitals, dry cleaning plants, laundries, milk plants, tire repair shops and many others find them ideal for their operations.

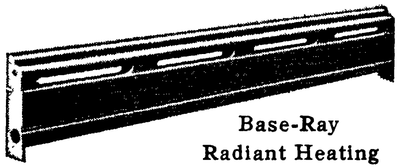
Burnham Corporation

BOILERS and RADIATORS

Irvington-on-Hudson, N. Y.

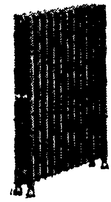
***There's a Burnham for Every Purpose*—Catalog No. 82 Sent on Request**





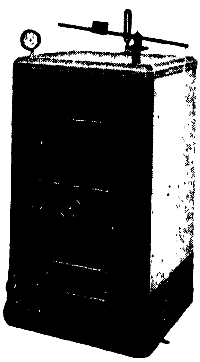
Base-Ray Radiant Heating

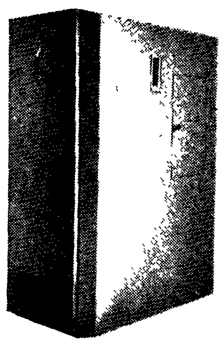
Hy-Power Model Base-Ray—Ratings —2.35 sq ft per lineal foot. Tappings—¾ in. at bottom only of both end sections. Sections are 7 in. high, 2 in. thick and in 12 and 24 in. length.

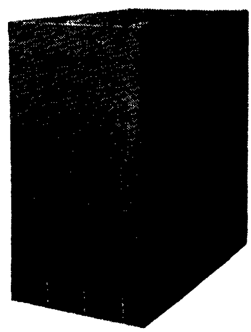


Burnham Radiant Raditor—Two heights. 20 and 23 in. Ratings, 2.25 sq ft per section and 3.40 sq ft per section.

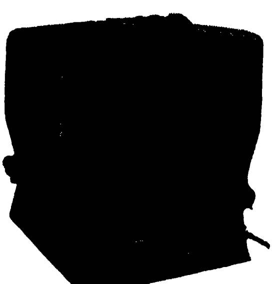
Burnham Slenderized Radiators—three to six tubes in all heights 19 in. to 33 in.



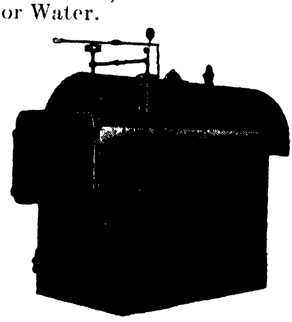




At left: No. 1, 2, 3 and 36 in. Series—All Fuel. 230 to 4920 sq ft. Steam and 370 to 7880 for Water.
Center: PACEMAKER Boiler for oil firing 270 to 710 sq ft for Steam and 490 to 1270 sq ft for Water.
Right: YELLO-JACKET Boiler (with extended Jacket) All Fuel Convertible 305 to 935 sq ft for Steam and 490 to 1600 sq ft for Water.



50 in. Twin-Section—4500 to 14,600 sq ft for Steam and 7200 to 23,360 sq ft for Water.



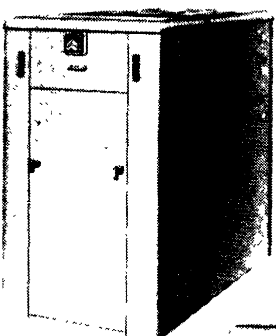
Welded Steel Boiler—Capacities from 2500 to 35,000 sq ft for Steam and 4800 to 56,000 sq ft for Water. Furnished for coal, oil or stoker firing.

Crane Co.

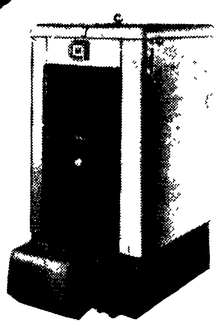
BOILERS, BASEBOARD PANELS, RADIATORS, FURNACES, VALVES, FITTINGS, PIPE, WATER AND STEAM SPECIALTIES, CONTROLS AND PLUMBING MATERIALS

General Offices: 836 South Michigan Avenue, **Chicago 5, Illinois**

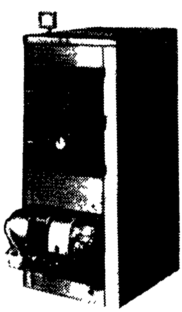
Nation-Wide Service Through Branches, Wholesalers, Plumbing and Heating Contractors



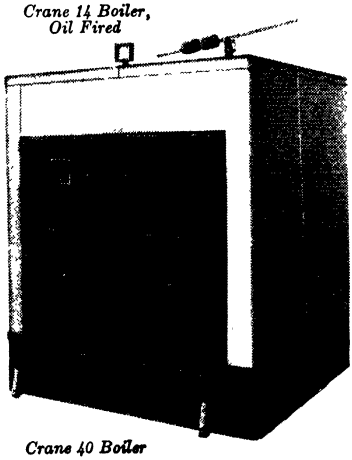
Crane 16 Boiler-Burner Unit



Crane 20 Boiler with Roberts-Gordon Gas Burner



Crane 14 Boiler, Oil Fired



Crane 40 Boiler

CRANE OFFERS EVERYTHING IN HOME HEATING

CRANE 16—A completely packaged boiler-burner unit. Patented Crane Sustained Heat Principle assures fuel economy. Oil burner is concealed beneath handsome jacket. For steam or hot water systems.

CRANE 20—A highly efficient boiler incorporating Crane patented water baffle and increased ceiling heating surface. For hand or stoker firing of coal, or for oil or gas conversion. Steam or hot water systems.

CRANE 14—A compact wet base boiler ideal for basement or utility room installation. May be had for steam or hot water system—to burn coal or coke, oil or gas.

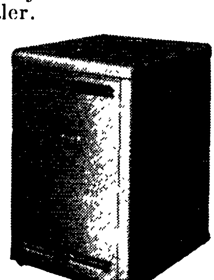
CRANE-LINE 2WG—Only 33 inches high, this efficient gas boiler will heat the average 5-room house. Designed for the most economical burning of gas. For hot water systems.

CRANE 30 & 40—Large boilers for schools, hospitals, public buildings, etc. For steam or hot water heating. Will burn coal or coke—oil or gas.

VALVES—FITTINGS—The Complete Crane line of valves and fittings offers all necessary piping items for any heating system.

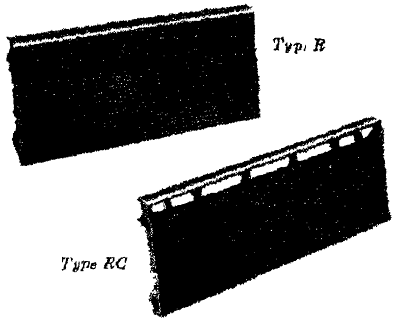
ALSO—A complete line of gas, oil and coal-fired Furnaces, Unit Heaters and Floor Furnaces.

For information on Crane Heating, consult your Crane Branch or Crane Wholesaler.



Crane-Line 2WG Deluxe Boiler

CRANE RADIANT BASEBOARD HEATING



Type R

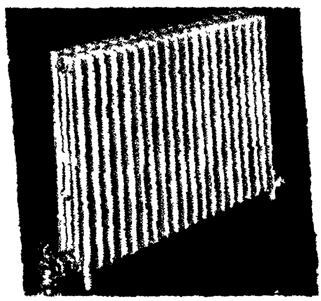
Type RC

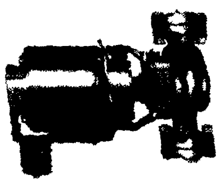
Type RC provides radiant and convected heat. Type R, solid front, provides radiant heat only. Both are 9 in. high.

Easy to install—special panel tool assures quick, positive assembly. Matching wood baseboard permits continuity of panels around wall—simplifies corner connections. When wall space is limited, i.e., kitchen and bathroom, Crane panels may be effectively installed at the ceiling.

CRANE COMPAC RADIATORS

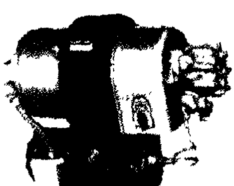
Slender in line—modern in design—give more heat—use less space. For free-standing or recessed installation, steam or hot water. Built in sizes to suit every need—in 3, 4, 5 and 6-tube types of 6 to 56 sections. Replace ordinary radiators with little or no change in piping.

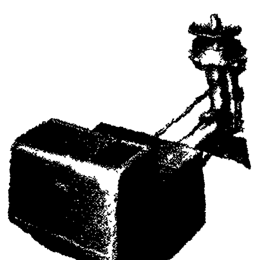




WATER SPECIALTIES

Hot water specialties to suit every installation. They include water heaters, circulators and flow control valves—all top-quality equipment.





OIL BURNERS

Newly engineered. Crane-Line oil burners provide heating comfort at low fuel costs. Simplified design and sturdy construction reduce maintenance.

GAS BURNERS

"Spreader Flame" principle of construction and operation. Produces the very best burning characteristics and eliminates losses due to incomplete combustion or excess air.

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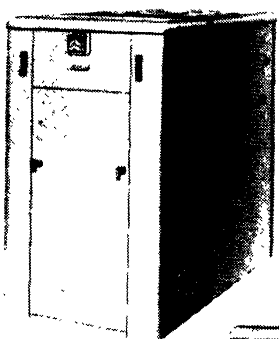
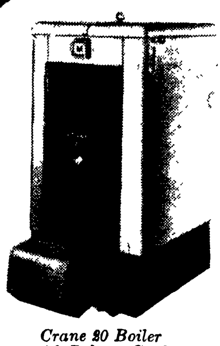


Crane 16 Boiler-Burner Unit



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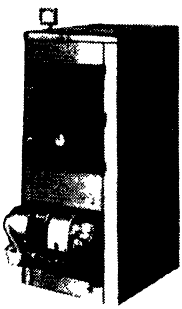
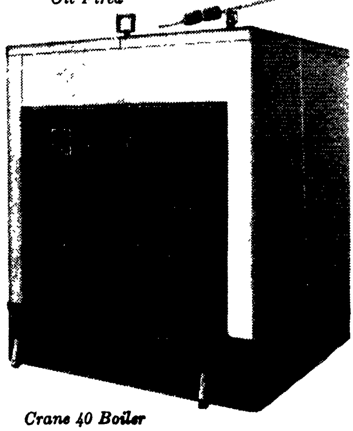


Crane 14 Boiler, Oil Fired



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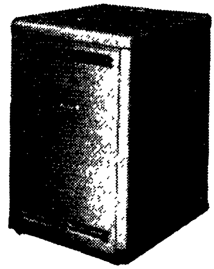

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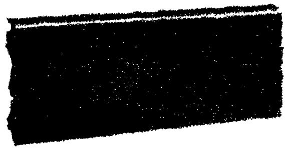
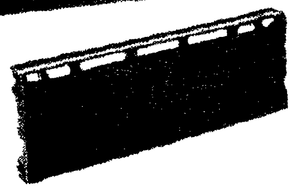


Type R



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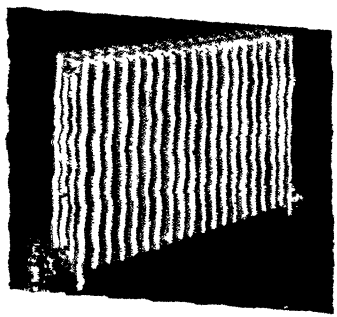

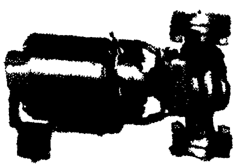

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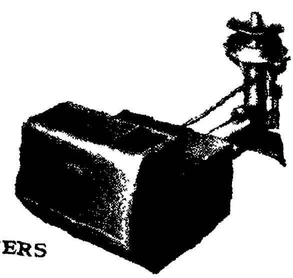


GAS BURNERS

"Spreader Flame" principle of construction and operation. Produces the very best burning characteristics and eliminates losses due to incomplete combustion or excess air.

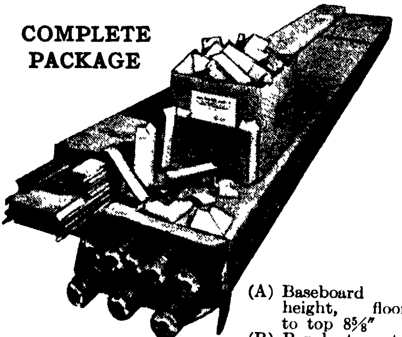
General Automatic Products Corporation

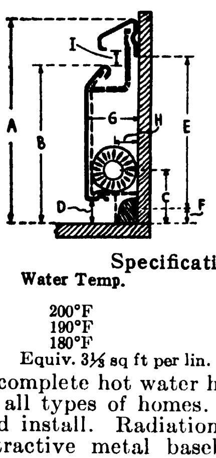
2300 Sinclair Lane **Baltimore 13, Md.**

Manufacturers of a Complete Line of Gas and Oil Heating Equipment

FLOORLEVEL BASEBOARD RADIATION

FLOORLEVEL BASEBOARD HOT WATER HEATING





(A) Baseboard height, floor to top 8⅝″
(B) Panel top to floor 6¾″
(C) Center of element to floor 2¼″
(D) Floor to bottom of panel 1″
(E) Screw holes in bracket, center to center 6¾″
(F) Floor to center of lower screw holes ½″
(G) Back of panel to wall 2⅜″
(H) Wall to center of element 1³⁄₁₆″
(I) Opening, top of panel to lower moulding 1″

Specifications

Water Temp.	Capacity (per lin. foot)
200°F	500 Btu
190°F	445 Btu
180°F	390 Btu

Equiv. 3⅓ sq ft per lin. ft. at 200° water.

A complete hot water heating system to fit all types of homes. Easy to handle and install. Radiation is concealed in attractive metal baseboard which replaces the normal wooden baseboard.

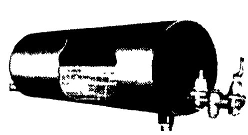
A COMPLETE PACKAGE

Packaged in 3 cartons . . . each package consists of enough material to do the average installation for the amount of square footage required. The packages are so sized in 6 assortments to fit any size installation. Carton No. 1 contains the fin type heating element, carton No. 2 contains the front paneling and top moulding and carton No. 3 contains the panel and moulding splices, wall brackets, end terminals, miter corners, element orifices and wood corner blocks.

"J" SERIES BOILER BURNER
(Oil Fired)

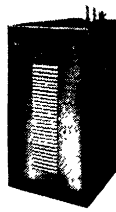
A compact unit especially designed for FLOORLEVEL installations from 200 to 650 sq ft of hot water radiation. A complete unit containing tankless domestic hot water, built-in combustion chamber and oil burner.

DE-AIRATOR TANK



A complete unit for the elimination of air in hot water heating system; also replaces the expansion tank.

"E" SERIES BOILER BURNERS
(Oil Fired)



For larger installations—440 to 1760 sq ft hot water radiation. Vertical water tube wet base boiler. *ASME* approved. All models equipped with 4½ gpm tankless domestic water coils. All units fully enclosed and well insulated.

CONVERSION OIL BURNER

All sizes with capacity of 1 to 20 gpm. G/A vibration free universal coupling.

WARM AIR CONDITIONER (Oil Fired)
Highboy and Lowboy models, 85,000 to 200,000 Btu capacity. Burner & controls.

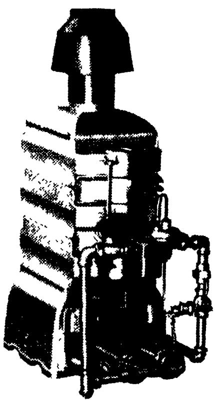
Hook & Ackerman, Inc.

63–56 Shakespeare St.
PITTSBURGH 4, PA.

9 East 40th St.
NEW YORK 16, N. Y.

HYDROTHERM

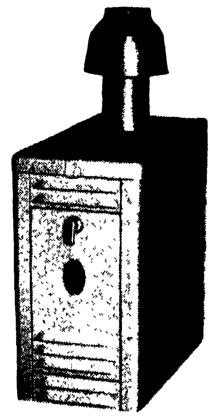
The Midget Automatic Gas Heating Plant
For residential, commercial and industrial hot water systems.



MODEL 2HW5 w/o Jacket
400 sq ft Installed Radiation
375 lbs, 18 in. x 27 in. x 30 in.

HYDROTHERM is engineered for maximum life and maximum fuel economy and fully *AGA* approved for use on Manufactured, Natural and LP gases. Outstanding for:

- **Radiant heating systems**
- **Convection heating systems**
- **Gravity hot water systems**
- **Volume hot water heating**
- **Booster service for 180 deg sterilizing water**



MODEL 2HW3
288 sq ft Installed Radiation
250 lbs, 13 in. x 26 in. x 26 in.

CONSTRUCTION:—All cast iron, the HYDROTHERM absorption unit has horizonta' sections, deep ribbed, overlapped and connected in zig-zag. This fully patented design is to give maximum heat transfer surface for a minimum water volume resulting in great efficiency and quick pick up. Absorption unit is furnished completely assembled and factory tested at 250 lb hydrostatic pressure. All controls including automatic gas control valve, Baso safety pilot, pressure regulator, aquastat and tridicator are enclosed in De Luxe jacket of Hammeroid finish.

CAPACITY RANGE OF HYDROTHERMS

	Hydrotherm Model No.	AGA ratings			Will Supply Sq Ft Installed Radiation Water 170° F	Capacity for Hot Water Storage Tanks Gallons per Hour for Temperature Rise Shown					Shipping Weight Lbs.
		Input Btu/Hr	Output Btu/Hr	Output Sq Ft Water		40°	60°	80°	100°	120°	
Single Un.	2 HW 2	45,000	36,000	240	180	106	70	54	42	35	160
	2 HW 3	72,000	57,000	384	288	170	113	85	68	56	250
	2 HW 5	100,000	80,000	530	400	238	158	119	95	79	375
	2½ HW 3	150,000	120,000	800	600	357	237	178	142	118	600
	2½ HW 4	200,000	160,000	1066	800	476	317	238	190	158	700
	2½ HW 5	250,000	200,000	1333	1C00	595	396	297	238	198	810
Dual Un.	2½ HW 6	300,000	240,000	1600	1200	714	474	356	284	236	1200
	2½ HW 7	350,000	280,000	1866	1400	833	554	416	332	276	1300
	2½ HW 8	400,000	320,000	2133	1600	952	633	475	380	316	1400
	2½ HW 9	450,000	360,000	2400	1800	1071	713	535	428	356	1510
	2½ HW 10	500,000	400,000	2666	2000	1190	792	594	476	396	1620

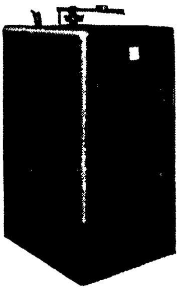
THE NATIONAL RADIATOR COMPANY

MODERN DESIGN HEATING EQUIPMENT

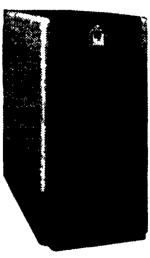
Johnstown Pennsylvania

Branch Offices: BALTIMORE, 2100 St. Paul Street • BOSTON, 620 Newbury Street • BUFFALO, 17 Wells Street • CHICAGO, 400 West Madison Street • DETROIT, 5736 Twelfth Street • NEW YORK, 60 East 42nd Street • PHILADELPHIA, 1218 Cherry Street • PITTSBURGH, 125 First Avenue • RICHMOND, 308 West Cary Street • SAN FRANCISCO, 681 Market Street • WASHINGTON, D. C., 4043 Georgia Avenue, N.W.

"100" Series



"200" Series

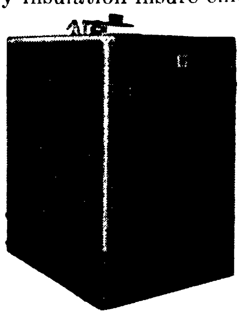


Oil Heating Unit

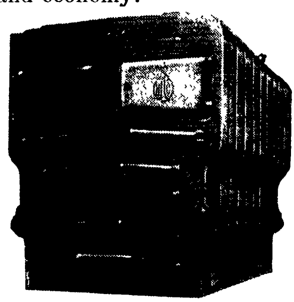
NATIONAL HEAT EXTRACTOR CAST IRON BOILERS. Designed to fit the requirements of automatic heating, National Heat Extractors are provided with many features to insure high operating efficiency and maximum fuel economy. These include extended heating surface, multiple-flue section construction, effectively insulated jacket and doors, and heat conserver baffles. Convertible from hand to automatic firing after installation and readily adaptable to any desired fuel or method of firing. Wide range of both storage and tankless domestic water heaters available.

NATIONAL OIL HEATING UNITS, cast iron or steel, provide complete "one package" equipment designed for maximum efficiency and top performance with this fuel. Complete automatic controls, prefabricated combustion chamber of proper proportions, quiet burner for rear firing and attractive all-enclosing jacket.

NATIONAL GAS BOILERS are modern, compact and designed exclusively for gas firing. Cast iron sections for long life and dependability. Tapered flues, long zigzag fire travel and heavy insulation insure efficiency and economy.



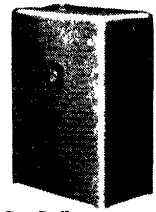
"300" and "400" Series



"500" Series

HEAT EXTRACTOR BOILERS

Boiler Series	Net I-B-R Ratings, Sq Ft: Steam	Net I-B-R Ratings, Sq Ft: Water
100	170 to 470	270 to 855
200	350 to 880	560 to 1560
300	700 to 2300	1120 to 3835
400	2500 to 6000	4000 to 9600
500	4000 to 10300	6400 to 16480



Gas Boiler

OIL HEATING UNITS

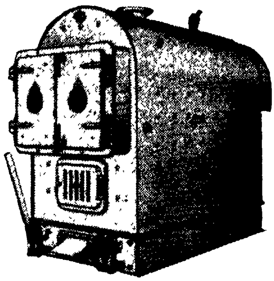
Type of Unit	Net Ratings, Sq Ft: Steam	Net Ratings, Sq Ft: Water
100 Series Cast Iron	230 to 470	430 to 855
200 Series Cast Iron	400 to 880	730 to 1560
Residential Steel	275 to 700	440 to 1120

GAS BOILERS

Boiler Series	Net Ratings, Sq Ft A.G.A. Approved: Steam	Net Ratings, Sq Ft A.G.A. Approved: Water
20 and 22	110 to 390	205 to 715
30 and 33	355 to 985	650 to 1725
40 and 44	445 to 1920	810 to 3250
55	1195 to 9690	2080 to 15505

26"-29"-39" Steel Boiler

NATIONAL STEEL BOILERS meet all the requirements of the *SBI* Testing and Rating Code and the *ASME* Boiler Construction Code. All are inspected and approved by a representative of the *Hartford Steam Boiler Inspection and Insurance Company*. The 18 in. and 23 in. Series RESIDENTIAL STEEL BOILERS (for oil or gas) are designed for smaller homes. The 26 in., 29 in., and 39 in. Series RESIDENTIAL STEEL BOILERS for Hand and Automatic Firing are designed for larger installations. The Commercial Series are adaptable to automatic or hand firing and are used for the largest installations.



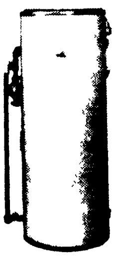
Commercial Steel Boiler

RESIDENTIAL STEEL BOILERS

Boiler Series	SBI Net Ratings, Sq Ft	
	Steam	Water
18" and 23"	275 to 700	440 to 1120
26", 29", and 39".	570 to 3000	910 to 4800

COMMERCIAL STEEL BOILERS

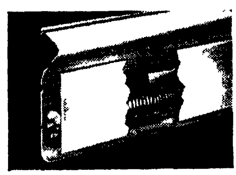
Boiler Type	SBI Net Ratings, Sq Ft	
	Steam	Water
Automatically fired ...	3000 to 35000	4800 to 56000
Hand fired .	2500 to 29170	4000 to 46670



The National Packet Flush Jacket

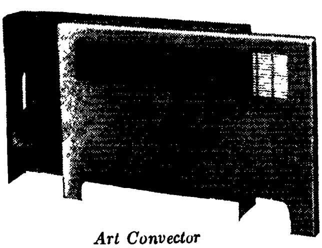
THE NATIONAL PACKET: A compact, all-in-one automatic oil-fired heating unit for small homes. An integral tankless heater provides domestic hot water. Unit also available with white enclosing cabinet. *SBI* Net rating 510 sq ft Water.

The National Packet used with National Art Baseboard provides a modern, low cost, easy-to-install complete hot water heating system for small homes, motels, diners, stores, etc.



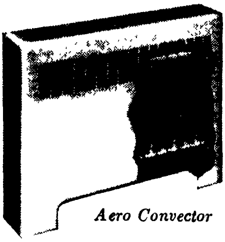
Art Baseboard

NATIONAL ART BASEBOARD—Replaces customary wooden baseboard. Two types, BF (flush-to-wall) and BR (recessed under plaster). Heating element consists of helical fin bonded to a copper tube. Designed for use with forced circulation hot water systems.



Art Convector

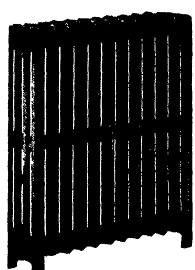
NATIONAL ART CONVECTOR—A nonferrous convector, for flush and semi- or full-recessed installation. Aluminum fins bonded to copper tubing comprise the heating element. Enclosures are made in a variety of types for residential or commercial installation.



Aero Convector

NATIONAL AERO CONVECTOR—Heating element is made of cast iron with fins cast integral with tubes. Adaptable to any type of heating system. Several types are available for residential or commercial installation.

NATIONAL ART RADIATORS—Blend inconspicuously with most decorative schemes. Compact proportions require minimum floor space. A wide variety of sizes and ratings can be furnished.



Art Radiator

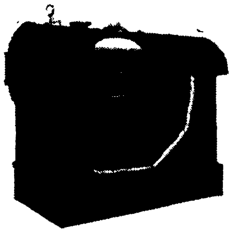
Pacific Steel Boiler DIVISION

Member SBI Reg U.S Pat Off.

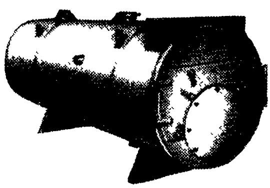
United States Radiator Corporation

Detroit 31, Michigan

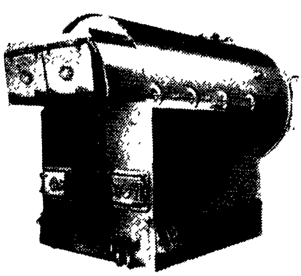
Sales Offices in Principal Cities



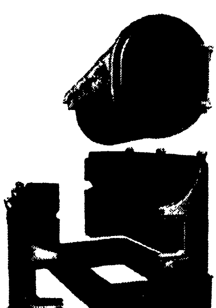
Pacific Standard Firebox Boilers



Pacific Scotch Marine Boilers



Pacific Front Smoke Outlet Boilers



Pacific Split-Firebox Boilers

COMMERCIAL BOILERS

A Complete Line of Low-Pressure Steel Heating Boilers For Commercial Application

All Pacific Commercial Boilers are built using the *ASME* Boiler Code Standards as minimums, and rated in accordance with *Steel Boiler Institute* code.

PACIFIC STANDARD FIREBOX BOILERS

Pacific Standard Firebox Boilers for mechanical firing, stoker, oil or gas are built in capacities of 2680 to 56830 sq ft for steam and in corresponding capacities for water. Design is of two types: Low Water Line and High Firebox.
Pacific Direct Draft and Smokeless Boilers for coal firing are built in capacities of 2200 to 35000 sq ft for steam and in corresponding capacities for water.

PACIFIC SCOTCH MARINE BOILERS

Pacific Scotch Marine Boilers for oil or gas firing. Capacities of 5470 to 42500 sq ft *SBI* rating steam and in corresponding capacities for water.

PACIFIC FRONT SMOKE OUTLET BOILERS

Pacific Front Smoke Outlet Boilers for hand, stoker, oil or gas firing are built in capacities of 4000 to 42500 sq ft for steam and in corresponding capacities for water.

PACIFIC SPLIT-FIREBOX BOILERS

Low Water Line and High Firebox of Pacific Boilers are built in three sections —shell, firebox and base—and require minimum building opening. Where necessary Pacific fireboxes can be split (as illustrated) to allow the boiler to be taken into the building in four sections through any ordinary door or window. No cutting, no welding is required in assembling any Pacific Boiler.

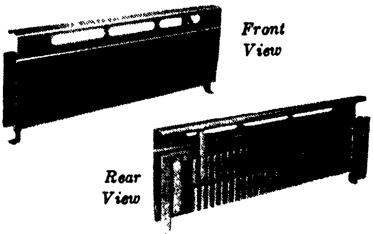
Descriptive Bulletins on Pacific Commercial Boilers will be mailed on request

Member I.B.R. Reg. U. S. Pat. Off.

UNITED STATES RADIATOR CORPORATION
OVER 60 YEARS A GREAT NAME IN HEATING

General Office, Detroit 31, Mich.
Branches and Sales Offices in Principal Cities

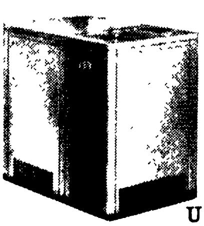
The United States Radiator Corporation carries a complete line of hand, stoker, gas and oil-fired boilers for steam or hot water heating, warm air furnaces, gas burners and oil burners, radiators, baseboard convectors, controls and heating accessories for residential, commercial and industrial heating.



U.S. Radiant Baseboard

Designed to operate with forced circulating hot water systems, U.S. Radiant Baseboard replaces ordinary baseboard. It is light in weight, of all-steel construction and is roughed-in and installed exactly as standard radiator practice. Radiant surfaces, waterways and finned surfaces of U. S. Radiant Baseboard are copper brazed into a single pressure-tight unit. Exterior surfaces are specially treated, then protected with grey zinc chromate primer.
Packed in individual cartons, distinctly marked, U.S. Radiant Baseboard is shipped in unit lengths of 2 to 12-ft inclusive in increments of 1-ft. Write for catalog AR-29G.

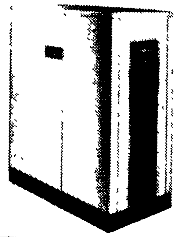
U.S. Solar Flame Oil Burners are available for steam and hot water boilers and warm air furnaces. 3 models. Capacities from 0.6 to 15 gal/hr.



U.S. Solar Flame Oil Burners

U.S. Gas-fired Boilers

U.S. Gas Boilers include a complete range of sizes and capacities from 480 to 12,480 sq ft for steam heating, and from 240 to 20,000 sq ft for hot water heating. U.S. Gas Boilers are furnished with necessary control equipment and are completely automatic in every operation. All units are *A.G.A.* approved.

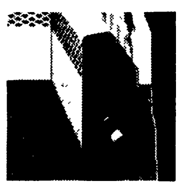


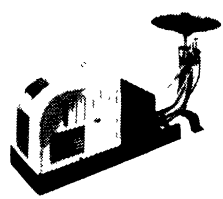
U.S. 12 All-Fuels Boiler

(*Oil ratings shown. For coal-fired ratings, write for Catalog A-399.*)

Boiler No.	I = B = R Net Ratings				I = B = R Gross Output 1000 Btu/hr
	D.C.I.R. Sq Ft		1000 Btu/hr		
	Steam	Water	Steam	Water	
12–30	230	430	55	65	86
12–40	310	570	74	86	114
12–50	390	715	94	107	143
12–60	470	855	113	128	171

U.S. Solar Flame Gas Conversion Burners are *A.G.A.* rated and are available for steam, hot water and warm air systems. 2 models. Capacities from 75,000 to 300,000 Btu's/hr.





U.S. Fin-Ray Baseboard Radiation

U.S. Solar Flame Gas Burners

U.S. Fin-Ray Baseboard Radiation for homes and offices is furnished in 1 in. and 1-1/4 in. pipe sizes.
For commercial applications U.S. Fin-Ray Commercial Radiation is furnished 1-1/4 in. and 2 in. pipe sizes.
Write for catalogs AR 319B (Baseboard) and AR 318 (Commercial) for data and specifications.

THINTUBE RADIATORS

3-Tube

Heights In.	Per Section Heating Surface
25	1.6 Sq Ft

4-Tube

22	1.8 Sq Ft
25	2.0 Sq Ft

5-Tube

22	2.1 Sq Ft
25	2.4 Sq Ft

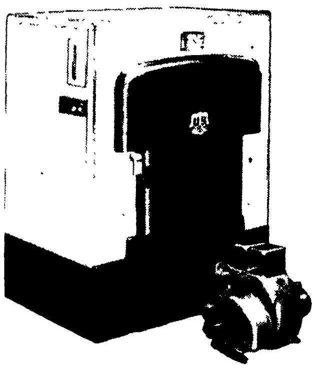
6-Tube

19	2.3 Sq Ft
25	3.0 Sq Ft
32	3.7 Sq Ft

1¾ in. Centers.

U.S. 2 Oil-Fired Boilers

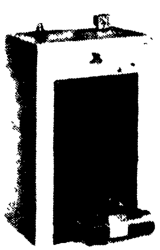
Boiler No.	I = B = R Net Ratings				I = B = R Gross Output 1000 Btu/hr
	D.C.I.R. Sq Ft		1000 Btu/hr		
	Steam	Water	Steam	Water	
2-03	350	645	84	97	129
2-04	500	910	120	137	182
2-05	650	1170	156	176	234
2-06	800	1425	192	214	285



U.S. 28 All-Fuels Boiler

(Oil ratings shown. For Stoker and Hand Fired ratings, write for Catalog A-417B)

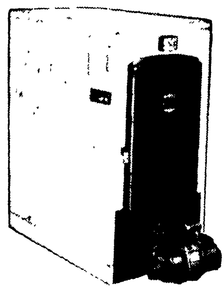
28-30	850	1510	204	227	302
28-35	1055	1850	253	278	370
28-40	1260	2185	302	328	437
28-45	1470	2530	353	380	506
28-50	1690	2880	406	432	576
28-55	1900	3210	456	482	642
28-60	2120	3560	509	534	712
28-65	2340	3900	562	585	780
28-70	2570	4255	617	638	851
28-75	2800	4600	672	690	920



U.S. 46 Oil-Fired Boiler

U.S. 46 Oil-Fired Boiler

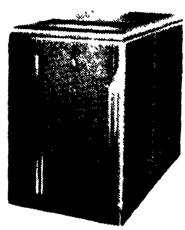
46-3	310	570	74	86	114
46-4	400	730	96	110	146
46-5	550	995	132	149	199
46-6	700	1255	168	188	251



All Fuels Boiler

(Oil ratings shown. For Stoker and Hand Fired ratings, write for Catalog A-417B)

21-30	400	730	96	110	146
21-35	500	910	120	137	182
21-40	600	1080	144	162	216
21-45	700	1255	168	188	251
21-50	800	1425	192	214	285
21-55	900	1590	216	239	318

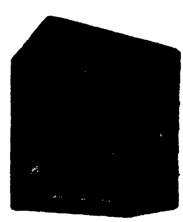


U.S. 25 Oil-Fired Boiler

U.S. 25 Oil-Fired Boiler

25-3	545	985	131	148	197
25-4	905	1610	217	240	320
25 5	1265	2195	304	329	439
25-6	1625	2775	390	416	555
25-7	1985	3345	476	502	669

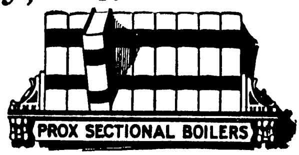
U.S. Vertical or Horizontal Unit Heaters



U.S. Vertical or Horizontal Unit Heaters are available with either 60 cycle AC or 25 cycle AC or DC motors in a wide range of sizes and capacities. Optional control equipment for special applications may be obtained at extra cost. Write for catalog.

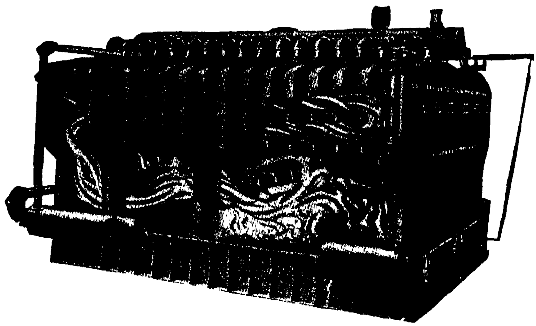
Frank Prox Company, Inc.

Office and plant
1201 So. First St.
Terre Haute, Ind.



PROX BOILERS

For Large Installations in Schools, Theatres, Apartments, Churches, Hospitals, Hotels, Etc. Write for complete catalog.



CAN BE FURNISHED WITH A 22 IN. HIGH CAST IRON BASE FOR AUTOMATIC FIRING

22 in. High base plus 38 in. from normal grate line to crown sheet allows you a total of 60 in. from floor to crown sheet making it convenient for the installation of stoker, oil burner, or gas conversion burner. Front base opening will be made to suit your specifications at no extra charge when so ordered.

6 POINTS PARAMOUNT IN CHOOSING BOILERS

1. **CONTINUOUS SERVICE**—Any heating plant will break if carelessly operated. Broken sections in Prox Boilers can be plugged off and heat maintained.
2. **FUEL ECONOMY**—Short, wide firebox design, full three layer fire travel, low stack temperature, self cleaning flues, conservative ratings.
3. **LONG SERVICE, SAFETY**—Prox cast sectional Boilers represent maximum permanence as compared to steel construction.
4. **QUICK, DRY—STEAMING**—Low water line, small waterways, quick circulation, dry steam assured by steam separating header over Prox Boilers.
5. **REPAIR ECONOMY**—Remove any section like tilting book from bookcase, eliminating tearing down and destroying expensive covering.
6. **INSTALLATION ECONOMY**—Take flow direct from large steam separating header, saving extra cost of additional header construction

STEAM WATER LINE (Max. 72 in.) (Min. 67 in.)			STEAM AND WATER			WATER		
Regular Boiler Number BB-60	Guaranteed Rating Steam	Returns (inches)	Grate Area (sq ft)	Heating Surface (sq ft)	Flows (inches)	Returns (inches)	Regular Boiler Number BB-60	Guaranteed Water Rating
S10	11440	1-3	24.50	570	1- 8	4-4	W10	17160
S11	12540	1-3	24.50	627	1- 8	4-4	W11	18810
S12	13640	1-3	24.50	684	1- 8	4-4	W12	20460
S13	14740	1-3	24.50	741	1- 8	4-4	W13	22110
S14	15840	1-3	27.60	798	2- 8	4-4	W14	23180
S15	16940	1-3	27.60	855	2- 8	4-4	W15	25414
S16	18040	1-4	27.60	912	2- 8	4-4	W16	27060
S17	19140	1-4	27.60	969	2- 8	4-4	W17	28710
S18	20240	1-4	30.60	1026	2- 8	4-4	W18	30360
S19	21340	1-4	30.60	1083	2- 8	4-4	W19	32010
S20	22440	1-4	30.60	1140	2- 8	4-4	W20	33660
S21	23540	1-4	30.60	1197	2-10	4-4	W21	35310
S22	24640	1-4	33.70	1254	2-10	4-4	W22	36960
S23	25740	1-4	33.70	1311	2-10	4-4	W23	38610
S24	26840	1-4	36.70	1368	2-10	4-4	W24	40260
S25	27940	1-4	36.70	1425	2-10	4-4	W25	41910

Shaw-Perkins Manufacturing Company

201 E. Carson St. **Pittsburgh 19, Pa.**

Cutaway—Shaw Model A

MANUFACTURERS OF

Shaw Panel Radiators—Wall Hung or Free Standing

Perkins Industrial and Ceiling Radiators

Shaw Baseboard Panel Radiators

Corner Radiators

Marine Radiators

FEATURES———

Long Life—Steam or water path 100 per cent non-ferrous. Strong box-type sectional construction. Entirely factory assembled.

Space Saving—Only three inches thick. High output.

Safe—High pressure test. Rounded corners and grilles.

Clean—Wide fin spacing. Smooth surfaces.

Comfortable—Warm Panel Heating. Gentle air circulation.

Shaw-Perkins Panel Radiators combine two of nature's basic elements—warm circulating air and mild radiant heat rays—in exact engineered proportions to produce activated, vitalized heat which permeates the entire room.

Air is heated in each section of Shaw-Perkins Panel Radiators by the full length rigid steel plates which are positively bonded to the copper tubing containing the steam or hot water. The side walls of each section are also positively locked to the copper tubing and conduct heat directly to the radiant panel. This unique construction assures the exact engineered proportions of warm circulating air and mild radiant heat rays which are necessary for health, well being and efficiency.

Shaw sizes from 3.9 sq ft EDR to 110 sq ft (1 lb steam); Perkins sizes to 187 sq ft (1 lb steam). As high as 582 sq ft in one unit (150 lb steam). Shaw heights from 8 in. to 26 in. in front or top air outlet; Perkins heights from 14½ in. to 32½ in. Supply and return tappings on same or opposite ends. Shaw and Perkins radiators may be used on hot water systems, steam or vapor systems, high pressure steam systems and high temperature water systems.

The many sizes and models of Shaw-Perkins Panel Radiators make it possible to select and to cover—with one specification—the correct Shaw-Perkins unit for any heating requirement—from one catalog—one set of heating tables—one source—one company. All sizes and types retain the same basic appearance which gives every Shaw-Perkins installation "the professional touch."

Represented in all principal cities. Send for catalog "Modern Radiation"

SPENCER HEATER

LYCOMING-SPENCER DIVISION

WILLIAMSPORT PENNSYLVANIA

SBI

Sales Representatives

Allentown, Pa.	Cincinnati, Ohio	Harrisburg, Pa.	Minneapolis, Minn.	Portland, Ore.
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Birmingham, Ala.	Detroit, Mich.	Knoxville, Tenn.	Philadelphia, Pa.	Seattle, Wash.
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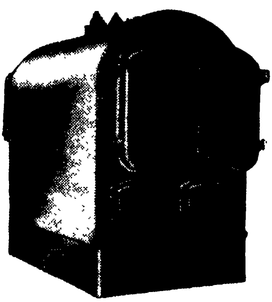
STEEL AND CAST IRON HEATING BOILERS

FOR EVERY BUILDING . . . FOR EVERY FUEL

"A", "C", and "R" Series fully approved by Steel Boiler Institute

All products manufactured in strict accordance with the *ASME* code and carry the code seal. Every Spencer meets rated specifications, is easy to install, and assures economical operation.



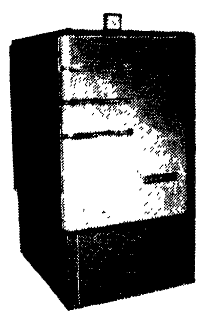
"A" Series

1,800 TO 42,500 SQUARE FEET, STEAM

Steel Commercial Heating Boilers—Exclusive peaked firebox design aids in making the "A" Boiler quick steaming and efficient. Space for either storage tank or instantaneous type service water coils. Larger sizes can be cut in half to move through narrow openings.

570 TO 3,000 SQUARE FEET STEAM

Steel Commercial and Residential Heating Boilers—For oil burner, stoker, or hand firing. With heavy-duty doors and frame precision-ground for airtight fit. Adaptable to front, rear, or side installation of oil burner or stoker. Available with domestic hot water coils and attractive insulated jacket.



"C" Series

320 TO 900 SQUARE FEET STEAM, NET LOAD

Steel Residential Heating Boilers—Excellent for small homes requiring economical heat and instantaneous hot water. Available with attractive beauty jacket.


"R" Series


"21" Series

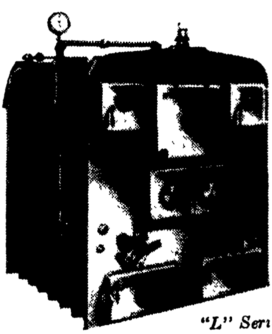
340 TO 1,000 SQUARE FEET STEAM,

Cast Iron All-Purpose Heating Boilers—Especially suitable for homes where owner intends to convert to different type of fuel or firing at a later date. Has attractive jacket and special glass observation ports in fire and ashpit doors. Precision-ground, iron-to-iron sectional fit requires no caulking.

290 TO 740 SQUARE FEET, STEAM
Cast Iron Sectional Magazine Feed Boilers—

Highly efficient for burning economical buckwheat anthracite or pea size coke. No motors or moving parts. Fuel feeds automatically down unique Spencer sloping grate with minimum attention. Permits conversion to oil heat.


"F" Series

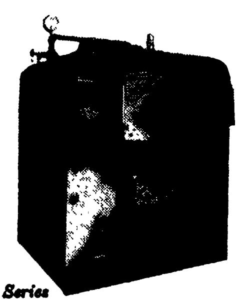

"L" Series

1430 TO 4,510 SQUARE FEET, STEAM

Cast Iron Magazine Feed Boilers—Residential or commercial. An economical heating unit for use with buckwheat anthracite. No motors or moving parts. Fuel feeds automatically down unique Spencer sloping grate, and requires only five to ten minutes' attention each day. May be converted to other types of firing if desired.

5,000 TO 18,740 SQUARE FEET, STEAM
Steel Tubular Magazine Feed Boilers—

For larger buildings, apartments, industry. One of the most economical boilers on the market. Designed for economical sizes of anthracite. Magazines hold enough fuel for 24 hours. Automatic feed, no moving parts. "V"-shaped duplex grate construction allows firing half of boiler during mild weather.

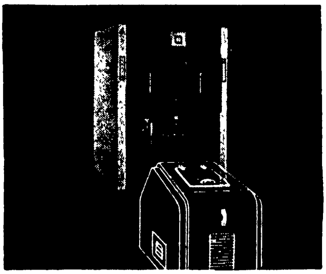

"M" Series

The H. B. Smith Company, Inc.

Westfield, Mass.

Branch offices and Sales Representatives in Principal Cities

A complete line of modern cast iron sectional boilers for residential, commercial and industrial heating and for domestic hot water supply



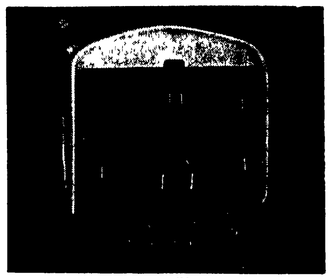
20 Mills

15-20-25 SMITH-MILLS BOILERS

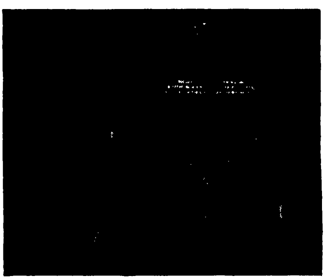
Capacities 200 sq ft to 2275 sq ft steam radiation. This complete line of modern push nipple boilers is available in models for oil, gas, stoker and hand firing. Provisions have been made for built-in domestic hot water heaters and controls.

MILLS WATER TUBE BOILERS
Series 24-34-44

Capacities 900 sq ft to 13,380 sq ft of steam radiation. Independent header type construction—tens of thousands of these famous Mills Boilers are installed in schools, hospitals, apartment houses, stores, and other commercial and public buildings. Models for hand and all types of automatic firing.



44 Mills



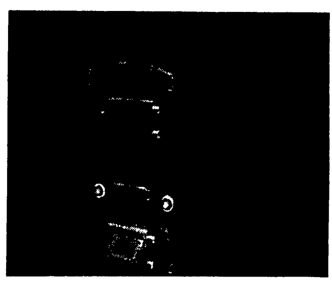
60 Smith

42 AND 60 SMITH BOILERS

May be used in batteries for heating loads up to and over 100,000 sq ft steam radiation. Many of these large units installed in industrial plants furnish steam for process requirements as well as for heating and domestic hot water.

SMITH HY-TEST BOILERS

Smith Hy-Test Boilers for hot water supply, are available in several models and many sizes for tank capacities to 20,000 gal. Constructed of the finest quality grey iron castings, these Hy-Test units are carefully tested at high pressures before shipment. The No. 17 series, for example, is tested at 350 lbs hydrostatic pressure—a higher test pressure than is customary for cast iron boilers.



17 HY-Test

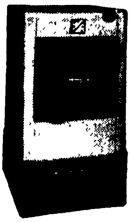
Complete catalog information describing Smith boilers is filed in current issues of Sweet's "Architectural" and Domestic Engineering Catalog Directory

Weil-McLain Company

Manufacturing Division: **Michigan City, Ind. and Erie, Pa.**
General Offices: **641 W. Lake Street, Chicago 6, Ill.**

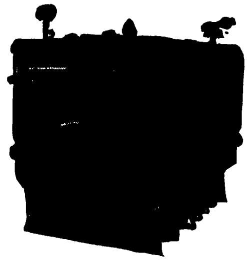
NEW YORK OFFICE: 501 Fifth Avenue

Weil-McLain Boiler and Radiator service is made conveniently available through local stocks carried by Weil-McLain Distributors in most of the important distributing centers.



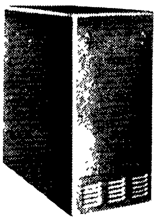
← Nos. 57, 67, 77, 87 All-Fuel Boilers

Conversion type boilers for hand or automatic firing. Connected Load Ratings: Steam 210 to 2300 sq ft, Water 340 to 3835 sq ft.



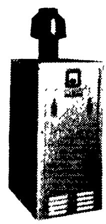
Square-Type Boilers →

Sectional boilers for larger installations 28, 40 and 44 Series. Connected Load Ratings: Steam 1,580 to 11,300 sq ft, Water 2700 to 18,080 sq ft.



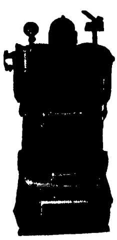
← No. B-5 and B-6 Oil Heating Units

Boiler burner units available with flush or extended jacket, Type A or Type N burner. Net I=B=R Rating: Steam 325 to 690 sq ft, Water 600 to 1235 sq ft.



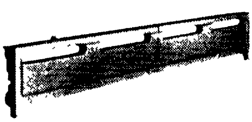
Type G and H Gas Boilers →

Jacketed gas boilers for natural, mixed, manufactured gas or liquefied petroleum gases. *A.G.A.* approved. Connected load ratings: Steam 390 to 1240 sq ft, Water 360 to 2160 sq ft.



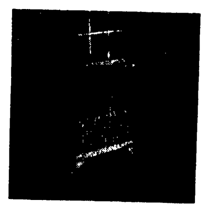
← Round-Type Boiler

Unjacketed Round Boiler with corrugated heating surfaces for hand or automatic firing. Connected Load Ratings: Steam 310 to 900 sq ft, Water 490 to 1590 sq ft.



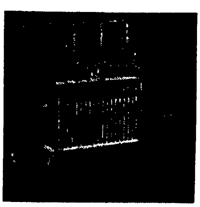
← Snug Baseboard Panels

Snug cast-iron baseboard panels giving combination of radiant and convected heat. A complete line of metal accessories for finishing is available. Height 7-¼ in. Rating 2.4 sq ft per lineal foot.



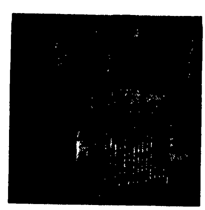
Raydiant "Recessed" →

A Radiant convector type all cast-iron Radiator. Made in "Recessed," also Partially Recessed types.



← Solray Radiator

Free standing, all cast-iron Cabinet-type Radiator with metal cover top. Three heights: 21, 24, and 27 inches. One depth in 18 in. height.



Junior Radiator →

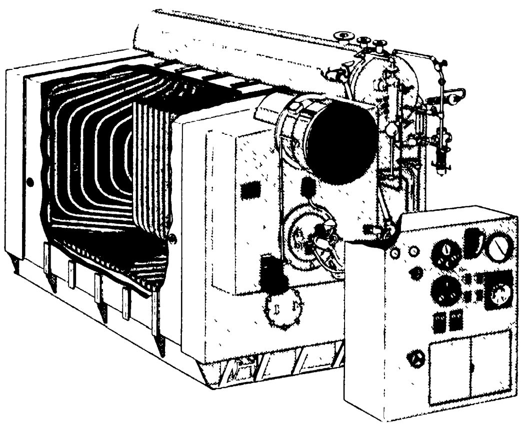
Smaller Tubular type Radiation which conserves space. Available in 1¾ in. centers in 3, 4, 5 and 6 tube widths and 13 to 32 in. heights.

The Babcock & Wilcox Co.

85 Liberty Street, **New York 6, N. Y.**

Offices in All Principal Cities

Water-tube Boilers for Stationary Power Plants, and Marine Services. . .Superheaters. . .Economizers. . .Air Heaters. . .Pulverized Coal Equipment. . .Chain Grate Stokers. . .Oil, Gas, and Multifuel Burners. . .Seamless and Welded Tubing and Piping. . .Refractories. . .Process Equipment.



B & W INTEGRAL FURNACE BOILER, TYPE FM

Complete Shop-Assembled Steam Plant

This compact, self-contained unit combines all the desirable advantages of "packaged" steam plants with the service-proved economy and dependability of larger B&W Integral-Furnace water-tube boilers that have been heavy favorites for years among power companies and all kinds of industrial plants.

The Type FM unit is expressly designed for small and medium sized plants, institutions, and buildings faced with problems of skilled operating labor, excessive fuel consumption, and costly maintenance. Even large operators may find their steam-load characteristics better suited to a battery of these smaller, shop-assembled units than to a single tailor-made installation of the same aggregate capacity.

The Type FM boiler is available in standardized sizes for steam requirements of 2800 to 25,000 lb per hr at pressures up to 275 psi. It is unusually sensitive to load changes; is fast-steaming; features automatic push-button operation with special provisions for safety; is suitable for outdoor as well as indoor installations with gas and/or oil firing.

Cleaver-Brooks Company

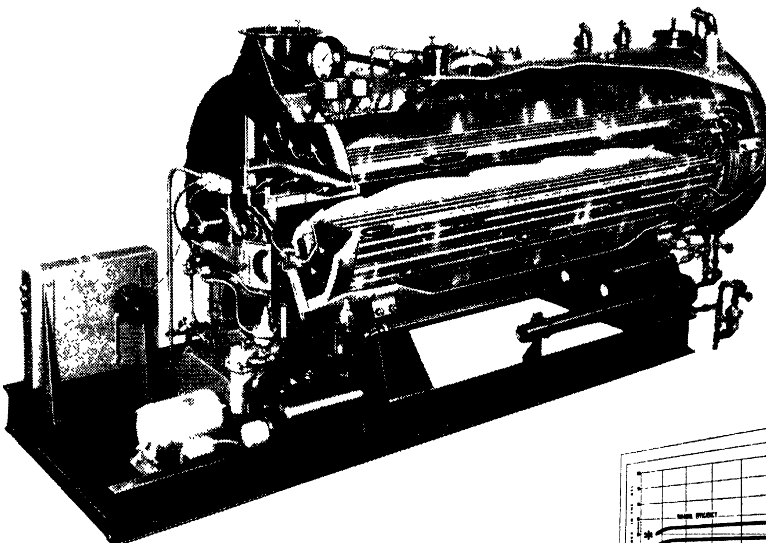
465 E. Keefe Ave., **Milwaukee 12, Wisconsin**

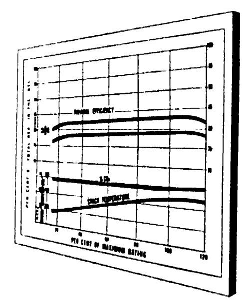
Cleaver-Brooks Steam Boilers—Self-Contained, Gas Fired. Complete 15 to 500-hp, Oil Fired. Factory Tested 15 to 250 lb psi, Oil & Gas Fired.

STANDARD SIZES

HP	15	30	50	80	100	125	150	200	250	300	350	400	500
Approx. rated capacity Lbs. Steam Per Hour 212°F.	515	1050	1725	2760	3450	4300	5200	6900	8600	10500	12000	13800	17000

BURN OIL. BURN GAS. BURN OIL & GAS WITH EQUAL EFFICIENCY. Diagrammatic illustration in cross section representative of the Four-Pass design found in Cleaver-Brooks models. The outstanding feature of the Cleaver-Brooks steam boiler is its proven high efficiency. Economy of operation is assured by establishing an efficient flame and then absorbing the maximum amount of heat in the four passes.





CLEAVER-BROOKS HELPS YOU SELECT THE EXACT STEAM BOILER YOU WANT TO MEET YOUR SPECIFIC NEEDS

JOB RATED—Each Cleaver-Brooks steam boiler is job rated to meet the steam requirements of your plant. A competent sales engineer helps analyze your load demands and takes into account these conditions:

• Maximum steam requirements in pounds per hour or boiler horse-power. • Whether your steam load is constant or fluctuating, and the degree of fluctuation. • The heating load requirements. • Standby requirements. • Steam pressure required at point of use. • Future growth or expansion of plant facilities. • Pop valve settings. • Per cent of total feed water returned as condensate. • Temperature of condensate. • Pressure of make-up water. • Type of fuel available. • Electric current characteristics. • Space and machinery arrangement. Suggested boiler room layouts. • Plant altitude above sea level.

Look for the "CLEAVER-BROOKS" listing under the boiler Section of your classified Telephone Directory.

Combustion Engineering-Superheater, Inc.

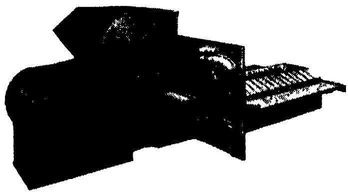
All Types of Fire Tube and Water Tube Boilers
Mechanical Stokers

Complete Steam Generating Units
Pulverized Fuel Systems

200 Madison Avenue, **New York 16, N. Y.**

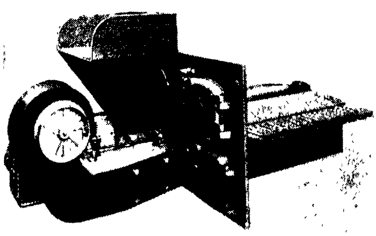
Offices in all principal cities of the United States and Canada

More than 20,600 C-E Stokers purchased to date



C-E Skelly Stoker

Type E Stoker



C-E Low Ram Stoker

C-E Spreader Stoker

C-E SKELLY STOKER—A compact, self-contained unit with integral forced-draft fan, designed to burn either bituminous coal or anthracite. Alternate fixed and moving grate bars assure lateral distribution of fuel. Automatic control is standard equipment. Approximate range—20 to 300 rated boiler horsepower.

TYPE E STOKER—A single retort, underfeed stoker designed to burn a variety of bituminous coals. Available with steam, mechanical or electro-hydraulic drive, it has a long established reputation for dependability. Approximate range—up to 600 boiler horsepower.

C-E LOW RAM STOKER—A single retort, stationary grate underfeed stoker for bituminous coals. Approximate range—20 to 200 rated boiler horsepower.

C-E SPREADER STOKER—A simple, rugged overfeed stoker designed to burn a wide variety of coals. Fines are burned in suspension and the coarser coal on a grate which may be of either continuous discharge or dumping type. Rate of coal feed and air supply may be regulated over a wide range and are adaptable to automatic control. Applicable to boilers from about 100 boiler hp up.

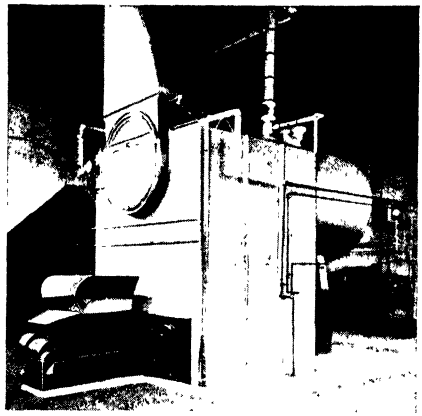
OTHER TYPES—In addition to the stokers described above, the C-E line comprises Traveling Grate and Chain Grate Stokers—including Coxe and Green designs—and the C-E Multiple Retort Stoker. These stokers are generally applicable to medium sized and larger boilers and collectively are suitable for all kinds of solid fuel. The C-E line is complete and designed to meet every need.

C-E BOILERS—Comprise all fire tube and water tube types—see typical examples on opposite page—including designs to suit all conditions of fuel, load and space. C-E Boilers range in capacity from 1000 to 1,000,000 (or more) lb of steam per hour.

Separate Catalogs describing each of these products are available. B 456-A

C-E PREMIER BOILER AND SKELLY STOKER

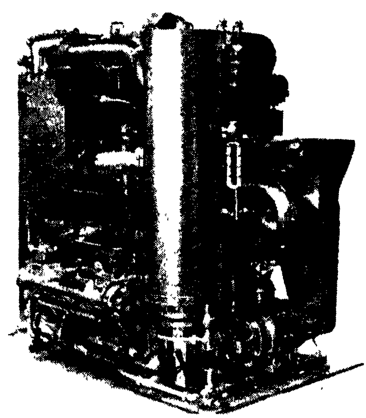
For small plants . . . from 75 to 375 developed hp . . . pressure to 150 psi or higher . . . burns bituminous coal or anthracite . . . oil or gas.
Economical and efficient, the Premier Boiler-Skelly Stoker Unit combines two service-proved components. The Premier Boiler has long been one of the most popular types in small plants. Compact and sturdily built, it can be installed in a minimum of space. Seams are fusion welded and stress relieved. The Skelly Stoker efficiently burns either bituminous coal or anthracite. In addition, provision is made for adding burners in the bridge wall for oil or gas firing . . . if desired.



C-E Premier Boiler and Skelly Stoker

C-E RE-CIRCULATION STEAM GENERATOR

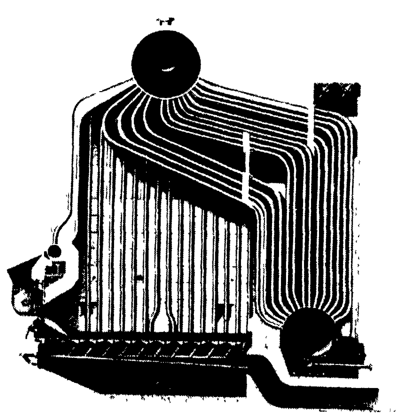
Completely automatic . . . full capacity and pressure in 3 minutes . . . sizes up to 6000 lb per hr . . . operating pressure to 300 psi. Gas and distillate fuel oil firing.
This complete steam generating plant has wide application wherever steam is required for processing or heating . . . in industry as well as in schools, hospitals and other institutions. Controlled forced re-circulation and a unique method of feedwater treatment assure minimum maintenance, high efficiency and maximum output per unit of space. Requiring a floor space only 5 ft by 7 ft and a height of but 7 ft, the unit needs no special foundation and is furnished complete with all auxiliaries. Push button controlled, it is ideally suited for unattended operation and for either continuous or intermittent loads.



C-E Re-Circulation Steam Generator

C-E VERTICAL-UNIT BOILER, TYPE VU-10

For medium-sized and smaller plants . . . pressure to 475 psi . . . capacity to 60,000 lb per hr, or more . . . suitable for any type of fuel.
A standard unit, the VU-10 is especially designed for industrial load conditions and plants having limited operating personnel. It is of symmetrical design, and steam is released evenly across the full width of the unit. Gas flow is uniform, heat absorption is efficient and draft loss is low; it is essentially a "quick steamer." The boiler is bottom supported with no outside supporting steel; there are no seals or slip joints at the grate line to cause air leaks. Type VU-10 boilers are adaptable to stoker firing or may be fired with oil or gas.



C-E Vertical-Unit Boiler, Type VU-10

Cyclotherm Corporation

Sales and Executive Offices

Oswego, New York

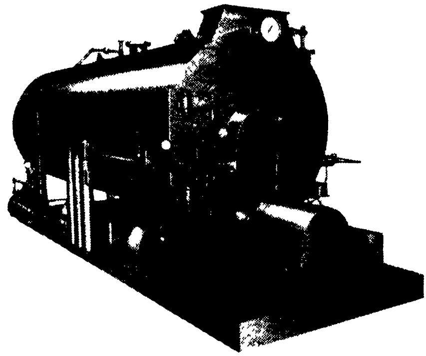
Factory Distributors throughout the United States

CYCLOTHERM STEAM AND HOT WATER GENERATORS

Operate on Cyclonic Combustion principle. Completely automatic with full safety controls. Oil, gas or interchangeable gas and oil firing. Only four connections required.

The Cyclonic Combustion operating principle causes the fuel to be burned while spiraling at high velocity around the inside wall of the combustion chamber, forming virtually a tube of flame. This provides uniform and highly efficient heat transfer over the entire furnace wall without hot spots. Complete utilization of heat transfer areas provides quick heat as well as greater than 80 per cent efficiency with the reduced maintenance of a two-pass design.

15 to 80 hp models are equipped with "on-off" control; 100 to 500 hp models have modulated control. 80 hp units with modulating controls furnished at customers request. Automatic combustion safeguard, low water and other safety controls are included.



Sizes from 18 to 500 hp

Pressure Range

Available in standard low pressure units or high pressures up to 200 psi. Hot water generators are also available in all sizes.

Standard Burner Arrangements

Fuel	Cyclotherm Model No.
Light Oil, 1 to 3	C-600 thru C-17,500
Heavy Oil, 5, 6 and Bunker C	C-2800 thru C-17,500
Gas, mfd., mixed and natural	C-600 thru C-17,500
Comb. lt. oil and gas	C-600 thru C-17,500
Comb. hy. oil and gas	C-2800 thru C-17,500

STANDARD RATINGS AND DIMENSIONS

Model No.	Max. HP Rating	Output BTU Per hr	Steam Units		Hot Water Units		Overall Dimensions (Inches)		
			Steam Per Hr. (Pounds)	Equivalent Direct Radiation (Sq Ft)	Gallons Per Hr (100°Rise)	Equivalent Direct Radiation (Sq Ft)	Length	Height	Width
C-600	18	600,000	600	2,500	743	4,005	71	44½	27½
C-1000	30	1,005,000	1,035	4,185	1,240	6,700	80¼	55	41
C-1400	40	1,340,000	1,380	5,580	1,655	8,930	110¼	55	41
C-2100	60	2,010,000	2,070	8,375	2,480	13,400	124	63¾	45½
C-2800	80	2,680,000	2,760	11,165	3,305	17,865	152	66⅜	55
C-3500	100	3,350,000	3,450	13,955	4,135	22,330	162	73½	64
C-4400	125	4,187,500	4,315	17,445	5,170	27,915	173½	84⅜	68½
C-5200	150	5,025,000	5,175	20,935	6,205	33,500	197½	91½	76½
C-7000	200	6,700,000	6,900	27,915	8,270	44,665	224½	98¾	81¾
C-8700	250	8,375,000	8,625	34,882	10,337	55,782	251½	98¾	81¾
C-10500	300	10,050,000	10,350	41,850	12,405	66,900	252	102½	88
C-12000	350	11,725,000	12,075	48,840	14,475	78,115	252	102½	88
C-13800	400	13,400,000	13,800	55,830	16,540	89,330	252	102½	88
C-17500	500	16,750,000	17,250	69,764	20,674	111,564	294	102½	88

The Dewey-Shepard Boiler Co.

Sales Office: 1311 N. Capitol Avenue

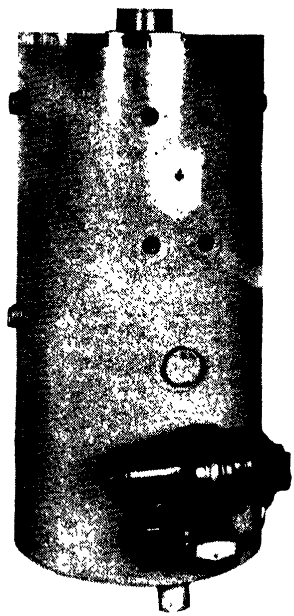
Indianapolis, Ind.

STEEL FIRE TUBE BOILERS & HEATERS
GAS & OIL FIRED

VERTICAL FIRE TUBE UNITS capacities 95,000 to 1,000,000 Btu
***ASME* CODE CONSTRUCTED**

Dewey-Shepard boilers & water heaters (oil & gas fired) are built on the patented tube with-in a tube principle. All units are jacketed with fiber-glass insulation ready for instant hook-up.

- Boilers automatically welded.
- Oil burners *Underwriters'* approved.
- Gas burners *A.G.A.* listed.
- Built-in coils optional.



NET RATING

Model	*Water, Sq Ft	Steam, Sq Ft	Btu
B	465	290	70,000
C	580	365	87,500
D	700	435	105,000
E	930	580	140,000
F	1,165	730	175,000
G	1,400	875	210,000
H	1,640	1,025	246,000
I	1,865	1,165	280,000
J	2,330	1,455	350,000
K	2,800	1,750	420,000
L	3,730	2,330	560,000
M	4,450	2,780	666,666

Boiler ratings do not include allowance for Built-in Coils.
* Based on heat emission rate of 150 Btu per sq ft only. For other emission rates see *SBI* rating code.

OIL OR GAS BOILERS

You Get 80% or More Heat Absorption with *the Tube within a Tube*
This "Tube within a Tube" action results in pre-heating the incoming cold water before it comes in contact with the metal surfaces to which radiant heat is being applied. By this application you offer your customers many advantages such as:

●LESS FUEL USED●LOWER INSTALLATION COST●NO LIME—NO SCALE● 20 PER CENT LESS HEAT LOSS THROUGH THE STACK●NO EXPANSION, CONTRACTION OR CONDENSATION● CLEANEST HEAT POSSIBLE● SUFFICIENT HOT WATER AT ALL TIMES

In the "Tube within a Tube" boiler there is one square foot of heating surface for every gallon of water. This method transfers every possible unit of heat to the water and results in EFFICIENT AND ECONOMICAL OPERATION.

Dutton Boilers

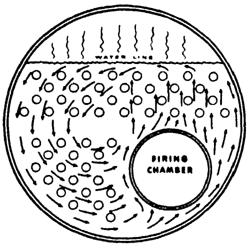
Division Hapman-Dutton Co.

639 Gibson Street, **Kalamazoo, Michigan**

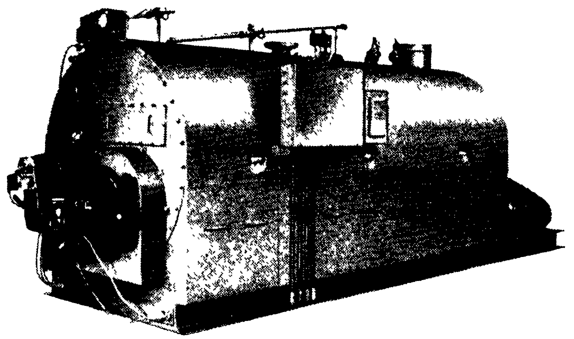
For Dutton Representatives, see "Boilers" in Classified Phone Book

Four types, many sizes . . . From 5 hp up . . . Firing with gas, light or heavy oil, stoker coal or by hand . . . For processing, power, high and low pressure steam and hot water heating . . . For Industrial Plants, Dairies, Laundry and Dry-Cleaning Plants, Chemical and Food Processing, Hotels, Institutions, etc.

Off-Center Firing With EconoTherm Packaged Models—Gas or Oil



EconoTherm water circulation diagram shows rotary motion.



Available in 9 sizes, EconoTherm is a compact power "package." Requires no heavy foundation or stack.

EconoTherm Models are fully automatic, self-contained, "packaged" units with full safety controls. Of 3-pass fire-tube design, they feature a modified Scotch Internal Furnace located off-center in the welded shell. Off-center firing creates a rotary water circulation, resulting in more uniform water temperature, fast steaming, a steady water line at all times, and high quality dry steam. Higher water column above furnace provides greater safety, extra efficiency.

Rotary combustion in furnace, large steam storage space, and five or more square feet of water heating surface per rated hp combine to deliver a guaranteed efficiency rating of 80 per cent with steam dryness rating of 99 plus 0/0 dry steam under proper operating conditions.

Induced draft fan pulls gases through boiler for positive draft and fast firing. Only a vent pipe is needed—no expensive chimney or stack! Boiler shell is one-piece welded construction, X-ray inspected, complying with all *ASME* requirements. Every unit gets full running test and is accompanied with certified *ASME* Data Report.

OTHER DUTTON TYPES

These include EconoMist automatic HRT with 5-way convertible firing (15 **to 105 hp), EconoMaster verticals (5 to 60 hp), and EconoMizer models (10 to 25 hp).**

ECONOTHERM—RATINGS AND DIMENSIONS

	ECONOTHERM Model Number	Sq Ft Water Heating Surface	Lbs Steam Per Hr	Equivalent Horsepower
4610 (oil)	4710 (gas)	125	773	25
4820 (oil)	4720 (gas)	250	1387	50
5835 (oil)	5735 (gas)	375	2456	75
5850 (oil)	5750 (gas)	544	3754	100
5865 (oil)	5765 (gas)	650	4485	125
5880 (oil)	5780 (gas)	814	5617	150
5890 (oil)	5790 (gas)	905	6210	175
58100 (oil)	57700 (gas)	1000	6900	200
58125 (oil)	57125 (gas)	1252	8639	250

Farrar & Trefts

Incorporated

ESTABLISHED 1863

20 Milburn Street, Buffalo 12, N. Y.

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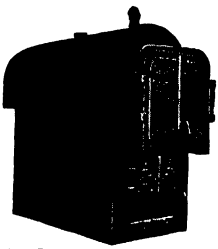


The Bison Compact Boiler
Series 100 and 900

Firebox Return Tubular Boiler
Series 500 and 600

The F&T Bison Compact Welded Heating Boiler is designed to have a large furnace volume, the proper volume of water, the right amount of steam liberating surface, the correct volume for steam storage and a balanced circulation. The result is a remarkably steady water line—**A Balanced Boiler.**

This boiler requires a minimum amount of floor space and is easy and inexpensive to install. It is reasonable as to cost and is economical in operation. Construction is in accordance with *ASME* Code for 15 lb working pressure and boilers are designed for hand or mechanical firing. Sizes from 2680 to 42,500 sq ft of steam radiation.

The Bisonette Compact Boiler has the same characteristics as the larger Bison Compact Boiler, and is designed for installation in large residences and small business establishments where the advantages inherent in a **Steel** boiler are desired.

Firebox Return Tubular Heating Boilers are Quality Boilers constructed to meet the high standards of Heating Engineers and to give unfailing service under all conditions. Being economical to install and operate, they are highly favored by architects and engineers for heating Schools, Hospitals, etc.

There are two types of Firebox Boilers, the Up-Draft Type and the Down-Draft Type. Both types are made of welded or riveted construction for heating purposes at 15 lb working pressure and riveted, or, welded, x-rayed and stress relieved for power purposes at 100, 125, and 150 lb working pressure in accordance with *ASME* Code. Sizes from 5,470 to 42,500 sq ft of steam radiation are designed for hand or mechanical firing.

Scotch Wet Arch Boilers are designed so that no refractory tile are required at the top of the rear combustion chamber. The steam space extends the entire length of the boiler and the furnace is entirely surrounded by water which permits immediate maximum heat absorption. This special design results in a boiler that is extremely efficient to operate and is maintained at a minimum cost.

The Scotch Wet Arch Boiler is a self-contained unit. It can be moved easily and can be installed on two saddles. No expensive foundation or pit is required. No external brickwork is needed. Because of its short length, low height and low water line, this compact boiler unit can be installed in small spaces where there is lack of headroom and where no other type will fit. It is designed for oil, gas or mechanical firing, in accordance with the *ASME* Code for 15 lb working pressure. Sizes range from 3,160 to 42,500 sq ft of steam radiation.

Ratings of all these boilers conform to *SBI*.

Write for Complete Catalog.

Reg. U. S. Pat. Off.

Fitzgibbons Boiler Company, Inc.

101 Park Avenue, New York 17, N. Y.

Sales Branches in Principal Cities

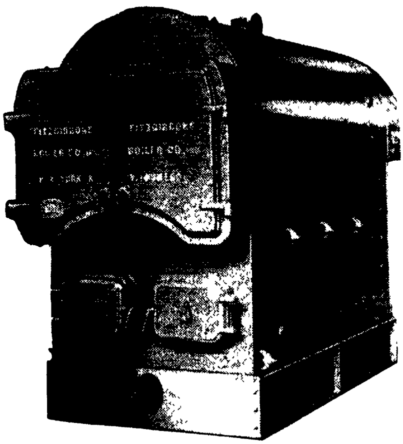
Member

Manufactured at Oswego, N. Y.

Steel Boilers since 1886

Reg. U. S. Pat. Off.

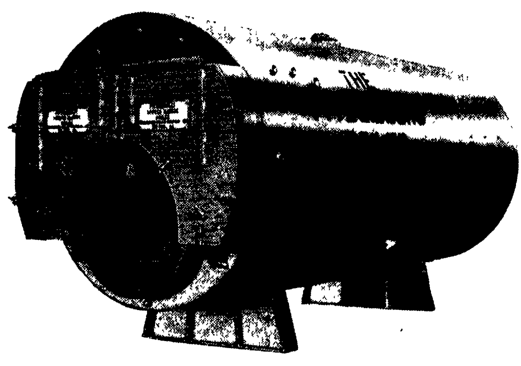
PRODUCTS—STEEL BOILER HEATING and POWER BOILERS for all fuels and all heating systems. Capacities to meet requirements of any building. Built and rated according to *ASME* and *SBI* Codes and "*Hartford*" inspected.



"D" TYPE STEEL BOILER

For mechanical firing with oil, gas or stoker in ratings from 2680 to 42,500 sq ft (steam), 4280 to 68,000 sq ft (water), 643 to 10,200 Btuh (1000's) EDR. 15 lbs Steam—30 lbs Water.
For hand fired coal in ratings from 2200 to 35,000 sq ft (steam) 3520 to 56,000 sq ft (water), 528 to 8400 Btuh (1000's) EDR.
The steel boiler for heating apartments, office buildings, theatres, schools, hospitals and other large commercial buildings. Provides year 'round service hot water, without the need of a separate storage tank, with the famous Fitzgibbons TANKSAVER®, a copper coil submerged in the boiler water.

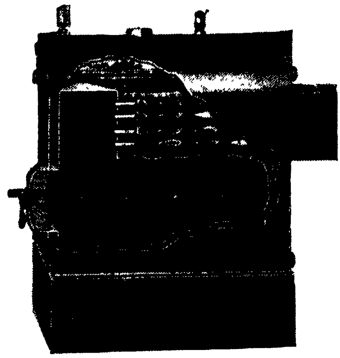
"SCOTCH MARINE" STEEL BOILER

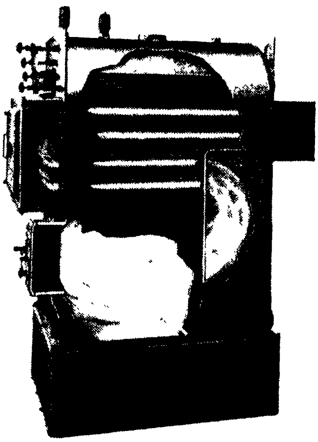


A Full Wet Back—Low Pressure Heating Boiler for oil and gas firing in sizes from 5470 to 42,500 sq ft (steam) and 8750 to 68,000 sq ft (water), 1313 to 10,200 Btuh (1000's) EDR. 15 lbs Steam —30 lbs Water. This compact, efficient Fitzgibbons boiler has the design advantages of complete water jacketing of all heating surfaces, including the rear furnace wall, thus eliminating usual rear dry-wall refractory lining and corresponding up-keep and repair expense.

"80" SERIES STEEL BOILER

For smaller commercial buildings and large residences for oil, gas or anthracite coal firing in sizes from 1100 to 3000 sq ft (steam), 1760 to 4800 sq ft (water) and 264 to 720 Btuh (1000's)—all ratings *SBI* net.
High in efficiency and fuel economy due to the generous firebox dimension and fast water circulation. Large tankless domestic hot water capacity year 'round, with Fitzgibbons TANKSAVER.®



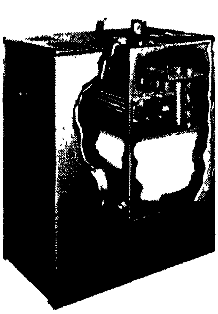


"R.Z.U." JUNIOR® STEEL BOILER

Especially recommended for bituminous coal stoker firing, although excellent for oil, gas and anthracite firing as well. In sizes from 1100 to 3000 sq ft (steam), 1760 to 4800 sq ft (water) and 264 to 720 Btuh (1000's)—all ratings *SBI* net.
Efficient firing with bituminous coal or other fuels is assured because of the ample tube area and diameter, low draft loss and generally large combustion space.

"400" SERIES STEEL BOILER

The aristocrat of steel boilers for residences. Quick heating, low fuel consumption. Abundant hot water, winter and summer, without the need of a storage tank with the Fitzgibbons TANKSAVER®. In sizes from 400 to 900 sq ft (steam), 640 to 1440 (water) and 96 to 216 Btuh (1000's).



"200" SERIES STEEL BOILER

A steel boiler for the smallest homes with all the famous characteristics of Fitzgibbons design and construction. Low in fuel costs, high in heating efficiency with the Fitzgibbons TANKSAVER® for tankless, year 'round domestic hot water. In sizes from 275 to 320 sq ft (steam), 440 and 510 (water), 66 and 77 Btuh (1000's).

Dutton Boilers

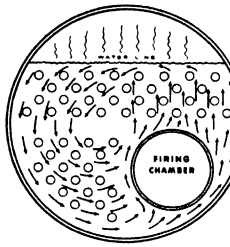
Division Hapman-Dutton Co.

639 Gibson Street, **Kalamazoo, Michigan**

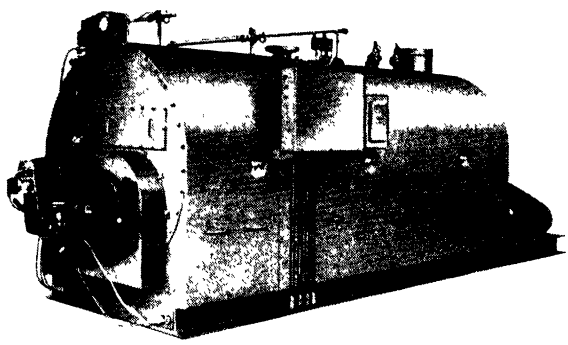
For Dutton Representatives, see "Boilers" in Classified Phone Book

Four types, many sizes . . . From 5 hp up . . . Firing with gas, light or heavy oil, stoker coal or by hand . . . For processing, power, high and low pressure steam and hot water heating . . . For Industrial Plants, Dairies, Laundry and Dry-Cleaning Plants, Chemical and Food Processing, Hotels, Institutions, etc.

Off-Center Firing With EconoTherm Packaged Models—Gas or Oil



EconoTherm water circulation diagram shows rotary motion.



Available in 9 sizes, EconoTherm is a compact power "package." Requires no heavy foundation or stack.

EconoTherm Models are fully automatic, self-contained, "packaged" units with full safety controls. Of 3-pass fire-tube design, they feature a modified Scotch Internal Furnace located off-center in the welded shell. Off-center firing creates a rotary water circulation, resulting in more uniform water temperature, fast steaming, a steady water line at all times, and high quality dry steam. Higher water column above furnace provides greater safety, extra efficiency. Rotary combustion in furnace, large steam storage space, and five or more square feet of water heating surface per rated hp combine to deliver a guaranteed efficiency rating of 80 per cent with steam dryness rating of 99 plus 0/0 dry steam under proper operating conditions. Induced draft fan pulls gases through boiler for positive draft and fast firing. Only a vent pipe is needed—no expensive chimney or stack! Boiler shell is one-piece welded construction, X-ray inspected, complying with all *ASME* requirements. Every unit gets full running test and is accompanied with certified *ASME* Data Report.

OTHER DUTTON TYPES

These include EconoMist automatic HRT with 5-way convertible firing (15 to 105 hp), EconoMaster verticals (5 to 60 hp), and EconoMizer models (10 to 25 hp).

ECONOTHERM—RATINGS AND DIMENSIONS

	ECONOTHERM Model Number	Sq Ft Water Heating Surface	Lbs Steam Per Hr	Equivalent Horsepower
4610 (oil)	4710 (gas)	125	773	25
4820 (oil)	4720 (gas)	250	1387	50
5835 (oil)	5735 (gas)	375	2456	75
5850 (oil)	5750 (gas)	544	3754	100
5865 (oil)	5765 (gas)	650	4485	125
5880 (oil)	5780 (gas)	814	5617	150
5890 (oil)	5790 (gas)	905	6210	175
58100 (oil)	57700 (gas)	1000	6900	200
58125 (oil)	57125 (gas)	1252	8639	250

Farrar & Trefts

Incorporated

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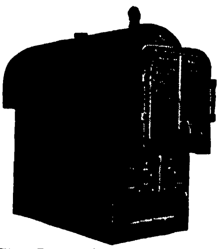
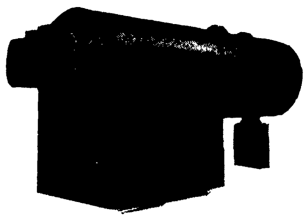


The Bison Compact Boiler
Series 100 and 900



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Manufactured at Oswego, N. Y.

Steel Boilers since 1886

Member

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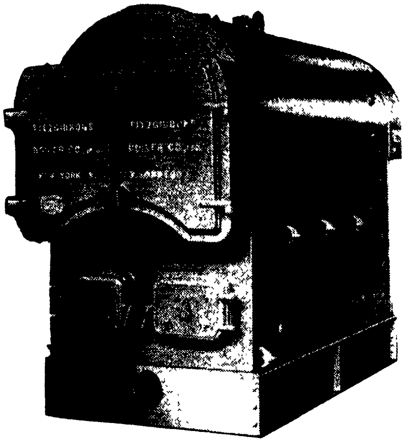

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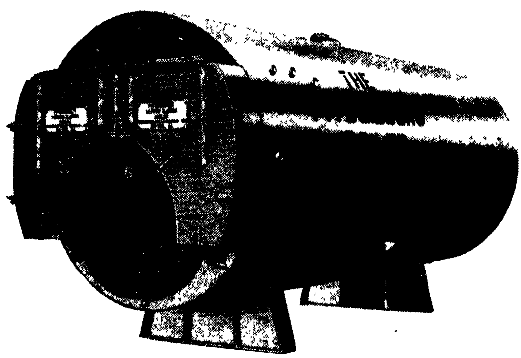

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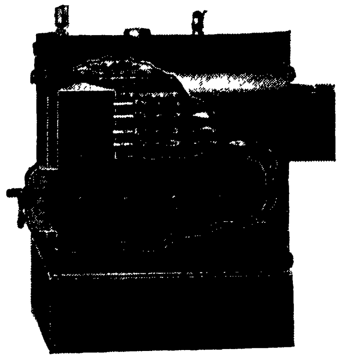

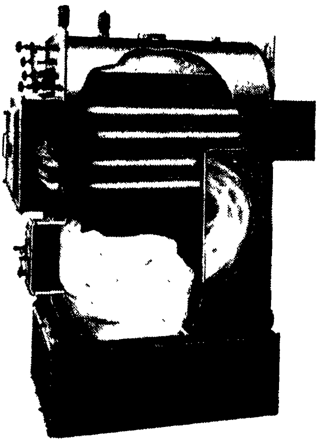

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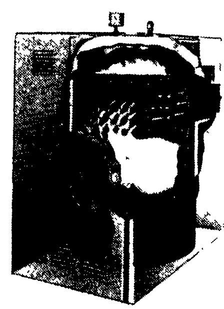

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The International Boiler Works Co.

500 Birch St., East Stroudsburg, Pa.

Sales Offices in Principal Cities

INTERNATIONAL Water Tube Boilers are Supplying DEPENDABLE—LOW COST HEAT AND POWER

for Apartment Buildings, Office Buildings, Schools, Theatres, Hotels, Greenhouses, Industrial Plants. . . and Steam for Low and High Pressure Process Work, throughout the Country.

WATER TUBE BOILER DESIGN ASSURES . . .

- **QUICK STEAMING** . . . due to rapid and directed water circulation.
- **MORE HEAT ABSORPTION** . . . due to extra long three pass gas travel, *across* the entire bank of water tubes.
- **EASY CLEANING** . . . free access to heating surfaces makes cleaning easy.

INTERNATIONAL Water Tube Boilers are cutting fuel costs in thousands of heating installations. Sizes and types for every requirement.

- Complete range of standard sizes Heating Boilers 400-70,000 sq ft steam, Power Boilers 10-600 bhp.
- *Steel Boiler Institute* ratings.
- Heating Boilers 15 lb steam-30 lb water.

  Power Boilers 100-125-150 psi.

  All Boilers *ASME* standard.
- For Oil, Gas, Stoker or hand fired Coal.
- Year round Domestic or Service hot water . . . from immersed copper coil, instantaneous or storage tank type.

FOR FURTHER INFORMATION

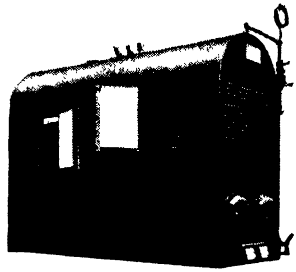
SEE OUR CATALOG IN SWEET'S

For 8 page Catalog and complete specifications,

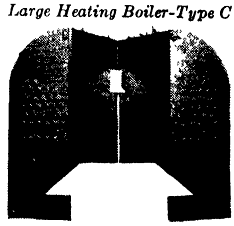
WRITE TO

THE INTERNATIONAL BOILER WORKS CO.

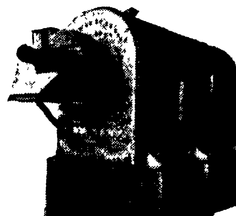
East Stroudsburg, Pennsylvania



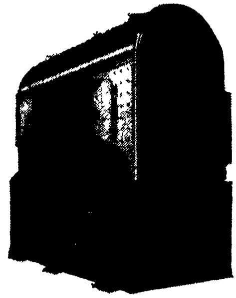
Large Heating Boiler-Type C



Twin Section Heating Boiler-Type C (Also available completely knocked-down-Type KD)



Induced Draft Power Boiler-Type IDH



Power Boiler-Type CR

Johnston Brothers, Inc.

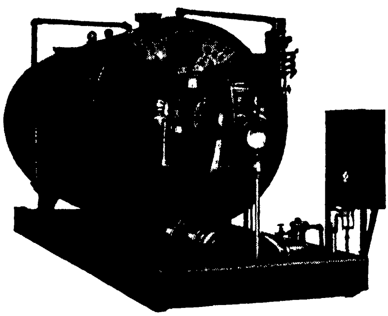
ESTABLISHED 1864

JOHNSTON

Ferrysburg, Michigan

Member SBI Reg. U. S. Pat. Off.

"PACKAGED"
STEAM BOILER UNIT
FULLY AUTOMATIC
HEAVY OIL



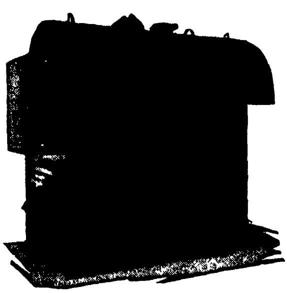
WATER-BACK TYPE,
FORCED DRAFT, THREE-PASS,
TROUBLE-FREE, EFFICIENT.

Catalog-Rating guaranteed and at least 25 per cent overloads readily obtained.

HIGH PRESSURE TYPE from 60 to 500 hp and pressures of 125, 150 and 200 lb psi. (250 lb available as a special)

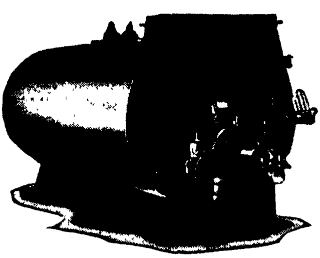
LOW PRESSURE TYPE for heating; 15 lb pressure, and EDR rating from 2190 sq ft to 42,500 sq ft steam.

ASME Code construction. Specification Forms in detail to exactly cover the requirements of any particular job will be furnished upon request and without any obligation whatever. Ask for Bulletin 507.



Firebox Heating Boiler, Compact Type. Oil, Gas, Stoker or Hand Firing. Three-Pass 15 lb pressure, *ASME* Code. Capacities 2190 to 42,500 sq ft, *SBI* rating. Ask for Bulletin 1500.

Also built for High Pressure (125 psi.) Ask for Bulletin 702.

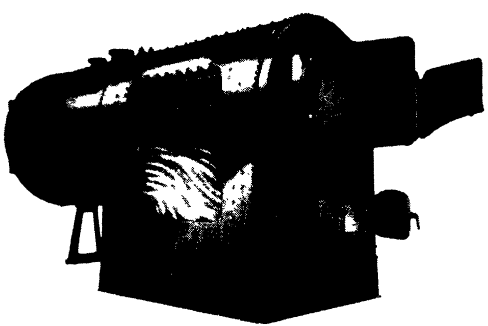


Standard Scotch Boiler Water-back Type, Natural Draft.

POWER PROCESS, HEATING.

25 to 300 hp and pressures 15 lb to 200 lb psi. For mechanical firing with Coal, Gas or Oil. Overloads up to 200 per cent of rating readily developed. Oil or Gas firing equipment and controls can be installed for completely automatic operation and in emergency or fuel scarcity, can be readily converted to Coal firing.

Ask for Bulletin 4000 and 9000.



400 and 500 Riveted Series for Heavy Duty

KEWANEE BOILER CORPORATION

Division of AMERICAN RADIATOR & Standard Sanitary CORPORATION

Kewanee, Illinois

BRANCHES IN 64 PRINCIPAL CITIES

SBI Reg U S Pat Off MEMBER

Steel Heating and Power Boilers, Storage and Domestic Water Heaters

FOR MORE THAN EIGHTY YEARS Kewanee Boilers have been outstanding for Long Life, Efficiency and Fuel Economy. Regardless of the type and size of building, or the fuel used, there is a Kewanee just right for the job. They can be depended upon to produce the steam shown by their rated capacity, and even carry "overloads" of 50 per cent without any sacrifice of efficiency. Kewanee Boilers are built in strict accordance with *ASME* and *SBI* Codes. The Kewanee series include:

HEAVY DUTY RIVETED FIREBOX TYPES: 1380 to 42,500 sq ft, 10 to 304 hp. Steam working pressures 100, 125 and 150 lbs. Single-pass tubes for rear smoke outlet, two-pass for front smoke outlet. WELDED BOILERS: 3000 to 42,500 sq ft. Type "C" with rear smoke outlet; welded firebox two-pass for front smoke outlet. RESIDENCE STEEL BOILERS: 330 to 3000 sq ft. Round and Square "R" with and without jackets. Cottage 510 sq ft water boiler for oil-gas. Water heating Coils either storage tank or tankless.

Boiler No.	576	577	578	579	580 480	581 481	582 482	583 483
SBI Rating Steam, Coal ..Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000
Horsepower ... Hp	25	29	32	36	43	50	61	72
SBI Rating Oil, Gas, Stoker Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150
Horsepower ..Hp	30	35	39	44	52	61	74	87
Width and Length In. x Ft In.	42x8–7	42x9–6½	48x8–11½	48x9–8	54x9–2½	54x10–6	60x11–1	60x12–9
Overall Height Shell ....In.	80½	80½	86½	86½	94½	94½	102	102
Height of Water Line ....In.	73	73	76	76	83	83	88	88

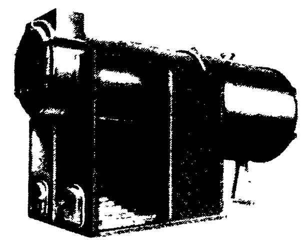
Boiler No.	584 484	585 485	586 486	587 487	588 488	589 489	590 490
SBI Rating Steam, Coal ..Sq Ft	12500	15000	17500	20000	25000	30000	35000
Horsepower ... Hp	89	107	125	143	179	214	250
SBI Rating Oil, Gas, Stoker Sq Ft	15180	18220	21250	24290	30360	36430	42500
Horsepower ..Hp	109	130	152	174	217	261	304
Width and Length In. x Ft In.	66x12–10	66x14–9	72x14–9	78x14–9	78x17–7½	84x17–9½	84x20–0½
Overall Height Shell ....In.	108	108	114	116	116	126	126
Height of Water Line ....In.	93	93	97½	100	100	108	108

Two-Pass Tubular
50 to 180 HP

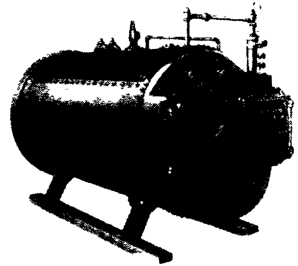
Six sizes for power or industrial process steam, for all fuels. 125 and 150 lbs W.P. All joints are fusion welded. X-ray checked and stress relieved. Quick steaming unit, compact yet extra rugged. Metal casing is designed for refractory lined firebox.

KEWANEE HI-TEST BOILER

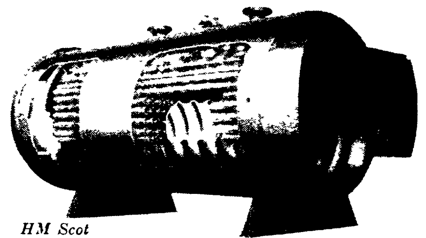
Boiler Number	HT 50	HT 60	HT 75	HT 100	HT 125	HT 150
Horsepower, Hand-fired	50	60	75	100	125	150
Mech.-fired	60	72	90	120	150	180
Pounds steam per hour,						
Hand-fired	1730	2070	2590	3450	4330	5180
Mech.-fired	2070	2470	3110	4140	5180	6210
Fur. vol. above grates. cu. ft	72	90	105	140	181	213
Cylinder diameter—front, in.	42	42	42	48	54	54
—rear, in.	66	66	72	78	84	84
Boiler length overall, ft.in.	12–5	13–5	14–3	15–5	16–2	18–6
Boiler height, floor to top in.	107	113	116	126	142	142
Water line height, in.	95	101	104	111	124	124



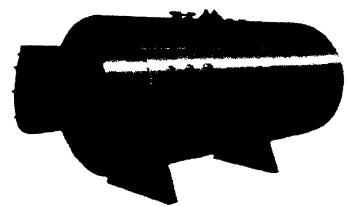
KEWANEE SCOTCH MARINE TYPES High and Low Pressure Boilers



Scottie Jr. 6 to 36 hp



HM Scot 52 to 304 hp



M-8100 Low Pressure

SPECIFICATIONS— Scottie Jr. 125 lbs. S.W.P.

Boiler No.		HM 6	HM 9	HM15	HM20	HM25	HM30
Ratings—Mechanical-Fired	Hp	7	12	18	24	30	36
—Lbs of Steam per hr		242	410	620	830	1050	1240
Ratings—Hand-Fired	Hp	6	9.9	15	20	25	30
—Lbs of Steam per hr		207	342	520	690	865	1050
Shell Diameter	In.	42	48	48	48	54	54
Length Overall, Oil-Gas	In.	82½	84½	105½	126½	127	144
Height Overall Shell	In.	54½	62	62	62	67½	67½
Height of Water Line	In.	46½	53	53	53	57½	57½

SCOTTIE JR. for 6 to 36 hp is an ideal unit for producing high pressure steam at low cost for small loads. Economical to operate with oil, gas or coal at 125 lbs steam working pressure. HM SCOT for 52 to 304 HP has full length corrugated furnace and heavy gage tubes completely surrounded by water. Burns oil or gas. Generates maximum heat from the fuel. 125 and 150 lbs working pressure.

M-8100 is low in first cost and economical to install. Low waterline makes it ideal where headroom is important. Entirely self-contained with return firetubes and tubular furnace. Oil or gas fired.

SPECIFICATIONS—HM SCOT SERIES—125-150 lbs S.W.P.

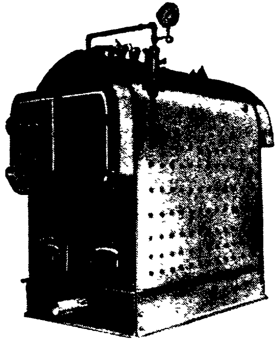
Boiler Number		HM-52	HM-61	HM-74	HM-87	HM-109	HM-130	HM-152	HM-174	HM-217	HM-261	HM-304
Rating, Steam Radiation	sq ft	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Horsepower		52	61	74	87	109	130	152	174	217	261	304
Thousands of Btu per Hour		1750	2040	2479	2916	3643	4373	5100	5830	7286	8743	10200
Boiler Diameter	in.	60	60	66	66	72	78	78	84	90	96	102
Boiler Length	ft in.	10-6	11-8½	12-0	13-8	14-10	15-3	17-3	16-4	17-6	18-6	18-6
Boiler Height, Floor to top of Shell	in.	73	73	79	79	85	91¼	91¼	97¼	103½	109½	115½
Water Line Height	in.	62	62	65½	65½	69½	73½	73½	77½	82¾	87½	92¾

SPECIFICATIONS—LOW PRESSURE SCOTCH MARINE TYPE BOILER

Boiler No.		M-8178	M-8179	M-8180	M-8181	M-8182	M-8183	M-8184
*SBI Rating, Steam Rad	Sq Ft	5470	6080	7290	8500	10330	12150	15180
Btu per hr	1000's	1313	1459	1750	2040	2479	2916	3643
*SBI Net Rating, Steam Rad.	Sq Ft	4500	5000	6000	7000	8500	10000	12500
Btu per hr	1000's	1080	1200	1440	1680	2040	2400	3000
Furnace Volume (SBI min)	Cu Ft	39.1	43.5	52.1	60.8	73.8	86.8	108.5
Shell Diameter	In.	54	54	60	60	66	66	72
Length Overall	Ft In.	9-2½	9-11	9-4	10-5¾	10-7	11-11½	12-9
Height Overall Shell	In.	66-¾	66¾	72¾	72¾	78¾	78¾	84¾
Height of Water Line	In.	54½	54½	59	59	64	64	69½

Boiler No.		M-8185	M-8186	M-8187	M-8188	M-8189	M-8190
*SBI Rating, Steam Rad	Sq Ft	18220	21250	24290	30360	36430	42500
Btu per hr	1000's	4373	5100	5830	7286	8743	10200
*SBI Net Rating, Steam Rad.	Sq Ft	15000	17500	20000	25000	30000	35000
Btu per hr	1000's	3600	4200	4800	6000	7200	8400
Furnace Volume (SBI min)	Cu Ft	130.2	151.8	173.5	216.9	260.3	303.6
Shell Diameter	In.	72	78	78	84	90	96
Length Overall	Ft In.	14-8	14-10½	16-6	18-2	18-8	18-0¾
Height Overall Shell	In.	84¾	90¾	90¾	97	103	109
Height of Water Line	In.	69½	71¼	71¼	74¾	80	84

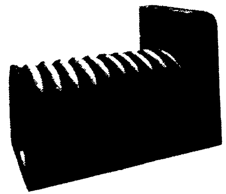
* Water radiation 60 per cent more than steam.



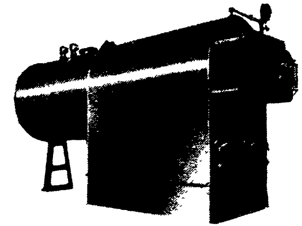
KEWANEE TYPE "C" HI-FIREBOX
with Corrugated Crown Sheet

Because of its ability to produce large amounts of steam quickly, while requiring a minimum of floor space, Kewanee Welded Type "C" has become the choice of engineers for heating larger buildings efficiently and at low cost.

The Corrugated Crown Sheet, developed by Kewanee, provides many advantages. The massive corrugations, made one-at-a-time, retain the same metal thickness throughout, adds strength, and provides additional heating surface which comes in contact with the hottest part of the fire.



KEWANEE "5000" Series Welded



This up-draft firebox boiler burns any fuel. 15 sizes for low pressure heating, features increased heating surface, steam space, furnace volume and water content, yet takes up less room than similarly constructed boilers.

SPECIFICATIONS—TYPE "C" HI-FIREBOX WELDED BOILER — 7L70 Series & 27L70 Series

Boiler No. Oil, Gas, Stoker		7L75	7L76	7L77	7L78	7L79	7L80	7L81	7L82	7L83	7L84	7L85	7L86	7L87	7L88	7L89	7L90
SBI Rating Steam	Sq Ft	3650	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
SBI Net Rating	Sq Ft	3000	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Width x Length	In. x Ft In	36x6–10½	36x7–9	42x7–10½	42x8–6½	42x9–2	48x9–4½	48x10–7	54x9–11	54x11–2½	60x11–6½	66x12–3½	72x12–1½	72x13–4½	78x14–9½	84x14–2½	84x15–11½
Overall Height Shell	In.	84	84	88½	88½	88½	92½	92½	106	106	115½	120½	130½	130½	138	162½	162½
Height of Water Line	In.	75	75	76½	76½	76½	79½	79½	91	91	100½	103	113	113	119	141½	141½
Boiler No. Hand-fired Coal		27L75	27L76	27L77	27L78	27L79	27L80	27L81	27L82	27L83	27L84	27L85	27L86	27L87	27L88	27L89	27L90
SBI Rating Steam	Sq Ft	3000	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
SBI Net Rating	Sq Ft	2500	2920	3330	3750	4170	5000	5830	7080	8330	10420	12500	14580	16670	20830	25000	29170

SPECIFICATIONS—WELDED FIREBOX BOILER — "5000" Series & "6000" Series

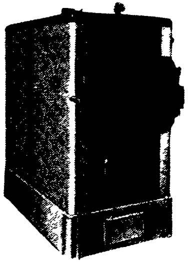
Boiler No. Updraft		5076	5077	5078	5079	5080	5081	5082	5083	5084	5085	5086	5087	5088	5089	5090
Downdraft		—	6077	6078	6079	6080	6081	6082	6083	6084	6085	6086	6087	6088	6089	6090
SBI Rating Steam, Coal	Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
SBI Net Rating	Sq Ft	2920	3330	3750	4170	5000	5830	7080	8330	10420	12500	14580	16670	20830	25000	29170
SBI Rating Steam: Oil, Gas, Stoker	Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
SBI Net Rating	Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Width x Length	In. x Ft In.	42x7–3	42x8–2	42x8–11	42x9–10	48x9–9	48x11–1¼	54x11–1	54x12–9	60x13–2½	60x15–3	66x14–5½	66x16–0½	72x16–8½	78x17–5½	78x19–8½
Overall Height	In.	81	81	81	81	86½	86½	92½	92½	100½	100½	105	105	111	117	117
Height of Water Line	In.	70	70	70	70	73½	73½	77½	77½	84	84	86	86	93	96	96

Rated Capacity for Water Boiler is 60 per cent greater than Capacity for Steam Boiler. Table for two series of Boilers lists maximum dimensions only.

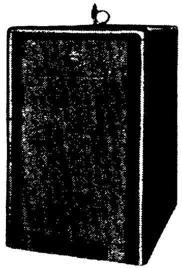
KEWANEE TYPE "R" RESIDENCE BOILERS

Built of the same husky steel plate and by the same skilled workmen as Kewanee's famous larger boilers, Type "R" Welded Series provide all the advantages of their well-known Dependability and Fuel Economy to fully meet heating requirements of homes and small buildings. Any Type "R" can be converted from mechanical to hand-firing or back again without making any change in the boiler proper. Water heating coils, tank or tankless models, can be installed at any time at small extra cost.

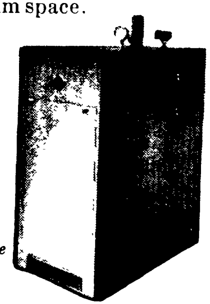
SQUARE-HEAT Type "R" is built in 8 sizes from 740 to 3000 sq ft steam, or 1180 to 3970 sq ft of water radiation. Can be furnished with attractive jacket. ROUND Type "R" comes in 4 sizes ranging from 330 to 900 sq ft steam, or 530 to 1440 sq ft water. Available with square, or insulated round jacket. COTTAGE vertical tube boiler for hot water heating is built for oil or gas firing only; 77,000 Btu or 510 sq ft water radiation. Compact two-tone green jacket occupies minimum space.



Square "R"



Round "R"



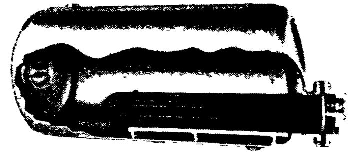
Cottage

SPECIFICATIONS Square-Heat Round "R" Cottage

Boiler No.		3R1	3R2	3R3	3R4	3R5	3R6	3R7	3R8	734	735	736	1737	510
Net Load Steam, Handfired	Sq Ft	740	910	1080	1230	1480	1810	2140	2480	330	450	570	...	....
Water	Sq Ft	1180	1450	1720	1970	2370	2890	3430	3970	530	720	910	..	...
Btu per Hour	1000's	178	218	261	295	355	434	514	595	79	108	137	.	..
Net Load Steam, Mechanical	Sq Ft	900	1100	1300	1500	1800	2200	2600	3000	400	550	700	900	
Water	Sq Ft	1440	1760	2080	2400	2880	3520	4160	4800	640	880	1120	1440	510
Btu per Hour	1000's	216	264	312	360	432	528	624	720	96	132	168	216	77
Heating Surface	Sq Ft	53	65	77	88	106	129	153	177	24	32	41	53	19
Width and Length	In x In.	30x30½	30x36½	30x42½	30x48½	34x42½	34x50½	34x58	34x67	d.24	28	31	31	20
Overall Height from Floor	In.	65	65	65	65	72½	72½	72½	72½	54½	58	60½	60½	56¾
Height of Water Line	In.	55½	55½	55½	55½	62½	62½	62½	62½	44	47	49	49	..

KEWANEE STORAGE WATER HEATERS

Ample hot water for steady service and sudden demands, using live or exhaust steam. 15 standard coil elements in 29 standard size storage tanks. Capacities 95 to 2240 gph.



TABASCO WATER HEATERS

Built of extra heavy steel plate, all-welded, for long life. 5 sizes to heat 165-700 gal raised 50 deg per hour. Burns any fuel. 100 lbs working pressure.

New KEWANEE COTTAGE BOILER

A Direct-Fired Domestic WATER HEATER

Provides plenty of hot water dependably and economically, for working pressures of 100 lbs; amply strong to safely withstand the high water pressures carried in most city mains. 140 gal raised 100 deg per hour. Handsome two-tone insulated jacket. Burns oil or gas only.

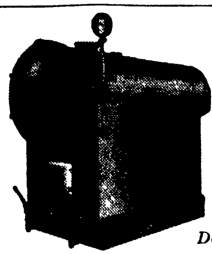
THE TITUSVILLE IRON WORKS CO.

TITUSVILLE, PENNSYLVANIA

DIVISION OF STRUTHERS WELLS CORPORATION

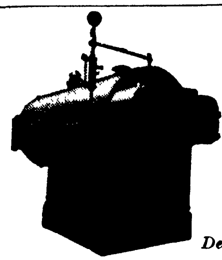
Designers and Manufacturers of

Dependable A.S.M.E. CODE BOILERS...since 1860



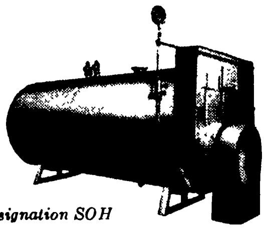
Type Designation CM

Titusville Compact Steel Heating Boilers built in 19 sizes ranging from 129 square feet to 2500 square feet heating surface, and maximum steam working pressure 15 psig.



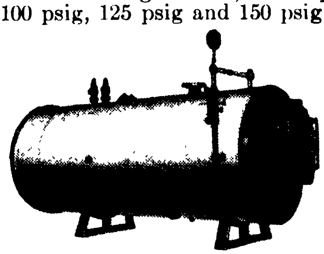
Type Designation WP

Titusville All-Welded Portable High Pressure Firebox Boilers built in 12 sizes ranging from 250 square feet to 2500 square feet heating surface, steam pressure 100 psig, 125 psig and 150 psig.



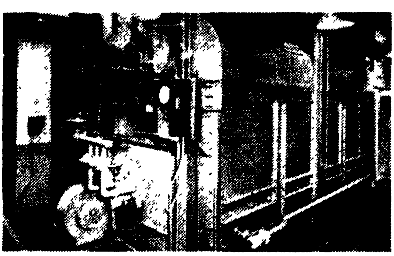
Type Designation SOH

Titusville Scotch Marine Heating Boilers built in 19 sizes ranging from 129 square feet to 2500 square feet heating surface, and maximum steam working pressure 15 psig.



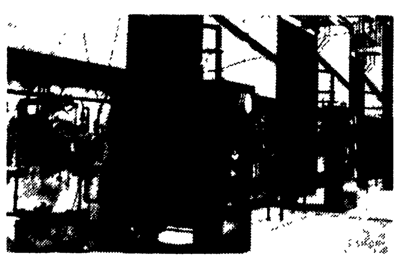
Type Designation Wee Scot to 50 HP, SP *above 50 HP*

Titusville Scotch Marine Power Boilers built in 13 sizes ranging from 97 square feet to 3000 square feet heating surface. Pressures 125 psig and 150 psig.



Type Designation Ticotherm

Titusville Ticotherm Steam Generators built in 13 sizes ranging from 1000 square feet to 5000 square feet heating surface. Pressures 160 psig, 200 psig, 250 psig and higher.



Type Designation TDL

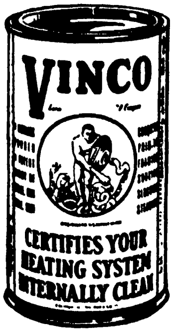
Titusville Three Drum Low Head Water Tube Boilers built in numerous sizes ranging from 729 square feet to 6109 square feet heating surface. Pressures psig, 160 200 psig, 250 psig and higher.

Descriptive technical literature is available on request.

The Vinco Company, Inc.

47 West 63 Street **New York 23, N. Y.**

Only a clean boiler can be an efficient boiler. A clean boiler means saving fuel, as well as safeguarding boiler metal.



Boiler Cleaner
3 and 5 lb. cans

A positively harmless insoluble powder cleaner for new, remodeled and old heating systems. A unique, scientifically processed compound on a special formula not to be confused with other powder boiler cleaners.

What Vinco Boiler Cleaner Does

VINCO removes oil, grease, scale, rust and dirt from the internal surfaces and from the boiler water *without the labor, expense, and uncertain results of blowing boilers over the top or of wasting returns.*

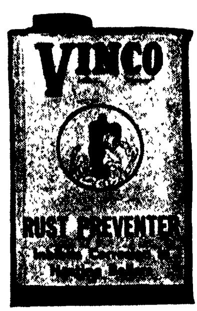
By this thorough cleaning Vinco prevents or cures foaming, priming, surging, and slow steaming.

How Vinco Boiler Cleaner Works

Each minute grain of VINCO powder adsorbs several times its own weight of oil, grease, rust and dirt. These larger grains of adsorbed impurities then settle and are drained through the bottom according to directions on each can.

Vinco Guarantees

1. VINCO contains no potash, lye, soda of any kind, oil, acid, or other harmful ingredients.
2. Purchase price is refunded if results are not as claimed when VINCO has been used according to directions.



Rust Preventer
1 qt. cans only

VINCO RUST PREVENTER

When used after VINCO Boiler Cleaner has removed oil, grease, rust, scale and dirt, it will add and keep the rust inhibiting factors at the optimal constant for a year or more. (Test kit below has complete instructions and chart.)

Vinco Field Test Kit No. 10

VINCO FIELD TEST KIT No. 10

for Testing and Treating Heating Boiler Waters

The kit enables the layman to make simple, rapid tests to diagnose and prescribe correct treatment of boiler waters right on the job.

A new time saving method that permits valid conclusions heretofore requiring complicated and often lengthy laboratory analysis and technique.

Each kit has sufficient material for complete tests on 100 jobs.

Refills cost about 2 cents for testing each job.

SPECIFICATIONS FOR COMPLETE VINCO TREATMENT OF NEW OR REMODELED STEAM, VAPOR, OR HOT WATER SYSTEMS

Do not use as a cleaning agent soda or any alkali, vinegar or any acid. Use Vinco.

1. **AFTER THE SYSTEM IS TESTED AND TIGHT, USE THE PROPER QUANTITY OF VINCO LISTED.**

After this first clean-out of any new or remodeled heating system, Vinco Boiler Cleaner need be used only if more piping, radiation, or another boiler is added to the original installation, or if the system is fouled by unwise cleaning or leak-sealing experiments.

2. After using Vinco Boiler Cleaner, Vinco Field Test Kit should be used to determine and apply the proper quantity of Vinco Rust Preventer. Vinco Rust Preventer should be applied annually or whenever the boiler water is drained for necessary repairs to the system.

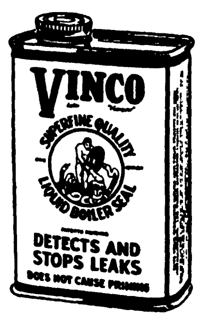
SPECIFICATION FOR OLD HEATING SYSTEMS THAT DO NOT PERFORM PROPERLY

Diagnose and treat according to Vinco Field Test Kit. If a test kit is not available, consult table of quantities on this page and follow directions on Vinco cans.

SPECIFICATION FOR HOT WATER SYSTEMS

If maintained below 200 F, use half quantities listed for treatment of steam systems to remove impurities. If maintained at approximately 200 F or above, use full quantities. Then use test kit to determine proper quantity of Vinco Rust Preventer.

Seal-Off—1 lb. cans 50 and 100 lb drums



Liquid Boiler Seal 1 qt. cans only

CONSULT THIS TABLE FOR NEW AND REMODELED HEATING SYSTEMS AND When a Vinco Field Test Kit No. 10 is not available if cleaning old heating systems.

QUANTITIES OF VINCO (IN POUNDS) REQUIRED FOR HEATING SYSTEMS

(Note that quantities are based on actual installed radiation, not on boiler capacity.)

Sq Ft of Radiation	For Steam or Vapor Systems, to prevent or cure priming or foaming. Also for Hot Water Heating Systems Maintained at approx. 200 F or above.	Annually, to remove rust scale, dirt and for Hot Water Systems below 200 F.
up to 350	3	1½
351 " 600	5	2½
601 " 1100	8	4
1101 " 1400	10	5
1401 " 1800	13	6½
1801 " 2100	15	7½
2101 " 2700	18	9
2701 " 3100	20	10
3101 " 3700	23	11½
3701 " 4200	26	13
4201 " 4600	28	14
4601 " 5000	30	15
5001 " 5300	31	15½
5301 " 5600	32	16
5601 " 5900	33	16½
5901 " 6200	34	17
6201 " 6500	35	17½
6501 " 6800	36	18
6801 " 7100	37	18½
7101 " 7400	38	19
7401 " 7700	39	19½
7701 " 8000	40	20
8001 " 8300	41	20½
8301 " 8600	42	21
8601 " 8900	43	21½
8901 " 9200	44	22
9201 " 9500	45	22½
9501 " 9800	46	23
9801 " 10100*	47	23½

***Above 10100 sq ft. use an additional pound Vinco for each additional 300 sq ft of actual installed radiation.**

REMOVE SOOT WITH VINCO SOOT-OFF SEVERAL TIMES A YEAR

Safely and thoroughly removes the insulating blanket of soot on fire pot, flues and chimney. It also insures against external corrosion (caused by dampness and soot forming sulfuric acid during summer layoff.) No dangerous chemicals.

VINCO SUPERFINE LIQUID BOILER SEAL

A different liquid seal. Unique in that it does not induce priming and foaming. It has no unpleasant smell. Makes lasting repairs of boiler and heating system leaks. Fine to tighten up new jobs. Directions simple.

Quantities—Steam and Vapor Systems—Use 1 quart VINCO Liquid Boiler Seal to each 6 sq ft *grate* area. **Hot Water Systems—**Use 2 quarts VINCO Liquid Boiler Seal to each 6 sq ft *grate* area.

McDonnell & Miller, Inc.

Safety Devices for Steam and Hot Water Heating Boilers . . . Liquid Level Controls

3500 North Spaulding Ave., **Chicago 18, Illinois**

Doing One Thing Well

The McDonnell line consists of (1) boiler water feeders, low water fuel cut-offs and combined feeders and cut-offs for low pressure steam boilers of all sizes and types, hand-fired or automatically fired; (2) pump controls, low water fuel cut-offs and low water alarms for steam boilers of any size or type with pressures up to 150 lbs; (3) safety relief valves (Btu-rated) for hot water heating boilers and domestic hot water heaters; boiler water feeders and low water fuel cut-offs for hot water heating boilers; (4) make-up water feeders for receiving tanks and other liquid level controls for special applications; (5) water level controls for humidity pans of warm air furnaces.

The selection of controls for usual applications is shown in the following service recommendation. Most of these products are described in the following three pages.

Service Recommendations

(1) STEAM BOILERS

Boiler Size	Maximum Steam Pressure	McDonnell product to use
HAND FIRED JOBS		
Up to 5000 sq ft	25 lbs	No. 47 Boiler Water Feeder
Above 5000 sq ft	35 lbs	No. 51 Boiler Water Feeder
Any size	75 lbs	No. 53 Boiler Water Feeder
AUTOMATICALLY FIRED JOBS		
Boiler Water Feeder—Low Water Cut-off Combinations		
Up to 5000 sq ft	25 lbs	No. 47-2 Feeder Cut-off Combination
Above 5000 sq ft	35 lbs	No. 51-2 Feeder Cut-off Combination
Any size	75 lbs	No. 53-2 Feeder Cut-off Combination
LOW WATER FUEL CUT-OFFS		
Any size	20 lbs	No. 67 Low Water Fuel Cut-off
Any size	50 lbs (Steam or water)	No. 63 Low Water Fuel Cut-off
Any size	150 lbs	No. 150 or No. 157 Low Water Fuel Cut-off

(Industrial or Process boilers—Specific recommendations on receipt of complete data.)

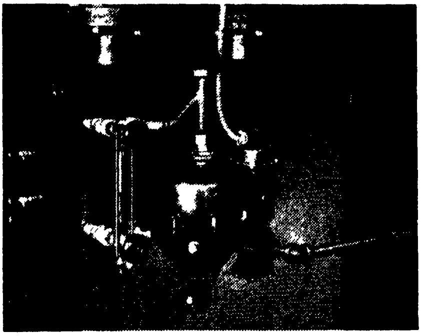
(2) HOT WATER HEATING BOILERS

Opening Pressure	Btu. per Hour Capacity	McDonnell Product to use
30 lbs.	242,900	No. 33 Safety Relief Valve
30 lbs.	496,700	The 233 Safety Relief Valve Assembly
30 lbs.	769,400	The 333 Safety Relief Valve Assembly

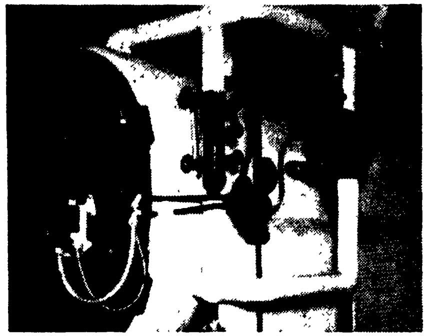
(3) DOMESTIC HOT WATER TANKS AND HEATERS

Opening Pressure	Btu per Hour Capacity	McDonnell Product to use
45 pounds	321,300	No. 3345 Safety Relief Valve
75 pounds	308,700	No. 3375 Safety Relief Valve
100 pounds	423,900	No. 33100 Safety Relief Valve
125 pounds	513,900	No. 33125 Safety Relief Valve
150 pounds	605,700	No. 33150 Safety Relief Valve

Boiler water feeders, feeder-cut-off combinations, and low water cut-offs . . . for low and moderate pressure steam boilers



McDonnell No. 47-2 for heating boilers under 5000 sq ft capacity. Maximum steam pressure, 25 lbs.



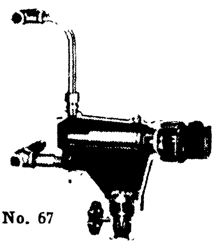
McDonnell No. 51-2 for heating boilers over 5000 sq ft capacity. Maximum steam pressure, 35 lbs.

McDonnell No. 47-2, shown installed above, maintains a safe water level in boiler by feeding water whenever necessary. If emergencies such as priming or foaming permit water to fall to ¼ in. in gauge glass, cut-off switch cuts current to burner until emergency has passed. Has "Quick-Hook-Up" for fast, accurate installation right in gauge glass tappings; "cool" feed valve; extra-deep sediment chamber; *ASME*-approved blow-off valve. Also available for hand fired boilers without No. 2 switch, as No. 47 Feeder.

McDonnell No. 51-2, shown installed above, is same as No. 47-2 described at left, except it has greater feeding capacity for larger boilers, and is installed with 1 in. equalizing pipes instead of "Quick-Hook-Up." Also available for hand fired boilers without No. 2 switch, as No. 51 Feeder.

For boilers operating at higher pressures, from 35 to 75 lbs, use McDonnell No. 53-2 (or No. 53, without switch.)

McDonnell No. 67 Low Water Cut-off
For automatically fired steam boilers of any size
Maximum steam pressure, 20 lbs.

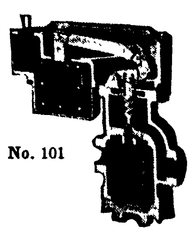


No. 67

Has the McDonnell "Quick-Hook-Up" for quick, easy and trouble-proof installation in gauge glass tappings; deep sediment chamber with large quick-opening blow-off; packless, non-binding construction; adjustable terminal box to make wiring neat and easy; dependable snap-action twin switches. One switch can be used to sound low water alarm, or control McDonnell No. 101 Electric Water Feeder described below. Second switch cuts current to burner if water level drops to ¼ in. in gauge glass.

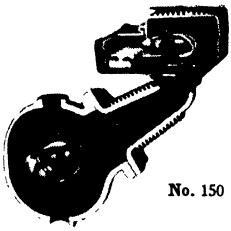
McDonnell No. 101 Electric Boiler Water Feeder
For boilers up to 5000 sq ft capacity

For use with No. 67 Low Water Cut-off or with McDonnell "built-in" Low Water Cut-offs which are standard equipment on many modern heating boilers. It converts the cut-off into a feeder cut-off combination as described at top of this page.

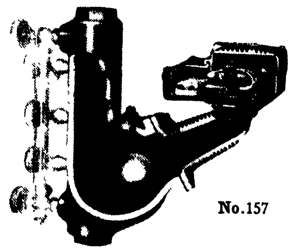


No. 101

McDonnell Pump Control, Low Water Fuel Cut-off and Alarm Switch . . . for steam boilers of any size; maximum steam pressure, 150 lbs.



No. 150



No.157

The McDonnell No. 150 Pump Control, Cut-off and Alarm switch (and its equivalent, No. 157) is built down to the last detail to stand the gaff of high pressure and temperature. It is equipped with two switches. One closes on small float drop to control electric boiler feed pump, or electric valve in line to steam pump. Second operates on greater drop to stop burner and complete low water alarm circuit. *Underwriters' Laboratories* approved. No. 150 has automatic reset; for manual reset order No. 150-M.

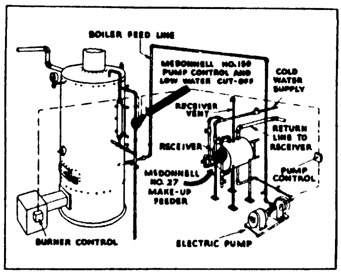
No. 157 is same as No. 150, but has integral water column which greatly simplifies installation, assures ideal reproduction of boiler water level in float chamber, and provides effective direct blow-down. No. 157 has automatic reset; for manual reset order No. 157-M.

Operation of No. 150 or 157

Typical hook-up of the McDonnell No. 150 is shown at right. When water level drops, No. 150 starts pump and then stops it when normal level is restored. If emergency occurs, cut-off switch stops burner; terminals for low water alarm are also provided.

This method holds boiler water level within the close limits necessary to attain not only safety, but also highest steaming efficiency. This has obvious advantages over the ordinary method of simply returning condensate whenever a certain quantity has accumulated in receiver. Note McDonnell No. 27 make-up water feeder on receiver to maintain minimum water supply.

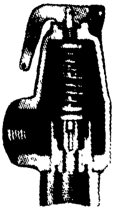
Drawings are available for all operating conditions—including two or more boilers supplied by one pump.



McDonnell No. 63 Low Water Cut-off for automatically fired hot water heating boilers.

In any hot water heating system there is a possibility of low water after a prolonged opening of an adequate relief valve as the result of some emergency condition. The protection against this, in automatically fired boilers, is a low water fuel cut-off, and for some time heating engineers have been using the McDonnell No. 150 to provide this protection because our low pressure low water cut-off was not rated for the higher pressures of hot water boilers. As a result the No. 63 Low Water Cut-off has been developed for pressures up to 50 lbs. It provides an equally dependable low water cut-off for hot water boilers at a more moderate price than that of the No. 150 which must be designed for higher pressures. The McDonnell No. 63 is also used on steam boilers where hook-up with 1 in. equalizing pipes is desired.

No. 63



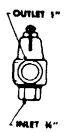
No. 33

McDonnell No. 33 Series Safety Relief Valves

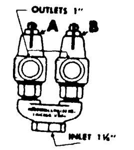
McDonnell Safety Relief Valves are built to comply with current *ASME Boiler Code*, and have been tested and rated by the *National Board of Boiler and Pressure Vessel Inspectors*. The certified rating appears on the nameplate attached to each valve.

The No. 33 Series offers the ultimate in safety in guarding boilers and water systems against the dangers of over-pressure. They have low discharge rate which handles normal thermal expansion easily, and high rate for emergency conditions.

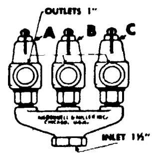
For Hot Water Heating Boilers



No. 33



233 Assembly



333 Assembly

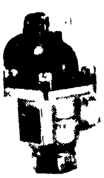
McDonnell No.	Opening Pressure	Btu/hr Capacity
33	30 lbs.	242,900

Valve	McDonnell No.	Opening Pressure	Btu/hr Capacity
A	33	30 lbs.	242,900
B	3333	33 lbs.	253,800
Total Capacity Btu/hr 496,700			

Valve	McDonnell No.	Opening Pressure	Btu/hr Capacity
A	33	30 lbs.	242,900
B	3333	33 lbs.	253,800
C	3336	36 lbs.	272,700
Total Capacity Btu/hr 769,400			

For Domestic Hot Water Tanks and Heaters

Valve No.	Opening Pressure	Btu Capacity	Inlet Pipe Size	Outlet Pipe Size
3345	45 lbs.	321,300	¾"	1"
3375	75 lbs.	308,700	¾"	1"
33100	100 lbs.	423,900	¾"	1"
33125	125 lbs.	513,900	¾"	1"
33150	150 lbs.	605,700	¾"	1"



No. 201

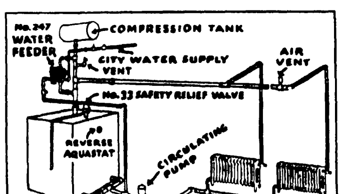
McDonnell No. 201 Temperature Relief Valve

For Hot Water Tanks and Heaters

A.G.A. Tested, Rated and Listed Heat Input 1,200,000 Btu/hr. Utilizes the well-known Vernatherm thermostatic element, which combines high accuracy, durability and ideal operating characteristics. Opens at 188F, reaches full discharge capacity at 208F. McDonnell design places this element right in the flow of hot water to fixtures, where it rapidly and more uniformly responds to variations in temperature. Suitable for operating pressures to 125 lbs.

McDonnell No. 33 Safety Relief Valve and No. 247 Feeder for hot water heating plants with boiler and radiation on same level.

The drawing illustrates how the McDonnell No. 33 and No. 247 team up to provide both efficient operation and protection for "same-level" systems. In this type of plant there is always a possibility of separation of the boiler from the connected load unless a means of maintaining a proper level is provided. The No. 247 water feeder does this by keeping the water level above the highest point in the main. Protection from excess pressure is provided by the No. 33 Safety Relief Valve for all boilers up to 242,900 Btu output. When the boiler is automatically fired the No. 247 can be used with low water fuel cut-off switch (No. 247-2).



Write for Literature—Condensed Catalog covering more popular McDonnell Products . . . Supplemental Catalog covering feeders, cut-offs and switches for all operating conditions . . . and Sample Specification Sheets.

Fluid Systems, Inc.

1881 DIXWELL AVE. • NEW HAVEN 14, CONN.

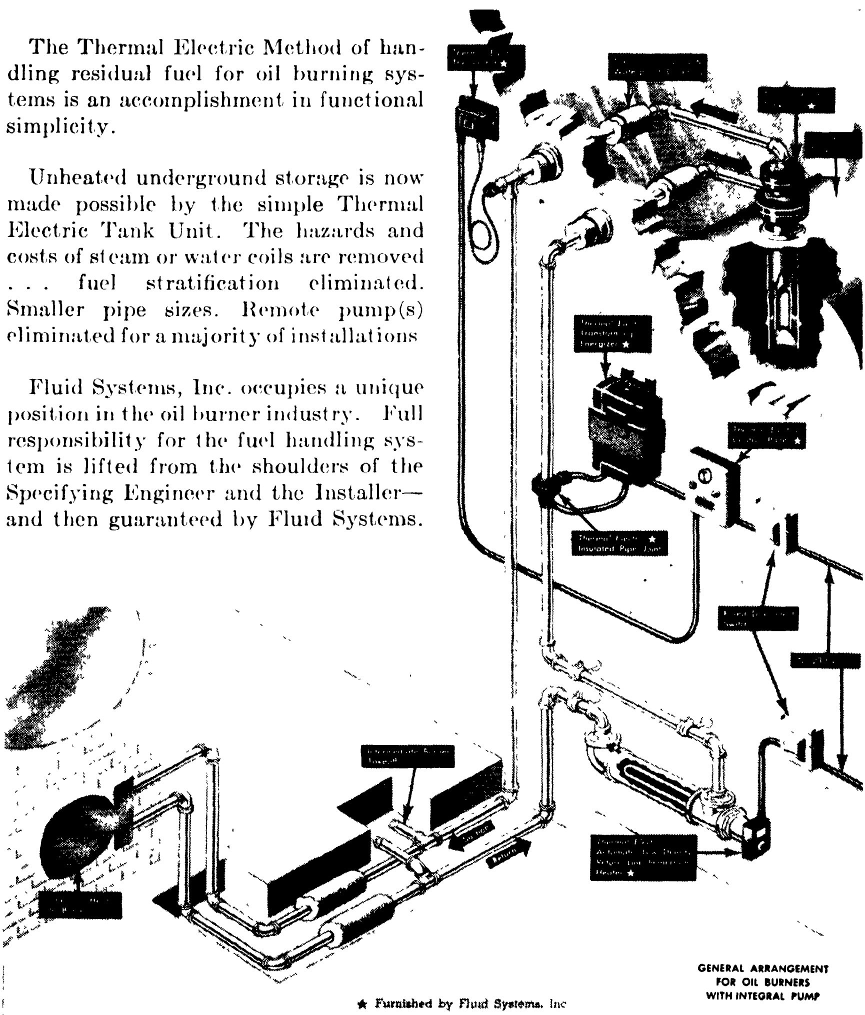
Fluid Systems, Inc. and Thermal Electric are trade marks registered U. S. Pat. Off. and Patent #2,224,403.

HYDRAULIC ENGINEERING by FLUID SYSTEMS, INC.

The Thermal Electric Method of handling residual fuel for oil burning systems is an accomplishment in functional simplicity.

Unheated underground storage is now made possible by the simple Thermal Electric Tank Unit. The hazards and costs of steam or water coils are removed . . . fuel stratification eliminated. Smaller pipe sizes. Remote pump(s) eliminated for a majority of installations

Fluid Systems, Inc. occupies a unique position in the oil burner industry. Full responsibility for the fuel handling system is lifted from the shoulders of the Specifying Engineer and the Installer—and then guaranteed by Fluid Systems.



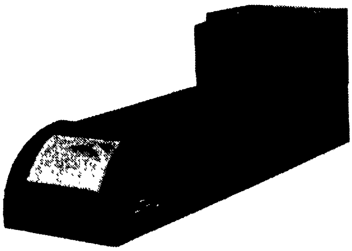
★ Furnished by Fluid Systems, Inc

GENERAL ARRANGEMENT FOR OIL BURNERS WITH INTEGRAL PUMP

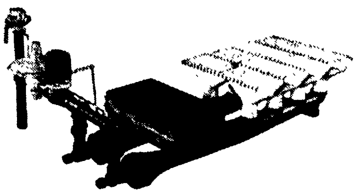
Sonner Burner Company

Designers and Manufacturers of Gas Conversion Burners and Unit Heaters

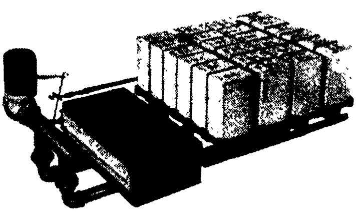
Offices and Factories: **Winfield, Kansas**



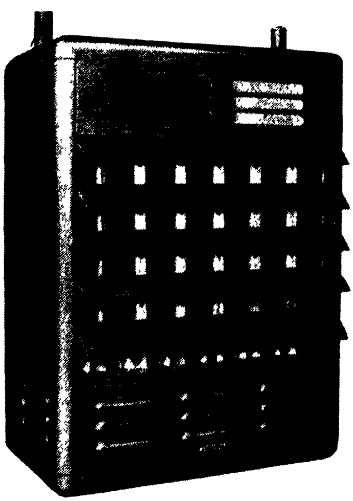
Type L-49 Heat Machine



Type LV SONNER Burner



Type D SONNER Burner



Unit Heater

Sonner Gas Conversion Burners are a product of a quarter century of specialization in design and manufacture.

A wide range of designs provides a Sonner Burner for domestic, commercial or industrial use. Consult Sonner Engineering for advice.

TYPE L-49 HEAT MACHINE is designed specifically for the conversion of residential heating plants to gas. Completely automatic burner is *A.G.A.* listed. Available in two sizes with capacities ranging from 75,000 to 300,000 Btu/hr. See Cat. 49-L-2R.

TYPE LV SONNER BURNER for large heating and power boilers, handles a wide range of fuels. Standard assemblies from 320,000 to 7,200,000 Btu/hr are described in Cat. 49-LV-1. Special assemblies are available if required.

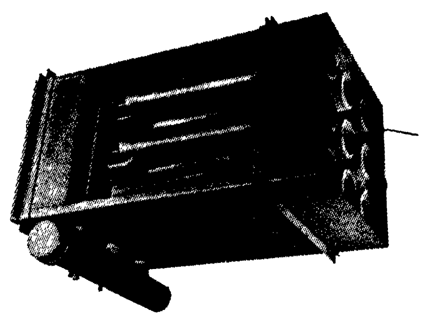
TYPE D SONNER BURNER offers complete flexibility for all types and sizes of boilers and plants. Type "D" Burner Units can be readily manifolded into exactly the correct size and shape of burner for any kind of firebox. See Cat. 49-D-1.

SONNER GAS FIRED UNIT HEATERS provide low cost automatic heating in commercial installations where space is at a premium. Approved by *A.G.A.* and *Underwriters' Laboratories.*

Available in two sizes, 75,000 and 100,000 Btu/hr input. See Cat. 51-U-1.

TYPE R GAS BURNER FOR SCOTCH MARINE BOILERS, brickset boilers, and other horizontal flame applications. Standard Assemblies from 480,000 to 5,760,000 Btu/hr described in Cat. 51-R-2. Special assemblies are also available if required.

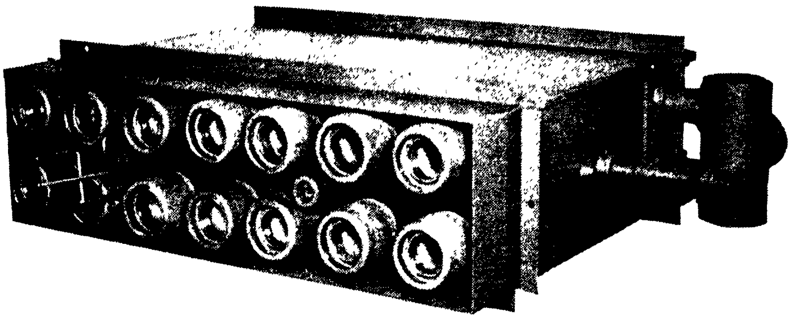
Send for SONNER Catalogs Today.



Type R SONNER Burner

The Webster Engineering Company

115 South Frisco St., P.O. Box 2168
Tulsa, Oklahoma
Division of
SURFACE COMBUSTION CORPORATION, TOLEDO, OHIO



WEBSTER KINETIC*

Low Pressure Gas and Standby Oil.

Consisting of a multiplicity of full venturi mixers with flame retention nozzles assembled in a metal casing complete with pilot and louvre, the WEBSTER KINETIC burner is presented as the latest addition to the Webster line of firing equipment. It may be used with natural, mixed and liquefied petroleum gases.

Being a multiple head assembly, the WEBSTER KINETIC burner can be supplied in any size or shape. The firing of Scotch Marine boilers with extremely low pressure gas without noise, vibration or electrical power and with minimum furnace draft was the sole objective when the WEBSTER KINETIC burner was developed. When design conditions preclude the use of a vertical burner in a steel firebox or sectional boiler, the WEBSTER KINETIC burner is highly recommended as the next best gas application.

* *Trade Mark*

WEBSTER ALSO MANUFACTURES:

Series F600 and Series 650 Vertical Gas Burners for Heating and Power Boilers.

Series 340 Gas Burners for Vertical Boilers.

Series VI and Webster Rectilinear High Pressure Inspirators for Boilers, Kilns, Stills, Dryers, etc.

Series 200 and Series R Combination Gas and Oil Burners for Power Boilers.

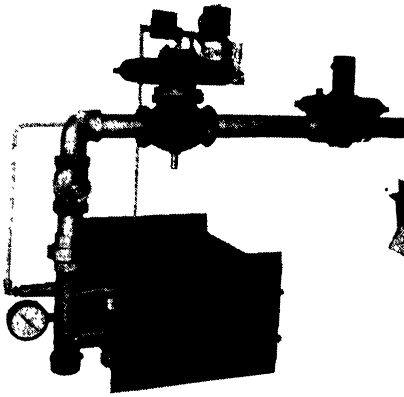
Sales and Service in all principal cities.

JOHN ZINK CO.

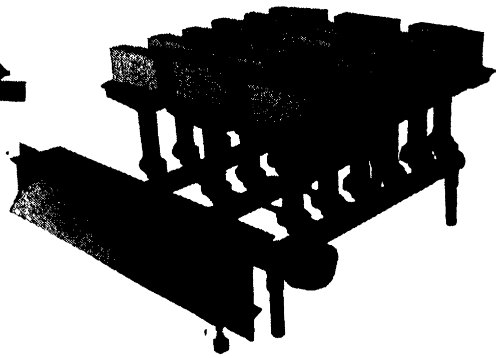
John Zink Company

4401 S. Peoria — **Tulsa, Okla.**

342 Madison Ave. — New York, N. Y.



STV



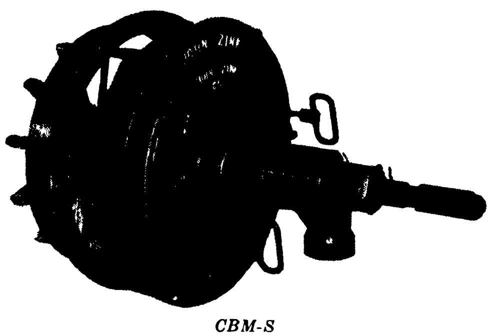
VR

SERIES STV GAS BURNER

Ideal for heating boiler installations. From one head to fifty heads. Quiet, efficient and economical operation. Write for literature.

SERIES VR VERTICAL GASBURNER

This burner is of the "upshot" or vertical radiant type with multiple burner heads so designed as to entrain a certain amount of primary air from the total air entering through the louvre. The "VR" burners are especially designed to operate in boilers having low draft and small combustion space. Write for literature.



CBM-S

SERIES CBM-S COMBINATION GAS AND OIL BURNER

The John Zink Series CBM-S Register Type Burner for either atmospheric or forced draft operation is specifically designed to provide short flame and high CO2 when firing either with oil or with gas at 2 oz to 3 lb.
Burners for Process industries, High Pressure Steam Generation, Direct Fired Air Heaters, for applications requiring heated streams of mixed flue gas and air.

John Zink Company

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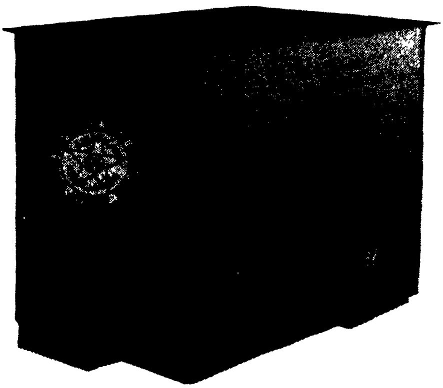
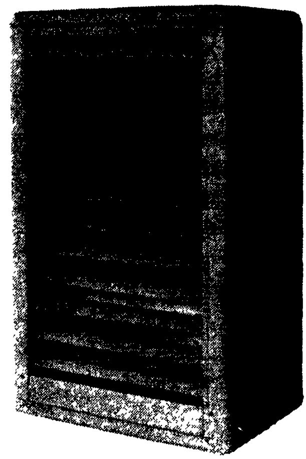




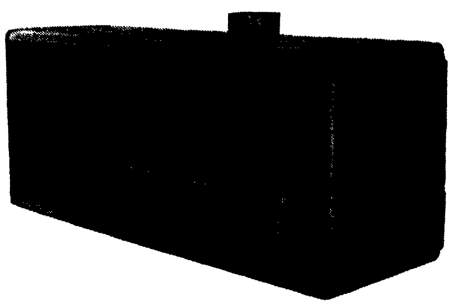
FLOOR FURNACES

A size for every home heating requirement. Small grille—fool proof—simple to operate—*AGA* approved—sturdy, construction.

Sizes—30,000; 35,000; 50,000 and 85,000 Btu input. Write for literature.

NEW SUSPENDED UNIT HEATER (Series U H S Fan Type)

AGA approved for Natural, Mixed, Manufactured or LP Gas. For clean, safe trouble-free heating of industrial and commercial establishments. A complete packaged unit that is fully automatic. Conserves valuable floor space. Designed for an attractive addition to any store or shop. Write for literature.



CENTRAL GAS HEATERS (Horizontal-Vertical)

A truly universal, efficient, simple and compact central heating unit. Fully automatic, it is a complete packaged unit ready for installation. May be installed in the basement, attic, service closet or utility room. It is designed as a winter air conditioner with a summer switch to provide air circulation during warm weather. Two models available. Write for literature.

Ace Engineering Company

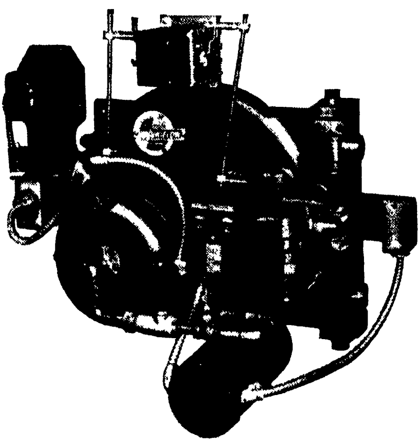
1435 West 15th St., **Chicago 8, Ill.**

"Custom Engineered Oil Burning Systems Since 1931"

The Ace Uniflow oil burner is built in ten sizes to burn all grades of fuel oil. The oil pump is available either as an integral part of the burner or as a separate unit, depending upon the job requirements. Standard models include both belt and direct drive and for operation with all electrical current characteristics.

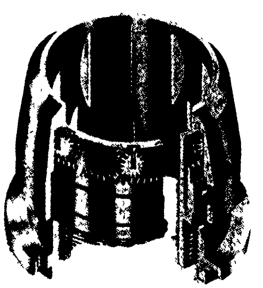
The Ace oil burner features as standard equipment the patented Ace Uniflow valve, which permits constant uniform flow and flame regardless of oil temperature or viscosity.

Approved by *Underwriters' Laboratories, Inc., New York Board of Standards & Appeals* and the *States of Massachusetts and Connecticut,* these burners are designed to oil fire commercial and industrial boilers in the capacity ranges as shown below.



TECHNICAL DATA AND SPECIFICATIONS

Burner Size	Maximum G. P. H.	Boiler Horse Power	Sq. Ft. Steam Radiation	Motor Size H. P.
12	8	24	3,300	[illegible]
14	15	45	6,000	[illegible]
15	22	66	8,800	[illegible]
16	30	90	12,000	[illegible]
17	45	135	18,000	1
18	70	210	28,000	1.5
19	85	255	34,000	2.0
20	100	300	40,000	3.0
22	130	400	52,000	5.0
24	165	500	64,000	5.0



ACE AIR NOZZLE

(Patent No. 2541347)

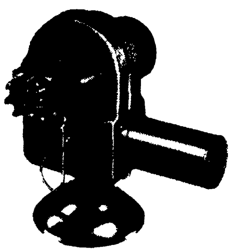
The Ace variable-vaned air nozzle permits accurate, efficient shaping of the flame to conform with the shape of the combustion chamber. The angle of the vanes can easily be changed in the field to any desired position to suit the boiler requirements and thus insure the maximum flame without oil impingement.

SEND FOR ACE MANUAL...
PREPARED SPECIFICALLY FOR ARCHITECTS AND ENGINEERS.

Automatic Burner Corp.

1823 W. Carroll Ave.
Chicago 12, Ill.

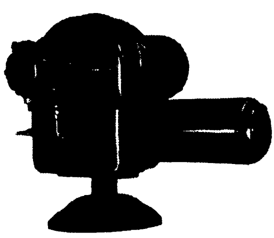
"THE STANDARD OF THE INDUSTRY"



MODEL 55

For Packaged Heating Units

This more compact pressure type burner is exceptionally easy to adapt to a wide range of heating units. It fits vestibules measuring only 9$\frac{15}{16}$ in. and mounts directly to the boiler by means of a flange. Among its many time-saving features are: 10 second nozzle removal, and a transformer which swings aside breaking electrical contact and permitting easier drawer assembly removal. Capacity 0.6 to 3 gph.



MODEL 53

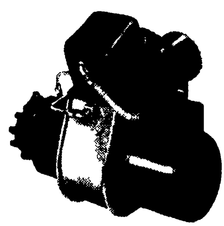
Industrial Oil Burner

Provides dependable, economical oil heat for small to medium sized factories, institutions, etc. Built-in safety combustion control eliminates need for any stack relay; positive, built-in automatic electric oil cutoff eliminates after-burn . . . has dual ignition and dual nozzles, hinged transformers, two-stage pump. Suitable for front or rear of furnace or boiler mounting. Capacity 9 to 16 gph.

MODEL 52A

Domestic Conversion Oil Burner

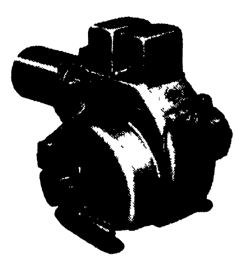
Famous for many years as the gun type oil burner that combines maximum efficiency and economy with smoother, quieter, trouble-free operation. Features exclusive ABC Choke, Spinner, Coupling and Oilairator mechanism for developing truly efficient combustion. Capacity 0.6 to 5.0 gph.



MODEL 54 For

Commercial and Large Residential Use

More compact yet more efficient for heating and hot water in stores, apartment buildings and similar installations. Incorporates an easily adjustable hinged damper on right hand side as well as usual air intake on pump side for more exact air adjustment at every capacity. Specially designed, double-inlet fan brings air into housing from both sides. With 2-stage pump. 4 to 10 gph.



Enterprise Engine & Machinery Co.

A SUBSIDIARY OF GENERAL METALS CORPORATION

18th & Florida Streets, **San Francisco 10, California**

Distributors in Principal Cities

ENTERPRISE HORIZONTAL ROTARY BURNERS—OIL, COMBINATION OIL—GAS

BURNER CAPACITIES

BURNER SIZE	MOTOR HP	OIL U.S. GALS. PER HOUR	1000 BTU GAS CU FT PER HOUR	BOILER HP	STEAM EDR 240 Btu	HOT WATER EDR 150 Btu
AA-3	¼	4	600	13	1800	2,880
A-3	½	7	1,050	23	3,200	5,120
C-3	½	15	2,250	50	6,950	11,120
E-3	½	20	3,000	67	9,310	14,900
F-3	¾	28	4,200	93	12,930	20,700
G-3	1	35	5,250	116	16,125	25,800
H-3	1	50	7,500	166	23,075	36,900
J-3	2	70	10,500	233	32,390	51,800
K-3	3	100	15,000	333	46,290	74,000
*L-3	3	135	20,250	450	62,550	100,000
*M-3	5	200	30,000	666	92,575	148,100

*Non-Pump only

All maximum capacities are necessarily approximate. Burners installed with adequate draft provisions and correct furnace volume, properly designed, will develop capacities indicated, at sea level.

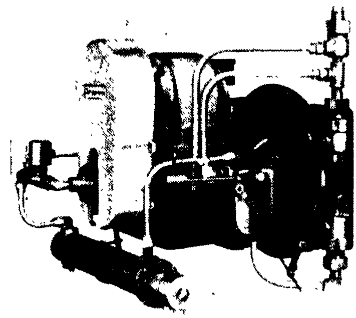
V-BELT DRIVE

All Enterprise Burners are equipped with V-belt drive, allowing for the selection of a standard motor to fit precisely the voltage and torque requirements. Any replacement or repair service of standard motors is readily available through nationwide service facilities of motor manufacturers at standardized prices. Enterprise V-belt drive provides for various fan and atomizer speeds as necessary to insure efficient operation for both normal and unusual combustion requirements.

OPERATION

Selection of burner type depends on the particular requirements. Industrial type burners are designed for manual or various forms of semi-automatic control. Full automatic burners are generally specified for commercial heating plants, public buildings, apartment houses, hospitals, and general industrial service, to obtain efficient, economical burner operation without the need for a constant attendant.

Manual, semi-automatic and full automatic burners are furnished as standard equipment in pump type, and non-pump type as shown in specifications chart. Non-pump type requires separate pump sets to furnish oil supply. Standard fuels are No. 5 oil, or No. 6 (Bunker "C") oil, preheated.



OPTIONAL EQUIPMENT

Gear Pump—for all sizes except L-3 and M-3
Metering Pump, dual type—for all sizes except L-3 and M-3
Fixed Fire Start—Sizes AA-3 through F-3
Low Fire Start—All sizes
Modulating Control—All sizes
Combination Oil and Gas—All sizes
Electronic Controls—All sizes

COMBINATION OIL—GAS BURNERS

Enterprise Combination Burners are designed to use whatever gas is readily available--natural, manufactured, or the by-product of a gas producing process. They are adaptable as well to the use of a wide range of fuel oils. Automatic burners are supplied with standard gas-electric ignition with constant gas pilot for gas fuel, and intermittent gas pilot for oil fuel.

METERING PUMP

The Enterprise Metering Pump automatically compensates for changes in viscosity, keeps the firing rate constant. It eliminates the need for compensating valves, metering valves, and in some cases relief valves. This is optional equipment, available for all sizes except L-3 and M-3.

Engineers will be provided with complete mounting dimensions, piping diagrams, wiring diagrams and combustion chamber designs for specific installation requirements. Consult with manufacturer before preparing specifications. Full information on request.

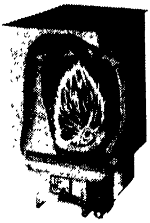
Enterprise Burners have been tested, approved and listed as standard by *Underwriters' Laboratories, Inc.*, and by other recognized boards and bureaus of safety measures and controls.

H.C.Little Burner Co.

Head Office: **San Rafael, Calif.**

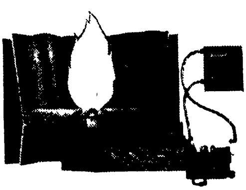
Factory Representatives in 18 Principal Cities

Trade Mark



Automatic electric ignition floor furnace Dual register wall discharge models also.

Automatic electric ignition or manual control circulating heaters.



Automatic electric ignition conversion burner.

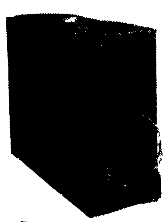
Domestic, Automatic, Natural Draft Electric Ignition Heating Equipment

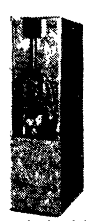
Complete line of domestic heating equipment, specifically designed to burn low cost furnace oil. Burners and complete units listed by *Underwriters' Laboratories*. Conventional thermostat control operates automatic electric metering valve and simultaneously turns on patented low voltage 3-volt ignition coil shown in burner. The floor furnaces, circulating heaters and wall furnaces shown offer high quality, low initial cost, automatic heat for small homes. The forced air furnaces offer basement type and first floor utility type automatic heating systems which are clean and quiet in operation. Special suspended ceiling furnace for industrial use saves floor space and offers individual zone thermostat control with directed forced air circulation.

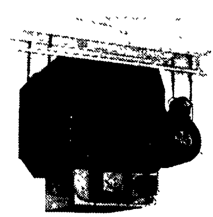
HOURLY CAPACITIES AND RATINGS OF UNITS.

Type of Unit	Sizes Available	Max. Cap. Gallons	Thou. Btu Input	Thou. Btu Output	Blower Size	Blower cfm @ ¼" SP	Motor Size hp	Dimen. H W.D. Inches	Usage
Automatic	#00	.25	35	26				9 7 22	Convert-
Conversion	0	.45	60	45				9 8 23	ing Domestic
Burners	9"	.62	87	65				9 9 25	Coal to
	1	.80	114	85				9 10 28	Oil Burn
	2	1 25	175	130				10 12 32	ing Fur-
	3	1.75	245	183				11 14 36	naces
	4	2.50	350	260				12 16 38	
Auto.	#70-47 CI	.48	65	50				45 23 23	Homes
Dual or	70-47 Dual	50	70	55				60 23 32	without
Floor	100-47 CI	.70	95	75				45 23 41	base-
Furnaces	100-47 Dual	.75	110	80				60 23 44	ments
Manual	#59	.45	62	43				31 28 20	Heating
or Auto.	71	.50	71	53				41 28 20	Small
Heaters	108	.75	108	77				43 36 21	Homes
Wall	#60	.37	53	42				51 29 20	No Base-
Furnaces	80	.60	84	65				80 30 20	ment
Auto.	A-AC-2	.75	105	84	9"	1000	1/6	45 20 54	Homes
Basement	B-AC	1.12	155	125	10	1500	1/4	57 23 65	with
Forced	D-MF #1	1.60	230	178	14	2400	1/3	60 41 65	Base-
Air Furnaces	D-MF #2	2.45	340	260	16	3600	1/2	66 47 77	ments
Auto.	A-UC	.75	105	84	9"	1000	1/6	71 21 37	Duct
Utility Forced	DU-44 CIBL	80	112	80	7	650	1/4	60 22 28	Distribution in
Air Furnaces	DU-46 CIBL	1.15	160	120	10	1300	1/4	79 28 35	Attic
Ceiling Furnace	CF-140	1.35	190	145	10"	1000	1/4	49 62 39	Garages, Stores & Factories

(Below) Automatic electric ignition wall furnace for homes with concrete floors, dual heat discharge.







L. to r. Automatic electric ignition furnaces: basement, first floor utility room, suspended ceiling furnace for industrial use.

S. T. Johnson Co.

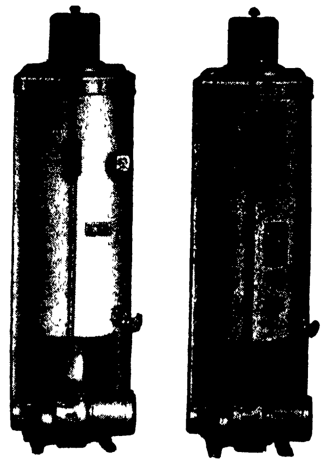
Builders of Domestic and Commercial Oil Burners

940 Arlington Ave., **Oakland 8, Calif.**
401 No. Broad St., **Philadelphia 8, Pa.**

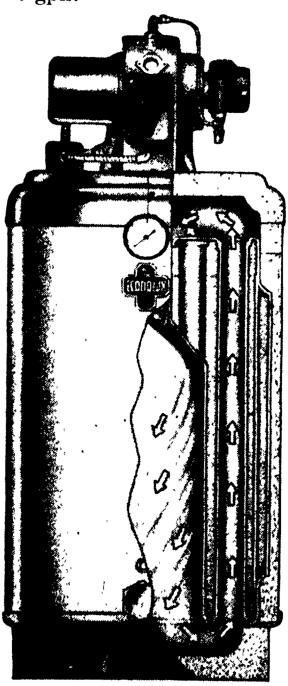
Self-storage water heaters, separate burner units, burner-boiler units, conditioned air units, range burners and various specialized items comprise the line-up of Johnson light-oil Burners.

There is a wide range of sizes and capacities in each classification with which heating engineers and contractors can successfully meet every type of problem. Every Johnson Burner is backed by an unbroken record of fine engineering and excellent craftsmanship that dates back to 1903.

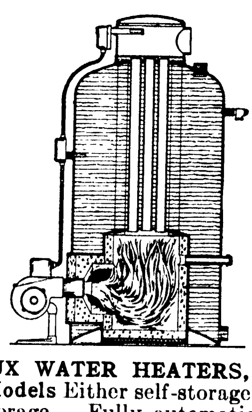
BANKHEAT BURNERS Fully automatic, pressure atomizing type. Sizes up to 20 gph.



HEATLUX HOUSE HEATERS (left) Fully automatic Bankheat Burners. Hot water heat. 2 sizes.
AQULUX WATER HEATERS (right) Fully automatic, self-storage. Capacity: 100 to 540 gph.



ECONOLUX HEATERS Fully automatic Steam and Hot Water Units, 150 to 830 Mbh.



AQULUX WATER HEATERS, Heavy-Duty Models Either self-storage or separate storage. Fully automatic. Low fuel consumption and costs. Oil Burning models up to 540 gph. Combination Coal or Oil models up to 2000 gph. Combination Gas or Oil models in all sizes.

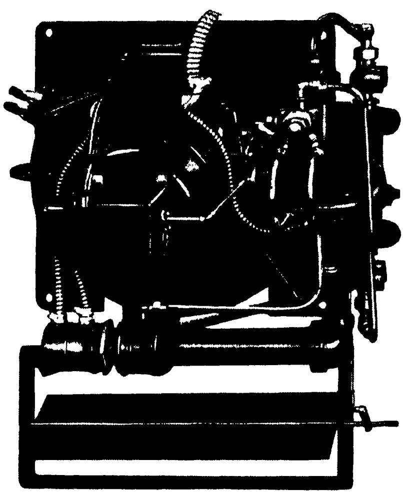
S. T. Johnson Co.

Builders of Heavy-Duty Industrial Oil Burners

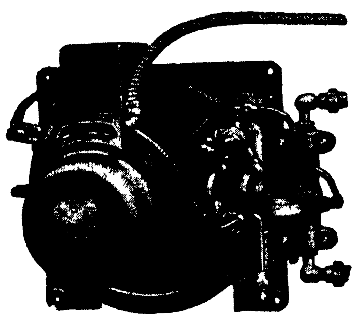
940 Arlington Ave., **Oakland 8, Calif.**
401 No. Broad St., **Philadelphia 8, Pa.**

Johnson Industrial Burners are designed to operate on Heavy Oils which produce extra heat at low cost. They increase the capacity of equipment formerly fired with coal and produce desired steam pressures more quickly. Automatic regulation permits the boiler to operate with maximum efficiency at any specified steam pressure, without watching, care or attention, thus reducing labor costs.

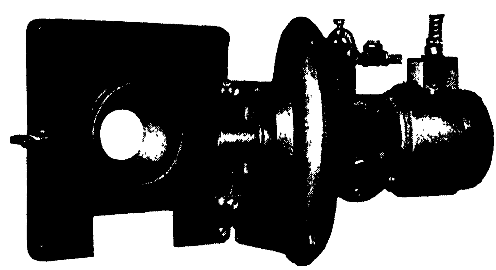
They have been installed with marked success in hotels, hospitals, factories, office buildings and other large structures all over America because they provide heating engineers with a wide range of capacities and with every desired feature of economy, performance and automatic control. In design and construction, Johnson Burners combine modern engineering techniques with the "know-how" and skill acquired in 49 years of practical experience in building fine oil burners.



TYPE 30 AVH—Fully Automatic. Preheater type. Burns No. 6 Oil. Six sizes, 20 to 300 horse power output.



TYPE 30 AV—Fully automatic. Burns No. 5 Oil. Six sizes, 2 to 100 gph.



TYPE 28 Manual and semi-automatic. With or without built-in pumps. Burns No. 5 and No. 6 Oils. Seven sizes, 2 to 135 gph. Illustration shows burner swung away from fire-hole-plate for easy inspection.

PETRO

Petroleum Heat & Power Company . . . . Stamford, Conn.

DOMESTIC OIL HEATING EQUIPMENT

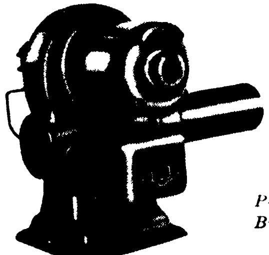
PRESSURE ATOMIZING DOMESTIC BURNERS—Applicable to steam, hot water, or warm air systems, for use with No. 2 or lighter fuel oils. Listed by *Underwriters' Laboratories*.

Models CA-1, CA-2. New type air distributor head for maximum combustion efficiency. Intermittent ignition with electronic or standard controls.

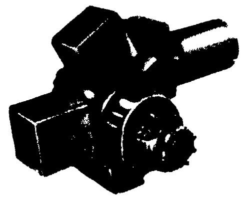
Models P-9-70A, P-9-70. Standard, reliable "gun" type burner with constant electric ignition. Choice of three package controls.

Models P-12-3, P-13-A3, P-13-3. For larger residences, stores, garages and commercial buildings. No. 2 or lighter oil.

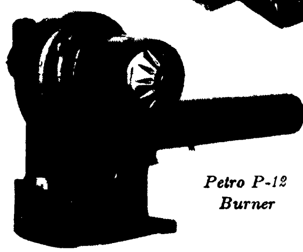
Burner Model Number	Nozzle Size gal per hr	Total Capacity Steam sq ft	Total Capacity Hot Water sq ft
CA-1	.75 to 2 00	300– 800	480– 1280
CA-2	2.00 to 4.00	800–1600	1280– 2560
P-9-70-A	1.00 to 1.50	400– 600	640– 960
P-9-70	1.65 to 2 50	660–1000	1050– 1600
P-21	2 00 to 4.50	800–1800	1280– 2880
P-22	3.00 to 6.00	1200–2400	1920– 3840
P-12-3	6.00 to 10 00	2400–4000	3840– 6400
P-13-A3	9.00 to 12.00	3600–4800	5760– 7680
P-13-3	12.00 to 18.00	4800–7200	7680–11520



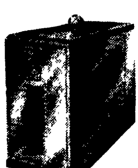
P-9-70 Burner



Petro CA Burner



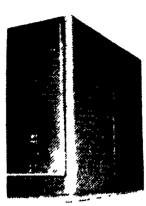
Petro P-12 Burner



Petro Boiler Unit Models HT-S and W

Petro Warm Air Conditioner —Model HT-A3

Front view of Petro Standard Automatic Boiler for Steam



Petro Series "A" Winter Air Conditioner

Petro Storage Type Hot Water Heater

PETRO MODEL HT OIL HEATING UNITS—The Petro Model HT Boiler normally delivers steam in as little as 43 seconds. With or without 3 gpm domestic hot water coil. Model HT-S3 rated 375 sq ft, and HT-S4, 450 sq ft, steam. Model HT-W3, 600 sq ft, and HT-W4, 720 sq ft, water.

The Petro Model HT-A3 Warm Air Conditioner is a compact electronically controlled unit. With or without 3 gpm domestic hot water coil. Rated 110,000 Btu at bonnet, 90,000 Btu at register. Capacity 1250 cfm.

OTHER PETRO HEATING UNITS Standard Model Petro Automatic Boilers, in two sizes, for steam and water, are steel boilers designed for small home field. Series "A" Winter Air Conditioner in three sizes from 75,000 to 115,000 Btu at register. Storage Type Water Heaters operate on No. 2 oil and heat 120 gph 100 F temperature rise.

Send for Petro Domestic Catalog.

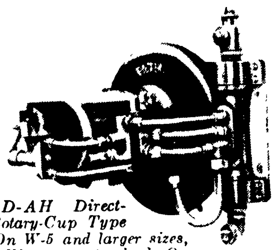
Petroleum Heat & Power Company

Main Office and Factory: **Stamford, Conn.**

"Since 1903" . . . Good Oil Burning Equipment . . . Fuel Oils

PETRO REG. U.S. PAT. OFF.

INDUSTRIAL AND COMMERCIAL OIL BURNING SYSTEMS

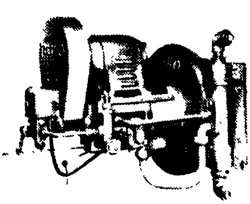


Model WD-AH Direct-Driven Rotary-Cup Type Burner On W-5 and larger sizes, Modutrol Firing is standard On smaller sizes, it is supplied as an extra.

PETRO MODEL W BURNER is available in two types: (1) Direct-Driven, which includes electric motor, fan, pump, rotary cup atomizer in one self-contained assembly, together with air and oil controls. (2) Belt-Driven, containing the same assembly as above except that the integral motor is replaced by motor mounted outside burner housing and arranged for belt drive. Removable rotary cup and nozzle permit changing shape of flame to suit any boiler furnace requirements.

Interlocking air and oil control mechanism permits any minimum or maximum operation required within the burner's range of operation. Counter-flow Angular Air Vanes at nozzle increase air and oil turbulence and aid efficient combustion of heavy fuel oils.

Special oil adjustment valve meters oil to rotary cup, yet permits manual operation without disturbing permanent burner adjustment.



Model WO-A Belt Driven, Rotary Cup Type Oil Burner.

PETRO'S THERMAL VISCOSITY CONTROL

A dependable and accurate control of viscosity—and hence delivered combustion efficiency—is through the heat applied to the oil. Petro's Thermal Viscosity System controls this heat-application at its source.

FOR UNHEATED COMMERCIAL OILS: Model W-A—Automatic ignition and operation with synchronized control of oil and air.

Model W-SA—Semi-automatic, i.e.: automatic variation of firing rate with manual ignition; also available for manual variation and manual ignition.

FOR PREHEATED OILS: HEAVY NO. 5, Models W-A-E and W-SA-E; No. 6 (BUNKER "C"), Models W-AH, W-SAH.

Models W-A-E; W-AH—Automatic ignition and operation with synchronized control of oil and air, and automatic control of oil heaters.

Models W-SA-E; W-SAH—Semi-automatic with oil heaters, i.e.: automatic variation of firing rate with manual ignition; also available for manual variation and manual ignition.

CAPACITIES at 70% Boiler Efficiency

Model	Motor H P.	Max. Gals. per Hour	Rated Cap. B.H.P.	Sq. Ft E.D R Steam Rad.
W-2½	⅛	11	34.4	4,810
W-3	½	15	47.0	6,560
W-4	½	25	78.5	10,940
W-5	1	35	110.0	15,300
W-6	2	50	157.0	21,880
W-7	2	70	220.0	30,600
W-8	3	100	313.6	43,750
W-8½	3	120	376.0	52,500
W-9	3	145	454.7	63,250

These burners are available for all types of electrical current supply.

PETRO MODEL GW COMBINATION GAS AND OIL BURNER, direct or belt driven. Independent combustion of each fuel—60 seconds to change from one to the other. Maximum combustion efficiency from either fuel.

CAPACITIES

Model	BTU	Oil* GPM	Gas* Cu. Ft/Hr
GW-4	3,750,000	25	3,750
GW-5	5,200,000	35	5,200
GW-6	7,500,000	50	7,500
GW-7	10,500,000	70	10,500
GW-8	15,000,000	100	15,000

* Based on oil of 150,000 Btu/gal and gas of 1,000 Btu cu ft.

Send for catalog of Petro Commercial and Industrial Oil Burners

Ray Oil Burner Co.

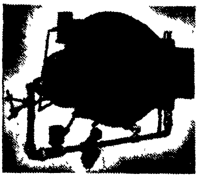
Since 1872

1301 San Jose Avenue
San Francisco 12, Cal.

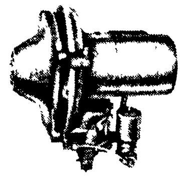
Distributors in
All Principal Cities of the World

Atlantic Seaboard Division
629 Grove Street
Jersey City 2, N. J.
Consult your local
Telephone Directory

Products: **A complete line of Horizontal Rotary and Pressure Atomizing Oil Burners; Combination Oil-or-Gas Burners; Industrial Gas Burners; Oil Burning Water Heaters; Winter Air Conditioning Units, Commercial Ranges.**



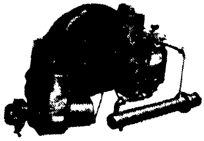
1 to 1000 Boiler hp Type AG. Manual, Semi-Automatic.



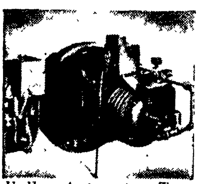
RCR for #2 oil, manual operation.

Steam Turbine Drive, Type TG, all grades oil. Tested and approved for U. S. Navy Service.

Four, 3 oven, fully automatic oil Ranges.

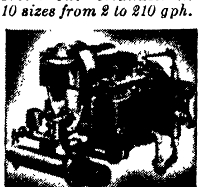


Type ARJP, Fully Automatic for heavy oil where gas for ignition not available.



Fully Automatic Type AR-134, for No. 5 oil.

Type AR144 Size 10 for No. 6 oil. Available in 10 sizes from 2 to 210 gph.



Type BR-141, Belt Drive Automatic. 2½ to 100 gph.

RAY HORIZONTAL ROTARY OIL BURNERS

Built in fully automatic, semi-automatic and manual types; in sizes from 1 to 1000 Boiler hp; to burn all grades of fuel oil.

Standard models include both direct and belt drives—the latter being recommended for use where other than 50 or 60 Cycles AC, or only DC is available. Types for straight electric or straight gas ignition; pump or gravity feeds. Direct drive types include a steam turbine driven model.

All fully automatic types for heavy oil incorporate the Ray Dual Pump and Reservoir, with the Ray VISCOSITY Valve, a patented, exclusive feature which automatically meters the correct amount of fuel at all times, regardless of changes in viscosity of the oil due to temperature variations. All larger sizes employ dual ignition system, consisting of dual high-voltage transformers, dual electric ignitors and dual gas valves. Fully automatic sizes 9 and 10 include as standard equipment electronic pilot and flame failure control.

RAY PRESSURE ATOMIZING OIL BURNERS

Fully automatic, for No. 2 oil or lighter. AC or DC; capacities to 18 gal/hr.

RAY WINTER AIR CONDITIONING UNITS

Built in four sizes, with input capacities of 105,000, 140,000, 230,000, 350,000, 450,000 Btu.

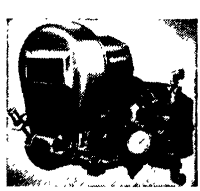
RAY OIL WATER HEATERS

Four sizes. Capacities: 35, 45, 60 and 75 gal. Maximum recover rates: 100 to 240 gph.

RAY OIL FIRED RANGES

Seven sizes for manual or fully automatic operation.

Ray Oil Furnace, Winter Air Conditioning Unit.



Small capacity (½ to 2½ gph) rotary burner.

Fully Automatic, Type JP, #2 or lighter oil

Ray Oil Burner Co.

Since 1872

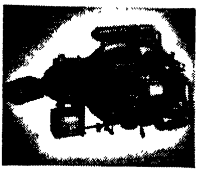
1301 San Jose Avenue
San Francisco 12, Cal.

REG. U. S. PAT. OFF.

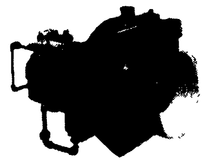
Atlantic Seaboard Division
629 Grove Street
Jersey City 2, N. J.

There is a RAY Burner for every Heating Purpose.

RAY COMBINATION OIL-GAS BURNER



ARC-131 Model 50 for Low Pressure Gas



Combination Oil and High Pressure Gas, Type AR HN.

Type XPJC for Low Pressure Gas

ARC-MODEL 50. This burner consists of a rotary type oil burner of Ray conventional design coupled with a gas housing for low pressure gas, from 2 in. WC to 10 in. WC. The gas burner parts provide for the introduction of gas in a manner which fully utilizes the energy in the high pressure air supplied by the rotary burner fan. Gas and primary air are intimately combined, in two stages, before mixing with the secondary air which is introduced through a vaned annular opening concentric with the nozzle. Change from one fuel to the other requires less than one minute. Available in ten sizes; in capacities to 31,500,000 Btu/hr.

TYPE HN for gas pressures above 1 lb/sq in. may be used alone or in combination with a Ray Oil Burner. Built in eleven sizes; in capacities to 43,000,000 Btu/hr.

TYPE XPJC. Combination gas-oil burner of the pressure atomizing type. Available in three sizes; in capacities from 560,000 to 2,500,000 Btu/hr.

HOURLY CAPACITY RATINGS of RAY OIL BURNERS

Burner Size	Motor H. P.	Oil Capacity *U. S. Gallons		Equivalent Boiler HP		Equivalent Lbs. Steam Generated		Heat Capacity Input Thou. Btu Oil or 1000 Btu Gas		Equivalent Sq. Ft Steam Radiation	
		Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
JP	1/4	1	3	3	9	110	325	140	420	438	1310
XPJ	1/4	1	4	3	12	110	430	140	560	438	1750
XPJ-1	1/4	3	8	9	25	325	865	420	1120	1310	3500
XP-2	1/2	8	18	25	56	865	1950	1120	2500	3500	7900
0000	1/10	0.5	2.5	2	8	60	290	75	375	234	1170
000	1/8	0.5	2.5	2	8	60	290	75	375	234	1170
00	1/4	1	2.5	3	8	110	290	150	375	469	1170
0	1/2	2	5	7	16	230	580	300	750	938	2350
1	1/2	4	11	13	35	460	1250	600	1650	1870	5170
2	1/2	5	15	17	50	580	1720	750	2250	2350	7030
3	3/4	8	22	27	72	930	2500	1200	3300	3750	10,300
5	1	10	33	35	110	1150	3800	1500	4950	4690	15500
6	1 1/2	12	50	40	165	1400	5800	1800	7500	5620	23400
7	2	15	67	50	225	1720	7750	2250	10000	7030	31400
8	3	25	100	85	335	2900	11500	3750	15000	11700	46900
9	5	35	150	120	500	4000	17400	5250	22500	—	—
10	7 1/2	50	210	165	700	5800	24400	7500	31500	—	—
12	7 1/2	75	320	250	1000	8700	37000	11250	48000	—	—

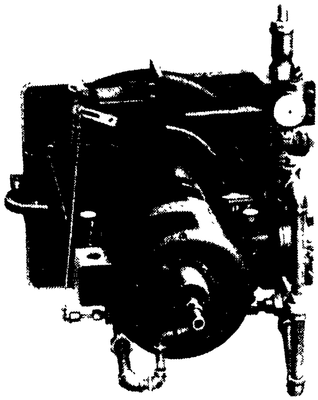
NOTE: These ratings are predicated upon specific conditions of draft and furnace volume. It may be permissible, under desirable conditions, to operate at higher rates, or advisable under restricted conditions, to operate at reduced rates.
Heating capacities are based upon 150,000 Btu per gal of oil for rotary burners and 140,000 Btu per gal of oil for pressure atomizing burners and upon an overall boiler efficiency of 75 per cent.

York-Shipley, Inc.

Main Office and Plants—**York 16, Pennsylvania**

OIL-FIRED EQUIPMENT FOR INDUSTRY

HORIZONTAL ROTARY BURNERS



Model AHPM direct-drive fully automatic pump-type burner with full modulating system, for heating or process work. Uses No. 6 oil.

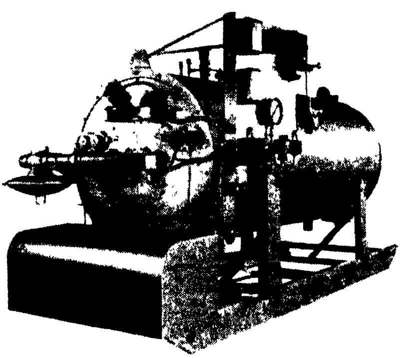
York-Shipley Horizontal Rotary Burners cover the range from 45 to 400 boiler hp, and are built in direct-drive and belt-drive types. Models are offered for use of fuel oils No. 1 through No. 6 (light distillate to most viscous bunker oils), for manual, semi-automatic, or full modulating control, with or without pumps.

The extremely wide range of models and sizes, high efficiency, and adaptability of this line assures easy selection of the *right* unit to fit *any* type or shape or make of boiler, with long life and low operating cost.

Three significant engineering developments provide *precision combustion*. *The Iris Shutter*, which gives absolute control of burner air, causing metered volume of burner air to remain constant; *The Flame-Former*, which shapes the flame to fit combustion chambers of any type; and *The Automatic Torch Lighter*, a pressure-type igniter for smooth and positive automatic starting.

All York-Shipley Industrial products are sold, engineered and installed by local distributors who have been selected because of their knowledge of engineering and of equipment using heavy oils.

STEAM-PAK GENERATORS



Model SPL-30-3 low-pressure boiler for heating applications. A three-pass, down-draft, horizontal fire tube design. Uses No. 3 oil.

Steam-Pak Generators are built *both* for low-pressure heating load and for high-pressure process load, in sizes from 15 hp up. Using fuel oils No. 3, 5, and 6, or gas or combination they require only a low-cost vent to remove products of combustion.

With about half the size, weight, and installation cost as compared to brick-set boilers of comparable capacity, Steam-Pak Generators require only four simple service connections, and are completely wired, piped, and pre-tested at the factory. Ready to operate when delivered.

The Steam-Pak Generator is an efficient unit, combining the oil burner and boiler in one package. It is designed to operate at peak efficiency through the entire firing range. This is accomplished with the Iris Shutter and modulating pump.

A COMPLETE LINE OF INDUSTRIAL COMMERCIAL AND RESIDENTIAL EQUIPMENT

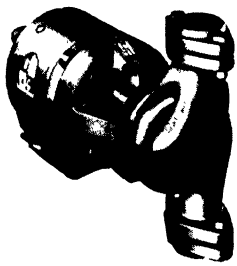
York-Shipley, Inc. builds a complete line of automatic heating equipment. Beside the Horizontal Rotary Burners and Steam-Pak Generators for industrial and commerical use described here, we offer the famous *York-Heat* boiler-units, winter air conditioners, water heaters, and conversion burners for either oil or gas.

Bell and Gossett Company

Morton Grove, Illinois

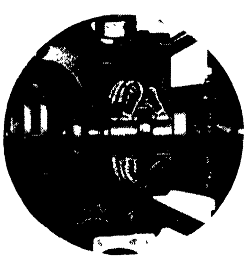
HYDRO-FLO HOT WATER SYSTEMS AND SPECIALTIES

B & G Booster Pumps



The B & G Booster is the basic unit of a B & G *Hydro-Flo* Forced Hot Water Heating System. It is built as a horizontally driven unit for sound engineering reasons which have demonstrated their practical value in thousands of installations. This construction makes possible many desirable, exclusive features. For example, the patented water-tight Seal eliminates need for a stuffing box.

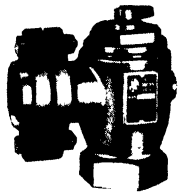
This Seal positively prevents entry of water into bearings. Shaft is of highly polished, hardened machine steel. The close-fitted Impeller makes every revolution count by holding water slippage to a minimum.



Oil Lubricated Bearings

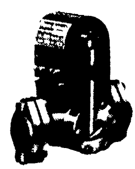
B & G Boosters have a genuine oil-circulating lubrication system—one of the greatest reasons for quiet, dependable, economical operation. Oil is drawn up from the oil well by wool fibre wicking and dropped on the horizontal bearing surfaces. Medium grade motor oil is used and only a few drops at infrequent intervals required.

SEE THE B & G HANDBOOK FOR COMPLETE DESIGNING DATA



B & G Flo-Control Valve

B & G Monoflo Fitting



B & G Motorized Valve

B & G Angle Flo-Control Valves

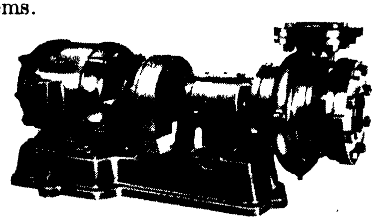
This valve, installed in the main, shuts off circulation to radiators when heat is not needed, permitting summer operation of a B & G Indirect Water Heater. It also helps maintain a uniform room temperature during the heating season.

B & G Monoflo Fittings

B & G Monoflo Fittings permit the use of a *single* pipe main instead of the conventional flow and return lines. They are installed at the junction of the radiator risers to the single main and assure the diversion of the proper amount of heated water into each radiator. Savings in space, labor and materials are obviously effected. Available in cast-iron and copper.

B & G Motorized Valves

Thermostatically operated valves used to control boiler water flow through individual circuits of zoned heating systems.



B & G Universal Pumps

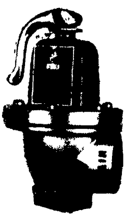
Designed for forced hot water heating systems in apartment and office buildings, factories, schools, etc. The installation can be operated as a large single zone or divided into several zones with circulation of pumped water in each circuit controlled by a B & G Motorized Valve, operated by a zone thermostat.

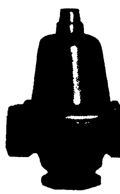
HOT WATER SYSTEMS AND SPECIALTIES

B & G Relief Valves

Designed and built to ASME requirements. Tested by National Board—labelled with ASME symbol.

For relieving excess boiler pressures in hot water heating systems, and in the lines of service water systems. B & G Relief Valves have the design features which assure dependable service.

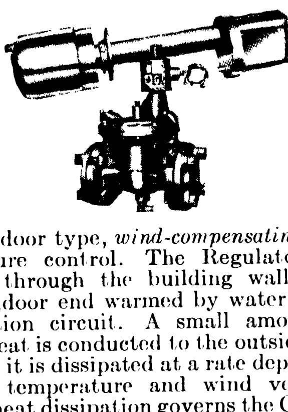




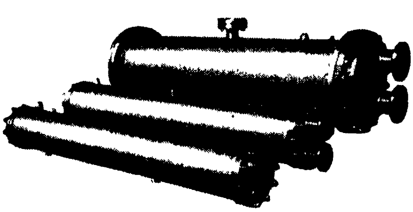
B & G Reducing Valves

Fast operating valves for keeping hot water heating systems properly filled. Easily adjusted to meet varying building heights. Also high pressure reducing valves for protection of plumbing fixtures.

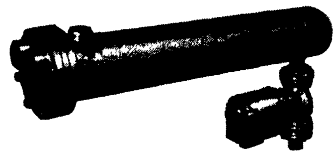
B & G Comfort Control System



Outdoor type, *wind-compensating* temperature control. The Regulator projects through the building wall, with the indoor end warmed by water in the radiation circuit. A small amount of this heat is conducted to the outside end, where it is dissipated at a rate dependent upon temperature and wind velocity. This heat dissipation governs the Control Valve, which permits hot boiler water to enter the system in required amounts.



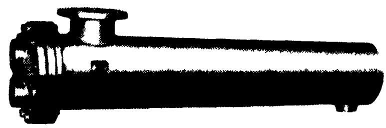
B & G Type "CWU" Radiation Heater



A "shell and tube" heat exchanger, installed below the water line of a steam boiler. Hot boiler water is pumped by a B & G Booster through the shell, thereby heating water for the heating system, which is pumped through tubes of Heater.

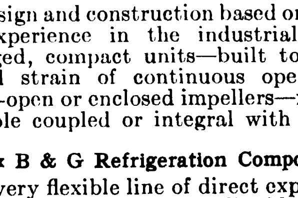
Pumping the water through both heater and heating system affords excellent temperature control, and permits use of much smaller pipe and fittings.

B & G Type "SU" Instantaneous Water Heaters



For heating water with steam. Ideal for industrial plants or wherever large volumes of hot water are required continuously for service water supply or process work. No storage tank required—the large heat transfer surface in these units heats water instantly as needed. Available in a wide range of capacities.

B & G Centrifugal Pumps



Design and construction based on years of experience in the industrial field. Rugged, compact units—built to withstand strain of continuous operation. Semi-open or enclosed impellers—motors flexible coupled or integral with pump.

← B & G Refrigeration Components

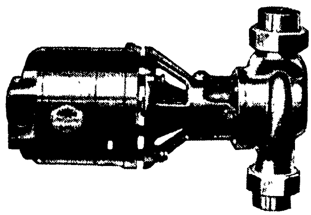
A very flexible line of direct expansion evaporators, condensers, liquid receivers, combination liquid receivers and subcoolers for refrigeration purposes are now available. Special alloys may be incorporated in the units for those critical heat transfer applications.

H. A. Thrush & Company

Peru, Indiana

Representatives in Principal Cities

FORCED CIRCULATING THRUSH FLOW CONTROL SYSTEM OF HOT WATER HEATING AND HEATING SPECIALTIES



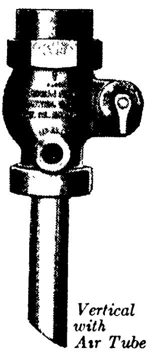
Horizontal Water Circulator

Patent Nos. 2,054,009, 2,111,441, 2,137,791, 2,207,183, 2,207,208, 2,257,867, 2,356,482, 2,358,670, 2,515,811

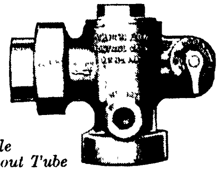
Thrush Flow Control System of Hot Water Heat assures continuous radiant heat whether used with radiant panels, convectors, radiant baseboards or radiators. Circulation is forced and operation is entirely automatic. It automatically compensates for outdoor weather changes. Hot water for kitchen, laundry and bath, both summer and winter are provided by the heating plant boiler. Piping plans and engineering assistance are available to the trade.

THRUSH WATER CIRCULATORS

Forced circulating pumps for Hot Water Heating and Domestic Water Systems. Insure uniform heating. Quiet and efficient, long lived and vibration-free. Sealed-in lubrication. Made in 6 sizes, 1 in., 1¼ in., 1½ in., 2 in., 2½ in., and 3 in. Hi-Head Horizontal Thrush Water Circulators, designed for use with radiant baseboards, convectors, radiant panels, etc., which require higher heads, are also available in 1 in., 1¼ in., and 1½ in. sizes.



Vertical with Air Tube



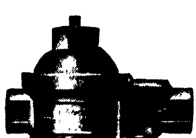
Angle Without Tube

THRUSH FLOW CONTROL VALVES

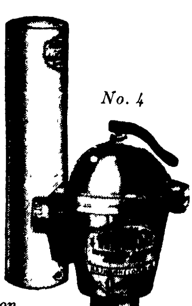
•

ANGLE or VERTICAL With or Without Air Vent Tube

Special check valves for use on automatically fired boilers, automatically control circulation when installed with a Thrush Circulator. Close tight when circulator is not running, prevent gravity circulation when heat is not needed in the radiators. Available with Air Tube which vents air from boiler into pressure tank, greatly improving heating efficiency by eliminating air from the system. 1 in. through 1½ in. valves also available with solder type outlet unions. Made in six sizes, 1 in., 1¼ in., 1½ in., 2 in., 2½ in. and 3 in.



No. 12—½ in. Automatic Filling Valve



No. 4

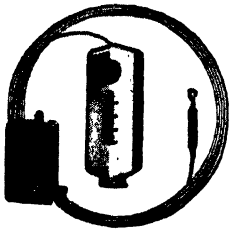
Bronze or Iron ½ in., ¾ in., and 1 in.

THRUSH AUTOMATIC FILLING VALVES

Whenever system pressure drops below 12 lbs, city water pressure will automatically open the valve and admit water until the system is filled.

THRUSH LOW PRESSURE RELIEF VALVES, PRESSURE TANKS

Relief Valves protect heating boilers from excess pressures. Large metal diaphragm assures positive closing and opening. Thrush Air Tight Pressure Tank conserves water and fuel. Heated water expands into tank.

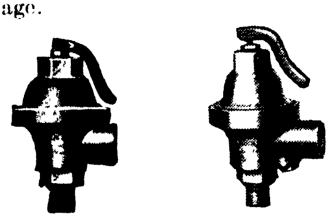


No. 200 *No. 201*

Patent reissue No. 19300

No. 201 THRUSH RADIANT HEAT CONTROL

Automatically maintains room temperature within a fraction of a degree. Controls both room and water temperature in the heating system, compensating to prevent variation in room temperature or a lack of radiant heat. No. 200 Thrush Relay Transformer supplies low voltage.

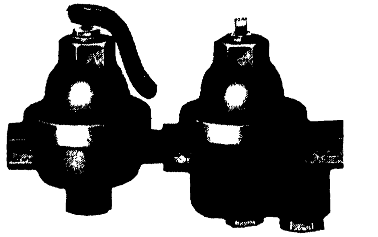


THRUSH HIGH PRESSURE WATER RELIEF VALVES

To protect hot water heaters, range boilers and automatic water heaters from excess pressure. Factory setting 85 lb. Adjustable at factory to 150 lb maximum.

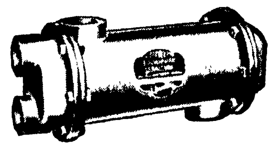
NO. 76 HIGH PRESSURE AND TEMPERATURE RELIEF VALVES

The relief valve illustrated at the right has an added safety feature. In addition to the relief of excessive pressure there is also provision for relief if excessive temperatures develop. It has a fusible plug in an opening in the casting. Another type with a fusible insert in the stem which projects into the hot water line, is also available. The fusible plug or insert melts at 210 F. Both types are designed for use in the hot water outlet line.



THRUSH DUAL CONTROL UNITS

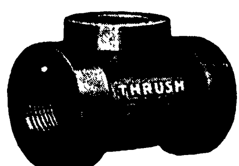
Provide automatic pressure relief, automatically fill and maintain water supply in hot water heating system. Built-in strainer. Made in four types, brass or cast iron, ½ in. or ¾ in.



Patent No. 2,180,620

THRUSH WATER HEATERS

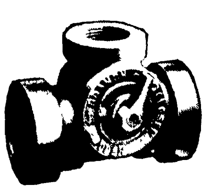
Highly efficient heat exchangers or converters. Fifteen sizes, for Hot Water or Steam. Pressure up to 150 lb water, 50 lb steam. Straight tubes readily cleanable. Provide Domestic Hot Water at low cost. Also used industrially for heating or cooling liquids.



Bronze or Iron

NON-ADJUSTABLE THRUSH SUPPLY TEES FOR ONE PIPE HOT WATER SYSTEMS

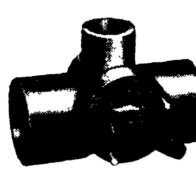
Assure positive diversion to radiator. Available in threaded cast iron or in bronze with solder connections for copper piping. Complete range of sizes.



C. I. Patent No. 2,404,996— Bronze Patent Applied for

ADJUSTABLE THRUSH SUPPLY TEES FOR ONE PIPE SYSTEMS

Provide exact balancing of forced circulating one-pipe hot water heating systems. Easily adjusted. When branch flow is cut down main flow is increased, not retarded. Available in bronze with solder connections for use with copper pipe and in threaded cast iron for steel pipe.



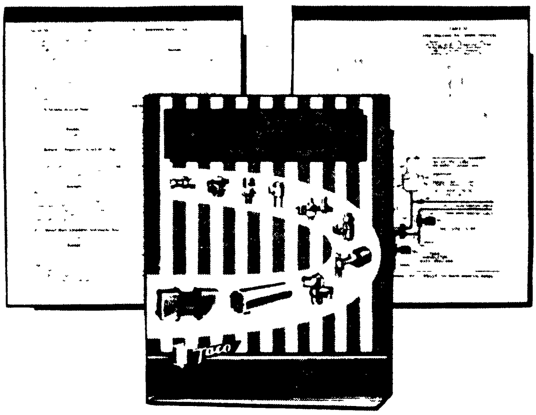
WRITE FOR NEW CONDENSED CATALOG

Taco Heaters, Incorporated

137 South St., Providence 3, R. I.

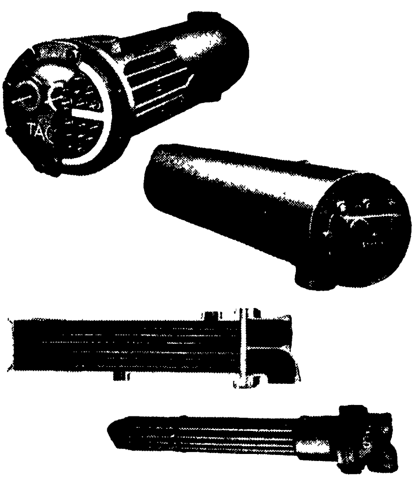
TACO HEATERS OF CANADA, LTD., 24 Adelaide St., W., **Toronto**

Accurate Design Information on Panel Heating
. . floor, ceiling, snow removal



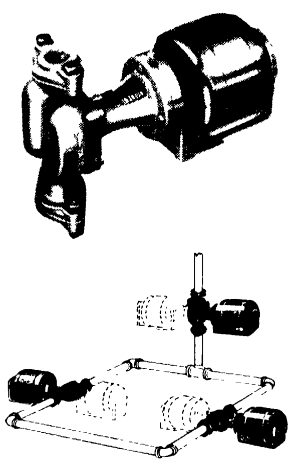
Simplified design tables, based on current scientific engineering practices, have been designed to avoid long and tedious calculations. Installations made in accordance with their recommendations, assure satisfactory and economical results.
Send for new 60 page catalog which has all this information on Taco products.

Heat Exchangers . . . Big or Small



There is a storage or tankless type Taco heat exchanger for every job—residential, apartment, commercial or industrial.

New Horizontal Circulators



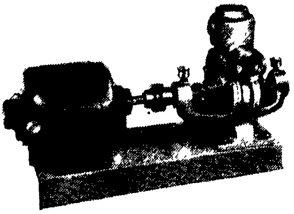
Taco circulators, both standard and high duty, are quiet, rugged and powerful. Packed in specially designed cartons for quickly changing flange sizes.

Aurora Pump Company

40 Loucks Street, Aurora (Chicago Suburb) Illinois

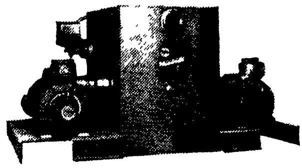
Manufacturers of
Turbine-Type, Vertical and Horizontal Centrifugal and Special Design Pumps

Distributors in Principal Cities



Apco Single Stage Turbine-Type Pumps. Also available in Two Stage.

Barring Unusual Conditions We MAINTAIN A COMPLETE STOCK FOR IMMEDIATE SHIPMENT

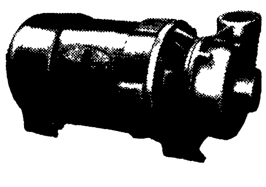


Packaged Duplex Condensation Return Unit with No. 4 Series Apco Pumps; designed for smaller jobs. **LARGER UNITS** *up to 150 gpm capacity, horizontal and vertical also available.*

APCO PUMPS are IDEALLY SUITED to MANY HEATING and AIR CONDITIONING DUTIES—CAPACITIES UP TO 150 GPM—HEADS TO 600 FT.

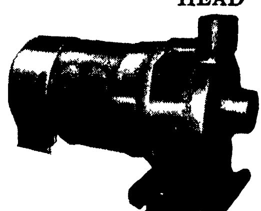
FEATURES—APCO pumps are distinguished for their ability to deliver small capacities against high heads; their ability to deliver with but slight change in capacity or efficiency against drastic head variations. They possess these advanced features—**SIMPLE—WEAR-FREE—COMPACT • HIGH EFFICIENCY • WILL NOT VAPOR BIND • HYDRAULICALLY BALANCED • ACCESSIBILITY • HIGH SUCTION LIFT** (28 ft at sea level) • **QUIET OPERATION • HIGH PRESSURE PER STAGE • DOUBLE SUCTION DESIGN • PRECISION SHAFTS • BALL BEARINGS** (support on both sides of impeller) • **RIGHT OR LEFT HAND OPERATION** (Changeable in Field without special parts) • **REPLACEABLE COVER PLATES • AVAILABLE IN VARIOUS CORROSION RESISTANT ALLOYS.**

LOW HEAD • SMALL CAPACITY—SIDE SUCTION PUMPS • MEDIUM HEAD



At Left
TYPE—SAC—Close Coupled Caps 5 to 85 gpm. Heads to 45 Ft. Sizes Available ¾", 1¼", 2"

At Right
TYPE—JMC—Close Coupled Caps, 5 to 150 gpm. Heads to 100 Ft. Sizes Available 1", 1¼", 1½", 2"



APPLICATIONS
Well suited as integral part of manufacturer's product such as air conditioning units, cooling towers, evaporator coolers, hot water circulators etc., also for general service.

CONSTRUCTION SPECIFICATIONS
Made in bronze fitted, all iron or all bronze construction.

CASINGS, vertically split—end suction. High grade material as specified. Casing wearing ring is standard on Type JMC.

MECHANICAL SEAL is standard on all sizes and is located in packing cover—easily replaceable.

IMPELLER is balanced, enclosed type of high grade bronze material.

SHAFTS—Stainless steel on Types JMC, SAC and JA. Can be furnished on Type SA at additional cost.

MOTOR Built to *NEMA* specifications and equipped with **STAINLESS STEEL** shaft.

BALL BEARINGS—Permanently sealed against dust and moisture—No lubrication required.

AURORA CENTRIFUGAL PUMPS are available in many types and sizes—all noted for their streamline co-ordination between impellers and shells.

Write for

CONDENSED CATALOG "V"

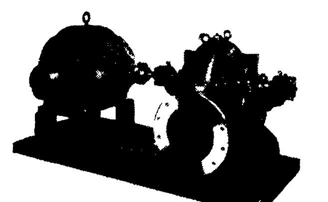
Buffalo Pumps, Inc.

450 Broadway, Buffalo, N. Y.

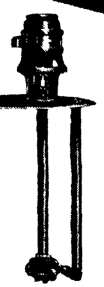
Manufacturers of a Complete Line of Centrifugal Pumps, Single and Double Suction, Single and Multistage, for All Types of Heating and Air Conditioning Installations. Write For Engineering Bulletins on Your Problem or Consult Your Nearest Engineering Representative Listed Below:

ENGINEERING REPRESENTATIVES

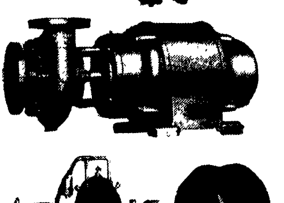
ALBANY 7, N. Y., Mr. R. B. Taylor, 966 Broadway; *ATLANTA, GA.*, Mr. J. J. O'Shea. 305 Techwood Dr., N. W.; *BALTIMORE 1, MD.*, Mr. C. A. Conklin III, 1014 Cathedral St ; *BOSTON 76, MASS.*, Mr. E. D. Johnson, 507 Main St., Melrose Station; *CANADA PUMPS*, Mr. A. S. Capwell, Kitchener, Ont., Canada; *CHICAGO 6, ILL.*, Emmert & Trumbo, 20 N. Wacker Drive; *CINCINNATI 25, OHIO*, Mr. S. O. Johnson, 2944 Colerain Ave.; *CLEVELAND 15, OHIO*, Weager & Sherman, 570 Hanna Bldg., Euclid at 14th St.; *CORPUS CHRISTI, TEXAS*, Langhammer-Rummel Co., 101 N. Alameda St; *DALLAS 1, TEXAS*, Mr. T. H. Anspacher, 615 Tower Petroleum Bldg.; *DENVER COLORADO*, Stearns-Roger Mfg. Co., 1720 California St.; *DETROIT 16, MICHIGAN*, Coon-DeVisser Co., 2051 W. Lafayette Blvd.; *GREENVILLE, S. CAROLINA*, Mr. Roy A. Stipp, P. O Box 1796, 104 E. Stone Ave.; *HOUSTON 5, TEXAS*, Mr. D. M. Robinson, 2609 Sunset Boulevard; *INDIANAPOLIS 4, INDIANA*, S. E. Fenstermaker & Co., 937 Architects & Builders Bldg.; *DAVENPORT, IOWA*, D. C. Murphy Co., 305 Security Bldg.; *DES MOINES 14, IOWA*, D. C. Murphy Co., 840-5th Avenue; *KANSAS CITY, KANSAS*, Mr. W. K. Dyer, Box 966, Post Office Building, Mission, Kansas; *LITTLE ROCK, ARKANSAS*, Mr. J. L. Brown, Terminal Warehouse Bldg.; *LOS ANGELES 13, CAL.*, Halladay & Knauff, 804 Pershing Sq. Bldg.; *LOUISVILLE 2, KENTUCKY*, Mr.H. M. Lutes, 633 South Fifth St.; *MEMPHIS, TENNESSEE*, Humphrey-Wynne Co., 620 Sterick Bldg.; *MIAMI, FLORIDA*, Mr. H. L. McMurry & Co., 46 N. E. 6th St.; *JACKSONVILLE, FLORIDA*, H. L. McMurry & Co., 25 Riverside Viaduct; *TAMPA, FLORIDA*, H. L. McMurry & Co., % Peninsular Warehouse & Ter. Co., 411 Hampton St.; *MINNEAPOLIS 2, MINNESOTA*, E. Floyd Bell, 2102 Foshay Tower; *NEW ORLEANS 12, LA.*, Devlin Bros., 1003 Maritime Bldg.; *NEW YORK 7, N. Y.*, Johnson & Norman, 39 Cortlandt St.; *NEWARK 2, N. J.*, Johnson & Norman. Room 205, 27 Washington St.; *OMAHA 2, NEBRASKA*, Wain Engineering Co., 2311 Douglas St.; *PHILADELPHIA 2, PA.*, Davidson & Hunger, 1200 Cunard Bldg.; *PITTSBURGH 22, PA.*, Mr. H. Lee Moore, 345 Fourth Avenue; *ROCHESTER 4, N. Y.*, R. D. Moyer, 846 Sibley Tower Bldg.; *ST. LOUIS 3, MO.*, Mr. J. W. Cooper, 2118 Pine St.; *SALT LAKE CITY 1, UTAH*, Stearns-Roger Mfg. Co., 405 Kearns Avenue; *SAN FRANCISCO 3, CAL.*, Mr. Chas. W. Lockhart, 1214 Central Tower Bldg.; *SAN ANTONIO, TEXAS*, Langhammer Rummel Co., 300 Blum St.; *TOLEDO 2, OHIO*, Mr. Carl Eyster, 1118 Madison Avenue; *WASHINGTON 5, D. C.*, Mr. G. S. Frankel, Mgr., 310 Woodward Bldg.



SINGLE STAGE DOUBLE SUCTION PUMPS. For clear water service from 10 to 10,000 gpm, and for heads up to 350 feet, these efficient pumps are widely used for air washers and other air conditioning units. Each pump is hydraulically balanced (water enters each side of impeller with equal pressure and volume), which contributes to the smooth operation and efficiency, as do simply formed water passages in the casing. BULLETIN 955-N.



AUTOMATIC SUMP PUMPS. Compact units, shipped complete, ready for quick installation. Ball bearing thrust carries weight of moving parts. Shaft is entirely enclosed—completely protected from sump water and from fouling with waste or stringy matter flowing into sump pit. All parts are readily accessible. BULLETIN 963-F.



CLOSE-COUPLED SINGLE SUCTION PUMPS. Extremely compact design permits installation vertically or horizontally. Available with either threaded or flanged connections. Suited to handling hot water with low submergence. Impeller overhung on motor shaft assures permanent alignment. BULLETIN 975-C.

SELF-PRIMING SINGLE AND DOUBLE SUCTION PUMPS. Positive prime is obtained almost instantly when pump is started, without use of foot-valve. All parts are accessible, and above liquid to be pumped. Can be automatically operated in remote locations. BULLETIN 970-A.

Chicago Pump Company

2330 Wolfram Street BRunswick 8-4110 Chicago 18

PRODUCTS—Return Line Vacuum Heating and Boiler Feed Pumps, Condensation, House, Booster, Fire Pumps, Circulating, Brine, Sewage, Bilge, Sludge, Pneumatic and Tankless Water Supply Systems and Automatic Alternator for Duplex Sets of Pumps.

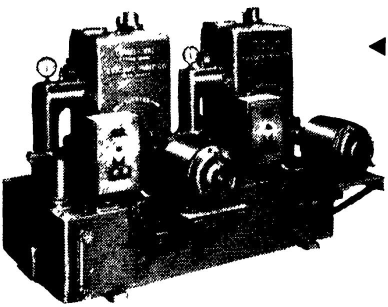


Fig. D-9900—Duplex "Condo-Vacs" with Duplex Double Automatic Control

"CONDO-VAC"

Return Line Vacuum and Boiler Feed Pump for Heating Systems

No vacuum on stuffing boxes, ample clearance in rotating member. It costs less to operate a "Condo-Vac." "Condo-Vac" reduces corrosion in piping and boiler to minimum—because pump does not take in air from atmosphere and entirely eliminates all air coming back from system. "Condo-Vac" is quiet, has a low inlet, entirely automatic, fool-proof, easy to maintain. *Ask for Bulletin 270.*

CLOSE-COUPLED PUMPS

Boiler Feed, Circulating, Tank Filling, Water Supply

Capacities range from 3 to 600 Gpm against heads up to 189 ft. Motors from ¼ to 20 hp. Discharge 1 to 3 in. Closed and open type impellers. *Bulletin 108.*

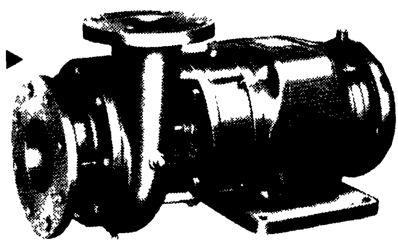


Fig. N-8905—Close-Coupled, side suction pump

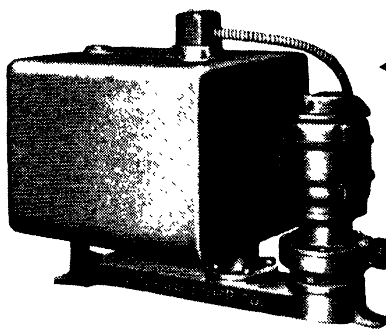


Fig. D-9200

"SURE-RETURN" CONDENSATION PUMPS

for Low and Medium Pressure, and Systems up to 75,000 Sq Ft Radiation

"Sure-Return" Condensation Pumps and Receivers are built for systems up to 75,000 sq ft of direct radiation and for low and medium pressures. Built in either single or duplex units. Duplex units are alternated in their operation by the Automatic Alternator. *Complete data in Bulletin 250.*

VERTICAL CONDENSATION PUMPS

for Low and Medium Pressure for Systems from 500 to 100,000 Sq Ft Radiation

The vertical condensation pump is designed to receive returns from lowest radiation. The receiver is placed underground—an ordinary hole sufficing if necessary—and requires very little floor space. Unit is shipped complete, easy to install, assembled so as to prevent steam leaks. Special bearings will stand up under hot water for several years. A special float mechanism is guaranteed not to leak or stick in stuffing box. *Complete data and description in Bulletins 245, 253 and 255.*

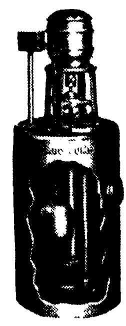


Fig. V-9600 Vertical Condensation Pump

Ingersoll-Rand

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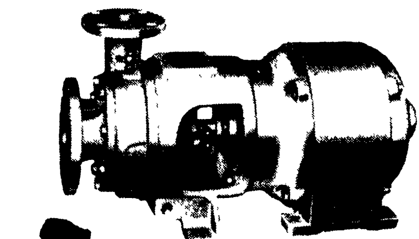
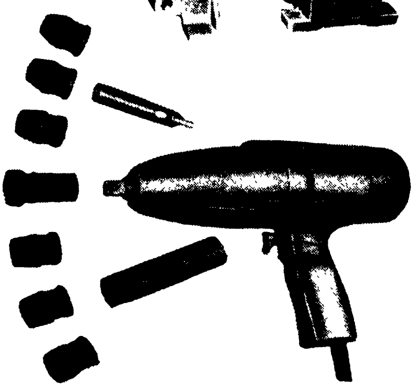
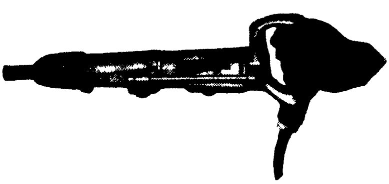
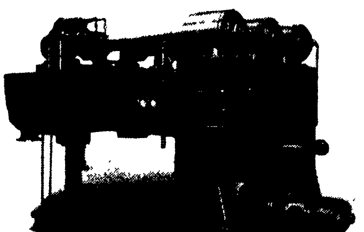








CENTRIFUGAL PUMPS

The MOTORPUMP is a compact, "package" unit, mounted integrally with motor on a rigid, oversize shaft and over-size bearings. It is highly adaptable to many services, needs no special foundations, and operates equally well in any position. It is available in several materials for pumping various liquids. Capacities from 10 to 1800 gpm, heads to 650 ft.

Other I-R pumps are offered for all hydraulic services, with any type of drive.

ALL PURPOSE PORTABLE TOOLS

Electric Impactools are light-weight, portable, tools for running and removing nuts, screws and studs, drilling, reaming, tapping, wire brushing, drilling brick and masonry, driving wood augers, hole-saw work, and the 101 jobs encountered in installation work. Plug into any wall outlet. Capacities: 3/8 in. drills, tapping to 3/4 in., running nuts to 5/8 in. thread size. 110 or 220 volts, universal motor.

LIGHT-WEIGHT JACKHAMER

The J-10 Jackhamer is the smallest of the self-rotating, rock-drill line, weighs 14 lbs, and is especially designed for maintenance and installation work. Its uses include drilling masonry for conduit, sprinkler hangers, foundation bolt holes, pipe lines and drains as well as tearing out brick-work for doors and windows and similar demolition jobs. The J-10 is air-powered.

STEAM-JET REFRIGERATION

Where refrigeration is needed down to 35°F, and a supply of steam is available, the I-R system of Stream-Jet Water-Vapor Refrigeration should be considered. In this system water is the only refrigeration medium. It is cooled by direct evaporation in a high vacuum created by steam-jet booster ejectors. There are no moving parts, no vibration, nor noise. Sizes run from 30 to 1000 tons of refrigeration and can be built to operate at any one of a wide range of steam pressures down to 1 lb per sq in.

I-R COMPRESSORS OF ALL TYPES AND SIZES

I-R Compressors are offered in all sizes and types from 1/2 to 3000 hp, in pressures from a few ounces to 15000 psi, and in stationary or portable models. Air-cooled units range from 1/2 to 100 hp. I-R Water systems are available for industrial and domestic uses.

Peerless Pump Division

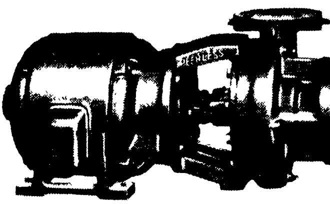
Food Machinery and Chemical Corporation

PLANTS: Los Angeles 31, Calif., Indianapolis 8, Ind.

Offices: New York; Atlanta; Indianapolis; Chicago; St. Louis; Tulsa; Dallas, Plainview, Lubbock, Texas; Albuquerque, New Mexico; Phoenix, Arizona; Fresno, Los Angeles, Calif.

PRODUCTS:

Vertical Pumps: Deep and shallow well pumps; water supply, boosting, circulating and waste disposal pumps; condensate pumps; sump pumps. All types of drive. **Horizontal Pumps:** General purpose horizontal split case and end-suction pumps; turbine vane type pumps; boiler feed pumps; condensate pumps; circulating pumps; process pumps. All types of drive. **Seals:** Mechanical shaft seals for rotative shaft equipment and pumps. From ½ in. to 5 in. in diameter.

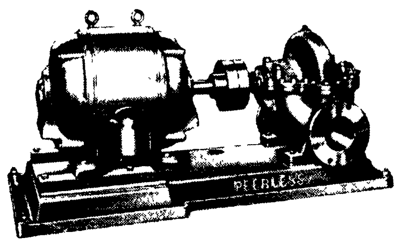


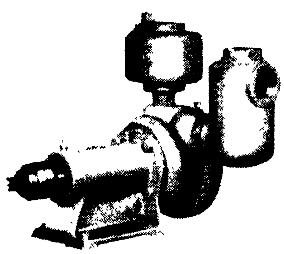
Peerless Type PE and Type PB general purpose, end-suction close and flexible coupled pumps (at left).

One of the most complete lines offered by any manufacturer. Capacities: to 5500 gpm. Sizes: ¼–150 hp. Heads: up to 260 feet. Single stage design. Compact, versatile, dependable and efficient. Good hydraulic characteristics. Completely described in Bulletin B-2300.

Peerless Type AS general purpose horizontal split case pump equipped with mechanical shaft seals (at right).

A packingless pump almost one half the size of conventional pumps offering same capacities and pressures. Saves space. Short shaft lengthens life. Smooth operation. Sizes. 1½ in. x 8½ in. through 4 in. x 13 in. Capacities: up to 750 gpm. Heads: to 230 feet; temperatures to 200 F. All types of drive. Completely described in Bulletin B-1350. Also larger size, designated as type "A," to 48 in. discharge for higher heads and capacities. Completely described in Bulletin B-1300.





Peerless Type TVE and TVB turbine vane type pumps for high temperatures and high heads (at left).

Standard and self-priming pumps for boosting, circulating and boiler feed application. Handle cold or hot water to 250 F. Heads: up to 800 feet. Capacities: up to 58 gpm. Sizes: Fractional to 20 hp. Compact, dependable, modern design. Completely described in Bulletin B-2205.

Peerless Hydro-Line compact, close-coupled vertical centrifugal pump for limited NPSH applications (at right).

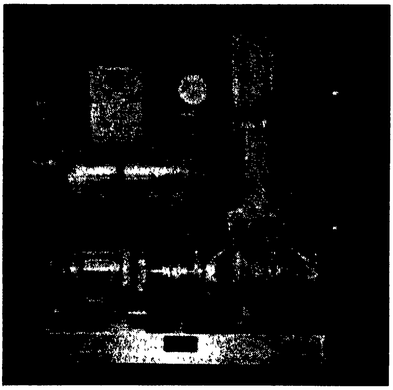
Ideally applied to condensate service. Provides capacities up to 5000 gpm. Pressures: up to 1500 feet. Temperatures: up to 250 F. Pump is enclosed in steel barrel. Easily installed as a complete unit. Ruggedly constructed for heavy-duty service. Designed for minimum required NPSH. Completely described in Bulletin B-592.

OTHER PEERLESS BULLETINS are available, completely describing each type of pump in the Peerless line. Request copies describing the type pump in which you are interested.

The Nash Engineering Company

234 Wilson Road **South Norwalk, Conn., U. S. A.**

Sales and Service Offices in all Principal Cities



Return Line Vacuum Heating Pump

Standard with the heating industry for over eighteen years. Removes air and condensation from return lines of vacuum steam heating systems, discharging air to atmosphere and returning water to the boiler.

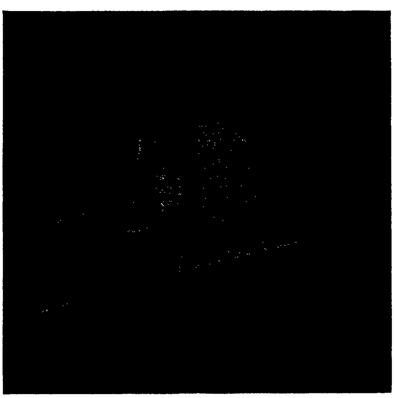
Two independent units are combined in a single casing—an air unit and a water unit. Impellers of both are mounted on the same shaft. Pump is bronze fitted throughout.

Supplied direct connected to standard electric motors, for belt drive, or for steam turbine drive. For continuous or automatic operation. Standard in capacities up to 300,000 sq ft E.D.R. Larger units special. Bulletins Nos. 307, 308, 309, and 310 on request.

Vapor Turbine Vacuum Heating Pump

Jennings Vapor Turbine Heating Pumps combine all advantages of the standard return line heating pump with a new type of drive, a specially designed low pressure turbine which operates directly on steam from the heating mains on any system, requiring a differential of only 5 in. of mercury, and returns that steam to the heating system with practically no heat loss.

This pump affords the safety and economy which goes with continuous condensation return and steady vacuum, and at no cost for electric current. Furnished standard in capacities up to 65,000 sq ft E.D.R. Larger units special. Bulletin No. 290 on request.



Condensation Pump and Receiver

Removes the condensation from radiators in return line steam heating systems, particularly radiators set below the boiler water line level, and pumps the condensation back to the boiler. Pump is bronze fitted with enclosed centrifugal impeller of improved design. By making the pump casing a part of the return tank, and bolting the motor base to the tank, floor space is conserved. The rectangular construction permits installation in a corner against the wall.

These pumps are furnished in standard sizes with capacities ranging from $1\frac{1}{2}$ to 225 gpm of water. For serving up to 150,000 sq ft of equivalent direct radiation. Bulletin No. 319 on request.

The Nash Engineering Company

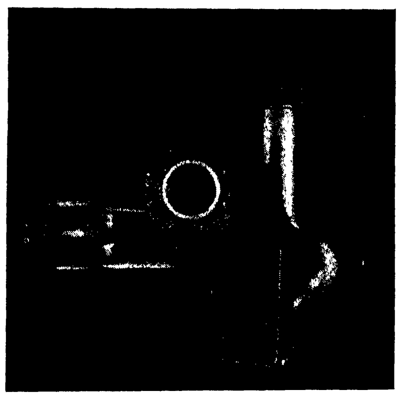
234 Wilson Road **South Norwalk, Conn., U. S. A.**

Sales and Service Offices in all Principal Cities

SEWAGE EJECTOR

For pumping unscreened sewage or drainage from basements below the street sewer level, handling crude sewage from low level districts, pumping effluent, sludge and other heavy liquids. The Jennings Sewage Ejector is of the pneumatic type. Air, compressed only to the pressure at which it is used, by a Nash Hytor Air Compressor, is motive power to pump the accumulated sewage from a pot to the sewer. There are no air storage tanks, reciprocating air compressors or screens, no air valves. Furnished in several standard sizes up to 1500 gpm against heads up to 100 ft. Bulletins on request.

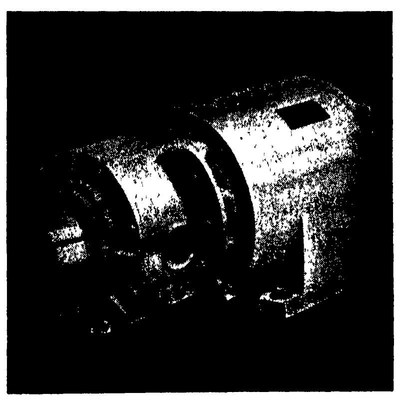
Suction Sump and Sewage Pumps



Jennings Sump Pumps are self-priming centrifugals for handling seepage water and liquids reasonably free from solids. Sewage Pumps are equipped with non-clog type impeller for liquids containing solids. Suction piping only is submerged. Centrifugal impeller and vacuum priming rotor are mounted on same shaft that carries rotor of the driving motor, forming a single moving element, rotating without metallic contact.

Will handle air or gas with liquid being pumped, and because of self-priming feature are installed entirely outside of pit, affording accessibility for inspection or cleaning. Capacities to meet all requirements. Bulletins Nos. 159, 161, and 338 on request.

Air Compressor and Vacuum Pump



Nash Air Compressors operate on a unique and different principle. The one moving part rotates in casing without metallic contact. There is nothing to wear, and no internal lubrication.

Nash Compressors deliver absolutely clean air; ideal for agitation of liquids, pressure displacement, and handling gases. Vacuum pumps ideal for priming pumps, blood sucking pumps in hospitals, and wherever non-pulsating vacuum is required.

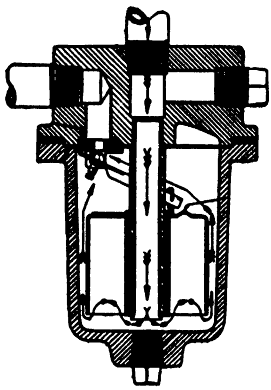
Pressure 75 lb or vacuum 27 in. of mercury. Furnished for any capacity; special for higher vacuums and pressures. Bulletins Nos. 282, 325, 331 and 337 on request.

The V. D. Anderson Company

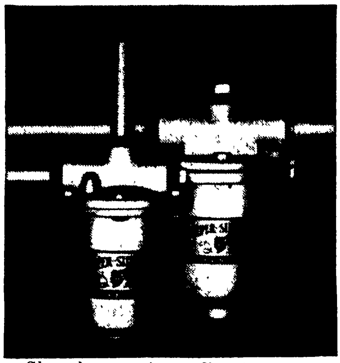
Representatives in all principal cities

1960 West 96th Street, **Cleveland, Ohio**

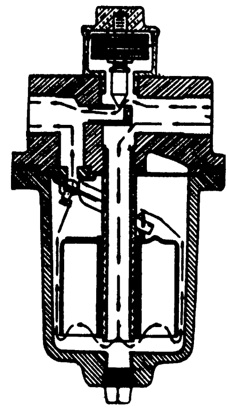
Manufacturers Of Steam Traps For Over 60 Years



No. 21, 22, 23, 24, 25.



Shows how trap is installed either as an elbow or straight in-line



No. 21B, 22B, 23B, 24B.

SUPER-SILVERTOP STEAM TRAPS

Are inverted bucket steam traps of an improved, thoroughly tested design. These steam traps automatically remove both condensate and air from the steam system—no manual operations are necessary.

APPLICATIONS

Super-Silvertops are used on any steam-using equipment where it is desired to remove condensate and air automatically in order to produce maximum heating efficiency.

FEATURES

Simplified Piping—Connected either as an elbow or straight-in-line—only one nipple needed since the U-tube is inside the trap. Saves as much as three elbows, three nipples and 60 minutes of time, compared to other inverted bucket steam traps.

Precision Parts Alignment—The bucket does not swing free. It is controlled in true engineering fashion, guided on a hexagonal tube. This makes a knife-edge contact, eliminating almost all friction. The unusual guide arrangement keeps all parts in proper alignment and prevents the bucket from hitting the sidewalls of the case. Positive closing of the valve is insured.

Self-Cleaning—The reversing of the condensate flow on entering the trap produces a scrubbing action. This stirs up any sediment and dirt which is then carried away in the discharging condensate.

No Restricting Passages—Even in the smallest sizes there are no narrow cored passages to become clogged with scale.

Cannot Air-bind—As the air is automatically discharged ahead of the condensate in each cycle of operation.

Vacuum or Pressure—These traps do not leak steam. No danger of vacuum being destroyed—trap is recommended for vacuum operation.

INSTANT HEAT

Anderson Super-Silvertops can be equipped with Thermal Air Eliminators or Combination open float and thermostatic steam traps can be used where instant heat is desired. **Other Anderson products are:** float-type steam traps, air release valves, and pipeline strainers, purifiers and separators.

Note: Write for Free Copy of Book "SOLVING STEAM TRAP PROBLEMS."

SIZES, LIST PRICES, AND CAPACITIES

Size No. Trap	110	118	119	120	21	22	23	24	25
Size Connections (See Note)	½" only	½" or ¾"	½" or ¾"	½" or ¾"	½" or ¾"	¾" or 1"	¾" or 1"	1" or 1¼"	1½" or 2"
Shipping Wt., Lbs.	2¾	5	6	7	8	14	16	38	75
Diameter, Inches	4	3⅜	3⅜	3⅜	4 15/16	5½	5½	8 3/16	9¾
Height, Inches	4 1/16	4 9/16	5¾	6 9/16	6¾	8 1/16	10 5/16	13 11/16	17 5/32
Max. Ga. Pressure	100	150	200	200	200	250	250	250	250
List Price	$10.00	$9.00	$10.00	$12.00	$14.00	$23.00	$32.00	$45.00	$60.00
WITH THERMAL AIR ELIMINATOR									
List Price	$11.60	$10.60	$11.60	$13.60	$15.60	$26.20	$35.20	$49.80	$66.40
Differential Pressure	**CAPACITY IN POUNDS OF WATER PER HOUR**								
1	137	161	225	390	775	1050	1500	2400	9200
5	305	360	510	970	1700	2300	3200	5274	21000
10	435	505	705	1190	2000	3200	4600	7500	29650
20	300	700	980	1650	2000	3500	6000	11100	40300
50	455	650	900	1400	1600	3100	4500	7500	26000
100	440	455	675	1200	1600	2450	4200	10000	25300
125		500	740	1310	1700	2900	3400	8000	19100
150	.	450	580	1000	1200	2000	3700	8600	20500
200		...	655	1100	1300	2200	4150	8900	16800
250						1900	4500	9600	13750

COMBINATION TRAPS

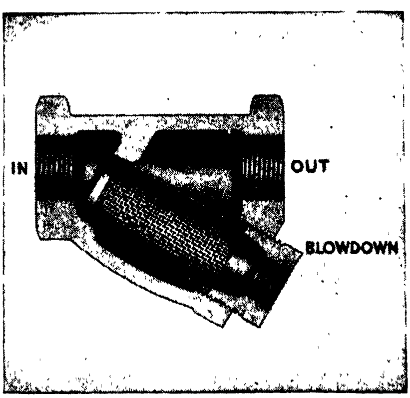
Size No. Trap	119B	120B	21B	22B	23B	24B
Size Pipe Connections	¾"	¾"	¾"	1"	1"	1¼"
Shipping Wt., Lbs.	7	8	9	15	17	39
Diameter in Inches	3⅜	3⅜	4 15/16	5½	5½	8 3/16
Height in Inches	8 1/16	8⅞	8¾	10 1/16	12 3/16	15 11/16
Max. Ga. Pressure*	50	50	50	50	50	50
List Price	$14.00	$16.00	$18.00	$27.00	$36.00	$49.00
Differential Pressure	**CAPACITY IN POUNDS OF WATER PER HOUR**					
1	315	480	865	1140	1590	2550
3	546	820	1504	1994	2654	4275
5	710	1170	1900	2500	3400	5500
10	905	1390	2200	3400	4800	7700
15	1032	1600	1950	3500	5600	10200
20	1180	1850	2200	3700	6200	11350
30	910	1310	1500	3700	4700	6500
50	1100	1600	1800	3300	4700	7800

Capacities based on continuous flow. When ordering be sure to specify maximum steam pressure.

Note: Pipe sizes shown in heavy type are standard and traps will be shipped tapped standard unless otherwise specified. Pipe sizes shown in light type furnished at no additional cost but only from Cleveland stock.

* *Combination Traps are available for pressures up to 150 psi. Prices, capacities and other information available upon application.*

ANDERSON STRAINERS



Cutaway view of Anderson strainer

Anderson Self-Cleaning Strainers remove scale, grit, and sediment from the line. Stainless steel strainer screens (brass on 2½ in. and 3 in.) are standard equipment and have an extremely large free area. Standard screen for steam and other gases has 1/64 in. openings—for liquids 1/32 in. openings. Other sizes of openings furnished when specified, prices on application. Made in sizes from ¼ in. to 3 in. inclusive with screw connections.

Armstrong Machine Works

851 Maple Street, Three Rivers, Mich.

Representatives in Principal Centers

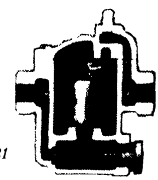
Steam Traps . . . Air Traps . . . Humidifiers
High Side Floats . . . Refrigerant Purgers

ARMSTRONG INVERTED BUCKET STEAM TRAPS

Armstrong offers a complete line of traps for draining low and high pressure steam headers, pipe coil radiation, unit heaters, air conditioning equipment, process equipment, etc., to gravity or vacuum return systems.

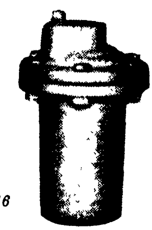
Automatic Air Discharge. Standard traps automatically discharge normal amounts of air along with condensate. Where large amounts of air must be vented quickly when steam is first turned on, Armstrong BLAST Traps with thermic bucket vents are recommended. Air handling capacity ranges from 500 to 1500 cu ft free air per hour.

NEW! Traps with integral strainers save fittings, labor. No. 880 with same capacity as No. 800 listed below is $11.50. → No. 881 with same capacity as No. 811 is $16.00. Both cost less than traps plus separate strainers.

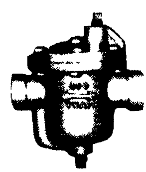


No. 880–881

Capacities, Prices, Dimensions
Note that capacities of No. 811, 812 and 813 traps are the same as No. 211, 212 and 213 respectively. On most installations, the engineer therefore has a choice of side inlet—side outlet or bottom inlet—top outlet body styles.



No. 211–216



No. 800–813

Side Inlet Traps	No. 800	No. 811	No. 812	No. 813
Pipe Connections	½″ or ¾″	½″ or ¾″	½″ or ¾″	¾″ or 1″
List Price (Regular)	$10.00	$14.00	$23.00	$31.00
List Price (Blast Trap)	$12.00	$16.00	$26.00	$34.00
Telegraph Code (Regular)	Aloe	Brown	Cherry	Dawn
Telegraph Code (Blast Trap)	Aloette	Brownette	Cherette	Dawnette
Height	5⅜″	6⅞″	8 11/16″	11¼″
Diameter	5″	5″	6½″	7¾″
Number of Bolts	6	6	6	6
Diameter of Bolts	¼″	¼″	⅜″	½″
Weight	5 lbs.	6 lbs.	15 lbs.	27 lbs.
Maximum Pressure, lbs.	150	250	250	250

Continuous discharge capacity in lb of water per hour at pressure indicated. For more complete information see the Capacity Chart in Armstrong Steam Trap Book. Lb. Pressure	No. 800	No. 811 / 211	No. 812 / 212	No. 813 / 213	No. 214	No. 215	No. 216
5	450	830	1600	2900	4800	7600	14500
10	560	950	1900	3500	5800	9000	17300
15	640	1060	2100	3900	6500	10000	19200
20	690	880	1800	3500	6000	8500	18500
30	500	1000	2050	4000	6800	9800	18000
50	580	840	1900	4100	6300	9000	18200
70	660	950	2200	3800	6000	9200	18300
100	640	860	1800	3600	6200	10400	18000
125	680	950	2000	3900	6700	10900	20000
150	570	810	1500	3500	5700	9500	18500
200	. . .	860	1600	3200	5300	9200	17500
250		760	1300	3500	5700	7000	19000

Bottom Inlet Traps		No. 211	No. 212	No. 213	No. 214	No. 215	No. 216
Pipe Connections	. . . .	½″	½″ or ¾″	½″ or ¾″	1″	1″ or 1¼″	1½″ or 2″
List Price (Regular)	. . . .	$13.00	$21.00	$29.00	$41.00	$55.00	$78.00
List Price (Blast Trap)	. . . .	$15.00	$24.00	$32.00	$44.50	$60.00	$85.00
Telegraph Code (Regular)	. . . . . .	Aspen	Birch	Walnut	Hemlock	Larch	Tamarack
Telegraph Code (Blast Trap)	. . . . . .	Aspette	Birette	Walette	Hemlette	Larette	Tamrette
Height	. . . .	6⅜″	8″	10¼″	12½″	14″	16¾
Diameter	. . . .	4⅛″	5″	6⅜″	7½″	8½″	10 5/16″
Diameter of Bolts	. . . .	5/16″	5/16″	⅜″	⅜″	½″	½″
Number of Bolts	. . . .	6	8	6	8	8	12
Weight	. . . .	6 lb	12 lb	21 lb	34½ lb	47 lb	78 lb
Maximum Pressure, lb.	. . . . . .	250	250	250	250	250	250

High Capacity, Compact Size. The high leverage of the patented free-floating lever makes it possible to open discharge orifices which are very large for over-all trap size.

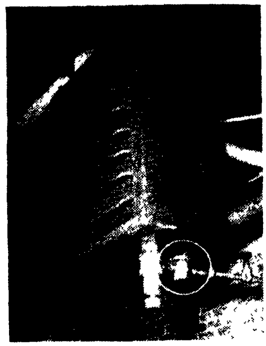
Positive Action. The discharge valve is either wide open or closed tight. Fast opening and closing prevent wire-drawing.

Self-Cleaning. Swirling action of condensate during discharge carries dirt through trap. There are no dead spots for dirt to collect.

High Quality. 18-8 stainless interior parts. Valve and seat are chrome steel, hardened, ground and lapped. Low pressure trap parts same material and quality as those used for 1500 psi, 900° F.

Armstrong Steam Trap Book. 44 pages of data on traps, selection, installation and maintenance. A usable handbook for any engineer dealing with traps. Free copy on request.

TRAPS FOR UNIT HEATERS



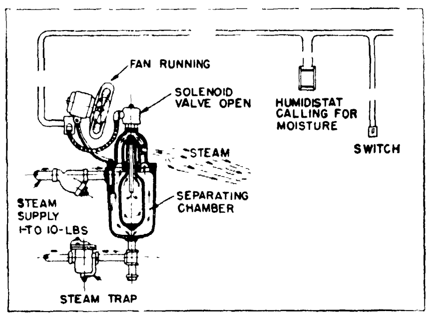
Armstrong inverted bucket steam traps are particularly suitable for draining low or high pressure unit heaters to vacuum or gravity return systems for these reasons:

1. Air and CO_2 are vented automatically each time the trap opens—keeps heaters hot, helps prevent corrosion.
2. Condensate discharges at steam temperature as it accumulates—keeps heaters at maximum temperatures.
3. No build-up of back pressure in return lines.
4. Armstrong traps seldom require maintenance.

A Bulletin with Btu output figures for 24 makes of unit heaters explains how to select traps. Available upon request.

ARMSTRONG STEAM HUMIDIFIERS

For Stores, Offices, Hospitals, Factories, Laboratories



Operation. These units provide automatic, closely controlled humidification by introduction of steam directly into the atmosphere. Installation is comparable to that of unit heaters. Steam at 15 psi or less is required.

A solenoid discharge valve on the humidifier is controlled by a sensitive humidistat. A fan mounted on the humidifier aids in steam dispersal, or a venturi nozzle can be supplied. Where an electric spark might represent an explosion hazard, compressed air operated models are available. In addition to the unit humidifier illustrated, there is a large model for installation in large air ducts and central heating systems. Capacities range from 31 to 630 lbs of steam per hour.

Advantages. Installation cost of Armstrong Humidifiers is as much as 80 per cent less than some types of equipment. The small C-2 unit lists at $182.25 complete with fan and motor, humidistat, strainer and steam trap. Operation, using steam at around $1.00 per ton, is economical. There is no extra load on the heating system. No dripping—any moisture in the steam is re-evaporated in the steam-jacketed separating chamber. Control is accurate within a few per cent R. H.

Bulletin No. 1773 gives complete data on required relative humidities, selection and installation of Humidifiers.

All Armstrong products are sold on a basis of satisfaction or your money back.

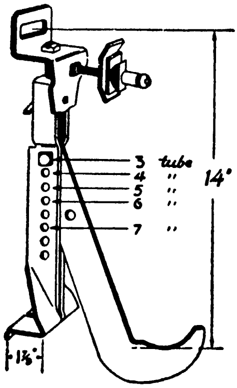
Carty & Moore Engineering Co.

Established 1919

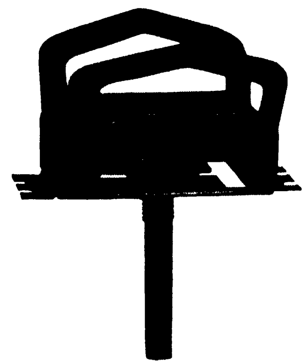
1150 W. Baltimore Ave. Detroit 2, Mich.

STEEL RADIATOR BRACKETS **STEEL CONCRETE INSERTS**

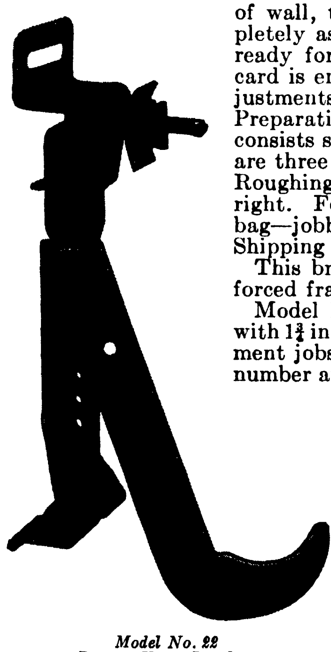
For over a quarter of a century Carty-Moore Speed Brackets have been recognized by engineers and contractors as a superior product and there are nearly a million in use today. Specify C & M Brackets on your next job and note the substantial savings in labor due to the quick-mounting features.



Model No. 22 Bottom Hung Speed Radiator Brackets



Model No. 44 Concrete Inserts



Model No. 22 Bottom Hung Brackets

Illustrated is the No. 22 Speed Bracket for hanging all types of wall, tube or thin-tube radiation. This bracket is completely assembled when shipped and all parts are furnished, ready for quick and easy installation. A brief instruction card is enclosed with each bracket explaining the simple adjustments required to hang any particular type of radiation. Preparation of a C & M Speed Bracket for hanging a radiator consists simply of selecting the right hold-back washer (there are three furnished) and bolting the hook in the proper hole. Roughing-in specifications are shown by the drawing on the right. For specific jobs 30 brackets are packed to a burlap bag—jobbers stock them six in an attractively-labeled carton. Shipping weight approx. 3½ lbs.

This bracket is also available with double hook and reinforced frame for double row wall radiation—specify No. 222.

Model No. 33—A completely assembled top hung bracket with 1¾ in. outset from the wall, especially approved for government jobs. When ordering top hung brackets, specify model number and the type of radiation to be hung.

Carty-Moore No. 44 Concrete Inserts have long been used by leading heating contractors and are designed to meet the most exacting requirements. The C & M Insert is made of heavy gage pressed steel stampings for ⅜ in., ½ in., ⅝ in. and ¾ in. nuts. A long travel slot permits ample adjustment yet the nut cannot pull out and the wide wing-spread allows the insert to become deeply imbedded in the concrete so it cannot tear out under heavy strain. 50-⅜ in. or ½ in. packed in a nicely-labeled carton—25-⅝ in. or ¾ in.

C. A. Dunham Company

DUNHAM

Heating systems and equipment

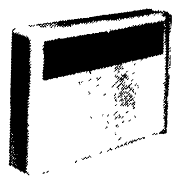
400 W. Madison Street, **Chicago 6, Ill.**

In Canada: C. A. Dunham Co., Ltd., Toronto
In England: C. A. Dunham Co., Ltd., London

Floor Cabinet Convectors; also available with lower inlet grilles and in Recessed type

CONVECTOR RADIATION

Designed for use with hot water or steam. Casings are No. 18 gage steel with removable fronts, and heavier gages for institutions Dampers optional. Capacities with steam at 1 lb, entering air at 65 deg F: 10 to 128 EDR. Cabinet dimensions: 4 to 10 in. wide; 18 to 64 in. long; 20, 24 and 32 in. high.

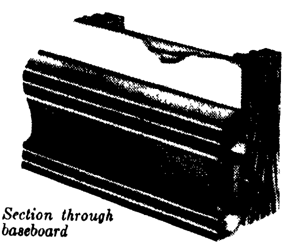


Wall Cabinet Convector; also available in Sloping Top type

Heating element of copper or aluminum fins on seamless drawn round copper tubes brazed to bronze headers; for working pressures up to 150 lbs. Tubes expanded after assembly to assure positive, permanent contact with fins. Heavy side plates protect fins from damage.

BASEBOARD RADIATION

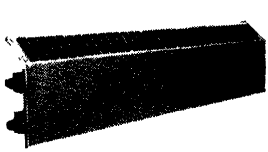
For forced hot water, steam or vapor heating systems. Baseboard consists of front and back enclosure sections of heavy gage sheet steel plus Dunham finned heating element. Enclosure sections are shipped in 10-ft lengths for cutting on job. Assembled by slipping front sections over flanges of back sections. Front sections are removable and are formed so that warm air is directed away from walls. Can be painted. Individual room dampers are optional. Finned heating elements available in 1 to 6 ft. lengths.



Section through baseboard

FIN-VECTOR RADIATION

New design. Extremely compact, light in weight, requires few supports, is easy to handle and assemble. Full range of sizes of covers and heating elements reduces on-the-job cutting. Tube ends designed for fast assembly; covers formed for quick, simple joining. Covers and elements made in lengths from 2 ft to 10 ft in 6 in. increments.



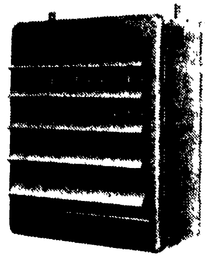
Sloping Top Fin-Vector

Covers in three styles: sloping top, flat top and expanded metal . . . each for one, two or three tiers of heating elements. **Elements:** tubes expanded into fins, no solder bond used. **Tubes:** seamless steel, 1¼ in. and 2 in. IPS; copper, 1¼ in. nominal. **Fins:** 3¼ in. × 3¼ in. and 4¼ in. × 4¼ in. with rounded corners, fins spaced 24, 32 and 48 per lineal foot.

DUNHAM UNIT HEATERS

Horizontal Discharge Type

Type V. Supplies proper volume of air at reasonably low outlet temperatures. Available in two models—H and HM. HM model, having less fin surface and lower final temperatures, is for higher steam pressures. Sturdy steel case assures maximum rigidity and freedom from vibration. Furnished with constant-speed, two-speed or multi-speed motors. Manufactured in 11 sizes. Capacities at 2 lb steam, 60 deg entering air: 15,600 to 360,000 Btu per hr.



Type V Heater

Vertical Discharge Type

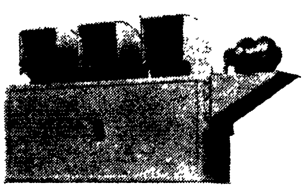
Type C Heater

Type C. Discharges air stream vertically downward. Outlet temperatures relatively low. Volume of air kept high to promote uniform heat distribution horizontally, as well as to equalize temperature differences between floor and ceiling. Four types of diffusers available.

Heating element of non-ferrous materials. Fins are copper or aluminum. Tubes are heavy copper—expanded into fins to provide a positive permanent mechanical bond—then silver soldered into cast bronze or copper headers.

Manufactured in 7 sizes. Capacities at 2 lb steam, 60 deg entering air: 32,800 to 492,000 Btu per hr.

Blower Unit Heaters



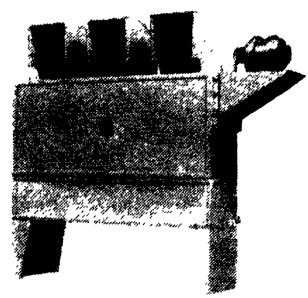
Wall type heater

Type R. Available with mixing dampers, by-pass dampers and filter sections for heating and ventilating large areas, or for heating only.

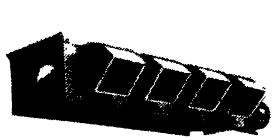
Belt-driven centrifugal type blower fans use constant speed 1750 rpm motors on 60 cycle or DC current. Fans are double width, double inlet type with forward pitched blades. Self-aligning, dust-proof ball bearings support fan shaft.

Heating element is replaceable tube type. Brass clamping nuts securely fasten seamless drawn copper tubes into semi-steel cast headers. All essential parts readily accessible, and can be removed through either end of heater casing.

Working pressure: up to 150 psi steam. Capacities at 2 lb steam, 60 deg entering air: 105,000 to 822,500 Btu per hr.



Floor type heater with mixing damper. Also available with elongated nozzles.



Ceiling type heater

DUNHAM HEATING SPECIALTIES

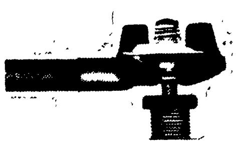
Radiator Traps

For all types of low pressure steam heating systems up to 25 lbs gage. Dunham design assures freedom from clogging; thorough draining of radiators by gravity; minimum wear on parts.

Body and cover of cast bronze; disc of Monel metal. Round, slightly raised valve seat minimizes depositing of incrustrants.

Size	Pattern	Tapping	*Cap. Sq Ft EDR
1 E	AP SW RH LH VST	½″	200
2 E	AP SW	¾″	400
3 C	AP	1″	700

* Ratings are based on ¼ lb condensation per sq ft of EDR per hour and a 1½ lb pressure differential.



Radiator Trap, No. 1E, 2E, 3C

Thermostatic Steam Traps

Operate in response to pressure caused by partial vaporization of liquid within thermostatic disc. Permanent adjustment for correct operation built into trap at factory. Body, cover, union nut and nipple are brass.

Valve and seat are special heat-treated stainless steel. Spherical valve is swiveled to insure tight seating without causing localized stresses on disc. Exceptionally large valve opening permits ready passage of water or dirt.

For 5 to 100 pounds working pressure— Made in ½ in., ¾ in. and 1 in. sizes, for capacities from 125 to 1780 lb Cond./hr.

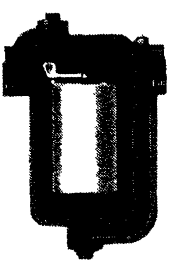
Thermostatic Steam Trap Type TH-IA

Float and Thermostatic Traps

Especially adaptable for service as a drip-trap for dripping ends of steam mains, unit heaters and all types of heat exchangers. Capable of handling large amounts of air and water. Float is made of copper. Float valve and seat are Monel metal. Body can be removed for inspection of working parts without disturbing piping connections. Operating pressures up to 15 psi gage. Sizes: ¾ in., 1 in., 1¼ in., 1½ in., and 2 in.; respective capacities: 800, 2000, 4800, 9600 to 20,000 EDR at ¼ lb condensate per sq ft per hr with 2 lb pressure differtial. Series 31 closed float traps available in same capacities.

Float and Thermostatic Trap, Series 30

Inverted Bucket Traps



Inverted Bucket Trap

For venting air and draining water from high pressure steam lines, heat exchangers, sterilizers and processing equipment. Traps increase heat transmission efficiency and hold equipment capacity to a maximum.

Body and cover of high tensile iron castings. Valve and seat (renewable and interchangeable) are special hardened, corrosion-resistant steel. Sheet copper bucket. Capacities range from 400 lbs to 5600 lbs of condensate per hour.

Operating Pressures: 20 to 150 lbs gage.

Sizes: Type OBS ½ and ¾ in.
Type OB ½ to 1¼ in. inclusive.

DUNHAM HEATING SPECIALTIES

Valves

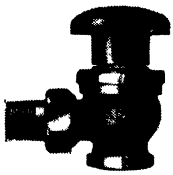
"Oriflex" Valves

Self-contained, adjustable orifice valves, packless bellows-type for proportioning steam supply to each radiator.

Valve need not be disconnected to make adjustments. Merely remove handle, insert key on adjustment stem and turn orifice to desired setting Sizes: ½ and ¾ in.

"Oriflex" Valve, Type 175

Packless Valves

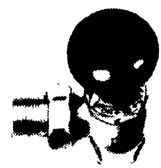


Packless Valve, Type 1140

For all types of low pressure steam heating systems. Body and bonnet are brass castings. Heavily constructed brass union nuts and nipples. Phosphor-bronze bellows sealing member gives maximum resilience and wear; prevents leakage; keeps steam, water and dirt from clogging and corroding spindle nut and screw. Non-rising stem. Heat resistant composition handle. Sizes: ½ to 1½ in. inclusive.

Spring Packed Valves

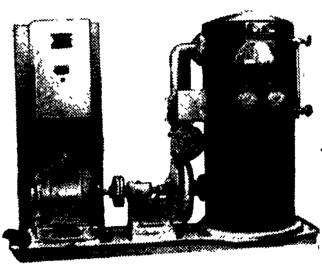
For low pressure steam heating services. Body is brass casting, rough finish. Heavily constructed brass union nuts and nipples. Non-rising stem. Valve opens fully on less than one turn—dial shows direction and amount of opening. Heavy spring keeps constant pressure on special graphite asbestos composition packing to maintain tight seal around valve stem. Sizes: ½ to 2 in. inclusive.



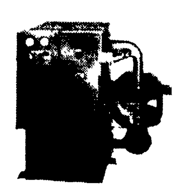
Spring Packed Radiator Valves, Type 740

Dunham Pumps

Vacuum Pumps



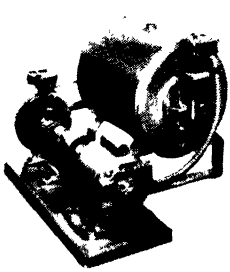
Type VR, Model B Pump, for capacities from 25,000 to 65,000 EDR.



Type VR, Model C Pump, for capacities up to 20,000 EDR.

Furnished as single (Type VR) or duplex (Type VRD) units. Separate accumulator tank available for handling low returns. Each pump has own control panel wired through to motor. Pumps efficiently maintain desired vacuum range on return lines and deliver condensate direct to low pressure boiler. Selector switch on panel permits pump to operate on fully automatic, continuous, or float control.

Condensation Pumps

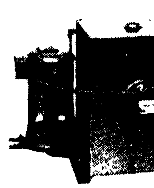


Single Condensation Pump, Type CH (CHH)

Complete, compact assemblies for automatically returning condensate to boilers for gravity heating systems or steam process equipment. Also manufactured as Boiler Feed Pump.

Type CHV—Single and Duplex—3450 rpm for pressures up to 20 lbs. Capacities 2,000 to 10,000 EDR. Low priced pump.

Type CH—Model D, Single and Duplex—1750 rpm for pressures 50 lbs and lower. Capacities 2,000 to 50,000 EDR. Rugged pump for heavy duty applications.



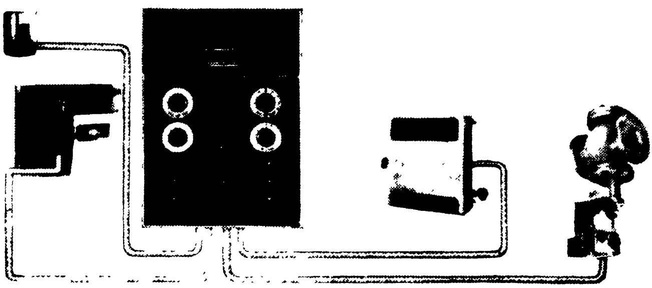
Single Condensation Pump, Type CHV

Type CHH—Model D, Single and Duplex—3450 rpm for higher pressures up to 70 lbs. Capacities 2,000 to 50,000 EDR. Rugged pump for heavy duty applications.

DUNHAM HEATING SYSTEMS

Vari-Vac Differential

Cuts fuel costs up to 40 per cent. Dunham Vari-Vac Heating Systems save up to 40 per cent on fuel costs . . . because Vari-Vac automatically provides the precise amount of heat required by utilizing a *continuous flow of steam at temperatures that vary with the weather.* For every size or type of building regardless of location or climatic conditions.



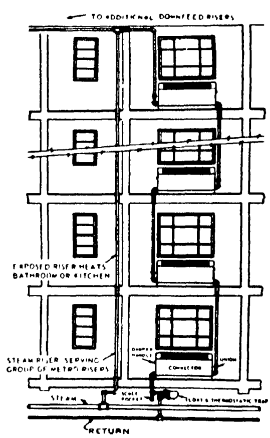
"Supreme" job, Type RSTT Control Equipment.

Metro "Single Riser"

Costs less to install, to maintain, to operate—

Provides an uninterrupted path for flow of steam from top to bottom of building. Continuous pipe runs down through overlying rooms . . . is offset in each room into a convector or baseboard radiator.

1. Eliminates all radiator branches.
2. Eliminates all traps and valves in occupied quarters.
3. Eliminates costly furring of masonry walls.
4. Eliminates expansion joints.
5. Eliminates settings for temporary heat.



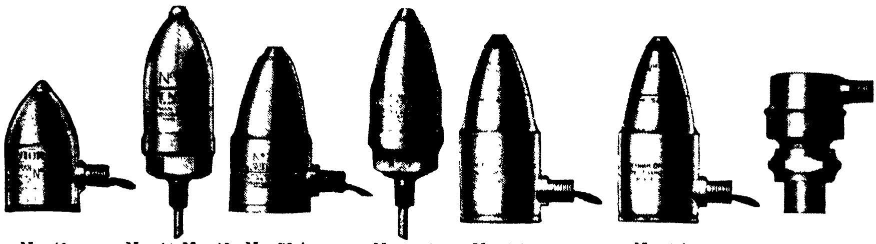
For complete catalog data on Dunham Products and Equipment write C. A. Dunham Company, 400 W. Madison St., Chicago 6, Illinois.

Hoffman Specialty Company

General Office and Factory
1001 York Street, **Indianapolis 7, Ind.**
Sales Representatives in Principal Cities

Manufacturers of Radiator Air Valves, Quick Vents and Air Eliminators for all types of Steam and Vacuum Heating Systems—Steam Traps of all kinds—Radiator Supply Valves—Vacuum and Condensation Pumps—and Hot Water Automatic Heat Control Systems.

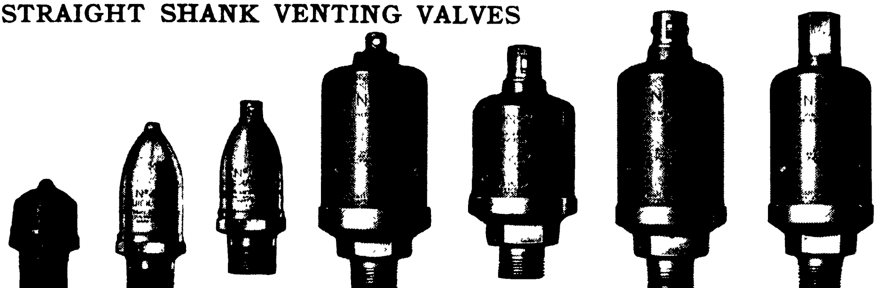
RADIATOR AIR VALVES FOR STEAM AND VACUUM SYSTEMS



No. 40 *No. 41 No. 43 and No 45* *No 70A* *No. 71A* *No. 2A* *No. 1A* *No. 3*

No. 40 Steam—Hoffman patented tongue syphon—⅛ in. connection—fixed port.
No. 41, 43 and 45 Steam—Straight shank for convectors—telescopic syphon—⅛ in., ¼ in., and ¾ in. male, ½ in. female connections.
No. 70A Steam—Tongue Syphon—Non-adjustable, single port—⅛ in. connection.
No. 71A Steam—Straight shank for convectors—telescopic syphon ⅛ in. connection.
No. 1A Steam—Tongue Syphon—ADJUSTABLE air opening—⅛ in. connection.
No. 2A VACUUM—Tongue Syphon—ADJUSTABLE air opening—⅛ in. connection.
No. 3 Steam—For Airline or PAUL systems—⅛ x ¼ in. conn.—union tailpiece.

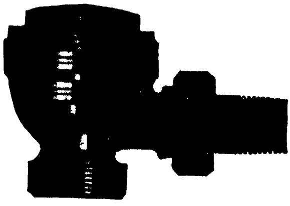
STRAIGHT SHANK VENTING VALVES



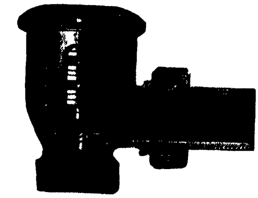
No. 4 *No. 4A* *No. 16A* *No. 74* *No 75* *No. 76* *No. 79*

No. 4 Steam Mains—Will not close against water—¾ in. male, ½ in. female connection.
No. 4A Steam Mains—Float closes against water—¾ in. male, ½ in. female connection.
No. 16A VACUUM Mains—Float closes against water—¾ in. male, ½ in. female connection.
No. 75 Steam Mains—Medium systems—has float—¾ in. male, ½ in. female connection.
No. 76 VACUUM Mains—Medium systems—has float—¾ in. male, ½ in. female connection.
No. 75A Steam Mains—Large systems at low pressure—has float—¾ in. male, ½ in. female connection.
No. 76A VACUUM Mains—Large systems low pressure—has float—¾ in. male, ½ in. female connection.
No. 74 Unit Heater Vent Valve—Operates 0 to 35 lb. Vents air at any pressure and whether rising or falling—can be used on steam mains—¾ in. male, ½ in. female connection.
No. 79 Hot Water Vent Valve—Positively removes air from piping of any hot water system. Max. pressure 75 lb—¾ in. male, ½ in. female connection at base and tapped at top for ¼ in. pipe connection.

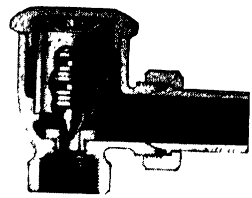
LOW, MEDIUM AND HIGH PRESSURE THERMOSTATIC TRAPS



Low Pressure



Medium Pressure



High Pressure

Low pressure traps have brass bodies, caps and union nut and tailpiece. 17C is made in Angle, Swivel and Vertical patterns. 8C is made in Angle and Straightway patterns. 9C is made in Angle pattern only. Thermostat and seat both renewable.

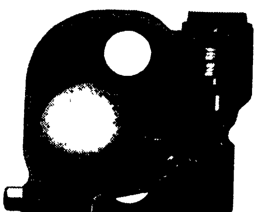
No. 17C Capacity 200 sq ft EDR 15 lb pressure ½ in. connection
No. 8C Capacity 400 sq ft EDR 25 lb pressure ¾ in. connection
No. 9C Capacity 700 sq ft EDR 25 lb pressure 1 in. connection

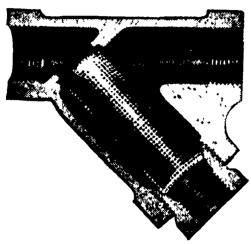
Medium Pressure Nos. 8 & 9 and High Pressure Nos. 8H & 9H have all bronze bodies and caps with union nut and tailpiece. Thermostats are 6 diaphragms of special non-corrosive metal. Thermostats and seats are renewable. ½ in. sizes are furnished in Angle, R.H., L.H. and Straightway patterns, others in Angle only. Medium Press. 50 lb limit. High Press. 125 lb.

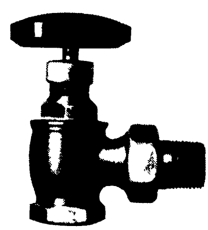
Capacities—Lb Condensate per Hour—Working Pressure—Lb per Sq In. Gage

Traps		5	15	25	50	Traps		25	50	100	125
8	⅜"	100	180	235	400	8H	⅜"	235	400	550	590
8	½"	125	225	300	490	8H	½"	300	490	650	720
9	¾"	225	350	450	650	9H	¾"	450	650	875	950
9	1"	325	500	625	850	9H	1"	625	850	1125	1250

FLOAT TRAPS, DIRT STRAINERS AND SUPPLY VALVES





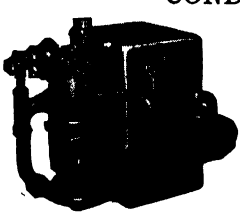


50 Series Float and Therm. Traps are available in large capacities and four pressures, 15, 30, 60 and 125 lb. Used for venting and draining risers, steam mains, unit heaters, blast coils, etc. 50 Series Traps are made for easy servicing with all working parts mounted on cover. Remove four bolts to expose all parts. Pipe sizes are from ¾ in. to 2 in.

Radiator Supply Valves are made in sizes from ½ to 2 in. in Angle, R.H., L.H., and Straightway patterns. Brass bodies, union nut and tailpiece. Nos. 180 and 185 have non-rising handles. Both are packless. No. 186 is especially suited to vacuum systems.

Hoffman Dirt Strainers are self-cleaning Y type. Brass strainer cylinders and cast iron body. Sizes ½ to 2 in. for 125 lb pressure. Should be used in line ahead of all float and thermostatic traps. Also available with monel metal strainers.

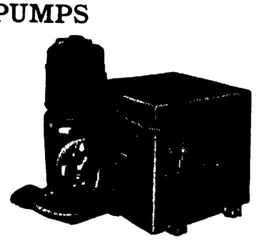
CONDENSATION AND VACUUM PUMPS



Hoffman pumps are available in varying capacities, D.C. and A.C. current, single, two, or three phase and in pressures up to 200 lbs.

Vacuum Pumps Single and Duplex Units for different Capacities and Sizes

Condensation Pump Single and Duplex Units for different Capacities and Sizes



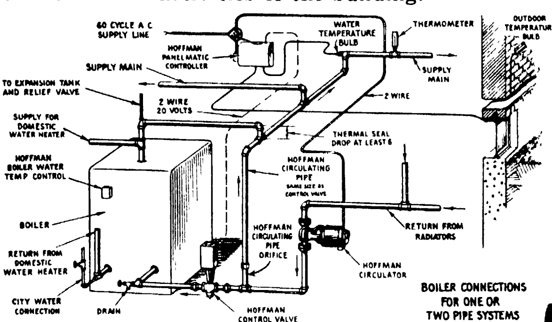
HOFFMAN "PANELMATIC" HOT WATER SYSTEM CONTROLS

Designed for all Types of Radiant Heating

The Hoffman method vastly improves the ordinary forced hot water system by the application of *Continuous Circulation.* This method permits a smoothly modulated regulation of the heat supply. The Control Valve closes and the circulating stream by-passes the boiler as long as the heat requirements are satisfied.

When the circulating water begins to lose heat, the Control Valve is slowly opened by the Controller, permitting hot boiler water to enter the system. Thus, the delicate precision of the Hoffman Controller smoothly varies the temperature of the *continuously circulating* water so that the heat supply is always equalized with the heat loss and room temperature remains constant throughout all changes in the weather.

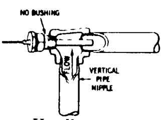
There is no type of building to which Hoffman Hot Water Controlled Heat cannot be applied. The system permits *zoning* of apartments, institutions, large residences and factories, thereby assuring a distribution of heat in direct relation to either personal temperature preference or to the functional activities of the building.



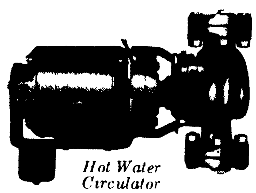
Boiler Connections For One or Two Pipe Systems

Outdoor Temp. Bulb

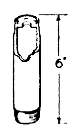
Control Valve



Hot Water Temp. Bulb

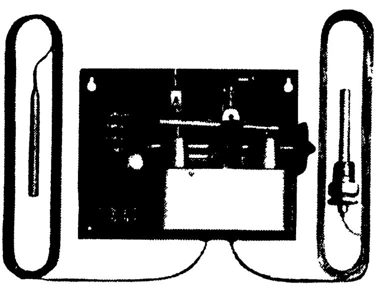


Hot Water Circulator



Orifice

HEATING COMFORT IS ACHIEVED BY THESE SIX HOFFMAN SPECIALTIES



The Panelmatic Controller

The brain of the Hoffman system. It automatically maintains a constant comfort condition regardless of the outdoor temperature. Its accurate balancing mechanism electrically opens or closes the Hoffman Hot Water Control Valve as required.

Outdoor Temperature Anticipating Bulb

This Bulb transmits changes in the outdoor temperature to the balancing mechanism of the Panelmatic Controller.

Hot Water Temperature Bulb

Accurately relays temperature changes of the supply main water to the balancing mechanism of the Hoffman Comfort Controller.

Circulating Pipe Orifice

A calibrated orifice used to balance the circuits through the boiler and through the Hoffman circulating pipe.

Hot Water Control Valve

Opens only sufficiently to supply the correct amount of water to maintain the proper water temperature being circulated through the system.

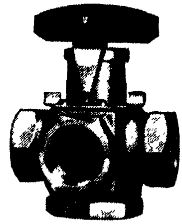
Thermometer. Should be installed about 6 in. from submerged Water Temp. Bulb.

Hoffman Hot Water Circulator

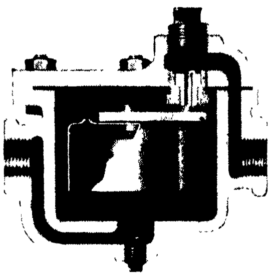
A centrifugal pump of prescribed capacity and low in power consumption. Usually installed in the return main and continuously circulates the water through the system.

HOFFMAN PANEL-FLO VALVES

for adjusting the flow of water through individual Panel Heating Coils



Gives a wide range adjustment of water flow to each coil without affecting the heat output of the other coils—*can cut flow through coil as much as 60 per cent.* . . WITH NO APPRECIABLE CHANGE IN CIRCULATING PUMP HEAD. Has indicating Dial for accurate setting of each coil—or adjusting any coil without affecting the other coils. 3/4 in. pipe connection.

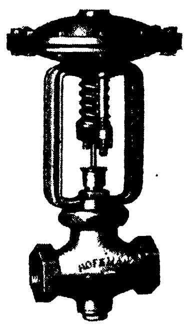


Series 600

HOFFMAN INVERTED BUCKET TRAPS

Working Pressures to 200 lbs

Simple mechanism assures high operating efficiency. Features include—straight-through pipe connection; simple seat adjustment; all working parts connected to bonnet and easily removed with it; stainless steel seat and pin.

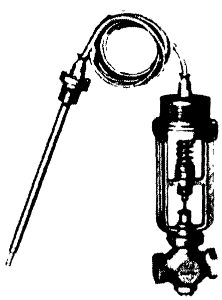


Series 710

HOFFMAN PRESSURE REDUCING VALVES

Series 710 Max. Press. 250 psi., Low Press. 5 to 80 psi.
Series 700 Max. Press. 200 psi., Low Press. 1 to 25 psi.

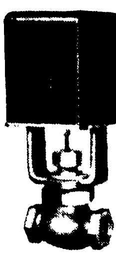
Choice of types specifically engineered for your needs—continuous service, or, for tight sealing against low pressure side when there is no demand—and with typical Hoffman features of design.



Series 1100

HOFFMAN TEMPERATURE REGULATORS

Series 1100—for steam pressure to 150 lb per sq in. Standard temperature range—80 F to 250 F. Other ranges available on special order. Self-motivated. Hydraulically formed bellows of selected material. Rugged construction. For water heaters, convectors, fuel oil preheaters and other similar applications.



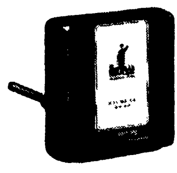
HOFFMAN "ZONET" ZONE CONTROL

Series 1200—Zonet systems or packages for Steam and Hot Water consist of heat anticipating Room Thermostat, motor operated Globe Type Valve, Transformer, Fuse Block and fuse, and in the case of steam zoning, a Vacuum Breaker. The motor operated, two position, single seated, packed globe type Valve (illustrated) is available in sizes from 1/2 in. to 6 in. Maximum service pressure: steam 100 lbs per sq in., water 150 lbs per sq in.

HOFFMAN ELECTRIC CONTROLS

Hoffman Room Thermostat

Hoffman offers a comprehensive selection of fine quality electric controls for steam, hot water and warm air heating systems. These controls are especially suited for use with other famous Hoffman heating system specialties and provide one source of supply and one responsibility for satisfactory performance. Illustrated are two popular control units, typical of the complete Hoffman line.



Hot Water Limit Control Immersion Type

ILLINOIS ENGINEERING COMPANY

General Offices
and Factory:
Chicago 8, Ill.

Representatives
In Principal Cities

ILLINOIS HEATING SYSTEMS

ILLINOIS HEATING SYSTEMS, with or without control, in five general types, cover the complete range of vacuum and vapor heating requirements; each admits of modification to meet any special condition in either. These systems include:

GENERATION Control Systems—Type C

For independent boiler installations affording maximum fuel economy.

CONTINUOUS FLOW Control Systems—Types A &M

For installations which require zoned control, or where steam is used for other than heating and domestic hot water service, or where steam is supplied by Central Station Service, Pneumatic or manual operation.

CYCLING FLOW Control Systems—Type E

For automatic or manual control of steam flow and pressures not only in vacuum systems but in one-pipe or two-pipe gravity installations as well.

VACUUM SYSTEMS

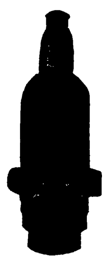
A two-pipe steam circulating system in which a standard vacuum pump is used to accelerate circulation, remove air and condensate from the system and return the condensate to the boiler.

Suitable for any type of building, for industrial plants, or for groups of buildings heated from a central plant.

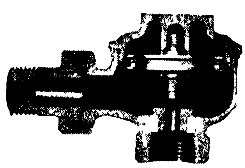
VAPOR SYSTEMS

A two-pipe system circulating steam at low positive pressures without any pump or mechanical vacuum producer. Recommended for Residence, Small Apartment, and similar service. Gives rapid, flexible steam circulation.

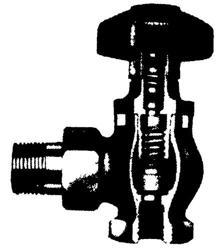
Series G



Selective Controller



Series G



Supply Valve

Illinois Float and Thermostatic Traps—Unsurpassed for draining ventilating units, unit heaters, and for dripping mains and risers—wherever it is desirable quickly to vent air from the main as well as handle the water of condensation in quantity, whether hot or cold.

Illinois Selective Pressure Control Systems—An entirely new and unique method of Steam Circulation Control . . . Heating Systems that set new standards in comfort, economy, simplicity and convenience of operation. Each system individually engineered to meet exact requirements. Recorded fuel savings, without sacrifice of comfort, warrant your investigation.

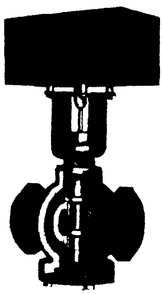
Illinois Radiator Supply Valve—Quick-opening, packless. Steam tight on 50 lb pressure. Large diameter of thread spool and machine cut threads make valve operation easy. Furnished in a complete line of sizes and patterns.

Illinois Thermo Radiator Traps—Illinois Thermo Radiator Traps for vacuum, vapor and low pressure heating systems. Has cone type valve. Flushes thoroughly and seats perfectly at all times. Valve and seat are of hardened steel alloy. The duplex diaphragm is of special phosphor bronze. Scientific design and rugged construction assure flexibility and long life.

ILLINOIS ENGINEERING COMPANY

General Offices
and Factory:
Chicago 8, Ill.

**Representatives
In Principal Cities**



Type E3

Illinois Motorized Valves (on and off)—Type E3—For automatic control of steam temperatures and pressures to prevent overheating and conserve steam; to control fluid levels; and to regulate flow in hot water heating systems. May be operated by any automatic contact device or by manual switches.

Furnished in three types.

Pressure Regulating Valve, Semi-Steel Bodies, Bronze Trim—Fig. 121—Furnished in either single seat or double seat type as service requires, for the control of steam, air or gas. Control spring is completely enclosed, protecting it from dirt and rust. Valves are furnished with proper size diaphragm and proper length spring to give satisfactory service under all operating conditions. Furnished also in weight loaded type, Fig. 71.

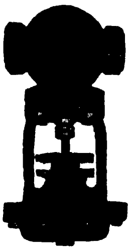


Fig. 121

Non-return Valves—Fig. 260—Placed between boiler and header to prevent return of steam to boiler. Sensitive in operation. Extra heavy semi-steel bodies with bronze trim for 250 lbs steam working pressure. Bronze dash pot and water sealed pistons prevent valve sticking. Globe and angle patterns from 4 in. to 12 in.

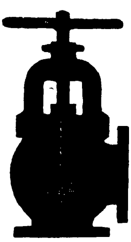


Fig. 260

Illinois Thermostatic Traps for High Pressures—Series HG—Maximum working pressure 150 pounds. Used where neat appearance and compactness are desirable, as for trapping sterilizers or water stills in hospitals; steam jacketed kettles, coffee urns, warming tables and for process work. Also used extensively for air vents on blast type drying heaters. Multi-diaphragm of phosphor bronze. Heavy duty bronze body. Made in three sizes.

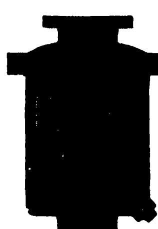
These traps are also furnished for medium pressures.

Steam and Oil Separators—Vertical Steam Separators—Eclipse steam separators are made in both horizontal and vertical type, standard or extra heavy.

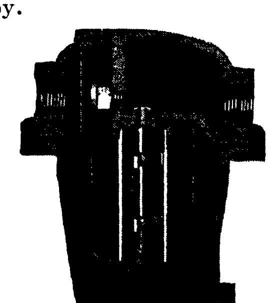
Eclipse oil separators are furnished in the horizontal type and have a removable baffle plate to facilitate cleaning of baffle and keeping the separator's efficiency the highest point.

Illinois Steam Trap—Series 30—Valve and stem are separate from the bucket and operated only by the bucket at extreme top and bottom of travel—result—*valve is always either full open or tight closed.* Provided with continuous thermostatic air vent. No wire drawing or cutting of valve and seat which are of hardened steel alloy.

Series HG



Vertical Steam Separators



Series 30

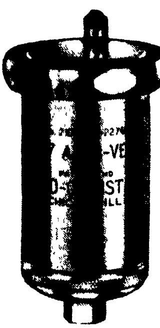
Maid-O'-Mist, Inc.

3217 No. Pulaski Road
Chicago 41, Ill.
Manufacturers of
Automatic Air Valves for
Hot Water Heating Systems

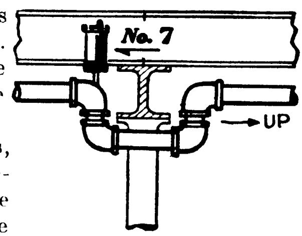
Products: Automatic air valves for hot water heating systems; automatic humidifiers for steam or hot water; automatic humidifiers for warm air furnaces; steam boiler water line controls; water line float control valves; liquid gas and air strainers.

AUTO-VENT

Air Eliminators



No. 7 Auto-Vent



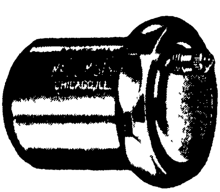
Venting trapped mains and circulating lines

Maid-O'-Mist Auto-Vents permit automatic venting of the air which causes circulation trouble and heating waste. More and more, contractors who value satisfied customers are installing these trouble stoppers.

No. 7 Auto-Vent. For mains, pipe lines, unit heaters, convectors, coils, etc. Designed for vertical mounting only. Size 4⅜ in. x 2¼ in. with ⅛ in. I.P. female connection. Made of brass and equipped with a self-closing, float operated valve. All working parts, including valve and copper float, mounted on a removable bonnet for quick servicing or replacement. Valve is equipped with a Monel metal spring and a Neoprene valve seat which is unaffected by high temperatures, oil, anti-freeze, etc. For pressure not exceeding 75 lbs.

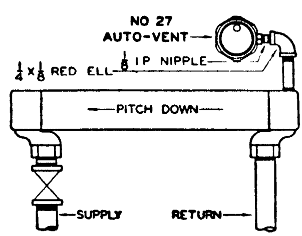
No. 77 Auto-Vent. Identical to No. 7 Auto-Vent, except for a ⅛ in., I.P. side opening to permit its use on overhead pipe lines, coils, etc., where head room is factor. Ideally suited for Diesel engine and cooling manifold use, where vibration demands secure mounting. No air chamber required.

Auxiliary Equipment: No. 7X—bonnet assembly for No. 7 and 77; No. 777—self-closing valve core; No. 7A—connector for safe waste.



No. 27 Auto-Vent

No. 27
Auto-Vent



For venting convector radiators

For convectors and free standing radiation. Designed for horizontal mounting only. Size 3 in. x 2½ in. with ⅛ I.P. female connection. Construction and internal working parts are same as in No. 7 Auto-Vent, but for horizontal mounting. Made of non-ferrous metals and designed for pressure not over 50 lbs. No air chamber required.

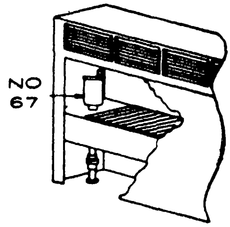
Auxiliary Equipment: No. 27X—bonnet assembly for No. 27; No. 7A—connector for safe waste.

Write for price sheet and descriptive literature.

No. 67
Auto-Vent

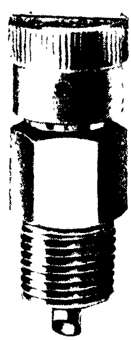
No. 67 Auto-Vent

For convectors and baseboard radiators. Designed for vertical mounting only. Size 3$\frac{3}{16}$ in. x 1$\frac{1}{2}$ in. with a $\frac{1}{8}$ I.P. male connection. Designed for limited space, small, self-closing, float operated valve may be installed in trouble spots previously neglected or improperly vented. Valve is equipped with a Monel metal spring and Neoprene valve seat, which is unaffected by high temperature, oil, etc. No air chamber required. For pressures up to 30 lbs. Where a Safe Waste is needed, specify No. 7A connector fitting.

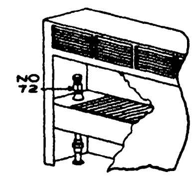


For venting convector radiators

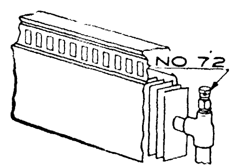
No. 72
Auto-Vent



No. 72 Auto-Vent



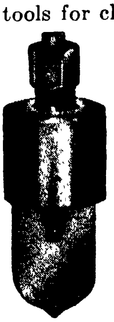
For convector radiators



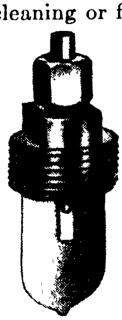
For baseboard radiation

For convectors, baseboard and free-standing radiation. Designed for both vertical and horizontal mounting. Size 1$\frac{1}{4}$ in. x $\frac{1}{2}$ in. with $\frac{1}{8}$ in. I.P. male connection. No. 72 Auto-Vent is fast venting valve of the expansion type. All expansion and contraction of the single, non-porous, composition disk is confined to four port vents. Internal siphon tube prevents water logging when installed in either vertical or horizontal position. Immediate drainage means fast disk drying, quick venting cycle.

Air in system enters valve and passes through the vent port to the atmosphere. Water following same path comes in contact with disk, which swells, closing ports. Entrapped air contracts disk and venting cycle is repeated. Cycle is continuous and no air chamber is needed. Manual venting features plus tight shut off are all controlled by valve cap adjustment. Valve cap is tamper-proof and can be removed and replaced without damage or special tools for cleaning or flushing, if needed.



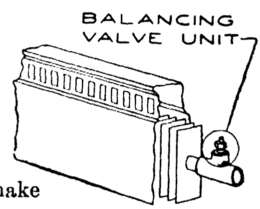
No. 14



No. 15

No. 14 and No. 15
Balancing Valve
Adapter Units

for Hot Water
Heating Systems



For baseboard radiation

Maid-O'-Mist Valve Adapter Units make any copper, bronze, or cast iron tee a balancing valve. Stocks of expensive square head cocks or valves no longer needed. These units, quickly soldered or sweat fitted into copper and bronze tees or threaded into cast iron tees, regulate hot water flow through radiators, convectors, baseboard panels, radiant coils, return mains, and branches. Can be inserted in side outlet or run of tee of same pipe size to complete either a straightway or angle balancing valve. Precision made of non-ferrous metals. Simple balancing requires only a screw driver.

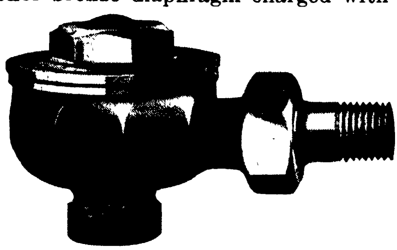
Jas. P. Marsh Corporation

Dept. 5, **Skokie, Illinois**

Branches in Principal Cities

Marsh products include: Pressure, Vacuum and Compound Gauges; Dial Thermometers; Steam Traps; Vents; Packless Radiator Valves and other heating specialties; Tri-trol Regulators; Electrimatic Refrigeration Water Regulators and Solenoid Valves.

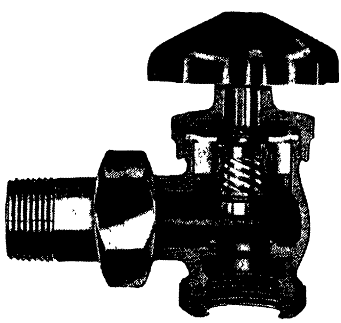
Radiator Traps—These highly efficient radiator traps are equipped with a phosphor bronze diaphragm charged with a



Thermostatic Diaphragm Radiator Trap

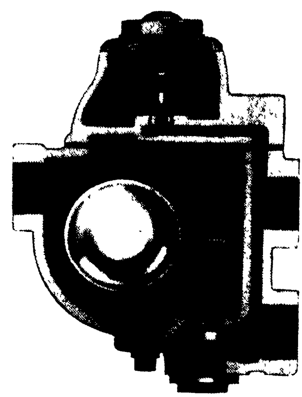
volatile fluid making them equally efficient for use on pressures below atmospheric up to 15 lbs pressure. Also other traps available up to 125 lbs pressure.

Packless Radiator Valves—The metal-to-metal seal of Marsh Radiator valves makes them truly packless. They contain no packing of any kind to wear, crack and deteriorate. Proved by many years of service. Easy to operate. Close on less than one turn. Individually tested. Adaptable for hot water as well as steam heating systems.



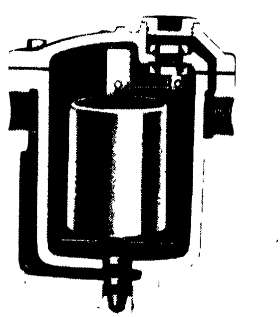
Packless Radiator Valve

Marsh Float and Thermostatic Traps—One of the many types of Marsh Heavy Duty Float and Thermostatic Traps is illustrated. These traps are designed for removal of air and condensate from steam mains, branches, or risers, unit heaters, steam coils, etc. The size and weight of the trap permits installation in the piping without any other means of support. Condensation is discharged through a float-operated valve located at the lowest point inside the trap body. Air vent is located in a by-pass in the cap or cover of the trap. Air passes through a passageway and out through the trap outlet. Construction permits removal of mechanism without disturbing the piping.



Float and Thermostatic Trap

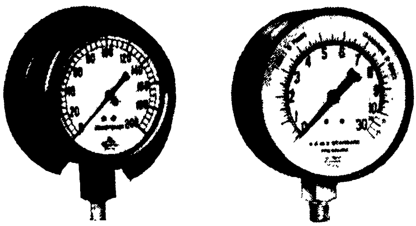
Marsh Inverted Bucket Traps are ideal for all types of hospital and kitchen equipment or similar service where a



Inverted Bucket Trap

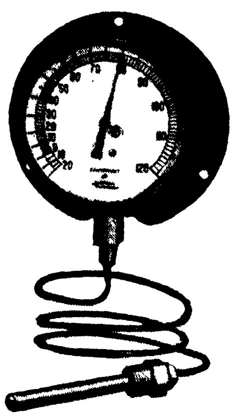
considerable volume of condensate is handled. Traps are self-venting and have large water capacity thus assuring unusually high efficiency in removing condensate, air and gases.

Marsh Pressure Gauges—The Marsh *ASME* standard, low pressure gauge will contribute to the economy and improve the operation of any type of steam boiler. It is finely built throughout and is available with the Marsh "Recalibrator" for quickly and easily resetting the hand to zero when the gauge is knocked out of adjustment.



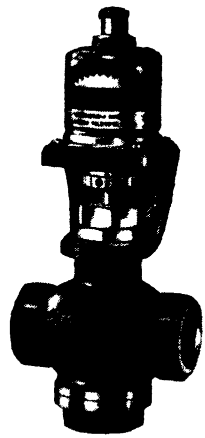
Marsh Gauges include vacuum and compound types in a wide range of designs covering all services and pressures. Over 75 years of gauge manufacturing has reached its highest achievement in the Marsh "Mastergauge" for use where high pressures and temperatures are present and where maximum stamina and accuracy are essential.

Marsh Dial Thermometers—The same basic refinements found in Marsh Gauges are found in Marsh Dial Thermometers Bourdon tube types are available in self-



contained and distant reading instruments, vapor-tension or gas filled. All ranges up to 400 F are covered. "Recalibrator" is standard in all bourdon tube types. Ask for catalog information.

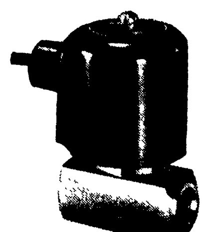
Marsh Electrimatic Refrigeration Valves—The Marsh Electrimatic line of refrigeration valves includes condenser water regulators for ammonia, Freon or methyl chloride service; temperature actuated valves, packless solenoid valves, and other related products.



Water Regulator

The popular Electrimatic Type WP direct acting piston type regulator is illustrated. All working parts are Monel and stainless steel and body is special, non-porous brass alloy. Advanced features are: stainless steel piston; heavier cushion spring; tight seal; water behind piston dampening vibration; Monel seat eliminating dezincification; sturdy, 2 ply, 300 lb-test bellows; open yoke permitting easy adjustment and rotatable to provide for mounting regulator in any position. Other types—pilot operated—available for heavy and extreme duty.

Marsh Electrimatic Solenoid Valves are made in both direct-acting and pilot-operated types in 3/16, 3/8 in. and 1/2 in. orifice sizes respectively. They are of packless, tight seating construction with impregnated coils to withstand frost and moisture.



Packless Solenoid Stop Valve

W. H. Nicholson & Company

Main Offices and Factory: 211 Oregon Street, Wilkes-Barre, Pa.

SALES REPRESENTATIVES IN U. S. A., CANADA, AND MEXICO AT:

Albany, N. Y.
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Birmingham, Ala.
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Denver, Colo.
Des Moines, Ia.
Detroit, Mich.
Evansville, Ind.
Fort Wayne, Ind.
Fort Worth, Tex.
Frederick, Md.
Greensboro, N. C.
Houston, Tex.
Indianapolis, Ind.
Kansas City, Mo.
Los Angeles, Cal.
Mexico, D.F.
Miami Springs, Fla.
Milwaukee, Wis.
Minneapolis, Minn.
Mobile, Ala.
Montreal, Que.
Needham, Mass.
New York, N. Y.
Oak Ridge, Tenn.
Omaha, Neb.
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Philadelphia, Pa.
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Ponca City, Okla.
Portland, Me.
Portland, Ore.
Richmond, Va.
Rochester, N. Y.
Salt Lake City, U.
San Francisco, Cal.
Seattle, Wash.
Spokane, Wash.
St. Louis, Mo.
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Syracuse, N. Y.
Toronto, Ont.
Tucson, Arizona
Vancouver, B. C.
Washington, D. C.
Yakima, Wash.

STEAM, WATER, AIR, OIL AND GAS SPECIALTIES

TRAPS
Thermostatic: Steam, to 225 lb.
Expansion: Steam, to 250 lb.
Weight-Operated: Steam, Air, Gasoline; to 1500 lb.
Piston-Operated: Steam, to 650 lb.
Radiator: to 15 lb.
SEPARATORS: Steam, Air, Gas to 250 lb.
STRAINERS: to 600 lb.
VALVES, Cylinder Control: Air, Gas, Oil, Steam, Water; Lever, Foot, Solenoid, Motor-Operated; to 5000 lb.
FLOATS, Welded: 2 in. to 14 in diam.; Pressures to 4800 lb.

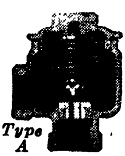
ENGINEERING BULLETINS AVAILABLE ON ALL NICHOLSON PRODUCTS

NICHOLSON INDUSTRIAL STEAM TRAPS

5 Types for Every Heat, Power and Process Application

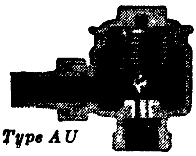
A survey of large users of Nicholson industrial steam traps showed these main reasons for the increasing standardization on Nicholson units for specified applications:

1) Operate on lowest possible temperature differential; no waterlogging.
2) Have 2 to 6 times average drainage capacity.
3) No need to change or adjust valves for varying pressures.
4) Record low for steam waste.
5) Maximum air-venting capacity.



Type A

Type AHV



Type AU

Nicholson installations have repeatedly shown production increases up to 30 per cent. Some typical unit applications: plastic moulding platens, dry kilns, steam mains, unit heaters, radiators, pipe coils, drips, hot water heaters, driers, jacketed kettles, cookers, coffee and hot water urns, dish heaters, vegetable steamers, bakers' proof boxes, steam tables, sterilizers, ironers, presses, mangles.

Types A, AHV and AU, shown on this page, are for pressures from vacuum to 200 lb. Bronze construction, aluminum painted. **BULLETIN 450**

Max. Capacity in Lbs per Hr. at Various Pressures—Types A, AHV, AU

Size, inches	1 lb.	2 lbs.	5 lbs.	10 lbs.	15 lbs.	20 lbs.	40 lbs.	50 lbs.
¼—⅜—½	865	1220	1915	2695	3290	3780	5290	5885
¾	865	1220	1915	2695	3290	3780	5290	5885
1	1020	1440	2270	3190	3900	4475	6250	6955

Size, Inches	60 lbs.	80 lbs.	100 lbs.	125 lbs.	150 lbs.	175 lbs.	200 lbs.
¼—⅜—½	6400	7310	8120	8925	9775	10,400	11,090
¾	6400	7310	8120	8925	9775	10,400	11,090
1	7590	8660	9600	10,570	11,590	12,310	13,100

See Note A p. 2

LIST PRICES—TYPES A, AHV, AU

Size, Inches	Types A and AHV			Type AU			Valve Orifice, Inches
	With Bellows of			With Bellows of			
	Bronze	Monel	Stainless	Bronze	Monel	Stainless	
¼—⅜—½	$17.60	$20.90	$24.90	$19.05	$22.40	$26.40	5/16"
¾	19.05	22.40	26.40	20.50	23.85	27.85	5/16"
1	21.30	23.65	28.65	23.45	26.80	30.80	23/64"

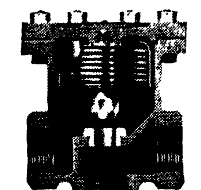
Nicholson Industrial Steam Traps, Types B and C

MAXIMUM CAPACITY IN LBS PER HR. AT VARIOUS PRESSURES

Size	1 lb.	2 lbs.	5 lbs.	10 lbs.	15 lbs.	20 lbs.	40 lbs.	50 lbs.	60 lbs.
½"	1140	1610	2530	3560	4350	4990	6980	7775	8460
¾"	1695	2385	3760	5290	6450	7420	10,350	11,540	12,560
1"—1¼"	2120	2985	4700	6610	8070	9260	12,960	14,430	15,710
1½"—2"	3200	4510	7100	9990	12,200	14,000	19,600	21,800	23,750

MAXIMUM CAPACITY IN POUNDS PER HOUR AT VARIOUS PRESSURES								CAST STEEL	
Size	80 lbs.	100 lbs.	125 lbs.	150 lbs.	175 lbs.	200 lbs.	225 lbs.	250 lbs.	300 lbs.
½"	9660	10,720	11,800	12,920	13,750	14,640	15,380	16,090	17,400
¾"	14,350	15,910	17,500	19,190	20,400	21,720	22,800	23,870	25,820
1"—1¼"	17,930	19,900	21,880	23,980	25,500	27,200	28,500	29,850	32,300
1½"—2"	27,120	30,100	33,100	36,300	38,600	41,100	43,200	45,150	48,900

Note A: Capacities shown are maximum with valve orifice wide open for 1 hr, with adjustment made for temperature and conditions of efflux. To permit intermittent discharge, and handling of peak loads at start of operations, select traps to handle not more than 50 per cent of capacities shown.

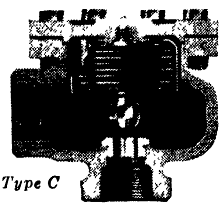


Type B

Types B and C are for pressures from vacuum to 225 lb; cast iron, aluminum painted. Type C is also furnished in cast steel, with stainless steel bellows, for pressures to 300 lb with superheat up to 500 deg total temperature. For complete details, **BULLETIN 450.**

LIST PRICES, TYPES B AND C

Size, Inches	Type B: Cast Iron With Bellows of			Type C: Cast Iron With Bellows of			Type C: Cast Steel With Bellows of		Valve Orifice Inches
	Bronze	Monel	Stainless	Bronze	Monel	Stainless	Monel	Stainless	
½"	$25.85	$29.20	$33.70	$25.85	$29.20	$33.70	$41.35	$45.45	⅜"
¾"	33.90	38.30	45.25	33.90	38.30	45.25	53.70	60.30	7/16"
1"—1¼"	40.40	43.95	50.15	40.40	43.95	50.15	63.00	69.65	½"
1½"—2"	59.80	64.45	71.10	59.80	64.45	71.10	91.00	97.65	¾"

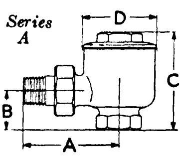


Type C

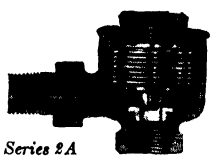
Types A, AU and C are made in angle type only, with horizontal inlet and vertical outlet. Type AU has union connection on inlet. All three types drain completely when cold and will not freeze. Type AHV is especially applicable to drainage problems where necessary or desirable to have all piping run horizontally or vertically, or close to floor, wall or pillar. Type B, made with horizontal inlet and optional horizontal or vertical outlet, offers either angle or horizontal straight-through connections. Types AHV and B traps are not freezeproof.

NICHOLSON TYPE R RADIATOR TRAPS

Series A

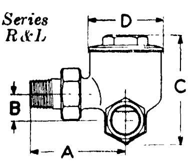


Thermostatic bellows type; feature balanced vapor-pressure principle and max. diam. valve orifice. Bronze bellows; brass body, cover, union, nut; nickel alloy valve, renewable stainless steel seat Two angle types for 200 and 400 sq ft EDR, vapor and vacuum; r and l-hand corner types for 200 sq ft. Pressure to 15 lb. **BULLETIN 744.**



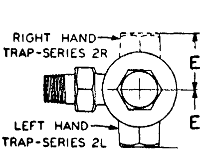
Series 2A

Series R&L



DIMENSIONS AND LIST PRICES, TYPE R TRAPS

Series	Size	List Price	A	B	C	D	E	Weight
2A	½"	$5.30	2¾"	1⅛"	2¾"	2⅛"		1⅜ lbs
2R & 2L	½	6.00	2¾	¾	3⅙	2⅛"	1⅝"	1½
4A	¾	7.95	3¼	1⁵⁄₁₆	3½	2⅛"		1¾



CAPACITIES IN SQUARE FEET EDR

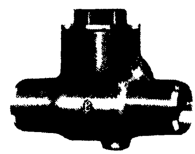
SERIES AND SIZE	PRESSURE DIFFERENTIAL—LBS. PER SQ. INCH: ¼	½	1	1½	2	5	10	15
½"-2A, 2R & 2L	85	120	165	200	235	370	530	640
¾"-4A	165	230	330	400	465	730	1050	1300

Note B: Ratings are in accordance with recommended standards of *Steam Heat Equip. Mfr's Assn.* Select trap directly from table for the lowest pressure differential that may exist in the system.

Ohio Brass Company

MANSFIELD, OHIO

EQUA-TEMP **Versatile New Radiant Heat Valve**



Provides balancing, tight shut-off, venting, draining and thermometer well all in one valve.

1. **ADJUSTABLE FLOW**—Easy adjustment of flow for balancing. Full open to tight shut-off possible.

2. **ABSOLUTE TIGHT SHUT-OFF**—Valve can be shut off completely by a 90 deg turn of stem. Synthetic rubber "O" ring on disc insures a leaktight closure.

3. **BUILT-IN THERMOMETER WELL**—O-B EQUATEMP has a thermometer well drilled into the stem, providing a handy means for accurate balancing.

4. **TAMPERPROOF CONTROL**—EQUATEMP settings require three separate Allen-type wrenches* (not normally available in the home). Same wrenches fit all size valves.

5. **VARIABLE VENTING OR DRAINING**—For convenience EQUATEMP can be installed in upright or inverted position. Top and bottom drain plugs allow venting or draining in either floor or ceiling installation.

* Handy key furnished with each 24 valves.

VENTING OR DRAINING IN UPRIGHT POSITION

Figure 1. Automatic vent installed in top drain. Figure 2. Bottom drain plug removed for draining of system. Valve may be operated with key from above.

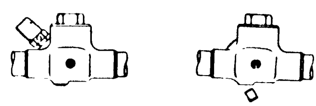


Figure 1 *Figure 2*

VENTING OR DRAINING IN INVERTED POSITION

Figure 3. Automatic vent installed in bottom drain. Figure 4. Top drain plug removed for draining of system. Valve may be operated with key from below.

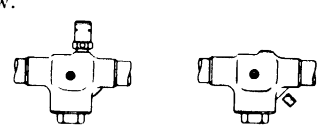
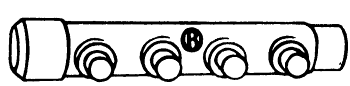
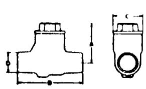


Figure 3 *Figure 4*

6. **EQUATEMP MANIFOLD FITTING**—Available with three or four outlets. May be combined to form manifolds of any desired number of valves.

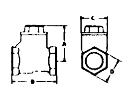


ROUGHING-IN DIMENSIONS



COPPER-TO-COPPER No. 37

Sizes Available	A	B	C	D
3/8	1 11/16	2 3/8	1 5/16	21/32
1/2	1 25/32	2 11/16	1 1/4	25/32
3/4	2 1/32	3 3/8	1 1/2	1 1/16



SCREW TYPE No. 38

Sizes Available	A	B	C	D
3/4	2 1/32	2 5/8	1 1/2	1 1/2
1	2 11/32	3	1 5/8	1 13/16
1 1/4	2 19/32	3 5/8	2	2 1/8
1 1/2	2 27/32	3 7/8	2 1/4	2 15/32

For additional information, please write to Ohio Brass Company, Mansfield (1), Ohio, for EQUATEMP Folder.

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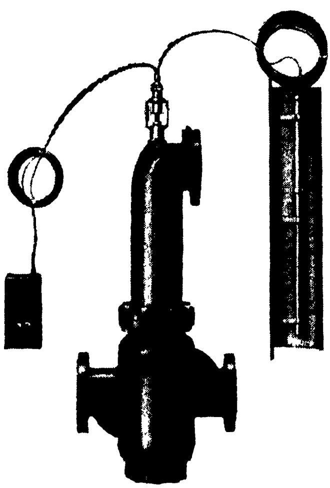
Empire State Bldg., **New York 1, N. Y.**

REPRESENTATIVES IN PRINCIPAL CITIES, FACTORY AT BETHLEHEM, PA.



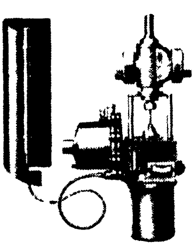
Sarcotherm Control for Hot Water and Radiant Heating System.

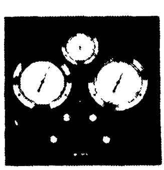
Fully automatic and completely integrated control systems for any type of heating, direct by outside temperature.

Hot Water and Radiant Heating

Sarcotherm provides a carefully engineered control system for radiant or panel heating which is fully modulating, and allows continuous circulation. The customary lag of conventional room thermostats is eliminated with the comfort control "Thermoray" sensitive to both convection and radiation.

For Forced Hot Water Heating System for garden apartments, housing developments, hospitals, and institutional buildings, Sarcotherm provides a completely zoned control system, including automatic night set-back and automatic morning pick-up.





Type "W" Sarcostat Control for Steam Heating System.

Steam Heating Systems

The Type "W" Sarcostat Control system automatically proportions the amount of heat supplied to the actual need for any given weather condition. The control is fully modulating, and complete program control panels are provided to meet any control cycle required.

Engineering Service

Consulting Engineers and Contractors are invited to consult with our Engineering Staff on any proposed control system. There is no obligation.

Sarco Company, Inc.

Empire State Bldg., **New York 1, N. Y.**
Branches in Principal Cities
SARCO CANADA LIMITED, 611 GERRARD ST., E., TORONTO 8, ONT.

PRODUCTS—A complete line of Specialties for Steam and forced hot water Heating Systems, and automatic control for same, combined with a competent engineering service to architects and heating engineers to assist them in providing modern heating.

SARCO RADIATOR TRAPS

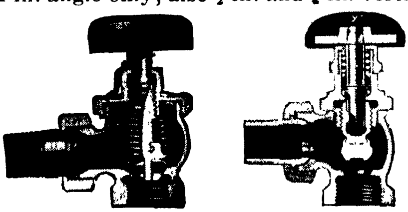
Radiator Trap, Type H

Type H is the standard radiator trap for vapor and vacuum systems. It is equipped with the well known Sarco heavy wall bellows, drawn from flat blanks and helically corrugated in our own plant. It operates noiselessly and positively at pressures from highest vacuum to 25 psi.

Body and cap are of brass, standard brass finish; self-aligning valve head and renewable seat of hard bronze; union connection on inlet.

In ½ in. size available in angle, straight or offset patterns, ¾ in. angle and straight, 1 in. angle only; also ½ in. and ¾ in. vertical. *Catalog HV-150.*

SARCO RADIATOR VALVES

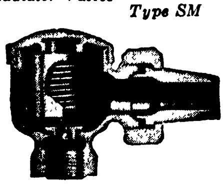


Radiator Valves
Type 46 *Type SM*

Sarco offers two types of valves; bellows packless type 43 wherein the valve stem is sealed by a standard Sarco bellows, positively preventing air leakage into the heating system; also "spring-packless" type SM. Both can be furnished with the modulating feature, including proportioning disc and indicating dial.

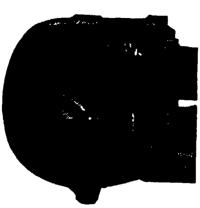
Valves are made in angle and straightway patterns, wheel handles or lock shield. Also available for hot water systems.

Bodies of all valves are brass, standard brass finish; outlet fitted with union connection; sizes ½ in. to 1½ in. *Catalog No. HV-150.*



N-100 Medium Pressure Trap

SARCO N-100 TRAPS



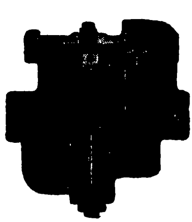
Float-Thermostatic Trap

Similar in style to Sarco Radiator Trap, the N-100 Thermostatic Trap is suitable for operation at pressures up to 100 psi. Has full length element protecting shield to prevent abrasive action on bellows. Shield also protects element against damage if removed while hot. Stainless steel renewable valve head and seat. Sizes ⅜″ to 1″. Also S-65 for pressures to 65 psi. *Catalog HV-190A.*

SARCO FLOAT-THERMOSTATIC TRAPS

Sarco offers a wide selection of Float-Thermostatic Traps, ¾ to 2 in. Available for pressures up to 200 psi. Traps 0–125 psi are equipped with built-in thermostatic air vents. High pressure 200 psi traps are equipped with external thermodynamic air by-pass. *Catalog HV-450A.*

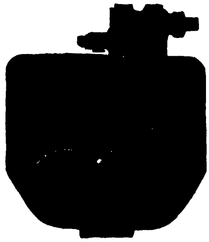
SARCO INVERTED BUCKET TRAPS



Inverted Bucket Trap

Sarco inverted Bucket Traps are also offered in a wide range of sizes and suitable for pressures up to 900 psi. Seats and valves are stainless steel and renewable. Traps are regularly furnished with built-in strainers. Built-in automatic air vents are available at extra charge. Sizes ½ to 2 in. *Catalog HV-350A.*

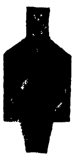
SARCO ALTERNATING RECEIVER


Alternating Receiver

A complete line of boiler return traps for vapor systems. Returns water of condensation to boiler automatically, thereby assuring positive return of water under all pressure conditions.

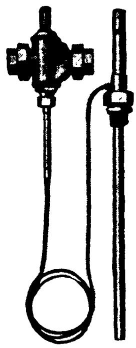
Made in four sizes up to 14,000 sq ft of radiation. *Catalog HV-165.* Same type available as a pumping trap for pressures to 100 psi.

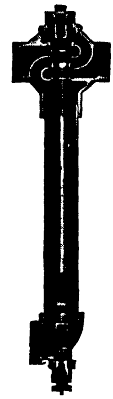
SARCO AIR ELIMINATORS



For venting air from vapor systems at one central point in the basement. Available in three sizes, for systems up to 15000 sq ft radiation. All are equipped with float valves to stop water escaping through the vent and with check valves to prevent ingress of air when system is under vacuum.

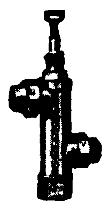
Also several types for hot water heating systems. *Catalog HV-170.*


Type TR-21 Standard for hot water storage tanks, fan units, etc.


Water Blender Type MB

SARCO SELF-CONTAINED TEMPERATURE REGULATORS

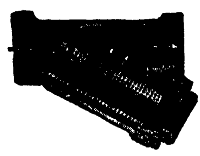
Sarco Temperature Regulators are simple, self-operated valves—the only self-contained units that use the uniform force of liquid expansion. No stuffing boxes to leak, no auxiliary "power" required; all moving parts are *inside* the equipment. Here again—a type and size for every purpose—for steam, gas, oil, water or brine for temperatures ranging from 0 to 300° F. *Catalog HV-600.*


Water Blender Type DB

SARCO WATER BLENDERS AND TEMPERING VALVES

For mixing hot and cold water to deliver automatically water at any desired temperature. Two models are available, type MB for showers, wash basins, etc., and type DB, a tempering valve for use with submerged heating coils or tankless heaters. *Catalog HV-800.*

SELF-CLEANING STRAINERS



For use in pipe lines carrying brine, steam, oil, gas, water, ammonia or air. Have large free screening area with minimum resistance to flow. Steam or air strainers can be cleaned by blowing through without disassembling. Made in cast iron, bronze or cast steel for pressures up to 600 psi, with brass, iron or monel screens. Available in sizes ¼ to 8 in. *Catalog No. HV-1200.*

Strong, Carlisle & Hammond Company

Cleveland, Ohio

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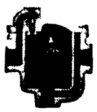

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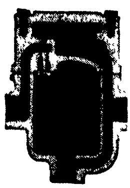

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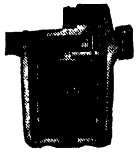

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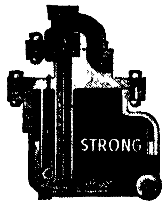

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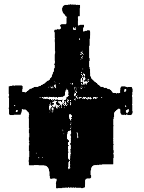

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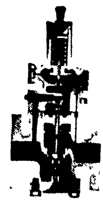

7

1. **Standard In-Line Traps** (Semi-Steel). Inverted bucket type. Hi-Cap design. Parts on cover for easy removal without breaking pipe connections. All stainless parts.

Trap No.	Pipe Size (inches)	Continuous at psi	Capacity lbs/hr	Weight lbs	List Price
070	½, ¾	125	490	3¾	$7.00
170	½, ¾	125	955	7	9.00
271	¾, 1	125	1230	7½	13.50

2. **"Y" Type Strainers** (Semi-Steel). Features perforated Monel screen and new V-shaped gripping lugs. 250 psi 450 deg steam. 400 psi cold, non-shock.

Pipe Size (inches)		Weight lbs	List Price
¼, ⅜	Sizes ¼-in., ⅜-in. fur-	1	$1.90
½	nished with 60 x 50	2¼	2.25
¾	mesh Monel cloth	3¾	2.70
1	screens. Sizes ½-in. to	5½	3.25
1¼	3-in. standard with .027	8¼	4.10
1½	perforated Monel.	11½	5.25
2		17	7.75
2½		22	18.00
3		32	21.00

3. **In-Line Blast Traps** (Semi-Steel). Inverted bucket (open-float) and thermostatic type. Hi-Cap design. Integral bi-metal thermal operated vent. For rapid heating on unit heaters, cookers, etc. Anum-Metl seat, stainless bucket and working parts.

Trap No.	Pipe Size (inches)	Continuous at psi	Capacity lbs/hr	Weight lbs	List Price
070-T	½, ¾	30	660	3¾	$8.50
170-T	½, ¾	30	1290	8	12.00
271-T	¾, 1	30	1760	9	17.50

4. **Bottom Inlet Traps** (Semi-Steel). Rugged, trouble-free inverted bucket type. All stainless working parts. Anum-Metl valve and seat. Self-cleaning action.

Trap No.	Pipes Size (inches)	Continuous at psi	Capacity lbs/hr	Weight lbs	List Price
171	½	125	1230	7½	$10.50
80	½	125	1700	16	17.00
81	¾	125	3300	25	22.50
82	1	125	6100	35	30.00
83	1¼	125	10,200	54	37.00
84	1½	125	15,400	82	55.00
85	2	125	32,000	117	73.00

5. **Open Bucket Traps** (Semi-Steel). For all steam service, and particularly suitable for pulsating pressures. Also for draining water or other liquids from lines on air or gas service. Anum-Metl seat guaranteed leakproof one year.

Trap No.	Pipe Size (inches)	Continuous at psi	Capacity lbs/hr	Weight lbs	List Price
30	½	125	1230	48	$22.50
31	¾	125	1560	53	28.00
32	1	125	2750	79	34.00
33	1¼	125	6100	120	47.00
34	1½	125	8700	165	60.00
35	2	125	11,500	197	85.00
36	2½	125	27,300	335	120.00

6. **Type O & K Pressure Regulators** (Semi-Steel). For steam, air and gas. Direct operated. Rugged construction. Fitted with special laminated, phosphor-bronze diaphragms and stainless valve and seat.

No.	Pipe Size (inches)		Weight lbs	List Price
Type O	⅜, ½, ¾	Initial pressure to 225 psi, 400 F, reduced ranges 0-200.	8	$15.00
Type K	½	Initial pressure to 225 psi, 400 F, reduced	15	19.00
(illustrated)	¾	ranges from 0 to 85	15	20.00
	1	psi.	18	27.00
	1¼	Type K has an inte-	40	38.00
	1½	gral strainer.	40	42.00

7. **Type C Pressure Regulators** (Semi-Steel). For installations requiring accurate and dependable regulation. Stainless trim, single-seated, piston-operated, pilot controlled, spring-loaded.

No.	Pipe Size (inches)		Weight lbs	List Price
Type C	½	From initial pressure	17	$36.00
	¾	to 250 psi, 450 F. For	17	39.00
	1	reduced pressure	17	43.00
	1¼	from 0 to 200 psi.	40	46.00
	1½		40	48.50
	2	Available in cast steel	48	60.00
	2½	series 30, 40 or 60	74	79.00
	3	flanges; pressure 600	98	95.00
	4	psi; temperature 750 deg.	150	160.00

Complete Catalog Available on Request.

WARREN WEBSTER & COMPANY

Pioneers of the Vacuum System of Steam Heating: : Since 1888

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THE COMPANY

Warren Webster & Company have specialized for sixty years in the field of steam circulation and steam distribution, particularly vacuum, vapor and low pressure steam heating of buildings, and medium pressure steam in industrial and process heating applications.

Webster "true perimeter" forced hot water heating utilizes Webster Baseboard Heating, with or without Webster Continuous Flow Control. It is particularly suited for residences and other one and two-story buildings.

The specialized experience of the Company is available through engineers at the Home Office and through the Representatives listed above. Webster Representatives are prepared to supply on request full technical, availability and price information on all Webster Systems and Products.

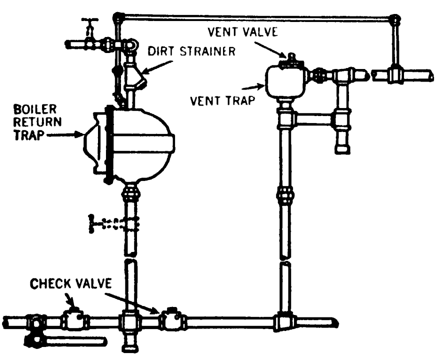
STEAM HEATING SYSTEMS

Webster Steam Heating Systems are low pressure, two-pipe systems in which steam is delivered to radiators and other heating surfaces through supply piping and water of condensation and air are removed through separate return piping. Webster Radiator Valves and Thermostatic Traps are installed respectively on the supply and discharge connection of each radiator. Webster thermostatic or float and thermostatic traps assure removal of water of condensation and air from the piping.

Available with vacuum return, or with open return (vented to the atmosphere) with either Condensation Pump or Boiler Return Trap and Vent Trap to return water to the boiler, or with Vent Trap alone where condensate is wasted to the sewer, or in appropriate small installations.

Webster Vacuum System—A conventional vacuum heating system in which the return mains are joined together and connected to the suction end of one or more vacuum pumps which remove air and water of condensation and assists circulation by maintaining a lower pressure in the return than in the supply piping.

Webster Type "R" System—A two-pipe, low pressure or vapor heating system. Water of condensation is returned to the boiler by gravity, prompt return being assured regardless of variations in boiler pressure through the operation of a Webster Boiler Return Trap and Vent

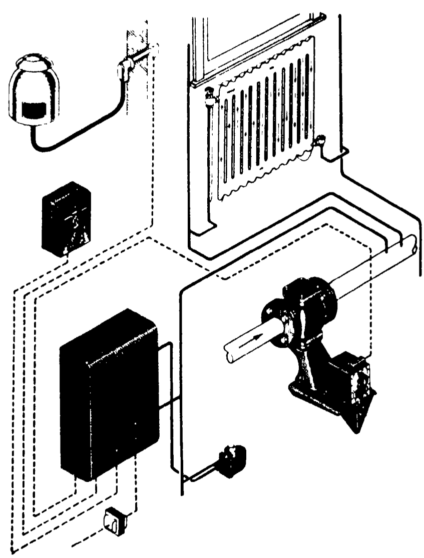


Basement installation of Webster Boiler Return Trap and Vent Trap for Vapor Heating, Webster Type "R" System.

Trap in combination. Equipment is available in sizes to care for systems ranging from the smallest to 16,000 sq ft EDR. Where desired, or where gravity return is not possible, a Condensation Pump may be substituted for the Boiler Return Trap Combination.

Webster Type "V" System—Employs only a Webster Vent Trap. For installations of 1000 sq ft EDR or less with oil burner, stoker or gas burner; with vaporstat having cut-in pressure of about ¼ lb and cut-out pressure of about ¾ lb (not pressurestat), lockswitch or protector relay and one or more key room thermostats. Vent Trap at ample height above water level. Boiler Protector, or at least a low water cut-out. Ask for Bulletin.

Webster Moderator Systems—These are all Webster Steam Heating Systems with vacuum or open return *to which are added* (a) accurately sized metering orifices in radiators and other heating surfaces to balance distribution and permit "partial filling" of all radiators practically simultaneously and at various rates of steam flow, (b) Automatic control by Outdoor Thermostat for variations in outdoor temperature, (c) Manual Variator to provide for convenient adjustments for heating up, reduced night heating, shut off, etc.



Typical arrangement of Webster E-5 Moderator System.

"E" Series Moderator Controls—In this series the Outdoor Thermostat and Variator position a motor-operated Steam Control Valve through an

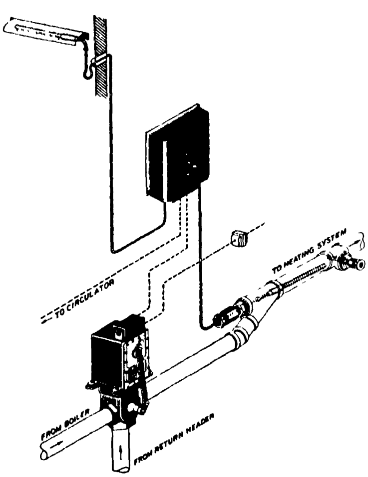
Electronic Differential Pressure Control Cabinet to produce continuous steam delivery and heating effect at the radiators with automatic variation in heating for changes in outdoor temperature and automatic adjustment to compensate for variations in steam supply pressure. Ask for Bulletin B-904A.

"EH" Series Moderator Controls—In this series the Outdoor Thermostat and Variator may control a motor-operated steam valve or directly control oil or gas burner or stoker through a cycling Control Cabinet. Steam delivery is intermittent but in short cycles so that heating effect is substantially continuous, particularly with cast iron radiation. Ask for Bulletin B-960.

Control Valves—Main steam control valve, throttling type, maintains proper pressure differential control in the system. Ask for Bulletin B-291.
Motorized valves provide shut-off service in steam or hot water heating, or throttling control in continuous flow hot water heating. Ask for Bulletin B-290.

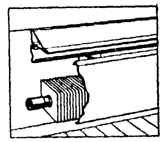
PERIMETER FORCED HOT WATER HEATING SYSTEMS

Webster Continuous Flow Control—for forced circulation Hot Water Heating Systems. Outdoor Thermostat and Variator control throttling-type valve or directly control oil burner, gas control or stoker motor. Water flows continuously through the system. Heating is continuous and adequate at all times. Applicable to Baseboard, Convector, Radiator or Panel Heating. Ask for Bulletin B-200.



Arrangement of Webster CF-2 Continuous Flow Hot Water Heating Control. Top, Outdoor Bulb; center, Control Unit with switch and Variator.

Webster Baseboard Heating—A patented forced circulation hot water heating system in which the heating element



fits behind a specially built metal baseboard. Air enters at the floor line, passes over the finned heating element, is warmed and comes out of slots at the top of the baseboard. The heating element is a copper tube with copper fins running in a continuous loop around the exposed walls of the house—a separate loop for each floor.

Uses less material and less labor than conventional radiator heating systems, while providing all the advantages claimed for forced hot-water, plus radiant effect from warmed baseboards and walls, plus natural convected air movement essential to comfort. Temperatures vary less than 2 deg from floor to ceiling. Ask for literature.

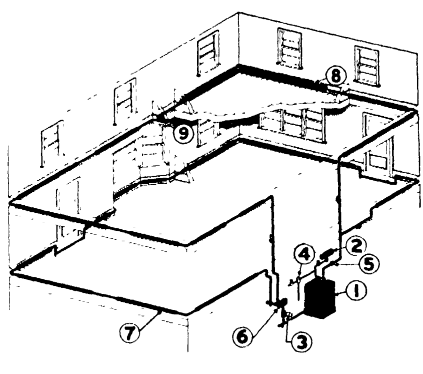


Diagram of typical installation of Webster Baseboard Heating in 2 story residence, showing: (1) Boiler; (2) Expansion Tank; (3) Circulator; (4) Pressure Reducing and Relief Valve; (5) Flo-Control Valve (If Required); (6) Webster Return Header Assembly Including Purge and Balance Valve and Air Vent; (7) Webster Heating Element; (8) Reduced Heat Damper, or By-Pass (Optional); and (9) Expansion Loop.

STEAM HEATING AND PROCESS SPECIALTIES

Radiator Valves—Choice of spring retained packing, Type BW-P or Sylphon Bellows Packless Series 600-S. ½ in., ¾ in., 1 in., 1½ in. sizes. In angle, right and left hand; straightway, with single or double union. Spring retained packing. For low pressure vapor and vacuum steam heating service. Ask for Bulletin B-705.

Thermostatic Traps—Series 7 and 5 for low pressure vapor and vacuum steam heating service, radiators and drips. ½ in., ¾ in., and 1 in. sizes. There are 6 body models in the ½ in. size alone. Maximum pressure, 25 lb per sq in. Series 7 (diaphragm type) described in Bulletin B-702. Series 5 (bellows type) described in Bulletin B-701.

Series 78 for discharge of air and water from heating coils of any apparatus using steam at pressures up to 150 lb per sq in. ⅜ in., ½ in., ¾ in., and 1 in. sizes. Ask for Bulletin B-1200.

Heavy Duty or Drip Traps—Series "26" Float-and-Thermostatic for heating and air conditioning. Most used sizes: 00026, 0026, 026. Pressures up to 15 lbs per sq in. Made for the pressure and capacity conditions encountered at all drip points. Series "79" Float-and-Thermostatic for process. For pressures up to 150 lbs per sq in. For use where large volumes of hot condensate must be handled more quickly than is possible by thermostatic traps alone. Ask for Bulletins.

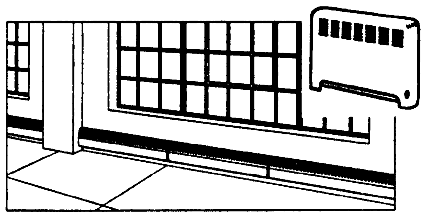
Strainers—**Dirt:** ½ in. to 6 in. sizes. Maximum working pressure 150 lbs per sq in. Placed ahead of traps in return lines of steam-using equipment and steam heating systems to catch dirt and other particles, preventing them from impairing the tightness of the traps. **Suction:** Maximum working pressures 15 lbs per sq in. Installed ahead of vacuum pump to prevent dirt from damaging pump. Ask for Bulletin B-709.

Boiler Protectors—One size, with ¾ in. connections with or without electrical cut-out switch. Maximum pressure 15 lbs per sq in. Maximum cold water main pressure, 150 lbs per sq in., minimum not less than 25 lbs per sq in. Prevents breakage in low pressure heating boilers when the water level becomes inadequate. Ask for Bulletin B-727.

Other Steam Heating Specialties—Double Service Valves, Check Valves, Lift Fittings, Gauges, Motorized Valves, Steam and Oil Separators, Sight Glasses, Vacuum Breakers. Ask for Bulletins.

RADIATION AND HEATING SURFACE

Webster System Convector Radiation—Non-ferrous convector radiation. Each Webster System Radiator includes a complete enclosure of furniture steel with baked prime coat. Choice of two types of enclosure, fully recessed, with metal front or free standing floor cabinet. Contained within the enclosure in a prefabricated unit, combining heating sur-

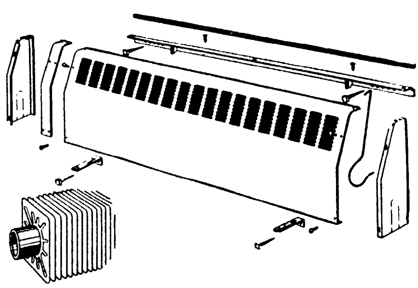


Wall-to-wall application of Webster Walvector. Inset, Webster System Convector Radiator.

face, valve, trap and uniono cnnections, shipped ready to connect to supply and return piping.

Webster System Radiation was first offered in 1932. Now, available in an improved design, using the same basic material, copper tubing and aluminum fins. Increased rigidity of the tubing and the development of a new method of manufacturing has produced a fin surface of unusual rigidity, free of expansion and contraction noises. Ask for Bulletin B-1500.

Webster Walvector—Elongated, non-ferrous convector. Heating element is made up of a specially annealed copper tubing with rib-reinforced, square pressed aluminum fins. Available in two fin sizes: 3 in. fin size in 2, 3, 4, 5 and 6 ft; 4 in. fin size in 2, 4, 6 and 8 ft lengths.



Exploded view of Webster Walvector including cutaway view of fin-and-tube type heating element.

Enclosure available in four types, for mounting along outside wall close to floor or under windows. Delivered as complete "package," it includes all components needed. Can be used as separate convector unit or for wall to wall application. Available without cover for low cost installation. For steam or hot water heating. Radiation may be orificed for steam installations. Ideal for educational or institutional buildings and other commercial structures. Ask for Bulletin B 1551.

WEBSTER-NESBITT UNIT HEATERS

Manufactured by John J. Nesbitt, Inc., Philadelphia 36, Pa., and distributed solely through Warren Webster & Company, Camden, New Jersey. Designed to circulate large volumes of air at comparatively low temperatures, assuring quick heating.

Ratings of Webster-Nesbitt Unit Heaters are based on tests made in accordance with standard test code of *Industrial Unit Heater Association* and A.S.H.V.E.

PROPELLER FAN UNIT HEATERS

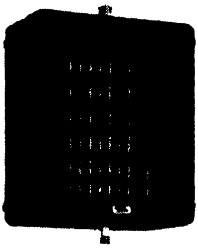


Fig. 1. Standard Propeller-Fan Type

Designed to incorporate four characteristics essential to both proper application and satisfactory performance: 1.) *Selective range of sizes.* Manufactured in nine sizes. Heating capacities from 34,700 to 338,000 Btu per hour. Air deliveries from 470 to 4800 cfm. 2.) *Quiet Operation.* All fans have blades of exceptionally large areas and of a shape to impart a gradual acceleration to the air stream. Ample spacing is maintained between the fan and heating element. Motors are of sleeve bearing type equipped with isolators. 3.) *Durable lightweight Heating Elements.* Extended fin-and-tube type, constructed of copper condensing tubes and plate-type aluminum fins. 4.) *Modern Casing Design.* Compact suspended type. Pub. W-N 126.

GIANT UNIT HEATERS

Sturdy blower-fan units for the economical heating of large areas. *Standard* (Non-*Thermadjust) Type.* Used principally where heating is by recirculation only, and where constant heat output is desired during operation. *Thermadjust Type.* Employs dampers in front of casing and over face of heating element to provide mixing of unheated and heated air, producing heat output in accordance with requirements and continuous circulation of air volume. *Valve Controlled Type.* Unit is of standard casing arrangement but equipped with Nesbitt Heating Surface and Steam-distributing Tubes for automatic control of heat output. Floor mounted, wall mounted, ceiling suspended, from 101,000 Btu/hr, 2450 cfm, to 1,008,000 Btu, 16,350 cfm. Pub. W-N 128.

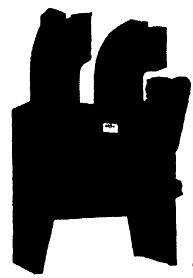


Fig. 2 Blower-Fan Type

LITTLE GIANT UNIT HEATERS

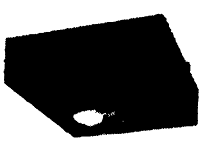


Fig. 3 Down-Blow Type

Adaptable to a wide variety of applications and field conditions. Seven basic sizes, each with a choice of two (some units three) heating elements. The three smaller sizes are of the blow-through type, having lower outlet velocities generally intended for the lower mounting heights of commercial installations. These sizes in down-blow type only. The four larger models are of the draw-through type; produce the high discharge velocities necessary to blow long distances. These four available for either horizontal or vertical down-blow.

Non-ferrous all-purpose heating elements designed for steam pressure up to 200 lb gauge, saturated, and sturdy casings of modern design. Heating capacities range from 28,500 to 348,000 Btu basic steam ratings. Pub. W-N 134.

SERIES "R" UNIT HEATERS

A neat, furniture steel cabinet enclosing a copper-tube, aluminum-fin heating element adaptable for steam or forced hot water systems; and two to five centrifugal fans belt-driven from an electric motor. A variable-pitch motor sheave permits low or high speed fan operation. Universal design offering wide flexibility and quiet operation. Available in four sizes. Air deliveries with standard drive range from 518 to 1890 cfm. Steam heating capacities from 158 to 588 EDR. Pub. W-N 133.

Fig. 4. Series "R" Unit Heater.

Yarnall-Waring Company

Manufacturers of YARWAY **Steam Specialties**

133 Mermaid Ave., **Philadelphia 18, Pa.**

YARWAY IMPULSE STEAM TRAPS

Construction—Made entirely of bar stock. Only one moving part, valve (F). For pressures to 400 lb, body, control cylinder, valve and seat are stainless steel, valve and seat heat-treated; bonnet is cold rolled steel, cadmium plated; cap is tobin bronze. For 600 lb, trap is all stainless steel.

Operation—At low condensate temperatures, bypass through control chamber (K) and center orifice of valve reduces chamber pressure and valve opens. At high temperature, condensate vaporizes in (K), increased volume builds up pressure and valve closes.

Light Weight—Need no support—½ in. trap weighs only 1⅜ lb. 2 in. weighs 8⅝ lb.

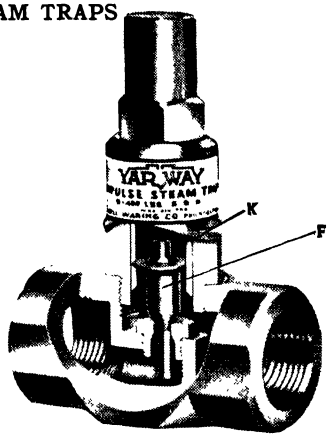
Small Size—½ in. trap 2¼ in. long –2 in. trap, 4¾ in. long.

Will not air bind.—Require no priming **Insure quick heating.**

Low Price—Often cheaper than repairing old traps.

Factory set for all pressures to 400 lb (or 600 lb) without change of valve or seat.

750,000 sold. Stocked by 200 distributors. Send for descriptive Bulletin T-1740.

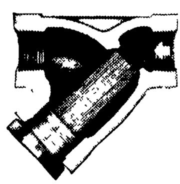


List Prices, Weights and Dimensions
No. 60 Series—to 400 lb; 120 Series—to 600 lb

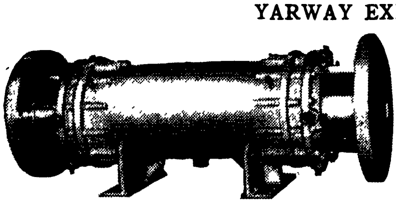
Size	Complete Trap Series 60	Complete Trap Series 120	Weight Pounds	Length Inches
½"	$15	$ 25	1¼	2⅝
¾"	22	37	2	3
1"	31	52	2½	3⅜
1¼"	48	80	4	3¾
1½"	68	114	5¾	4¼
2"	90	150	8½	4¾

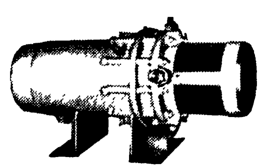
YARWAY FINE-SCREEN STRAINERS

Offer better protection against rust, scale and dirt for all steam equipment. Ten standard sizes ¼ in. to 3 in. Cadmium plated bodies. High grade Monel woven-wire screens. Many thousands in use. Also flanged strainers, ½ in. to 5 in. Write for Bulletin S-203.



YARWAY EXPANSION JOINTS





All-steel welded construction; light but strong. Chromium covered sliding sleeves. Cylinder guide and stuffing box integral, assuring perfect alignment. Internal limit stops. Gun-pakt and Gland-pakt types: Gun-pakt (illustrated) has fixed glands fitted with screw guns which permit addition of plastic packing while joint is under pressure. Sizes 2 in. to 24 in., single end or double end, flanged or welding ends; 150, 300 and 400 lb pressures. Choice of leading utilities and industrial firms. Send for Bulletin EJ-1912.

The Dole Valve Company

Main Offices and Factory: 1933 Carroll Avenue, **Chicago 12, Ill.**

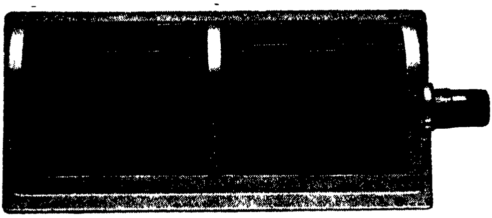
WATER MIXERS

THERMOSTATIC AIR CONTROL

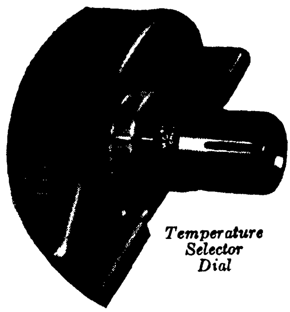
THE ALL STAR LINE

AIR AND VACUUM VALVES

"DOLE THERMOSTATIC AIR CONTROL"



FOR FORCED WARM AIR HEATING SYSTEMS



Temperature Selector Dial

PROVIDES INDIVIDUAL ROOM TEMPERATURE CONTROL

1. Operates thermostatically from room air temperature.

2. Extremely Sensitive: Modulates output to meet heat requirements.

3. Completely self-contained; no wires to run—no bulbs to locate—simple to install. Replaces standard forced warm air registers.

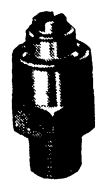
4. Simple setting of the thermo-dial assures room temperature as desired—corrects many unsatisfactory heating installations. Materially improves any forced warm air system. An automatic balancer.

5. A fully automatic zone control for every room. Dole Air Controls are available in two sizes and will fit the following stackhead openings:

10″ will fit—10″ x 4″, 10″ x 5″, 10″ x 6″.
12″ will fit—12″ x 4″, 12″ x 5″, 12″ x 6″.
With an adapter 12″ will fit—14″ x 4″, 14″ x 5″, 14″ x 6″.

An adapter is available for baseboard installation of these Controls.

DOLE AIR AND VACUUM VALVES



Dole No. 20 Fully Automatic Hot Water Air Valve

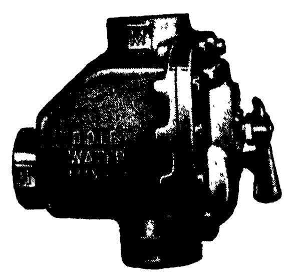
The Dole line covers every venting need on one pipe steam and hot water heating systems and offers a complete choice for every purpose.

DOLE WATER MIXERS

Dole Water Mixers provide safer, tempered domestic hot water on all tankless heater and storage tank installations. Available in 3 sizes, ½ in., ¾ in., and 1 in.

Dole No. 1A Vari-Vent Air Valve

Dole No. 3 Air Valve



Dole Water Mixer

The Fairbanks Company

393 Lafayette Street **New York 3, N. Y.**

Boston; Pittsburgh; Binghamton, N. Y.; Rome, Georgia

Dependable Service GUARANTEED From This Complete Line Of Bronze And Iron Body Valves Available From Your Local Distributor

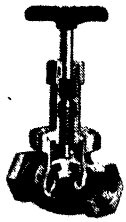


Fig. U-01
150 Lb. SWP

BRONZE GLOBE AND ANGLE VALVES

Pressures—125 through 300 Lbs S.W.P.
Ends—Screwed, Flanged, Solder, Brazed and Hose.
Discs—Bronze: Renewable Composition; Nickel Alloy Semi-Plug Disc and Seat; Stainless Steel Semi- and Full Plug Disc and Seat.
Needle Valves
Radiator Valves

BRONZE GATE VALVES

Pressures—125 through 300 Lbs S.W.P.
Ends—Screwed, Flanged, Solder, Brazed and Hose.
Stem Action-Rising with Solid or Split Wedges, Non-Rising with Solid Wedges.
Bonnets—Screwed, Union, Bolted and O. S. & Y.
Radiator Valves

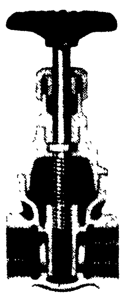


Fig. 0250
125 Lb. SWP

BRONZE CHECK VALVES

Horizontal, Angle and Vertical
Types—Swing and Lift
Pressures—125 through 300 Lbs S.W.P.
Ends—Screwed, Flanged, Solder and Brazed.
Discs—Bronze, Renewable Composition, Rubber Faced.

IRON BODY GLOBE & ANGLE VALVES

Bronze Mounted
Pressures—125 through 250 Lbs S.W.P.
Ends—Screwed and Flanged
Discs—Bronze, Renewable Composition, Nickel Alloy Semi-Plug Disc and Seat
Bonnets—O. S. & Y.
Automatic Stop and Check Valves

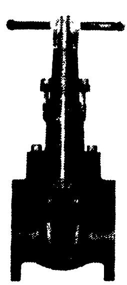


Fig. 0405
125 Lb. SWP

IRON BODY GATE VALVES

Bronze Mounted and All Iron
Pressures—125 through 250 Lbs S.W.P.
Ends—Screwed, Flanged and Hub
Stem Action—Rising, Non-Rising and Quick Opening
Bonnets—Screwed, U-Bolt, Bolted & O. S. & Y.
Underwriters' and AWWA Approved Valves

Fig. 0702
125 Lb. SWP

IRON BODY SWING CHECK VALVES

Horizontal—Bronze Mounted
Pressures—125 through 250 Lbs S.W.P.
Ends—Screwed, Flanged and Hub
Discs—Bronze, Bronze Faced and Rubber or Leather Faced

HAMMOND BRASS WORKS HAMMOND, INDIANA

HAMMOND VALVES

A complete line of packed type radiator valves for steam, gravity and circulator hot water systems—balancing elbows and fittings for forced hot water heating systems and floors, ceiling, and wall radiant panel heating.

Hammond Circulator Valves, Balancing Elbows and Fittings for Forced Hot Water Heating Systems

No. 201 No. 202 No. 301 No. 302 No. 305

Service Recommendations No. 201—Circulator-Male Union and Sweat Connection. Provides a "nearly tight" shut-off when closed and full flow when opened. A quarter turn of the handle permits the valve to be opened smoothly and easily.

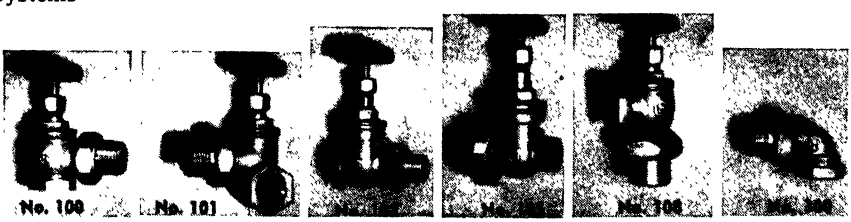
No. 202—Circulator-Male Union and Sweat Connection. For forced hot water systems only. Particularly suited for use with concealed or "convector type" radiators, which installation frequently requires a valve for the lower vertical radiator tapping. A quarter turn of the handle permits the valve to be opened smoothly and easily.

No. 301—Balancing Elbow-Male Union and Female Thread. No orifices or adaptors are needed when this elbow is used. The external adjustment allows for adjusting and readjusting while the system is in operation and eliminates inconvenience of draining.

No. 302—Union Elbow-Male Union with Sweat Connection. Can be used on copper tubing gravity jobs and also where the supply valve is adjusted, but we do not recommend adjusting any system at the inlet.

No. 305—Balancing Fitting-Male Union and Female Thread For concealed or convector type radiators.

Hammond Radiator Valves and Elbows for Steam and Gravity Hot Water Heating Systems



No. 100 No. 101

No. 100—Steam Angle. Can also be used for vapor.

No. 101—Steam Corner. Same as No. 100. Specify whether right or left hand wanted. Valve illustrated is left hand.

No. 102—Gate Union. For hot water installations and steam or vapor heating systems. Especially suitable for use with unit heaters; ideal for general shut-off valve service at other points throughout heating systems including boilers and hot water heaters. Specify when ordering for hot water system, as such valves are furnished with a small hole drilled through the disc to provide for slow circulation when valve is closed.

No. 105—Convector Single Union Gate. Same as No. 102, but single union does not permit removal of radiator while system is in operation.

No. 108—Convector Female Union Steam Angle. Same as No. 100, but should only be used for convector radiators equipped with inside leg tappings. The female union eliminates the necessity for exact roughing in measurements to tapping. Variation of the nipple inside leg will allow for adjustment. Shipped with bonnet loose to facilitate installation.

No. 300—Union Elbow—For gravity hot water heating installations.

See Hammond Catalog for complete line of valves.

Warehouse stocks located in 29 cities throughout the United States for your convenience **and** prompt delivery, at our regular prices and terms.

Homestead Valve Manufacturing Co.

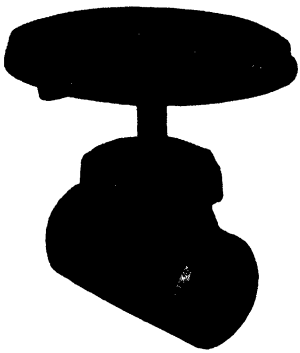
"Serving Since 1892"

P.O. Box 127 **Coraopolis, Pa.**

Makers of Radiant Heating Valves and All Kinds of Plug Valves

HOMESTEAD-RADIANTROL VALVE

The Combination Balancing Valve and Air Vent for Radiant Heating Systems



In the simplest sense, the Homestead Radiantrol Valve performs the same function for each radiant heating panel as the manual valve on a radiator, or the shutter on a hot air grille. It controls the flow of hot water to each panel or room in accordance with the varying heat requirements caused by differences in types of floor covering, changes in wall or ceiling construction, addition or elimination of storm sash, individual likes and dislikes, etc.

The Homestead Radiantrol Valve is an unique combination balancing valve and air vent for radiant heating systems. Saves up to $25 per panel in installation costs through:

1. Eliminating cost of making special valve "wells" and covers.
2. Eliminating need for separate riser air vents.

Controls heat in each panel; and balances heat between various panels.

Provides control of room temperatures to meet varying requirements.

Is designed for simplicity and convenience of adjustment; and

Engineered for ease and low cost of installation.

Homestead Radiantrol Valves are butterfly type valves, which permit slight passage of water in the "closed" position.

A quick quarter-turn fully opens or closes the valve.

Graduations between "open" and "closed" positions give instant choice of amount of hot water for each panel. In "open" position valve provides nearly full pipe area flow.

Chamber in neck of valve body is especially designed to collect trapped air without restricting flow.

Vent screw on valve cap provides simple means of removing trapped air from system.

The Radiantrol Balancing Valve is a "must" for any radiant heating system of more than one panel in order—

1. to insure proper flow needed to offset the various heat losses encountered; and
2. to remove air from each heating panel.

Valve bodies are made of wrought iron, steel, or brass pipe with beveled or sweat ends for welding or soldering.

Valve floor plates may be had with locks to prevent tampering; or with positive adjustable stops to limit opening and save rebalancing system.

Radiantrol Valves are available in either foot-control or hand-wheel types; sizes ½ in., ¾ in., 1 in. and 1¼ in.

Write for complete catalog, Reference Book 39-7. No obligation.

Jenkins Bros.

100 Park Avenue, **New York 17, N. Y.**

BRIDGEPORT, CONN.; BOSTON, PHILADELPHIA, CHICAGO, SAN FRANCISCO, ATLANTA

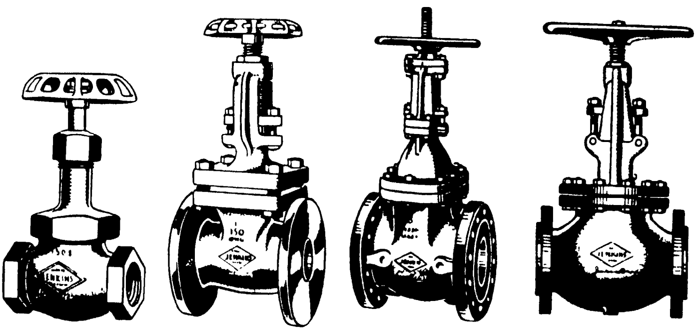
LOOK FOR THIS DIAMOND MARK

REG. U.S. PAT. OFF. TRADE JENKINS MARK

Jenkins Bros

Leading Supply Houses Everywhere Stock Jenkins Valves

Jenkins VALVES for LIFETIME SERVICE



Bronze *Stainless Steel* *Iron* *Steel*

FOR EVERY NEED

JENKINS CATALOG LISTS OVER 500 VALVES

Consult Jenkins Catalog for complete details on more than 500 different valves that cover practically all industrial plumbing and heating, and engineering requirements. Below is a brief list.

All-Iron Valves,—globe, angle, gate; **Angle Valves,**—bronze, steel, and iron body with bronze mounting or trimming.

Blow-Off or **Y Valves,**—bronze and iron.

Electrically Operated Valves, Gates, Globes, Angles, Fire Line Valves; Floor Stands; Foot Valves for gasoline service.

Gate Valves,—bronze, iron, steel; with solid wedge or double disc parallel seats; with removable bonnet and renewable bushing.

Globe Valves,—bronze, iron and steel; one piece and union bonnets; renewable and integral seats; rubber composition or metal discs and plugs.

Horizontal Check Valves.—bronze, iron and steel; **Hose Valves; Indicator Posts; Lock Shield Valves.**

Needle Valves; Non-Return Valves; Quick-Opening; Self-Closing Valves.

Radiator Valves; Rapid Action Valves; Regrinding Valves; bronze and iron body with bronze trimming; renewable plug seats and bevel seats of a special nickel alloy in globe, angle, check and swing check patterns.

Selclo Valves; Stop and Check Valves,—combination or automatic equalizing; Swing Check Valves,—bronze, iron and steel.

Stainless Steel Valves,—globe, angle, gate and check.

Underwriters' Pattern Valves,—check and gate; **Whistle Valves; Waterworks Valves.**

Carey PRODUCTS

The Philip Carey Mfg. Company

Lockland, Cincinnati 15, Ohio

District Offices In All Principal Cities

CAREYDUCT is recommended whereever quietness, ease of installation, fire safety, fume resistance and good appearance are desirable or essential. Widely used in air conditioning systems. Careyduct has proven itself on some of the largest governmental, industrial and commercial installations in the country.

Write for engineering performance and installation data.

ACOUSTICAL. Careyduct is a natural sound absorber and non-conductor of sound. Quiets fan noise; won't pick up and "telegraph" other outside noises.

INSULATED. High-efficiency insulation assures delivery of hot or cold conditioned air to outlets with minimum change in temperature.

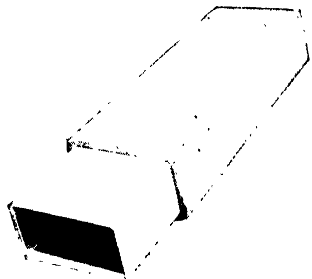
AIRTIGHT. Won't "breathe" or vibrate at high velocities. Slipjoint construction prevents leakage.

SAVES SPACE. Being 40 per cent to 50 per cent quieter than ordinary duct, Careyduct handles higher velocities, permitting the use of smaller sized ducts.

***EASY TO INSTALL.** Prefabricated Careyduct units are easy to install—particularly in tight places. Simple low cost fittings can be made in the shop or on the job.

FIREPROOF. Being 100 per cent asbestos construction Careyduct won't smoulder or burn. Approved by *Underwriters' Laboratories, Inc.*

* *Installation under jurisdiction of International Sheet Metal Workers, A. F. of L.*



GOOD LOOKING. Surfaces are smooth and free from unsightly raised seams or joints. No stiffeners or braces. Blends well with modern interiors.

5 TYPES OF CAREYDUCT

Insulated and Acoustical (I. & A.) Type. Built of asbestos sheets with an inner core and an outer jacket. Combines duct, insulation and acoustical treatment into one unit.

Single Wall (S.W.) Type. This is an all asbestos duct designed for ventilating and heating where a high degree of insulation is not essential. It is ideal for residential heating.

Asbestos - Cement (K.D.C.A.) Type. Made of asbestos-cement wallboard in different thicknesses. Shipped knocked down ready for assembly.

Firefoil Panel (K.D.F.) Type. Fabricated from Firefoil and shipped knocked down ready for assembly. For high temperature work.

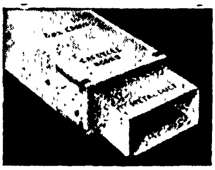
Acid and Fume (A. & F.) Type. Built of asbestos sheets laminated together and treated with two coats of acid, alkali and water resistant coating. Ideal for laboratories or industrial work.

The Philip Carey Mfg. Company

Lockland, Cincinnati 15, Ohio

District Offices In All Principal Cities

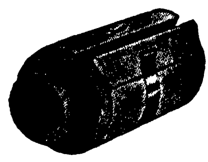
PRODUCTS



CAREYCEL FOR AIR DUCTS

Uses: A fireproof, low cost, high efficiency asbestos board for insulating ducts and all types of air conditioning equipment. Use 1 inch thickness up to 85F air temp and 80 per cent relative humidity. Recommendations for more severe conditions on request.

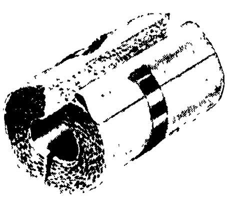
Description: Composed of 12 to 14 laminations of indented (not corrugated) asbestos felt per inch of thickness. Weight: approximately 1½ lb per board foot. Sheet Size: 36 in. x 36 in., or cut to order. Blocks: 6 in. x 36 in. Thickness: ½ in. up.



CAREY IMPERVO FOR COLD PIPES

Uses: A high efficiency insulation for cold or ice water pipes—keeps the water cold and prevents sweating. Double ¾ inch thickness recommended for ice water pipes.

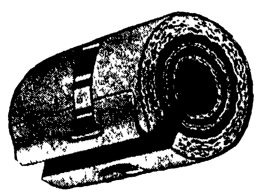
Description: Laminated insulating felt with waterproof liner and jacket. 36 in. long in ½ in., ¾ in., double ½ in. and double ¾ in. thick sections, finished with cotton duck jackets and bands.



CAREYCEL FOR HEATING SYSTEMS

Uses: Pipe coverings and blocks for pipes, boilers, ovens and other apparatus where the temperature doesn't exceed 300 F. Use 1 inch thickness for temperatures up to 300 F.

Description: Pipe covering sections 36 in. long by 1 in. thick, finished with cotton duck jacket and bands. Blocks: 6 in. x 36 in. Sheets: 36 in. x 36 in., or cut to order. Thickness: 1 in. up.



CAREY PROTECTO TO PREVENT FREEZING

Uses: Designed especially to reduce the danger of freezing of exposed water pipes.

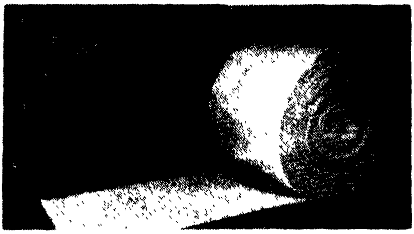
Description: Consists of two inner layers of hair felt, a waterproof felt liner and an outer layer of insulating felt (wool felt). For severe conditions—exposure down to 0 F—use two-inch thickness. 36 in. long sections with cotton duck jacket and bands. Standard thickness—approximately 1¼ in.

Gustin-Bacon Manufacturing Company

210 W. 10th St. **Kansas City, Mo.**

Distributors in All Principal Cities (Consult Classified Phone Directory)

ULTRALITE glass fiber Duct Insulation and Duct Liner



Characteristics of Ultralite. Ultralite Duct Insulation and Ultralite Duct Liner are composed of long, fine, textile-type glass fibers, bonded with a thermosetting resin. Both are manufactured and shipped in blanket-like rolls. Both are immune to fire, rot, corrosion, age, rodents and insects, odors, vibration, and are low in moisture absorption. Ultralite is resilient—quickly returns to original dimensions after pressure or bumps during or after installation.

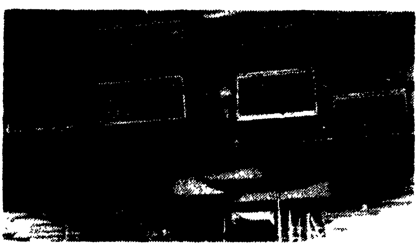
Application Characteristics. Ultralite Duct Insulation and Ultralite Duct Liner are exceptionally light in weight. Can be cut readily with a knife, quickly run around curves and corners without special fitting. Can be adhered with adhesives, metal screws and washers, even staples. Ultralite is not unpleasant to handle. Extreme simplicity of application keeps applied costs down.

Ultralite Duct Insulation (Thermal). A highly efficient "wrap-on" type thermal insulation weighing only 1 oz per board foot. Available plain or with your choice of a number of facings already adhered to insulation (when a vapor barrier and/or a base for finished appearance on duct runs is required). Shipped in compressed rolls. (See *Sweet's File* (*Arch.*) or write for *AIA File* 37-D-2)

Thermal Efficiency of Ultralite Duct Insulation is shown in table at right. Tests were conducted by independent laboratories with *ASTM* approved, guarded hot plate methods.

THERMAL EFFICIENCY

DENSITY	"K" VALUE AT 60°
¾ #/cu ft	.253
1 " "	.246
1½ " "	.235
2 " "	.233
3 " "	.217



Ultralite Duct Liner (Acoustical). A sound-absorbing insulation to be applied to interior of duct. Effectively absorbs objectional fan, air-rush and transmitted noises. Also an excellent thermal insulation that can be used on the exterior of duct. Weighs only 2 oz per sq ft in the ½ in. thickness. Won't break, chip or dent. Available in ½ in. and 1 in. thicknesses, coated one side with a fire resistant coating. Can be adhered to flat metal sheets and fabricated with the metal through brakes and shears. (See *Sweet's File Architectural* or write for *AIA File* 37-D-2)

ACOUSTICAL TABLE

Density lbs/cu ft	Thickness	Sound absorption Coefficients at Frequencies 128	256	512	1024	2048	4096	NRC
3	½"	.11	.46	.42	.67	.80	.80	.60
	1"	.15	.53	.65	.90	.87	.90	.75

Owens-Corning Fiberglas Corporation

General Offices—**Toledo 1, Ohio.**

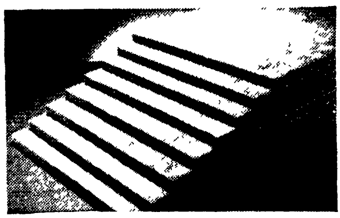
Pacific Coast Division, Box 89, Santa Clara, California.

OWENS CORNING FIBERGLAS

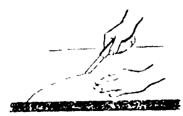
DUCT INSULATION

For highly efficient thermal and acoustical insulation of ducts and duct systems, there are three distinct styles of Fiberglas* Insulations—each designed for specific duct insulation application requirements.

FIBERGLAS COATED DUCT INSULATION is versatile, lightweight Fiberglas PF (Preformed) Insulation both surfaces of which have been uniformly coated with a finish which improves handleability. Can be used on interior and exterior duct surfaces. Standard size 24 by 48 in., in various thicknesses. Thermal conductivity (k) is approximately 0.23 Btu at 75 F mean temperature; the insulation is recommended for temperatures up to 450° F.



CUT WITH A KNIFE—Fiberglas Coated Duct Insulation may be cut accurately with a knife to conform to irregular shapes and curved surfaces, is easily installed and provides a neat and lasting insulation for hot or cold ducts.



FIBERGLAS PF INSULATION—PF Insulation is recommended for even greater economies on concealed duct work. It is also used for ducts which require a canvas or plaster finish. Available in densities from 2½ to 10½ lb per cu ft, PF Insulation is useful to 600° F and has physical characteristics comparable to Fiberglas Coated Duct Insulation. (Fiberglas PF Insulation is not suitable for application to duct interiors.)

FIBERGLAS AEROCOR* FLEXIBLE DUCT INSULATION—Aerocor is made of superfine glass fibers nearly thirty times as fine as human hair, lightly bonded into a fluffy blanket form. It is exceptionally efficient and useful up to 600° F. Aerocor is especially adaptable for inexpensive applications on the exterior of concealed ducts and for all ducts of a circular or elliptical section. It provides savings in material and application costs.

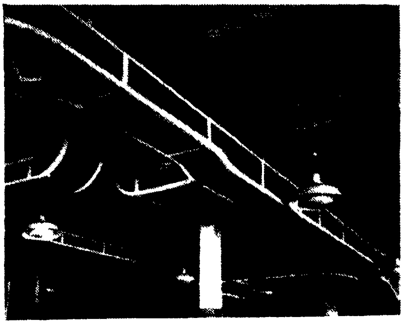
WHERE TO BUY FIBERGLAS INSULATIONS—Check yellow pages of your phone book for name of your local Fiberglas applicator. If not listed, write Dept. 44, Owens-Corning Fiberglas Corporation, Toledo 1, Ohio.

**FIBERGLAS (Reg. U. S. Pat. Off.) and AEROCOR are trade-marks of Owens-Corning Fiberglas Corporation, Toledo, Ohio, for a variety of products made of or with fibers of glass.*

Grant Wilson, Inc.

141 West Jackson Blvd.
Chicago 4, Illinois

ASBESTOS PROTECTED DUX-SULATION



Specifications

THERMAL: All sheet metal duct work from the unit (heating or cooling) shall be insulated on the outer surface with Asbestos - Protected Dux - Sulation as manufactured by Grant Wilson Inc. to the thickness specified (or indicated) on the drawings, applied in accordance with the manufacturer's directions. Grant Wilson Dux-Sul Glue shall be used for applying Dux-Sulation and Dux-Sul Tape shall be used for sealing the joints and corners.

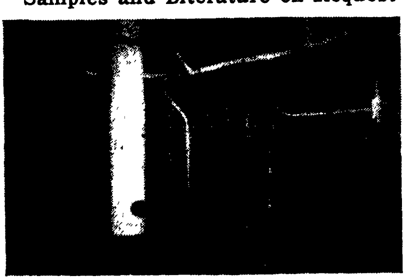
ACOUSTICAL: All sheet metal duct work from the unit (heating or cooling) to the registers or grilles shall be lined on all four (4) inner surfaces with ½ in. thick (or thickness shown on drawings) Asbestos - Protected Dux - Sulation as manufactured by Grant Wilson Inc. Application shall be in accordance with the manufacturer's directions.

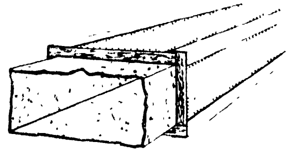
Acoustical Values

70 Per Cent Reduction in Loudness

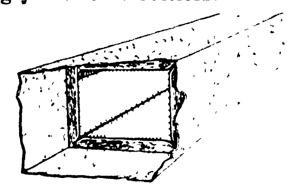
Frequency	Feet
1035	9.5
2048	9.7
4100	11.4
517	15.2
259	31.0
129	34.0

Samples and Literature on Request





THERMAL INSULATION: Dux-Sulation saves 75 per cent of heat otherwise lost through ducts. It has a K factor of 0.27 Btu and is composed of flexible fibres fabricated into a strong felt with millions of dead air spaces. A heavy Asbestos membrane, woven just below the outer surface, adds fireproofing qualities. Dux-Sulation comes in ½ in. and 1 in. thicknesses (100 sq ft rolls), complete with adhesive and Asbestos strips for sealing joints and corners.



ACOUSTICAL INSULATION: Dux-Sulation, when applied to the inside of Air Conditioning and Ventilating Ducts, reduces noise travel 70 per cent in less than 10 lineal feet. It has a low frictional resistance coefficient of F = 0.0001322.

Surface Temperature of ½ in. DUX-SULATION applied to Metal Duct

Outside Duct Temperature	Room Temperature—Deg F			
	30	50	70	90
40°F	33	48	63	78
60°F	37	53	68	83
80°F	42	57	72	88
100°F	47	62	77	93
120°F	52	68	82	97
150°F	60	75	90	105

Dew Point Temperature—Deg F

Relative Humidity	Room Temperature—Deg F (Dry Bulb)			
	30	50	70	90
20%	0	12	28	44
40%	10	27	45	64
60%	18	37	56	75
80%	25	44	64	84

Note: As determined through using the two Tables above, the Surface Temperature of the Dux-Sulation must be HIGHER than the DEW POINT to prevent condensation.

Durant Insulated Pipe Company

1015 Runnymede Street, P. O. Box 88
Palo Alto, California

Eastern Manufacturing and Sales Associate
DURANT INTERNATIONAL CORPORATION
Williamstown, New Jersey

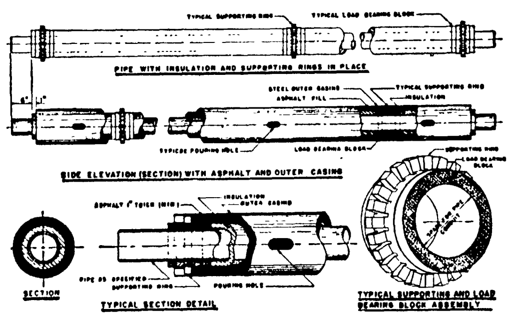
TRADE-MARK
REG. U. S. PAT. OFF.

T. M. REG. U. S. PAT. OFF.

REPRESENTATIVES IN PRINCIPAL CITIES

DURANT Pre-Sealed INSULATED PIPE

Provides Positive Protection for Underground or Overhead Conveyance of Steam, Hot Water or Refrigerants



Construction and Piping Details of D. I. P.

The patented construction principle used in the manufacture of **DIP** has provided effective, reliable and enduring insulation for the conveyance of hot or cold liquids and gases on many types of installations during the past twenty-five years.

While each installation is custom designed to meet specific conditions and requirements, DURANT catalogs a wide variety of factory-built fittings—such as, ells, tees, expansion loops and bends—ready for easy installation in the field.

DIP Research has recently developed a new insulating anchor which isolates the pipe electrically from the anchor plate, eliminating potential corrosion by electrolysis.

The **DIP** *Engineering Staff is always ready to confer on your piping problems, and to supply detailed information or construction data.*

DURANT Processes and Products are protected by Registered U. S. Patents and Patents Pending

Therm-O-Tile

REG. U.S. PAT. OFF.

H. W. Porter & Co., Inc.

817-G Frelinghuysen Ave., **Newark 5, New Jersey**

REID HAYDEN, INC.

Baltimore, Md. Richmond, Va. Charlotte, N.C.

Permanent Protection and Insulation for Underground Pipe Lines.

Also sold and installed by Johns-Manville Construction Units in all principal cities.

The Outstanding Advantages of Therm-O-Tile are:

1—PERMANENTLY higher efficiency. A permanently DRY conduit. DRY insulation.

2—PERMANENCY means: "much longer life."

3—Positive sealing throughout. Positive internal drainage.

4—Arched construction. Stronger than required by ASTM.

5—"Spread - footing" foundation. The "sidewalk" makes installation easier.

6—Surrounded by sealed air.

7—All loads transmitted directly to an unyielding base.

8—Correct slope is PERMANENT, hence no condensation pockets.

9—PERMANENCE assures lowest ultimate cost. Maximum economy.

With wet insulation, efficiency drops drastically, and unless the conduit is built on an unyielding foundation there may be sagging and collection of water in pockets.

Always Drained. The Therm-O-Tile concrete base contains a drainage channel—see photograph—which carries off all water that may enter the conduit from any source, thereby keeping the insulation PERMANENTLY dry. Drainage is entirely internal and ample to keep the pipe space always dry. Open to thorough inspection at any time at manholes.

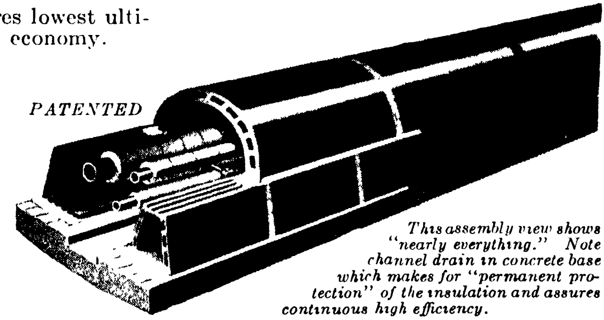
"Spread - Footing" Foundation. The Therm-O-Tile foundation base is a thick concrete slab poured directly in the trench bottom. Settling and sagging are thus positively prevented. The original

THERM-O-TILE

STEAM

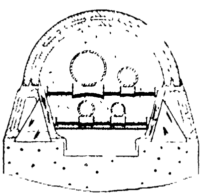
CONDUIT

SYSTEMS

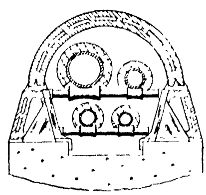


This assembly view shows "nearly everything." Note channel drain in concrete base which makes for "permanent protection" of the insulation and assures continuous high efficiency.

PERMANENT Therm-O-Tile has long been well known to all leading heating and ventilating engineers, but we wish, again, to emphasize the importance of "Permanent protection." It is not difficult to provide TEMPORARY protection and insulation under ground. Threads and joints don't fail immediately. Foundations don't sag immediately. But unless the job is properly done INITIALLY it won't be long before water seeps in and ruins the insulation.



Showing the Use of Filler Type Insulation.



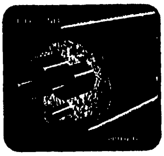
Showing the Use of Sectional Pipe Covering.

correct slope is PERMANENTLY held so that condensate pockets cannot form. Steel reinforced or placed on piles when installed over filled or boggy ground to insure PERMANENCY.

Numerous Conduit Sections. Base and top sections of the Therm-O-Tile envelope are made in a number of different sizes. These make 27 conduit sections available. For complete information ask for Therm-O-Tile Bulletin.

Cost is Competitive. Despite the superior features that are obtained in Therm-O-Tile, it is nevertheless competitive in total first cost.

Manufacturers - Engineers - Contractors. Write or call the nearest Porter-Hayden or Johns-Manville Technical Service Unit for recommendations, estimates of cost, and complete specifications for the conduit and insulation on any underground pipe line project.



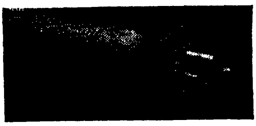
The Ric-wiL Company

PREFABRICATED INSULATED PIPING SYSTEMS

UNDERGROUND OR OVERHEAD

Union Commerce Bldg., Cleveland, Ohio • **Agents in Principal Cities**

Ric-wiL produces complete insulated piping systems designed and engineered to specific operating requirements for transmission (with the lowest possible thermal loss) of steam, hot or refrigerated liquids or gases.

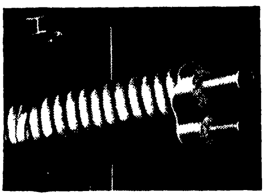


HEL-COR Insulated Pipe Unit—for Underground Lines

Prefabricated units with single or any specified combination of pipes, in complete sealed 21-ft sections Helical corrugated conduit, asphalt and phenolic-resin coated, wrapped with asphalt saturated asbestos felt.

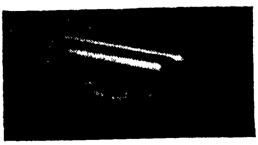
Cast-Iron Insulated Pipe Units

Heavy, corrosion-resistant cast iron is used as the protective housing for this pre-assembled underground system. Std. units are 18 ft, 6 in. long, with all accessories furnished. Units are connected with standard mechanical joint, solid sleeves, or flanges.



HEL-COR Insulated Pipe Unit—for Overhead Lines

Pipe and insulation are housed within a 16-gage spirally corrugated ingot iron conduit, hot-dip zinc galvanized and coated on the inside with a corrosion-resistant phenolic resin. Available in single or multiple pipe systems, prefabricated in 21-ft lengths with all joint materials and accessories.



Standard Tile Conduit (Sectional)

Vitrified and glazed *A.S.T.M.* Standard Tile housing—acid and weather proof. Foundation type drain supporting pipe with correctly engineered pipe support. For single or multiple pipe system—sectional pipe covering or filler type insulation. 2 ft standard lengths.

Super-Tile Conduit (Sectional)

Similar to Standard Tile but with double-strength walls for heavy overhead loads. Will support static load of 6 tons per wheel under actual installed conditions. Heavy duty tile base drain.

Cast-Iron Conduit (Sectional)

Heavy duty reinforced cast iron conduit for use in shallow underground installation close to or under rail traffic. Durable watertight vibration proof clamps insure tightness. 4 ft lengths.

Tile Conduit-Universal Type (Sectional)

For use where installation conditions require concrete pads. Side walls are double-cell vitrified trapezoidal design. Arch may be Standard Tile, Super Tile or Cast Iron.

For full technical information on Ric-wiL products and services, call or write the Ric-wiL office nearest you or Dept. 16-Z in Cleveland, Ohio.

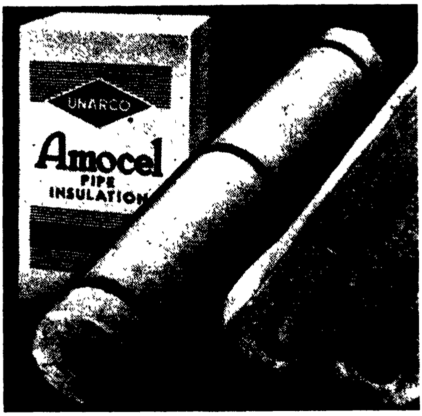
Union Asbestos & Rubber Company

332 South Michigan Ave., **Chicago 4, Illinois**

AMOCEL AND

UNIBESTOS

INSULATIONS



For Temperatures up to 1200 F
For convenience, high thermal efficiency, and the economies of single-layer appli cation, always specify "Unarco." Three great insulations to serve you, all made from Amosite—the strong, light, long-fibre asbestos that withstands moisture, heat, acid fumes; has high structural strength and impact resistance.

For temperatures to 750 F, Use Unarco UNIBESTOS NO. 750

For temperatures to 1200 F, Use Unarco UNIBESTOS NO. 1200

Both types of Unibestos are regularly furnished in 3-foot lengths as cylinders or half-rounds in popular thicknesses to 4 in. for pipe from ½ in. to 24 in. Greater thicknesses and larger sizes available to 44 in. O.D. Unibestos 1200 is also made in blocks 36 in. long; widths to 36 in. in 6 in. increments; thickness to 3 in.

For temperatures to 600 F, Use Unarco AMOCEL, made for pipes from ½ in. to 12 in. Furnished in 3-ft half-rounds with jackets and bands.

Unibestos and Amocel are easy to cut and to fit; may be removed and reapplied repeatedly without loss of thermal efficiency. Carried in stock in principal industrial centers from coast to coast.

Write for quotations and literature.

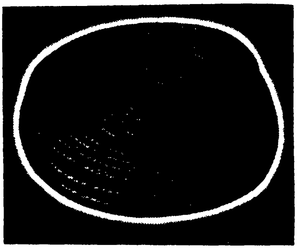
A COMPLETE INSULATION SERVICE

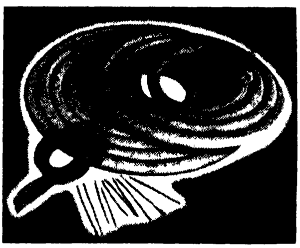
Unarco insulations also include high-temperature cements; tailored-to-fit insulations of asbestos, glass fibres, glass cloth; and flexible insulations which wrap-on, lace-on, slip-on—all made at Unarco plants in Illinois, New Jersey, North Carolina, and Texas. Technical data and recommendations on request.

Union Asbestos & Rubber Company

332 South Michigan Avenue
Chicago 4, Illinois

PACKINGS, GASKETS, AND TEXTILES





Unarco Asbestos Packings, Gaskets, and Textiles comprise a wide variety of types, forms, and sizes for precise, economical selection for any job.

HIGH-PRESSURE PACKINGS. Spiral, coil, or ring-form. Plain or semi-metallic asbestos. With or without a core of red rubber. $\frac{1}{8}$ in. through $\frac{1}{2}$ in., in increments of $\frac{1}{16}$ in. Good for 500 F; pressure to 300 psi.

BRAIDED PACKING. Plain or wire-inserted asbestos yarns. Plaited, or braid-over-braid. Unimpregnated, or with lubricant or coating for use with air, hot or cold water, oil, weak acids, steam. $\frac{1}{8}$ in. through 2 in. square, in increments of $\frac{1}{16}$ in.

TWISTED AND BRAIDED PACKING. For valve stems: long-fibre or wire-inserted asbestos yarns individually lubricated or graphited. Twisted, $\frac{1}{16}$ in. through $\frac{1}{2}$ in. Round braid-over-braid, $\frac{1}{8}$ in. through 1 in. For gasoline: as above, but specially treated, and also made in square $\frac{1}{8}$ in. through 1 in.

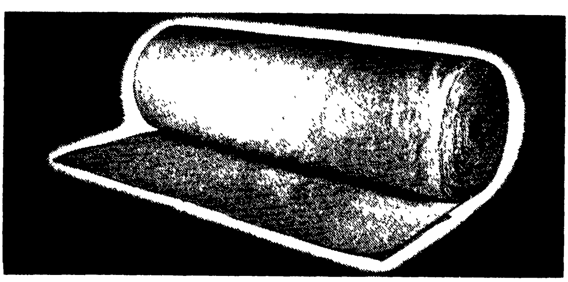
TWISTED ASBESTOS ROPE. High-grade asbestos rovings. $\frac{1}{4}$ in. through 2 in.

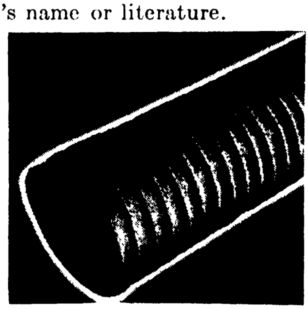
SHEET PACKINGS. Compressed, graphited, asbestos fibre sheets for high pressures and temperatures. 50 in. x 50 in. Thickness, $\frac{1}{64}$ in. through $\frac{1}{8}$ in. Also, cloth woven from asbestos-metallic yarns, coated. 40 in. wide. Thickness, $\frac{1}{32}$ in. through $\frac{1}{8}$ in.

UNARCO ASBESTOS GASKETS AND TEXTILES

High-pressure Manhole and Handhole Gaskets: cut and folded from closely woven wire-inserted asbestos yarns Treated for heat-resistance. Regular shapes. Oval or round. $\frac{1}{8}$ in. through $\frac{1}{2}$ in. Asbestos-Lead Tubular Gasketing: asbestos-metallic cloth around a hollow lead tube. Optional dimensions. Versatile Asbestos-Metallic Gasketing Tape: widths, $\frac{1}{2}$ in. through 3 in.; thickness, $\frac{1}{8}$ in. through $\frac{1}{2}$ in.

Unarco Textile Products include: fibre; yarn; cord; tape, plain or brass-wire-inserted; tubing; and woven cloth—for a multitude of uses. Write for nearest distributor's name or literature.





"Z" crete

Division Zonolite Company

135 S. LaSalle St.
Chicago 3, Ill.

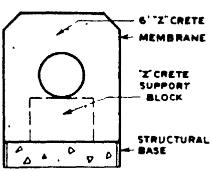
UNDERGROUND PIPE INSULATION

Z-CRETE* is a Lightweight Insulating Concrete which is poured directly and monolithically around heated underground piping.

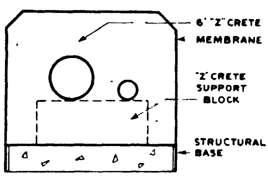
GENERAL DATA The improved Z-CRETE system of underground insulation is a field fabrication adaptable to any size, number or arrangement of pipes. Six inches of insulation is ordinarily used on the outside of all pipes. A minimum of four inches is economical spacing between piping.

Z-CRETE insulation serves as a monolithic, jointless insulating filler around pipes. A curing and waterproofing membrane completes the installation.

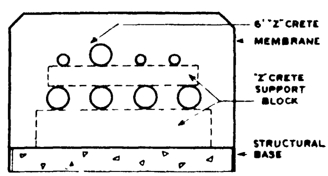
Z-CRETE installations have no joints to leak heat or collect water—they are as permanent as the earth itself. Anchors, guides and expansion devices (except loops) are the same as used in conventional systems. The curing membrane provides external water protection and imparts a tensile strength to the conduit.



Single Pipe Conduit



Two Pipe Conduit



Multiple Pipe Conduit

TYPICAL Z-CRETE CONDUITS

Commonly used installations of Z-CRETE are shown above. The single pipe and double pipe conduit consist of a structural concrete pad, pre-cast Z-CRETE support blocks on which the pipes rest, a pour of Z-CRETE insulating concrete and a curing and waterproofing membrane. The multiple pipe conduit at the right has a second pre-cast support for the upper group of pipes.

REINSULATION OF EXISTING CONDUITS

The ruinous effect of water on steel pipe and insulation is well known. Z-CRETE insulation restricts moisture, keeps the conduit dry. Inorganic Z-CRETE concrete naturally resists deterioration.

Most open boxes or voids may be filled with Z-CRETE insulation without using rollers, rods, etc. The mass of water resistant Z-CRETE restricts the movement of moisture, so special drainage provisions are not ordinarily required. Z-CRETE is sold and installed by licensed applicators of Zonolite Company under U. S. Patent No. 2355966—Canadian Patent No. 439356. There is an applicator near you.

For further data or information, write Dept. HVG-2.

***Z-Crete is a registered trade mark of Zonolite Company.**

April Showers Company, Inc.

4126 Eighth Street, N.W. **Washington 11, D. C**

RIGHT AS RAIN

April Showers

PATENTED AUTOMATIC ROOF COOLING

(Trade Mark Reg. U. S. Pat. Off.)

AUTOMATIC EVAPORATIVE ROOF COOLING

Distributors and Dealers in Principal Cities

ROOF

COOLING

Spray Method

GREATER COMFORT AND BETTER WORKING EFFICIENCY WITH ECONOMICALLY WATER COOLED ROOFS

The advantages in preventing Solar Infiltration by SURFACE COOLING, are today widely recognized by economy-minded, efficiency-wise Architects, Air Conditioning Engineers and Top Management.

The APRIL SHOWERS System is fully automatic. The Sun operates it. Requires very little water and City water, under normal city pressure, is usually adequate. No runoff. Installation and Maintenance Costs are low. Only finest materials are used. Precision workmanship throughout.

APRIL SHOWERS are now in use on 7,860,000 sq ft of Industrial Roofs

A few of these installations are:—Country Life Press, Garden City, L. I., Hallicrafters Co., Chicago, Ill., Lilly Tulip Cup Co., plant (Westinghouse air conditioned), Augusta, Ga., Aerojet Engineering Corp., Azuza, Cal., Westinghouse Electric Co., Hyde Park, Mass., General Electric Corp., Providence, R. I., Bulova Watch Co., Providence, R. I.

Recent installations include:—Westinghouse Electric Corp., Baltimore, Md., 33,480 sq ft, Eastman Kodak Company, Rochester, N. Y., 156,000 sq ft, The Jacobs Mfg. Co., West Hartford, Conn., 85,760 sq ft, Weldon, Williams Lick, Ft. Smith, Ark., 38,000 sq ft, Lily Tulip Cup Corp., Springfield, Mo., 270,000 sq ft, Dixie Cup Company, Easton, Pa., 25,800 sq ft, City of L. A., Pierce Agricultural College, Cal., 20,000 sq ft, Val-O-Will Farms, Lake Geneva, Wis., 55,000 sq ft.

In addition, the Federal Government has over one million square feet of April Showers installations. Among them is the 500,000 sq ft roof the U. S. Naval Ordnance building in Indianapolis, Ind.

APRIL SHOWERS also works efficiently in conjunction with any True air condition system by reducing (in cases where requirements necessitate the use of an exorbitant tonnage) the size of the True system necessary to give the required temperatures for Comfort and Efficient Working Conditions. At the same time, it helps by reducing operating and maintenance costs. Moreover, it adds years of life to the roof itself by preventing the sun from more speedily evaporating the protective roof-coating oils.

Consider APRIL SHOWERS for your next project or remodeling job. We will be pleased to work with you on any problems that may arise. NOTICE:—Our newly developed Spray Head, for use on Dwellings with flat or peaked roofs, are now in distribution.

APRIL SHOWERS controlled roof cooling is protected by U.S. Patents.

Write for Descriptive Literature *No obligation for Estimates*

A FEW CHOICE DEALERSHIPS STILL AVAILABLE . . . WRITE

American Structural Products Company

Toledo 1, Ohio—Subsidiary of Owens-Illinois Glass Company

INSULUX GLASS BLOCK

Insulux Glass Block Give Better Control of Interior Conditions

What they are—Insulux Glass Blocks* are hollow, hermetically sealed units containing a partial vacuum. Properly used they aid control of interior conditions to a point where initial and operating costs of heating or cooling equipment are reduced.

Conductivity

Coefficient of Heat Transmission—The "U" factors for panels of Insulux Glass Block are as follows:

Nominal Block Size	"U"
6″ sq.	0.60
8″ sq.	0.56
8″ (with glass fiber screen)	0.48

See page 197 of this GUIDE for complete data.

The "U" factor for a panel of 8 in. glass block is just one-half of that for single glazing. The reason lies in the two heavy glass surfaces separated by partially evacuated and hermetically sealed dead air space.

Surface Condensation

Because of the low over-all, air-to-air heat transfer, the exterior air temperature which will produce condensation on a glass block panel is much lower than that for ordinary windows. This permits higher humidities where needed, for air conditioning for both comfort and industrial processes. Being glass, they cannot rust, rot, nor corrode . . . are not subject to deterioration caused by moisture.

Infiltration

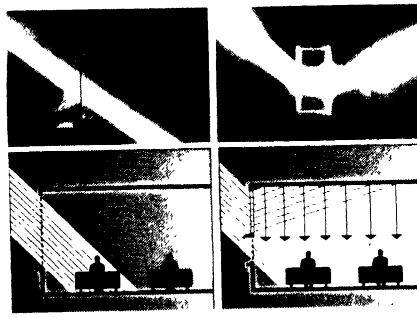
A panel of Insulux Glass Block provides a wall of glass and mortar which seals the building against infiltration. Dust, drafts and air and vapor leakage are minimized. Natural ventilation requirements can be met by installing windows, either inset into the panels or directly below or above the panels.

Solar Heat Gain

A comparative test showed over two times as much solar heat through steel sash as through glass block panels. However, as with sash, glass block transmit less solar heat when properly oriented and shaded. Complete instantaneous solar heat gain data for all block designs and for all exposures are given on pages 294 and 295 of this "GUIDE."

Design, Sizes, Erection

Insulux Glass Block are made in a variety of face designs that distribute light and limit sight in varying degrees. They are made in three standard sizes, 6 x 6, 8 x 8, 12 x 12 in. Actually each of these dimensions is a quarter of an inch less to allow for mortar joint. All blocks have a standard thickness of 3⅞ in. Complete technical data, description, and details will be gladly sent. Just address American Structural Products Company, Box 1035, Toledo 1, Ohio.



Photographs and illustrations above show how Insulux Glass Block effectively controls daylight. To the left you see what happens when light beams strike an ordinary window. Notice uncomfortable, harsh brightness near windows, extreme contrast in other parts of room. To the right, notice how the built-in prisms in Insulux Glass Block No. 363 throw light UP, and spread it. Result is even, diffused light over all parts of the room.

* Reg. U. S. Pat. Off.

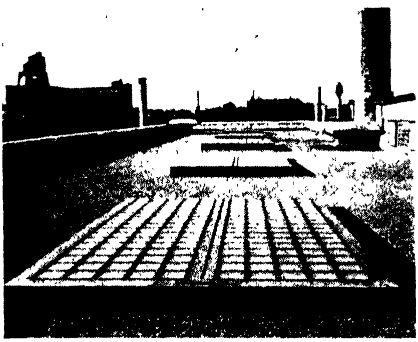
American 3 Way-Luxfer Prism Co.

431 S. Dearborn St.,
Chicago 5, Ill.

3-WAY Daylighting Products

270 Park Ave.,
New York 17, N. Y.

Products: **American 3-Way Rooflights** made of glass blocks of special strength and design in 3 arrangements for adaptation to all types of skylights in commercial, industrial, and institutional buildings. American Skylights and Skylight Ventilators for special applications. American Magnalite Diffusing Glass Blocks for skylights, and for doors, ceiling lights, screens and partitions.



American Glass Block Skylight

Benefits: **American 3-Way Rooflights** make possible many benefits, including maximum diffusion of soft light over greater area, increased structural attractiveness, easy cleaning, long life, dust and air resistances, lower heat transfer, and reduction of solar heat transmission.

Construction: **American 3-Way Rooflights** use specially designed, semi-vacuum glass blocks approximately 9 in. sq, 2½ in. thick, set in a 3⅞ in. thick, reinforced concrete grid. Each glass block is made of water-white crystal glass, designed solely for skylight use. Elements incorporated assure maximum light diffusion. Each glass block is sealed in place with permanent Tees-Ess compound applied in fluid form at about 280 F, insuring homogeneous, weatherproof seal. Non-ferrous metal reglets set and anchored in the reinforced concrete grids around the outer rows of glass blocks provide arrangement for weather-tight flashing connections with any type of roofing as per standard flashing detail.

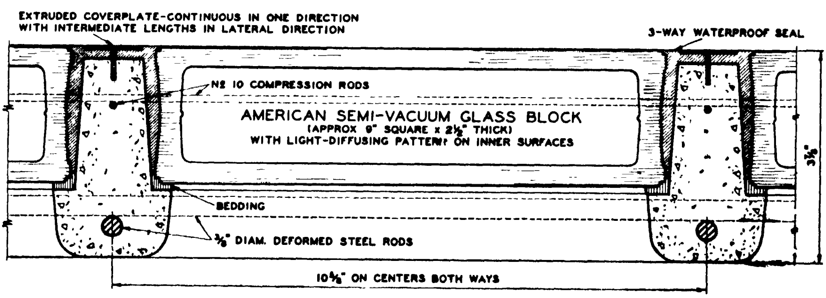
IDEAL FOR AIR CONDITIONED BUILDINGS

Reduces Heat Transfer: Tests using methods suggested by the A.S.H.V.E. Conductivity Test Code show that Glass Block Rooflights have about two-and-one-half times the insulating value of sheet metal skylights.

Reduces Solar Heat Transmission: Reduction in total solar heat gain as compared with ordinary windows is indicated by relative values given in Tables 23, 24, and 27 in Chapter 12.

Reduces Condensation: Due to the nature of the grid construction where insulating materials are employed with semi-vacuum glass blocks assemblies, there is little or no tendency for condensation to form on the underside.

Light Up—the **AMERICAN** ***way!***



Pittsburgh Corning Corporation

Room U52, 307 Fourth Avenue, **Pittsburgh 22, Pa.**

PC GLASS BLOCKS

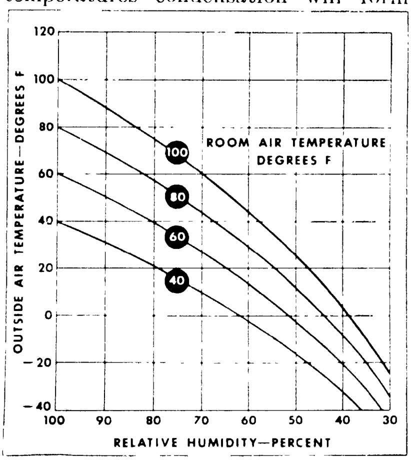
Distribution by Pittsburgh Plate Glass Company; W. P. Fuller & Company on the Pacific Coast; Hobbs Glass Ltd. in Canada; and by leading distributors of building materials everywhere. Also makers of FOAMGLAS.

THERMAL INSULATION

PC Glass Blocks allow the economical use of large glass lighting areas, reduce heat loss in cold weather and materially aid air-conditioning. This is because each PC Glass Block contains a sealed-in dead-air space that is an effective retardant to heat transfer. Tests by nationally recognized laboratories have established the value of glass blocks for insulation. See pages 294 and 295 of this Guide.

SURFACE CONDENSATION

Due to high insulating value, condensation will not start forming on the room side of glass block panels until outside air has reached a temperature much lower than that necessary to produce condensation on single-glazed windows. The accompanying chart shows at what temperatures condensation will form.



Outdoor temperature required to produce condensation on the room side surface of PC Glass Block panels.

For example, with inside air at 70° F and relative humidity at 40 per cent condensation will not begin to form on the interior surfaces of a panel of single cavity glass blocks until an outdoor temperature of minus 14° F is reached. Under similar conditions, with single-glazed sash, moisture will begin to form when the outdoor temperature reaches 33° F. PC double cavity blocks (LX patterns)—in which a fibrous glass screen is inserted between the halves of the block—provide even better insulation value, with less chance for condensation even at high temperature and humidity levels.

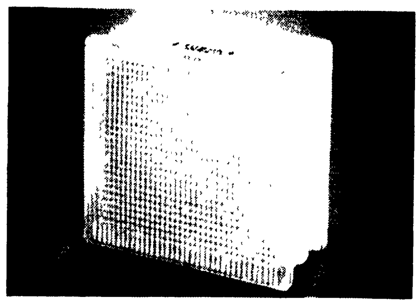
SOLAR HEAT GAIN

The use of glass blocks for light-transmitting areas results in a marked reduction in total solar heat gain as compared with ordinary windows. This factor is of considerable advantage in buildings that are properly air-conditioned, but does not eliminate the need for adequate ventilation or shading in non-air-conditioned rooms.

For data on solar heat gain through glass blocks see tables 23 and 24 in the solar radiation section of this Guide—chapter 12. The tables are for standard pattern glass blocks.

PC GLASS BLOCKS AID AIR-CONDITIONING

Two of the chief aims of air-conditioning—temperature control and cleansing of air—are aided by the use of PC Glass Blocks. Heat loss is less in winter—heat gain is less in summer. Solar heat transmission and radiation are reduced. Neither dirt nor drafts can filter in, for each panel is a tightly sealed unit.



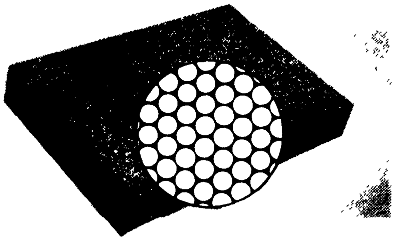
PATTERNS, SIZES, INSTALLATION

PC Glass Blocks are available in decorative and functional patterns—the latter designed for special control and direction of transmitted daylight. They are made in three sizes: 5¾ in. x 5¾ in., 7¾ in. x 7¾ in. and 11¾ in. x 11¾ in. (generally referred to as 6 in., 8 in. and 12 in.). All are 3⅞ in. thick. Special shapes are available for turning corners and for building curved panels. Any mason can install PC Glass Blocks; no special tools are required. For complete information, write the Pittsburgh Corning Corporation, Pittsburgh 22, Pa.

Pittsburgh Corning Corporation

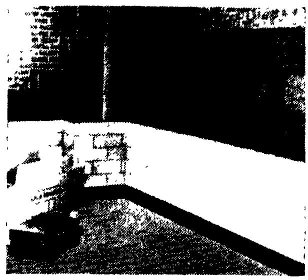
Room T52, 307 Fourth Avenue **Pittsburgh 22, Pa.**

FOAMGLAS—***the long life insulation***

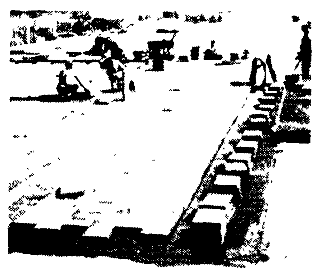


This is FOAMGLAS®. The entire strong, rigid block is composed of millions of sealed glass bubbles. They form a continuous structure which has unusually high resistance to moisture, vapor and acid atmospheres, is incombustible, vermin-proof and odorless. In those closed glass cells, which contain still air, lies the secret of the material's long life insulating efficiency.

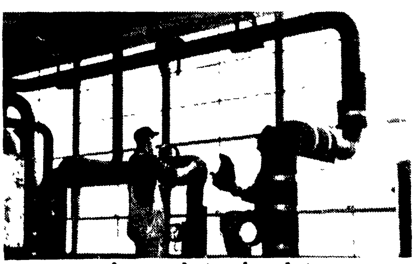
In FOAMGLAS you find a unique combination of properties which make it a truly effective and economical insulation—whether for buildings or for indoor and outdoor pipe lines and equipment. FOAMGLAS is a cellular glass material which effectively retards heat travel wherever it is used. Its exceptionally high resistance to moisture, vapor and many other destructive elements enables FOAMGLAS to *retain* its original insulating efficiency. Freedom from repairs, maintenance and replacement keep FOAMGLAS insulating costs low. You can get complete, up-to-date information on FOAMGLAS in our current literature. We shall be glad to mail you *free* copies of our booklets—and a sample of the material.



On new and existing walls, and as free-standing partitions, FOAMGLAS supports its own weight. It remains in place . . . insures a long lasting barrier to vapor transfer and heat and cold.



On roof decks, rigid blocks of FOAMGLAS provide a firm level base for roofing felts. The long life insulation value of FOAMGLAS can be depended upon to help maintain desired indoor temperatures and humidities.



FOAMGLAS is used to insulate process equipment and pipe lines. It is available in standard flat blocks and curved segments to fit equipment, and in preformed sections for standard pipe sizes and fittings.

FOAMGLAS used under concrete wearing floors—or around the edge of floors on ground—reduces heat loss, increases comfort. Its high compressive strength supports heavier than normal floor loads.

When you insulate with FOAMGLAS, the insulation lasts!

Armstrong Cork Company

Building Materials Division

Lancaster **Pennsylvania**

Offices

ALBANY	CINCINNATI	HOUSTON	MINNEAPOLIS	ROCHESTER
ALLENTOWN	CLEVELAND	INDIANAPOLIS	NEW ORLEANS	ST. LOUIS
ANCHORAGE	COLUMBUS	JACKSONVILLE	NEW YORK	SAN FRANCISCO
ATLANTA	DALLAS	KANSAS CITY	OMAHA	SEATTLE
BALTIMORE	DENVER	LOS ANGELES	PHILADELPHIA	SPOKANE
BIRMINGHAM	DETROIT	LOUISVILLE	PITTSBURGH	SYRACUSE
BOSTON	EVANSVILLE	MEMPHIS	PORTLAND	TACOMA
BUFFALO	HARRISBURG	MILWAUKEE	PROVIDENCE	TULSA
CHARLOTTE	HARTFORD		RICHMOND	WASHINGTON, D. C.
CHICAGO				WILMINGTON

Distributors

CHARLESTON 23, W. VA. Capital City Supply Co.
EAU CLAIRE, WIS Horel-George Co.
EL PASO, TEXAS Case Industrial Service Company
FORT WAYNE, IND. . . The Baldus Co.
GRAND RAPIDS, MICH. Tony Batenburg Insulation Co.
GREEN BAY, WIS. Northwestern Asbestos and Cork Insulation Co.
JAMESTOWN, N. Y. Laco Roofing & Asbestos Co.
JOPLIN, MO. Joplin Cement Co.
MANITOWOC, WIS. Northwestern Asbestos and Cork Insulation Co.
PHOENIX, ARIZ. Barrett & Holmes
SAN ANTONIO, TEX. . General Supply Co., Inc.
SOUTH BEND 23, IND.... Midland Engineering Co
SALT LAKE CITY, UTAH Asbestos Engineering & Supply Co.
SPRINGFIELD, MO.. Southwestern Insulation Co.

Armstrong's Contract Service

Armstrong's Contract Service provides trained engineers, supervisors, and installation crews that are thoroughly experienced in the application of both high- and low-temperature insulations to ducts, piping, and equipment. Backed by more than 40 years of service in the insulation field, this nation-wide organization will see that your job is done right from start to finish. Your nearest Armstrong office will furnish prompt estimates and technical assistance without obligation.

The following products are available for installation by Armstrong's Contract Service or for your own application:

Armstrong's Corkboard

Armstrong's Corkboard is the standard low-temperature insulation. Properly applied to ducts and other air-conditioning apparatus, it will greatly improve the over-all thermal efficiency and the operating characteristics of the system. It will also prevent moisture condensation and consequent drip from air-conditioning ducts and equipment surfaces.

The exceptional moisture resistance and durability of Armstrong's Corkboard assure lasting insulating efficiency. Strong and light in weight, this material is easy to handle on the job. It is readily cut and worked with ordinary tools and may be shaped to fit curved surfaces. It provides a firm bond with all conventional finishes. Conforms to Federal Specification HH-C-561b. Sizes: 36 in. long; 12 in., 18 in., 24 in., and 36 in. wide; 1 in., 1½ in., 2 in., 3 in., 4 in, and 6 in. thick. *K factor:* 0.27.

Armstrong's Cork Covering

Armstrong's Cork Covering has the same high insulating efficiency and general characteristics as Armstrong's Corkboard. The use of Armstrong's Cork Covering on cold lines prevents from 80 to more than 90 per cent of the refrigeration loss occurring when lines are left uninsulated. Pipe and fitting covers are machined to accurate size and finished with a heavy mastic coating which provides a seal against air and moisture penetration. Pipe covering is made in 36 in. half-sections to fit all standard pipes and tubes from ¼ in. o.d. up. Fitting covers are made for all sizes of valves, elbows, tees, and other fittings. Covers for special sizes and shapes made to specification on request. *Three thicknesses:* Light Duty Thickness (1.2 in. to 1.93 in.) for temperatures from 35 F up; Standard Thickness (1.7 in. to 3.5 in.) for temperatures from 0 F to 35 F; Heavy Duty Thickness (2.63 in. to 4.19 in.) for temperatures from −25 F to 0 F.

Heat Insulations

In most areas, Armstrong is the sole distributor of Keasbey & Mattison's complete line of heat insulations. These materials are available for direct sale or as installed by Armstrong's Contract Service. They include 85 per cent magnesia block and pipe covering, air cell block, sheet, and pipe covering, high-temperature block and pipe covering wool felt, hair felt, etc. All are available in standard sizes and thicknesses.

For detailed technical information, samples, and descriptive literature, ask any office or distributor Specifications appear in Sweet's Catalogs for Architects and Engineers.

Mundet Cork Corporation

7105 Tonnelle Ave. **INSULATION DIVISION** **North Bergen, N. J.**

Manufacturers of Corkboard, Cork Pipe Covering, Compressed Machinery Isolation Cork, Natural Cork Isolation Mats, and all kinds and varieties of Cork Specialties, also 85 per cent Magnesia Insulation in pipe covering & block form.

Complete Insulation Services for High and Low Temperature.

Mundet Branches

Atlanta, Ga.			New Orleans 16, La.
Baltimore 30, Md.			New York 17, N.Y.
Boston (No. Cambridge) 40	Dallas 10, Tex.	Jacksonville 6, Fla.	Philadelphia 39, Pa.
Charlotte, 6, N.C.	Detroit 21, Mich.	Kansas City 7, Mo.	St. Louis 9, Mo.
Chicago Ill.	Houston 1, Tex.	Knoxville 16, Tenn.	San Francisco 7, Calif.
Cincinnati 2, Ohio	Indianapolis 4, Ind.	Los Angeles (Maywood)	Washington, D.C.

Mundet Distributors are Located in the Following Cities—Names and Addresses on Request

ARIZONA	Phoenix, Tucson	MONTANA	Anaconda	TEXAS	El Paso
COLORADO	Denver	OHIO	Toledo	UTAH	Salt Lake City
CONNECTICUT	Hartford	OKLAHOMA	Oklahoma City	VIRGINIA	Norfolk, Richmond
D. C	Washington	OREGON	Portland	WASHINGTON	Seattle, Tacoma
IOWA	Amana	RHODE ISLAND	Providence		
MINNESOTA	Minneapolis	SOUTH DAKOTA	Brookings	W. VIRGINIA	Charleston
MARYLAND	Baltimore	TENNESSEE	Johnson City, Knoxville, Memphis, Nashville	WISCONSIN	Appleton

NEW YORK, Averill Park, Buffalo, Plattsburg, Rochester, Utica, Westbury, L. I.

Natural Cork—Cork in its natural state consists of minute hermetically sealed cells containing "dead" air. Approximately 200,000,000 cells per cubic inch. Cell walls are resinous, resilient, and impervious to the passage of air. There is no "free" air to conduct heat or moisture and no capillary attraction.

LOW TEMPERATURE INSULATION

Mundet "Jointite" Corkboard

Natural cork is ground into 1/4 in. to 5/8 in. granules and compressed under heat in moulds to produce Mundet flat or shaped corkboard. Air spaces between granules are eliminated by the pressure and the milled resin in the cell walls cements the mass into a homogeneous structure retaining the properties of natural cork.

Section of Mundet Moulded Cork Pipe Covering with Fitting. The pipe covering is made in sections 36 in. long, to fit all sizes of pipe.

Mundet Corkboard meets U. S. Government Master Specifications. Its heat transmission is guaranteed not to exceed .29 Btu when tested in accordance with Bureau of Standards regulations. In actual cold storage practice, this figure may be safely reduced to .27 Btu. Sold in standard 12 in. x 36 in. sheet. Standard thicknesses, 1/2 in., 1 in., 1 1/2 in., 2 in., 3 in., 4 in., 6 in.

Mundet "Jointite" Cork Pipe Covering

Protects all types of low temperature lines. Made in 3 thicknesses, with complete line of standard covers, suitable for pipes carrying sub-zero to 50 F temperature.

HEAT INSULATION

85% Custom-Molded Magnesia

The new Mundet plant for the manufacture of 85 per cent magnesia insulation makes available the most modern plant facilities for the production of heat insulation in pipe covering and block forms.

Mundet Cork Vibration Isolation

Machinery vibration encountered in heating and ventilating work is effectively controlled by the use of Mundet Natural Cork Isolation Mats. We also manufacture sheet isolation cork for heavier machinery loads.

Engineering and Specification Service

Our engineering department is at the service of Architects and Engineers, to assist and advise in the preparation of specifications. This service is available without obligation.

Mundet Contract Service

Covers the complete installation of our products, in accordance with best established practice. Divided responsibility is avoided. Materials and workmanship are guaranteed. Send for catalog.

The Celotex Corporation

General Offices

120 South LaSalle Street, **Chicago 3, Ill.**

CELOTEX

REG U. S PAT. OFF

Celotex Insulation Board Products are made by felting long, tough cane fibres into strong, rigid boards. Manufactured under the patented Ferox* Process. Ferox-treated Celotex board has been demonstrated by laboratory tests and years of use to be protected effectively against dry rot and termite attack. Integrally waterproofed.

Celotex Insulating Sheathing—For use in frame construction under wood siding, wood or asbestos shingles, stucco, or masonry veneer. *Double-Waterproofed*—integrally treated, then asphalt-coated on all surfaces. Exceeds government vapor permeability requirements.

Sizes: 4 ft wide x 8, 9, 10, 12 ft long x 25/32 in. or 1/2 in. with square edges. 2 ft x 8 ft x 25/32 in. with V-type tongue and groove on long edges.

Celotex 4 ft wide 25/32 in. Sheathing, applied vertically without corner bracing, greatly exceeds racking strength requirements set forth in FHA Technical Circular No. 12 (racking strength at least equal to horizontally-applied wood sheathing with let-in bracing is minimum requirement).

Celotex Insulating Lath—Insulation and continuous plaster base in one material. All edges beveled for additional plaster reinforcement at joints. Long edges shiplapped. Sizes: Regular or Vapor-Seal, 18 in. x 48 in. x ½ in.

Celotex Roof Insulation

Regular—Natural cane fibre board surface. Provides excellent bond for pitch or asphalt. Meets requirements for Roof Insulation Board (Class C) of Federal Specifications LLL-F-321b, Commercial Standard CS42-49 of U. S. Department of Commerce, and *ASTM* Specification C208-48. Sizes: 23 in. x 47 in., 24 in. x 48 in. Thicknesses: ½ in., 1 in., 1½ in., 2 in.

Preseal—Coated with special asphalt on all surfaces and edges for additional moisture protection. Sizes: 23 in. x 47 in., 24 in. x 48 in. Thicknesses: ½ in., 1 in., 1½ in., 2 in.

Preseal "30"—Extra high quality board with conductance "C" before asphalt coating of 0.30 Btu per inch nominal thickness. Same moisture-resistant asphalt coating as Preseal. Size: 24 in. x 48 in. Thicknesses: 1 in., 1½ in., 2 in.

Vapor-Seal—Has ½ in. x ½ in. offsets on all edges which form network of channels next to deck, serving to equalize air pressure therein. Asphalt-coated on all surfaces and edges. Thermal conductance, before coating, is 0.30 Btu per inch nominal thickness. Size: 24 in. x 48 in. Thicknesses: 1 in., 1½ in., 2 in.

Flexcell* Expansion Joint Filler—Cane fibre felted into strong, resilient boards, then saturated with durable asphaltic compound. For expansion joint uses, perimeter insulation for concrete floors at grade, sill sealer, plate sealer, and vibration isolation. Thicknesses of ¼ in. to 1 in. in various lengths and widths.

Cemesto* Structural Insulating Panels—Completely fabricated panels for walls, roof decks and partitions. Consists of cane fibre board core surfaced on both sides with layer of asbestos-cement board. Sizes: 4 ft wide x 4 ft to 12 ft lengths. Special sizes available. Thicknesses: 11/16 in., 1⅛ in., 1 9/16 in., 2 in.

Celo-Block*—Cold storage insulation made by laminating ½ in. low density cane fibre boards with waterproof mastic and surfacing front and back with waterproofing asphalt. Sizes: 12 in. x 36 in., 18 in. x 36 in. Thicknesses: 2 in., 3 in.

Celotex Rock Wool Products—Regular Blankets, full- or semi-thick, 15 in. x 96 in., 15 in. x 48 in., 15 in. x 24 in. Utility Blanket, 15 in. x 96 in. Reflective Blankets, standard thickness, 15 in. x 96 in. Also Loose, Granulated and Hand-Pouring Rock Wool.

Q-T* Ductliner—Sound absorbing material designed for duct lining in air-conditioning systems. Made of rock wool and special binder. Withstands air duct humidity, is fire-resistant and will not support combustion. Thermal conductivity of 0.30.

Celotex Sound-Conditioning Products—Complete line of specialized acoustical materials to comply with every requirement, specification, or building code. Distributors in principal cities.

Celotex Interior Finishes—Tile Board, Finish Plank, Building Board—triple-duty products that build, decorate, insulate. Variety of sizes and finishes.

* Reg. U. S. Pat. Off.

For detailed information on all Celotex products, see Sweet's Files or write The Celotex Corporation

Insul-Mastic Corporation
OF AMERICA

1162 Oliver Building **Pittsburgh 22, Pa.**

Representatives in Principal Cities

INSULATION—CONDENSATION PREVENTION VAPORSEALING INSULATION—CORROSION PREVENTION

INSUL-MASTIC—A highly viscous, semi-plastic material for the performance of the above functions. It is spray applied; adheres to surface at all angles; and, when dry, remains flexible but extremely tough.

COMPOSITION—Insul-Mastic was developed with the idea of topmost quality. Therefore, Gilsonite, or "mineral rubber" was used as the basic material. This thoroughly saturated hydrocarbon is almost chemically inert and extremely hard to displace by chemical reaction with acids, alkalis or weathering. To this is added a proper balance of high grade asphalt. This combination assures maximum service at extreme temperatures. For long life and ease of application, three important fillers are used in Insul-Mastic; Mica Flake, Asbestos Fibre and Ceramic Clay. See *National Bureau of Standards* report showing proof of excellent results with mica flake used to increase the life of coatings.

QUALITIES—Extremely resistant to most acids and alkalis. Not affected by temperatures between −40 F and 300 F. Flexible, stands normal expansion, contraction and bending of surface beneath it. Adheres to any dry, dust free surface at any angle including ceilings. Impervious to moisture; the moisture vapor penetration rate per ⅛ in. thickness, per 100 sq in., per 24 hr is 0.01 grams.

APPLICATION—Insul-Mastic is spray applied under heavy air pressure. Insul-Mastic licensees in principal cities have trained crews to do this work. Coatings are applied at the rate of six to eight gal per 100 sq ft for corrosion prevention and vapor-sealing and at the rate of 20-30 gal per 100 sq ft for insulation and condensation prevention. Used as it comes from the drums, no heating required.

APPEARANCE—Insul-Mastic is black, but may be colored with Insul-Mastic aluminum spray or Insul-Mastic colored vinyls. Slate granules of various colors may be blown into the coating while it is wet.

LIFE—Accelerated weather tests identical to that of the *Bureau of Standards* place Insul-Mastic's life at over 50 years in outside weather.

INSULATION—Insul-Mastic Type "D"—To the high quality material described in the opposite column, Insul-Mastic adds 65 to 75 per cent granulated cork. This forms the sprayable insulation known as Insul-Mastic Type "D." This insulation is capable of stopping 65 per cent of heat flow through metal plates. The K factor is 0.36 per sq ft per inch thickness. No mechanical means of attachment are needed. Insul-Mastic Type "D" insulation may be used indoors or outdoors without covering. It also prevents corrosion and deadens sound.

CONDENSATION CONTROL—Insul-Mastic Type "D." Insul-Mastic Type "D" also controls condensation when applied to pipes carrying cold liquids or to ducts and panels subject to chilling. Here again it adheres at any angle without mechanical support. Pullman cars, freight cars, skyscrapers and others employ Type "D" for this purpose.

VAPORSEALING INSULATION—Insul-Mastic Reinforced With Glasfab Membrane—For keeping soft or semi-rigid insulation dry and protected. The system gives the insulation a tough, water repellent covering that is not likely to be broken in spite of the soft material beneath it. Insul-Mastic and Glasfab take the place of asbestos cement and chicken wire which is brittle, requires two coats plus asphalt; and to the insulation adds water which may never be driven off. Insul-Mastic and Glasfab is a flexible jacketing which can withstand abuse such as ladders, foot traffic or falling objects. The Glasfab Membrane is imbedded into a tack coat of Insul-Mastic and then sprayed with a ⅛-in. coating of the mastic.

CORROSION PREVENTION—Installations of all types from small pipes to large tanks can be kept from corroding by a coating of Insul-Mastic. In chemically laden air or constantly moist conditions, this protection is particularly necessary; and Insul-Mastic's record in paper pulp mills, oil refineries and other industries where corrosion was a great problem confirms the durability of this coating.

INSULITE DIVISION • Minnesota and Ontario Paper Company
General Office: **500 Baker Arcade Bldg. Minneapolis 2, Minnesota**

INSULITE®

STRUCTURAL INSULATION BOARD

For 38 years engineers and architects have specified Insulite materials for structural uses, interior finish, and for other thermal insulation and sound control work. Insulite materials have proved their merit through actual performance on the job.

STRUCTURAL MATERIALS

CONDENSATION CONTROL—To prevent condensation within walls, authorities recommend "sealing the warm side and venting the cold side" of the wall. Sealed Lok-Joint Lath provides the necessary vapor barrier on the warm side of the wall, thereby reducing the flow of vapor into inner wall areas. On the cold side of the wall, vapor-permeable Bildrite Sheathing allows surplus vapor to escape toward the outside.

Bildrite Sheathing*—A tough, durable, insulating sheathing material made from new wood fibers. Waterproofed throughout by an integral asphalt treatment. Bildrite (4 ft width) has more than twice the bracing strength of horizontal wood sheathing. 25/32-in. thick. Sizes: 2 x 8 ft. (V-joint on long edges) . . . 4 x 8 ft to 4 x 12 ft. (square edges). Thermal conductivity: 0.36 Btu per inch thickness.

Sealed Lok-Joint Lath*—An insulating plaster base, made from tough Northern wood fibers. Horizontal joints are reinforced by patented metal "Loks." Waterproofed throughout by an integral asphalt treatment. Asphalt vapor-barrier protects against harmful condensation.

Thicknesses: ½, ¾, and 1 in. Size: 18 x 48 in. Also available without vapor seal in ½-in. thickness.

THE INSULITE "WALL OF PROTECTION"

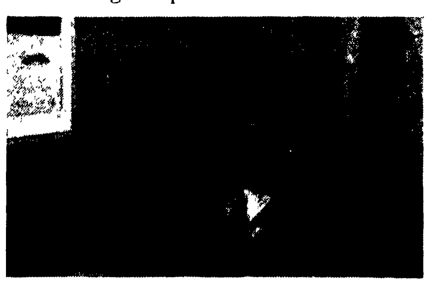
Bildrite Sheathing (outside) and Sealed Lok-Joint Lath (inside) form the approved Insulite "Wall of Protection." Transmission coefficient (U) for this construction with wood siding exterior is 0.15 Btu/hr/sq ft/°F.

This value is typical of the results gained by using Insulite materials in frame construction. For further (U) values see Chapter 9, pages 186 and 187.

SHINGLE-BACKER

Shingle-Backer is a fast applying, insulating under-course material for double-coursed shingled walls. Shingle-Backer's long 4-ft panels take the place of low-grade, wood under-course shingles. Designed to make faster, easier shingled walls, without waste. Made from waterproofed, 5/16-in. insulation board, asphalt-impregnated throughout. No building paper is needed. 5/16 in. thick, 48 in. long. Two widths: 13½ in. for 12-in. shingle exposure; 15½ in. for 14-in. shingle exposure.

Applying Bildrite Sheathing



Applying Lok-Joint Lath

ROOF INSULATION

Insulite Roof Insulation is fabricated from either Ins-Lite or Graylite insulation board. The ½-in. thickness has square edges. The 1, 1½, and 2-in. thicknesses are multiple layers stapled together. Available with either square or offset edges. Size: 23 in. x 47 in.

INSULATING WOOL

Insulite Insulating Wool is made from famous "Fiberglas," consisting of millions of long glass fibers bonded together with a thermo-setting resin. Available in the following forms: roll blankets, batt blankets, utility batts, and pouring wool.

INDUSTRIAL INSULATION

Lowdensite Industrial Board—A 10 to 14 lb density board with average tensile strength of 100 lb/sq in., and average conductivity of 0.31 Btu/hr/sq ft/°F/inch thickness.

Ins-Lite* Industrial Board—A 14 to 18 lb density board with average tensile strength of 250 lb/sq in., and average conductivity of 0.34 Btu/hr/sq ft/°F/inch thickness.

Graylite* Industrial Board—Differs from Lowdensite and Ins-Lite in that it's integrally treated with asphalt,

Applying shingles and Shingle-Backer

which provides increased strength and moisture-resistance. A 16 to 20 lb density board with average tensile strength of 300 lb/sq in., and average conductivity of 0.36 Btu/hr/sq ft/°F/inch thickness.

HARDBOARD PRODUCTS

Insulite HardBoard is a rigid, durable, wood-fiber material with tremendous strength. Available in a range of densities from 55 to 68 lb per cu ft. Thicknesses from ⅛ to 5/16 in. Sizes from 4 x 3 ft to 4 x 12 ft. Many colors and designs available.

INTERIOR FINISHES

Graylite Building Board—A rugged, durable board, integrally treated with asphalt for maximum strength and moisture resistance. Thermal conductivity is 0.36 Btu per in. thickness. Sizes: 4 x 6 ft to 4 x 12 ft. Thicknesses: ½, ¾, and 1 in.

Primed Graylite Building Board—Identical to regular Graylite Building Board, but has a prime coating for easy painting. Sizes: 4 x 6 ft to 4 x 12 ft. Thicknesses: ½ and ¾ in.

Lusterlite Interior Board—A smooth, tough-surfaced board, factory-painted in White or Light Ivory colors. Easy to clean or repaint. Sizes: 4 x 6 ft to 4 x 12 ft. Thicknesses: ½ in.

Durolite Interior Board—Rugged, durable interior board with factory-painted finish. Easily cleaned or repainted. Highly resistant to scuffing and abrasion. Available in Ivory, Pale Green, and variegated Woodtone colors. Sizes: 4 x 6 ft to 4 x 12 ft. Thickness: ½ in.

Smoothlite Interior Board—A natural-colored, factory-coated board with glossy finish. 68 per cent light reflection. Sizes: 4 x 6 ft to 4 x 12 ft. Thickness: ½ in.

Wevelite Interior Board—A practical, low-cost interior board with factory-painted finish in Ivory-White color. Easy to clean and repaint. Sizes: 4 x 6 ft to 4 x 12 ft. Thickness: ½ in.

Tileboard (Lusterlite)—Same smooth, long-lasting finish as Lusterlite Interior Board. Flanged tongue-and-groove joint permits easy, secure fastening with staples, nails, or an adhesive. Sizes: 12 x 12 in , 16 x 16 in., 16 x 32 in. Thickness: ½ in.

Plank (Durolite ½ in. and ¾ in)—Same tough, scuff-resistant surface as Durolite Interior Board. Flanged tongue-and-groove joint permits easy, secure fastening with staples, nails, or an adhesive. Durolite ¾ in. Plank has greater strength and insulation value—requires no furring strips. Colors: Durolite ½ in.—Ivory, Pale Green, and variegated Woodtones . . . Durolite ¾ in.—Ivory, Pale Green, and Light Woodtone. Sizes: Durolite ½ in.—8, 10, 12, and 16 in. wide; 8, 10, and 12 ft long . . . Durolite ¾ in.—16 in. wide and 8 ft long.

Fiberlite Acoustical Tileboard—A low-cost, highly efficient acoustical tile, factory-painted in White. Extremely high light-reflection (80 per cent). Beveled edges on all sides. Sizes: 12 x 12 in., 16 x 16 in., and 16 x 32 in. Thicknesses: ½ and ¾ in.

Acoustilite Perforated Tileboard—A rugged, sound-absorbing tileboard for residential and commercial interiors. Each tile unit contains 484 cleanly-drilled holes. Available with either flanged tongue-and-groove joint (for staple or nail application), or beveled butt-edge joint (for adhesive or nail application). ½ in. thick, 12 x 12 in. square. Factory-painted white finish has high light-reflection.

* Reg. U. S. Pat. Off.

Johns-Manville

Executive Offices: 22 East 40th Street, **New York 16, N. Y.**
Offices in All Large Cities

Home Insulation

Applying Longfibre Super-Felt batts in new home

For Existing Homes and Buildings: J-M Type A "Blown" Rock Wool

J-M Type A Rock Wool is blown pneumatically into the spaces between studs in outer walls and between roof rafters or attic floor joists. Insulation thickness in walls corresponds to stud depth, approximately 3⅝ in. The uniform fill assures maximum thermal efficiency. This type of insulation is installed only by Approved J-M Franchised Home Insulation Contractors, whose trained crews are equipped with the necessary apparatus.

For complete information on J-M "Blown" Rock Wool Home Insulation and its application, write the address above or call your local Johns-Manville office.

For New Construction: Super-Felt* Longfibre Batts and Blankets

J-M Superfelt Batts and Blankets are made of the revolutionary *Longfibre* rock wool. Instead of the short, coarse fibres inherent in other types of mineral wool, J-M produces long, fine fibres. These fibres are felted into batts and blankets that are stronger, lighter in weight, more resilient and with greater uniformity throughout

FUL-THIK BATTS are fabricated of *Longfibre* rock wool to full stud thickness which assures maximum comfort and fuel savings. Each batt has a vapor-seal backing paper with extended tacking flanges for over-lapping at the framing members which protects against passage of abnormal humidity. Furnished in sizes 15 x 24, 15 x 48, 19 x 24, 19 x 48, 23 x 24 and 23 x 48 inches. Also furnished Semi-Thik in the same sizes.

THICK BATT BLANKETS are quality home insulation blankets fully enclosed in a permeable Kraft paper wrapping. In this blanket the *Longfibre* rock wool is firmly felted as in the Ful-Thik Batt, then encased for convenience in handling. It is also backed with a heavy vapor barrier with reinforced tacking flanges. Furnished in sizes 15 x 48 and 23 x 48 in.

MEDIUM ROLL BLANKETS are used where insulating requirements are not as exacting and where first-cost economy is a factor. Made of firmly felted *Longfibre* rock wool, these easy-to-install rolls are fully enclosed and have a heavy vapor-seal backing with reinforced tacking flanges. Furnished in sizes 15 x 64, 23 x 64, 15 x 96 and 23 x 96 in.

Airacoustic* Sheets for lining Air-Conditioning Ducts

Airacoustic Sheets, for duct linings of air conditioning systems, are flameproof, highly sound-absorbent and moisture-resistant, with a surface which will not materially increase friction losses in the duct system. Airacoustic Sheets are furnished 24 x 36 in., ½, 1 and 1½ in. thick.

Pipe and Boiler Insulations

Pre-Shrunk Asbestocel*

Cellular type of insulation for pipes carrying low pressure steam or hot water.

Made up of alternate layers of plain and corrugated, specially-treated, moisture-resistant, asbestos felts. Three finishes: Glazed White for quick application, will not carry flame; asbestos paper; and regular canvas cover.

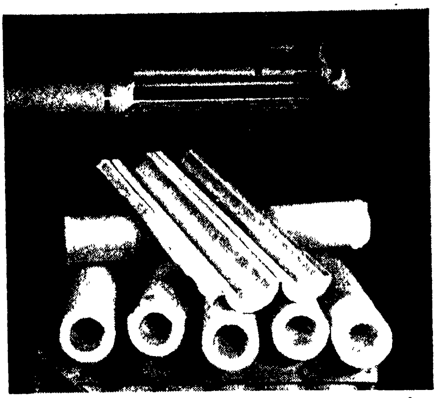
Furnished in 3-foot sections, in following thicknesses: Coarse Corrugated, 2 to 8, ¼ in. plies and Fine Corrugated, 6 plies per inch of thickness.

* Reg. U. S. Pat. Off.

PIPE AND BOILER INSULATIONS, Cont'd

J-M 85% Magnesia

Recommended as the most widely used insulation of the molded type for temperatures up to 600 F. Pipe insulation is furnished in sectional or segmental form for all standard pipe sizes†, in thicknesses up to 3 in. Flat blocks are 3, 6, 9, 12 in. wide and 18 and 36 in. long. Also furnished in curved blocks. Other sizes and greater lengths available on special order.



J-M 85% Magnesia Pipe Insulation

Pre-Shrunk Wool Felt

J-M Pre-Shrunk Wool Felt is equally effective and durable on either hot or cold water service piping. Prevents sweating on cold water pipes. Made of a specially indented wool felt and provided with a dual service liner.

Supplied in canvas finish or weatherproof jacket in 3-ft sections in thicknesses of ½ in., ¾ in., 1 in., single layer; 1 in. and 1½ in., double layer, for all standard pipe sizes.† Temp. limit 225 F.

Asbesto-Sponge* Felted

Recommended on all high pressure steam piping at temperatures to 700 F where insulation may be subjected to rough usage or where both maximum efficiency and durability are desired.

Furnished in 3-ft sections from 1 in. to 2½ in. thick, single layer; over 2½ in. thick in double layer, for all standard pipe sizes.†

* Reg. U. S. Pat. Off.

Superex* Combination

Superex Combination Insulation (an inner layer of high temperature Superex and an outer layer of 85% Magnesia) is recommended where temperatures exceed 600 F. Both Superex and 85% Magnesia insulations are furnished in sectional and segmental pipe covering, as well as in block forms.

Asbestocel*

Asbestocel Sheets and Blocks are used for insulating low pressure boilers, feed water heaters and warm air ducts. Temperature limit 300 F. Furnished 6 to 36 in. wide by 36 to 96 in. long, from ½ in. through 4 in. thick.

Rock Cork*

Rock Cork is made of mineral wool and an asphaltic binder molded into sheets and pipe insulation for all low temperature service to *minus* 300 F. It is strong, durable, and will not support vermin. Because of its unusual moisture resistance, its high insulating value is maintained in service.

Furnished in sheets 18 in. by 36 in., in 1, 1½, 2, 3, and 4 in. thicknesses. Lagging, for curved surfaces, supplied 18 in. long by 1½ through 4 in. thick, 2 to 6 in. wide, depending on diameter. Pipe covering furnished in Ice Water, Brine, and Heavy Brine thicknesses, for all commercial pipe sizes.† Discs also available to a 36 in. max. diameter.

Zerolite*

Zerolite is a resin bonded, mineral wool insulation for temperatures to *minus* 300 F. In addition to possessing the same basic characteristics as Rock Cork, Zerolite is highly fire-retardant, resists petroleum and organic solvents, and has the added advantage of 6 to 10 per cent lower conductivity. Furnished in sheets 18 in. by 36 in. (18 in. by 18 in. available in 1 in. thickness only), in 1 in. through 4 in. thickness. Lagging furnished in same sizes as Rock Cork.

Details on Request

For further information about J-M Insulations and J-M Application Service, write Johns-Manville, 22 East 40th Street, New York 16, N. Y.

† Also available in sections to fit straight runs of copper pipe or tubing with nominal diameters of ⅜ in. and larger.

Kimberly-Clark Corporation

Neenah, Wisconsin

KIMSUL INSULATION

New York 17, N. Y., 250 Park Avenue
Atlanta 3, Georgia, 22 Marietta St., N. W.
Chicago 3, Illinois, 8 S. Michigan Avenue
San Francisco 4, Calif., 155 Sansome Street

KIMSUL* is unique among building insulations and acoustical materials because it is made of many individual plies—each one a continuous separate layer of soft, clean, creped, asphalt-treated cellulose fibers. Each ply is controlled carefully in manufacture both as to thickness and crepe structure. The result is a flexible blanket, inherently uniform in thickness—an important factor in a thermal or sound insulating material. The Kimsul plies and cover of the creped Pyrogard* or creped Reflective Vaporseal are held together with rows of strong stitching that prevent sifting and settling. No heat-leaking thin spots, no money-wasting thick spots in the Kimsul blanket. Reduced to 1/5 its installed volume for easier shipment, handling and storage, the Kimsul blanket is expanded in installation. The stitching controls expansion to the density of maximum efficiency.

KIMSUL INSULATION IS NOW MADE IN TWO GRADES—REFLECTIVE AND REGULAR—SEE PAGE OPPOSITE

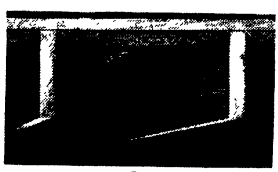

1.


2.


3.

No. 1. Flexible—fits into corners, tucks behind pipes, electrical wiring and other "tight spots." No areas unprotected.
No. 2. Clean—no sharp particles to irritate, nothing to sift; stitched ply construction prevents settling or sagging.
No. 3. Caulkable—one ply or many plies may be compressed to high density in narrow or wide joints, sealing out cold air and sound. Kimsul asphalt-treated wood fiber does not break up during caulking or tamping.


4.


5.


6.

No. 4. Insulated Fastening Edge—The many layer Kimsul blanket is extra wide to provide fully insulated fastening edges, and to ensure completely filling spaces where framing may be slightly off center.
No. 5. Over-Framing Compressibility—Kimsul is easily compressed over framing members. Especially valuable for 48 in. wide Kimsul—suitable for mass or prefabricated construction.
No. 6. Any Width, Any Length—it's easy to cut exact lengths or narrow widths. Avoids muss and fuss. Workmen do a fast, neat job—with Kimsul.

* T. M. Reg. U. S. Pat. Off.

Now Available—Reflective KIMSUL* Insulation

The text of these two pages applies to both Regular and Reflective Kimsul Insulation. The differences in the two types are: Reflective Kimsul has a cover of aluminum foil that acts both as a vapor barrier to shut out condensation, and as a highly efficient reflective surface to turn back radiant heat. Reflective Kimsul has strong reflective tacking flanges for easy, secure attachment to framing. For heat flow downward, Reflective Kimsul provides additional insulating value. Regular Kimsul has the Pyrogard fire-resistant cover.

Fire-Resistant—Special permanent chemical treatment makes Kimsul resist fire.

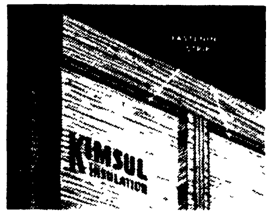
Pyrogard Fire-Resistant Cover*—(safety feature of Regular Kimsul) resists flame-spread.

Creped Aluminum Foil Cover—(a feature of Reflective Kimsul) reflects heat, shuts out condensation.

Moisture-Resistant—asphalt treatment of each ply sheds water.

Resists Mold, Rot, Vermin—The materials of which Kimsul is made offer no subsistence to vermin or insects. Special chemical treatment resists mold and fungus.

"k" Factor—0.27 Btu/sq ft/hr/°F.

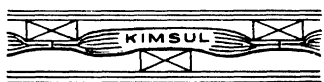


Reflective KIMSUL Installed. Edge of fastening flange folded over face of framing completes the vapor seal.

Air Space—is a prime requisite. Use a vapor-permeable building paper under exterior finish. Ventilation in attics and floors should never be omitted. Use approximately one sq ft of louver area for 1000 sq ft of ceiling area.

SOUND CONTROL

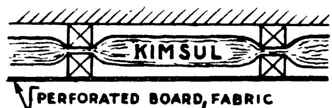
Sound Deadening (one room to another). Kimsul flexible blanket used in staggered stud construction.



Lath and Plaster

1) Absorbs sound from diaphragmatic action of wall panels.
2) Absorbs sound which leaks through joints, thus maintaining original sound resistance of partition.
3) Cushions wall surface.
4) Prevents accidental bridging.

Sound Absorption (within a room). The blanket design makes Kimsul inexpensive as a sound absorbing element. See coefficients below.



T.M. Reg. U. S. Pat. Off.

CONSTRUCTION DETAILS AND HEAT FLOW (U) FACTORS

Kimsul Grade	Heat Flow Horizontal		Heat Flow Up		Heat Flow Down		Heat Flow Down	
	"U"	%	"U"	%	"U"	%	"U"	%
UNINSULATED	.25		.69		.48		.28	
REG. COMMERCIAL THICK	.15	40	.25	64	.20	58	.16	43
" STANDARD THICK	.12	52	.17	75	.15	69	.12	57
" DOUBLE THICK	.08	68	.10	85	.09	81	.08	71
REFLECTIVE MED. THICK	.10	60	.15	78	.10	79	.09	68
" DOUBLE THICK	.08	68	.10	85	.08	83	.07	75

"k" Factor (KIMSUL plies) 0.27* authority of J. C. Peebles, Armour Institute, 1938 (Does not include value of reflective cover.) *Factors expressed in Btu/hr/sqft/°F; k, per inch of thickness; "U" per assembled section of construction. Calculation based on FHA Technical Circular No. 7, dated Jan. 1949. % = % of uninsulated heat flow stopped in the insulated construction.

For further information write to

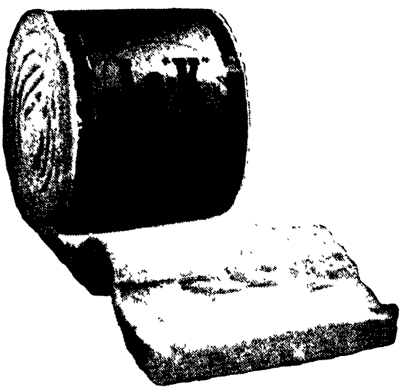
KIMBERLY-CLARK CORPORATION—NEENAH, WISCONSIN

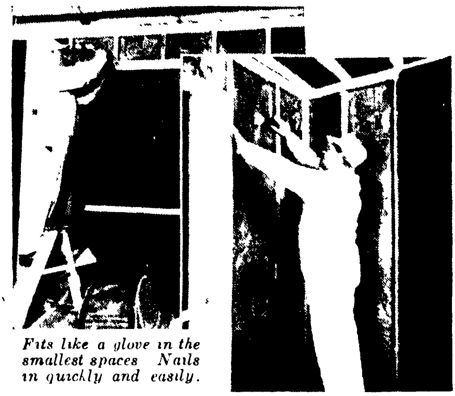
Lockport Cotton Batting Co.

Lockport, New York

THERMAL **Lo-K** *ACOUSTICAL*

INSULATION





Fits like a glove in the smallest spaces Nails in quickly and easily.

Comes in four featured types to meet every insulation need: (1) ***Standard,*** open, blanket roll, backed by tough, waterproof, asphalt-coated kraft paper to form an effective vapor barrier. (2) ***Enclosed Blanket.*** Insulation is completely enclosed in envelope made of asphalt-coated paper on one side and a porous or "breather" type paper on other side. (3) ***Open Aluminum Foil***—providing all the features of open Type 1 plus the extra value of aluminum foil backing. Forms an effective vapor barrier—stops 90 per cent of radiant heat. (4) ***Enclosed Aluminum Foil.*** Superior in insulation plus values and thermal efficiency. Provides greater convenience, comfort, economy and performance.

Thermal Conductivity—The "k" value for cotton is 0.24 Btu/hr/sq ft/degree F/inch. (*See table.*)

Light Weight—Weight of 1 cu ft is ⅞ lb. (*See table.*)

Flame-Proofed—Withstands 1800° blow torch heat.

Moisture-Resistant—Chemical treatment, combined with natural protective coating on cotton fibres, enables cotton to effectively resist moisture. Prevents rot and mildew.

Smooth Texture—Cotton contains none of the sharp particles that irritate the skin.

Flexible—Cotton batt may be expanded or contracted to fit any enclosure.

Easy to Warehouse and Handle. Offers far more "compressibility." Requires one third the trucking and warehouse space of ordinary insulation.

Simple and Economical to Install. Saves from 25 to 40 per cent in costs.

Designed to Maintain Maximum Utility. Resists all types of deterioration. Won't sag or settle. Packaged in Rolled Form.

Thicknesses—inches: 1, 1½, 2, 3, 3⅝. Width—16, 20, 24 in. centers. Lengths: Standard from 12 ft up.

INSULATING VALUE OF VARIOUS INSULATORS[a]

The coefficients of conductivity (k value) are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in of thickness

Type of Insulation	Wgt. per Cu Ft	k^b Value
Cotton Insulating Batt	.875	0.24
Rock Wool: Fibrous material made from rock	10.00	0.27
Mineral Wool: Fibrous material made from mineral slag	.	0.27
Glass Wool: Fibrous material made from glass slag ...	1.50	0.27
Rigid Insulation made from sugar cane fibre...	13.50	0.33
Chemically treated wood fibre between layers of paper	3.62	0.25
Eel grass between layers of paper	3.40	0.25
Stitched and creped expanding fibrous blanket.	1.50	0.27
Shavings: Various from planer...	8.80	0.41
Corkboard: No binder added ...	7.00	0.27
Rigid insulation made from wood fibre	15.90	0.33
Rigid fibre board made from shredded wood and cement	24.20	0.46

[a]Compiled from Chapter 9.
[b]"k" indicates temperature conductivity.

Owens-Illinois Glass Company

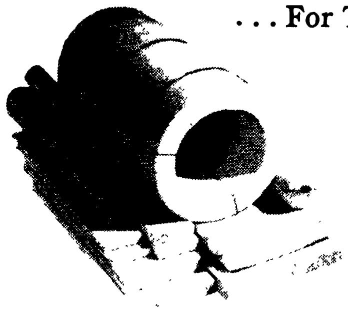
Kaylo Division Toledo 1, Ohio

Sales Offices

ATLANTA	CHICAGO	DETROIT	NEW YORK	PITTSBURGH
BOSTON	CINCINNATI	HOUSTON	OKLAHOMA CITY	ST. LOUIS
BUFFALO	CLEVELAND	MINNEAPOLIS	PHILADELPHIA	WASHINGTON

KAYLO® Heat Insulation

. . . For Temperatures Up to 1200° F.



PHYSICAL CHARACTERISTICS

DENSITY Approximately 11 lb per cu ft.

FLEXURAL STRENGTH 50 lb per sq in.

COMPRESSIVE STRENGTH (at 5% deformation)

Before heating	150 lb per sq in.
After heating for 24 hours	
at 750° F	144 lb per sq in.
at 1000°F	123 lb per sq in.
at 1200°F	117 lb per sq in.
After boiling for 24 hours (while wet)	74 lb per sq in.

LOSS IN WEIGHT

After heating for 24 hours	
at 750° F	5.5%
at 1000° F	7.9%
at 1200° F	9.8%
After boiling for 24 hours (after drying)	0.2%

RESISTANCE TO ABRASION

(Conventional tumbling test-loss in weight after 10 minutes)

Before heating	2.2%
After heating for 24 hours	
at 750° F	3.7%
at 1000° F	5.7%
at 1200° F	6.9%

DIMENSIONAL STABILITY

Linear shrinkage after heating for 24 hours	
at 750° F	0.8%
at 1000° F	0.9%
at 1200° F	1.5%
Elongation after saturation (max.)	0.04%

MOISTURE ABSORPTION (volume)

After 6 hours exposure in atmosphere of 120° F and 90% Relative Humidity	0.9%

CONDUCTIVITY (K)

At 100° F mean temperature	0.41
At 500° F mean temperature	0.54

KAYLO HEAT INSULATION, a hydrous calcium silicate, is effective up to 1200° F. It performs efficiently through the hot water and low pressure steam range, also through temperatures in the super-heated steam range. Therefore, a single material can be used for high temperatures which usually require combinations of two different insulating materials. Kaylo Heat Insulation is incombustible and insoluble in water. Its light weight simplifies handling, shipping and application. High strength makes breakage almost negligible. Kaylo Heat Insulation can be cut, scored or sawed with ordinary tools. The material is non-irritating to the skin and non-toxic.

KAYLO PIPE INSULATION is made to Simplified Dimensional Standards of thicknesses and diameters for snug nesting, when necessary. Coverings are sectional for tube and pipe sizes ¾ in. to 12 in.; tri-segmental up to 23 in.; quad-segmental up to 41 in.; K-segmental (18 in. wide segments) up to 72 in. in diameter.

KAYLO HEAT INSULATING BLOCK is made in all standard sizes, up to 18 in. wide and in thicknesses from 1 in. to 6 in. Where necessary, on special order, the block can be shiplapped for broken joint, single-layer application in thicknesses of 3 in. or greater.

WRITE FOR LITERATURE ON KAYLO HEAT INSULATION

Sprayo-Flake Insulation Co.

INdependendencce 3-3300
2727 Irving Park Road, **Chicago 18, Ill.**

"Sprayo-Flake" is a multiple purpose insulation providing the utmost in thermal insulation—noise reduction—condensation control—sealing against air infiltration, etc.

VERSATILITY: Sprayo-Flake Insulation is a versatile product produced from fibrous insulation materials and adhesives suited to simultaneous Sprayo-Flake Insulation Gun application to structural surfaces having regular or irregular centers, spacing or contours.

SPRAYO-FLAKE PROCESS: It is made in 10 standard types and is unique in that it is fabricated and applied in one single, efficient, economical operation on the job. The Sprayo-Flake Insulation Process consists of forcibly projecting, through a specially constructed Sprayo-Flake Insulation Gun, dry fibrous materials simultaneously with an atomized adhesive. The adhesive primes the surface being treated and coats the fibers as they leave the nozzle of the Sprayo-Flake Insulation Gun, causing them to build up in a homogeneous, light weight, bonded, cellular insulation coating on the structural surface treated.

"Air - Gun - Applied" Continuous "Bonded-On" Insulation Coatings for Steel, Brick, Aluminum, Cement and Cinder Block, Cement and Asbestos Board, Gypsum, Precast Cement Tile, Wood Sheathing and Roof Decks, Concrete Slabs and Walls etc., in thickness of ½ in., 1 in., 1½ in., etc.

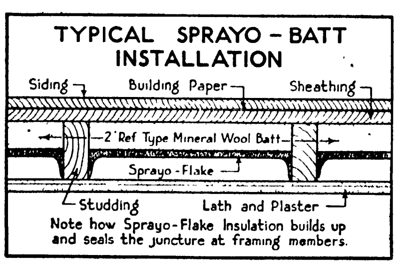
THERMAL INSULATION APPLICATION: On Masonry Construction Sprayo-Flake Insulation is the ideal insulation for masonry walls because it is bonded securely to the wall surface treated providing highly efficient lifetime insulation and at the same time sealing the wall against air infiltration. The emulsified asphaltic adhesive used in applying Sprayo-Flake Insulation is one of the finest damp-proofing agents known to the construction science.

ON STEEL CONSTRUCTION Sprayo-Flake Insulation is especially well adapted to use on all types of metal buildings. Being plastic in nature, it bonds and conforms to the surface of the outer covering without cutting and fitting. These properties provide substantial savings in both installations and maintenance throughout the life of the building. Seals joints—eliminates condensation.

Sprayo-Flake triple-purpose insulation combining dampproofing, insulation, and vapor seal in one tailor-made application on wood furred brick wall.

Sprayo-Flake insulation on air conditioning and heating and air conditioning ducts.

Sprayo-Flake insulation on Concrete Walls and Ceilings below grade with Aluminum Overcote on Walls and Ceiling of Fur Storage Vault.

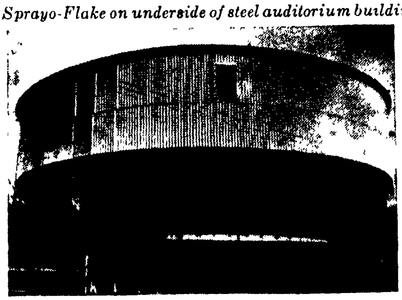


Sprayo-Flake 'K' factor 0.10 Btu

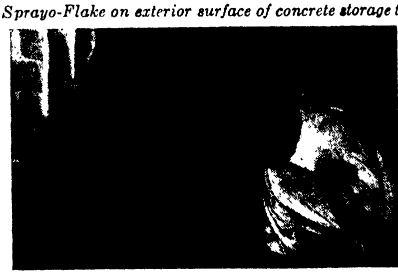
Sprayo-Flake Insulation Co.

INdependence 3-3300
2727 Irving Park Road, **Chicago 18, Ill.**

Sprayo-Flake on underside of steel auditorium building



Sprayo-Flake on exterior surface of concrete storage tank



Sprayo-Flake process seals all joints between masonry and framework.

The Sprayo-Flake Bonded Insulation Mat is applied to the required thickness to the inside surface of the metal roof decking or sheet metal walls. Sprayo-Flake Insulation is usually applied around exposed purlins in a ¼ in. to ⅜ in. thickness.

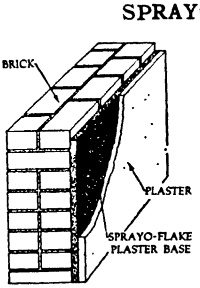
ON WOOD CONSTRUCTION Sprayo-Flake Insulation is bonded directly to wood sheathing, roof decks, etc. The Sprayo-Flake Insulation Mat covers all cracks and crevices and it not only provides a highly efficient insulation but it also seals the exposed surfaces against cold and warm air infiltration.

SPRAYO-FLAKE "SPRAYED-ON" INSULATION is ideal for Housing Projects, Homes, Industrial Plants, Defense Projects, Storage Tanks, Dust Collectors, Fur Storage Vaults, Auditoriums, Hospitals, Laboratories, Schools, Hotels, Flight Wind Testing Tunnels, Radio and Television Broadcasting Studios, Churches, Office Buildings, Commercial Buildings, Apartment Buildings, Cold Storage Plants, Libraries, Warehouses, Cafeterias, Machine Shops, Banks, Public Utility Buildings, Dairy Buildings, etc.

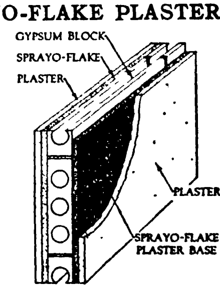
SOUND INSULATION: Sprayo-Flake Insulation is a very effective sound deadening insulation when applied directly to one side of the sound transferring membrane. Tests were made by the C. F. Burgess Laboratories on floors and walls before and after being treated with Sprayo-Flake Insulation in actual structures.

Sprayo-Flake data booklet 10A on request.

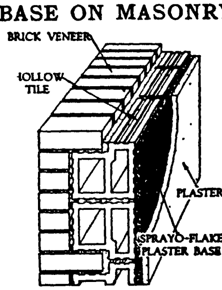
SPRAYO-FLAKE PLASTERBASE ON MASONRY WALLS



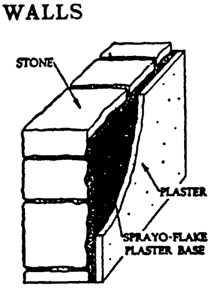
Brick Wall



Gypsum Block Wall



Brick Veneer Over Hollow Tile



Stone Wall

The Pacific Lumber Company

100 Bush Street, San Francisco 4, Calif.

35 East Wacker Drive, Chicago 1, Ill.

PALCO INSULATION WOOL®

PROVIDES EFFICIENT INSULATION

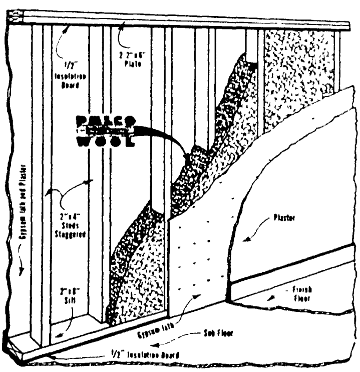
FLAME PROOF SAFERIZED PROCESS

INSULATION THICKNESS—Ceiling insulation should be at least 4 in. thick. We recommend 6 in. thickness, giving 50 per cent more insulation at little added cost. Radiant heating system manufacturers insist on 6 in. thickness for maximum efficiency and economy. Insulation in walls should be full stud thickness. Below are typical ceiling sections showing U values for various thicknesses of PALCO WOOL Insulation:

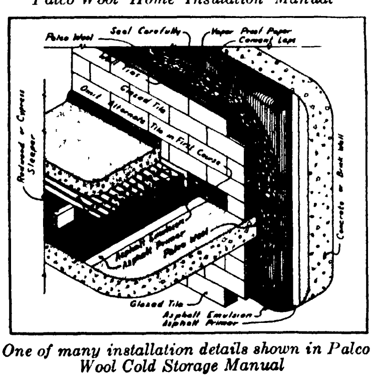
U = 0.047 — Ceiling joists, sheetrock, full 6 in. thickness PALCO WOOL Insulation over joists.

U = 0.0631 — Ceiling joists, sheetrock, full 4 in. thickness PALCO WOOL Insulation over joists.

U = 0.080 — Ceiling joists, sheetrock, 3 in. thickness PALCO WOOL Insulation between joists.



Typical sound deadening application from Palco Wool Home Insulation Manual



One of many installation details shown in Palco Wool Cold Storage Manual

9 PROVED QUALITIES

High thermal efficiency—K factor of only 0.26 Btu.

Non-settling and non-compacting.

Flame proof—fire resistant.

Moisture resistant.

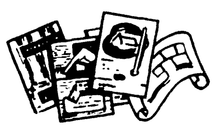
Odorless—won't give off odors.

Permanent—will outlast structure.

Non-attractive to vermin or insects.

Economical—initial low cost, high insulating efficiency.

Resilient—springy structure gives better sound absorption.



WRITE FOR THESE TECHNICAL AIDS

HOME INSULATION MANUAL—home, commercial sound deadening applications.

COLD STORAGE MANUAL—construction details, all types of cold storage.

LOCKER PLANT MANUAL—plans, material lists, construction details.

FARM PRODUCE STORAGE PLANS for fruit and vegetable storage.

ENGINEERING REPORTS—on-the-job findings on actual installations.

Wood Conversion Company

Dept. 220-2 First National Bank Building
St. Paul 1, Minnesota

NEW YORK CHICAGO DENVER BOSTON KANSAS CITY DETROIT ATLANTA

EFFICIENT INSULATION FOR EVERY NEED

For many years a leader in the insulation field, Wood Conversion Company manufactures a complete line of flexible, fiber and rigid insulation for all industrial and domestic purposes. This insulation is the product of scientific research, and is especially designed to embody the most desirable qualities for every use. Backed by the name of Weyerhaeuser, Wood Conversion Company insulation assures high efficiency and long, satisfactory service.

Balsam-Wool* Sealed Blanket Insulation is a flexible insulation. The insulation blanket is made from new wood fibers, completely enclosed on both sides and edges with heavy asphalt saturated and coated kraft liners.

Years of practical application and constant testing are behind each of these Balsam-Wool features—

- Integral, continuous vapor barrier
- Sturdy wind barrier
- Special spacer flange
- Double bonding of mat to liner
- Rot and termite treatment
- Highly fire retardant

In addition, Balsam-Wool is manufactured under rigid quality control; proved by more than a quarter century of experience.

K-25* Fiber Pneumatic System—The modern, high-speed, automatic way to insulate domestic refrigerator cabinets and doors. Fluffed to proper low density in the manufacturer's plant, K-25 is blown into cabinets and doors under high pressure, forming a tightly felted insulating mat without joints, laminations or voids.

Tufflex*—A soft, felted blanket material made from fleecy wood fiber, combines high insulating efficiency with toughness and exceptional resistance to heavy impact blows. Tufflex is light in weight and nonabrasive—will not tear or pull apart even when cut into narrow strips. Tufflex is available in rolls or sheets of various thicknesses and widths.

Applying Nu-Wood Interior Finish

Applying Balsam-Wool Sealed Insulation

NU-WOOD* STRUCTURAL INSULATION

Nu-Wood Interior Finish—A multiple-purpose wood fiber material available in tile, plank and board. Nu-Wood insulates, decorates and quiets noise. Nu-Wood's colors are soft and harmonious—will not fade. The Nu-Wood interior finish line includes Sta-lite, an insulating interior finish with more than 70 per cent light reflection.

Nu-Wood Sheathing—A strong, structural insulating sheathing in ½ in. and 25/32 in. thickness, asphalt impregnated, offering complete moisture protection.

Nu-Wood Roof Insulation—Will fill all requirements of Federal specification LLL-F-321b. Furnished in any practical thickness.

* Reg. U. S. Pat. Off.

M-47-7169

American Flange & Manufacturing Co. Inc.

30 Rockefeller Plaza, New York 20, N. Y.
Plaza 7-2200

Ferro-Therm

Reg. U. S. Pat. Off.

STEEL INSULATION

FULLY PROTECTED BY U. S. AND FOREIGN PATENTS ISSUED AND PENDING

Ferro-Therm installed in a cold storage room

Ferro-Therm Steel Insulation, made from rigid steel sheets with a special alloy coating, reflects 90 per cent to 95 per cent of all radiant heat. This high reflectivity, combined with extremely low heat storage capacity, provides maximum insulating efficiency in a minimum overall thickness.

Saves Pay Space and Weight

In cold storage construction, the number of sheets of Ferro-Therm depends on the temperature to be maintained and the U value required. The k value of Ferro-Therm, based on tests, is listed in the Data Book of the *American Society of Refrigerating Engineers* as 0.226 Btu per (hr) (sq ft) (°F temperature difference). Laboratory tests and thousands of applications have demonstrated that a wall of Ferro-Therm will provide insulating efficiency equivalent to a wall of mass insulation approximately twice as thick.

Assures Rapid Pull Down of Temperature

The low heat storage capacity of Ferro-Therm is extremely important in achieving rapid pull down of temperature, and in saving refrigeration costs for the initial and each subsequent cooling of space. Specifically, the heat storage capacity of a single sheet of No. 38 gauge is 0.029 Btu per (hr) (sq ft) (°F temperature difference). This is approximately 1/16 of the heat storage capacity of 1 sq ft of 1 in. thickness corkboard.

Permanent, Fire-Proof Insulation

Ferro-Therm construction eliminates trapping of moisture condensate, with subsequent deterioration of the construction. As it is all-metal, Ferro-Therm cannot be penetrated by rodents, vermin or termites, and is absolutely non-combustible. The value of Ferro-Therm for fire protection is apparent.

125° Below Zero Maintained in Altitude Test Chambers

Ferro-Therm has proved its superiority in buildings, cold storage rooms, refrigerated cabinets, locker rooms, dry ice containers, refrigerated railway car construction, ovens, high-temperature storage tanks—in fact, practically every type of application where high insulating efficiency with economies in space and weight are a requisite. The most notable demonstration of Ferro-Therm performance has been its selection for the insulation of altitude chambers for the testing of Army and Navy aviation equipment and personnel. In these chambers, temperatures as low as −125°F were maintained, with a temperature drop of +70°F to −70°F in 10 to 12 min.

Ingersoll Products Division
Borg-Warner Corporation

321 S. Plymouth Court, **Chicago 4, Ill.**

Koolshade* sunscreen is a bronze miniature venetian blind with fixed horizontal bars set at an angle which will keep the greatest possible amount of solar heat load outside of windows.

By reflecting, absorbing, and radiating most of the sun's heat rays outside the window, Koolshade keeps rooms up to 15° cooler and has been shown to account for as much as 75% of the cooling necessary in air conditioning installations. 100 sq ft of Koolshade is equal to 1 ton of refrigeration and costs ⅓ to ½ as much.

Koolshade provides insect protection equal to ordinary insect screen.

Koolshade is framed and applied to windows like ordinary insect screens. Ingersoll distributors can apply Koolshade in patented Koolshade aluminum frames, in Koolshade Quik-on frames which have only top and bottom frame rails held apart in tension by installation hardware or in wood or metal frames

Ingersoll's Sungard* economy sunscreen stamped from aluminum sheet, is intended for industrial applications and performs the same heat load reducing function as Koolshade at lower cost. Sungard is not insect proof

Sungard is framed and applied by Ingersoll Distributors like Koolshade.

An Example† of cooling load reduction based on conditions prevailing at peak solar load at 40° latitude:

We have 16 windows on East side of building—each window 4 ft x 6 ft for a total of 384 sq ft of window area. Use 10 a.m. Peak Load:

	Btu's
Solar load transmitted through bare windows 116 Btu x 384 sq ft	44,544
Solar load transmitted through KOOLSHADE 11.5 Btu x 384 sq ft	4,416
Amount of Solar Heat stopped by KOOLSHADE	40,128

Expressed in tons of refrigeration (12,000 Btu's—1 Ton)

Heat stoppage equals Tons 3.3444

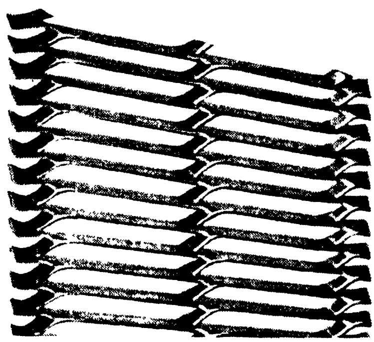
Follow same procedure for 15 West Windows 4 ft x 6 ft—total square footage 360 sq ft—and using 3 p.m. Peak Load:

	Btu's
Solar Load through Bare Windows	61,200
Solar Load through KOOLSHADE	7,920
Reduction	53,280
in Tons	4.440

South windows consist of 7—4 ft x 6 ft and 2—3 ft x 6 ft for total square footage of 204 sq ft—Figuring on 12:00 noon Peak Load:

	Btu's
Solar Load through Bare Windows	15,096
Solar Load through KOOLSHADE	1,224
Reduction	13,872
in Tons	1.156
Total tons of refrigeration saved	8.9404

Ingersoll Koolshade



Ingersoll Sungard

Write to Ingersoll Products Division, Borg-Warner Corporation, Dept. V for Technical Information.

† Example based on test data from ASHVE GUIDE, 1940.
* Koolshade and Sungard are trademarks (Reg. U.S. Pat. Off.). They are the property of the Ingersoll Steel Division Borg-Warner Corp

Infra Insulation, Inc.

525 Broadway, **New York, N. Y.**

Telephone: WOrth 4-2241

THERMAL FACTORS OF INFRA INSULATION AND DRY ROCKWOOL EQUIVALENTS

Type 6	Type 4 Jr., in ONE-INCH Space
Up-Heat C.089 R11.23 = 4$\frac{3}{5}$″ dry rockwool.	Up-C.194R 5.15 = 2$\frac{1}{8}$″ dry rockwool
Wall-Heat C.073 R13.69 = 5$\frac{5}{8}$″ dry rockwool.	Wall-C.150R 6.66 = 2$\frac{3}{4}$″ dry rockwool
Down-Heat C.044 R22.72 = 9″ dry rockwool.	Down-C.097R 10.30 = 4$\frac{1}{5}$″ dry rockwool

Infra Insulation, Type 6, is a tough 3-aluminum-sheet insulation, with 6 heat-ray-reflective surfaces, plus two outer reflective spaces, and four rows of inner, alternating, reverse triangular air spaces. (Total, 6 reflective spaces.) It is easily and quickly installed between wood joists, furring strips, and between steel trusses, metal beams, and girders. The normal installation rate between wood joists is 2,000 sq ft a day. The 1$\frac{5}{8}$ cu ft carton contains 500 sq ft, weighs 40 lbs. Standard width, for 12 in., 16 in. and 24 in. centers. Standard Type 4, for 12 in., 16 in., 24 in. centers.

Heat flow through wall space, which is air, is 5 per cent to 7 per cent by Conduction; 15 per cent to 28 per cent by Convection; and by Radiation, 65 per cent to 80 per cent.

ABSORPTIVITY and RADIATION:—Each of Infra's 6 aluminum surfaces THROWS BACK 97 per cent of the heat rays which strike it. As emittive surfaces, they emit ONLY 3 per cent of heat actually absorbed by conduction, convection and radiation. Practically all building materials, including ordinary insulations, absorb and emit 90 per cent of heat rays, against Infra's 3 per cent.

CONDUCTION:—One sq ft of Type 6 Infra weighs only 1$\frac{1}{4}$ oz, has only 1$\frac{1}{4}$ cu in. mass. The ratio is 1 of Infra mass to 431 of low conductive air. Ordinary insulation, when dry, has a ratio of 1 of mass to 23 of air.

CONVECTION and VAPOR:—Each of the 3 tough aluminum sheets of Infra has ZERO permeability to all gases, including heated air, cold air, and water vapor.

NON-CONDENSATION-FORMING:—The 3 aluminum sheets together with TWO inner accordion separators, which permanently prevent metal to metal contact, form FOUR inner rows of alternating, reverse, triangular, reflective air pockets. The construction, and the lack of weight for heat exchange, make Infra Insulation absolutely non-condensation forming. Neither can it absorb nor store moisture.

Infra Insulation uses 99.5 per cent pure .0009 in. and .002 in. aluminum, made in accordance with special Infra emissivity requirements. It has 17 lbs and 52 lbs bursting strength (Mullen test), 800 per cent to 2600 per cent greater than ordinary foils. The tearing strength is 28 grams and 80 grams (Elmendorf test). The special fiber separators are permanently flame-proof, mold-proof, and vermin-resistant.

FIRE:—Since Infra emits practically no heat rays, and since aluminum's melting point is 1250° F, it has actually prevented the spread of fire.

SANITARY:—Infra is sanitary, inhospitable to vermin, and DOES NOT RETAIN odors. Mechanics like to work with Infra. It is CLEAN, free of DUST or lint, with permanent freedom from floating particles.

Infra Insulation Type 6

INFRA, TYPE 4 JR. is half ($\frac{1}{2}$) inch in depth. Can be used in 1 in. spaces between furring strips; in brick or masonry walls; around metal ducts; and under floors, ordinary or radiant heated.

Write for new 56-page authoritative manual, "Simplified Physics of Vapor and Thermal Insulation."

ALFOL INSULATION

Manufactured by

Reflectal Corporation

155 East 44th Street
New York 17, N. Y.

HEAT INSULATION for ALL PURPOSES

Easy to Apply

Handy Package—250 Sq Ft of Insulation

Alfol Building Blanket—for all Types of Building Structures.

Alfol Building Blankets consist of spaced layers of Alfol Aluminum Foil Insulation attached along the edges to a liner sheet of heavy vapor proof paper.

Packaged in handy rolls of 250 sq ft each for use on 12 in., 16 in., 20 in. or 24 in. centers. Weighs less than $^{1}/_{10}$ lb per sq ft.

- **High Insulating Efficiency**
- **Positive Vapor Barrier.**
- **Low Heat Storage Capacity.**
- **Negligible in Weight.**
- **Moisture Proof.**
- **Durable.**
- **Odorless and Clean.**
- **Easily Applied.**
- **Low Cost.**

Specifications

Description	Widths	Net Area per Roll	Net Weight per Roll
Type I — 1 Layer ALFOL	12″-16″-24″	250 sq. ft.	17 lbs.
Type II — 2 Layers ALFOL	16″-20″-24″	250 sq. ft.	19 lbs.
Type III — 2 Layers ALFOL	12″-16″-24″	250 sq. ft.	20 lbs.
Type IV.— 3 Layers ALFOL	16″-20″-24″	250 sq. ft.	23 lbs.

A L S O

● ALFOL PREFABRICATED INSULATION PANELS

For Tanks, Towers, And All Types of Heated Equipment.

● ALFOL ASBESTOS

Alfol Aluminum Foil Insulation Laminated To Asbestos For Ovens, Ranges, Boiler Jackets, Hot Water Heaters, And All High Temperature Insulation Purposes.

● ALFOL JACKETING

Heavy Reinforced Paper Combined With Alfol Aluminum Foil. Provides Insulation And Vapor Barrier In One Convenient Form. For Refrigerator Cars, Trucks, Buses, Trailers, Etc.

Write for complete information, catalogs and prices.

Silvercote Products, Inc.

161 East Erie Street, **Chicago, Ill.**

SILVERCOTE *Reflective* **INSULATION**

Silvercote Heat Reflective Surfaces—The silver-like surface of Silvercote reflective insulation consists of a polished, heat reflective coating applied to a special kraft paper. The importance of using a Silvercote radiant heat reflective surface in modern building construction is obvious when it is realized that from 50 to 80 per cent of the heat transferred across a normal air space is in the form of radiation.

REFLECTIVE SHEET INSULATIONS

Silvercote Duplex—A thin flexible vapor barrier and insulation consisting of two sheets of Silvercote paper bonded together with asphalt. This material, containing two exposed Silvercote surfaces, weighs approximately 50 lbs per thousand sq ft and is manufactured in 500 sq ft rolls in widths of 36 in. or 52 in. to span two 16 in. or 24 in. standard framing spaces. These widths permit bow-in of the Silvercote over the room side of the framing members to form an air space between the insulation and the interior finish. The water vapor permeability of Silvercote Duplex is 0.23 grains per sq ft per hour per inch of mercury vapor pressure difference.

Silvercote Simplex—An economical vapor permeable reflective insulation designed for use where a vapor barrier is not required. Silvercote Simplex is a single sheet of special kraft paper coated on both sides with the Silvercote surface. It weighs approximately 30 lbs per thousand sq ft and is manufactured in 500 sq ft rolls in 36 in. or 52 in. widths to span two 16 in. or 24 in. standard framing spaces. These widths permit bow-in of the Silvercote over the exterior side of the framing members to form an air space between the insulation and the exterior sheathing. The water vapor permeability of Silvercote Simplex is 99.2 grains per sq ft per hour per inch of mercury vapor pressure difference.

REFLECTIVE BLANKET INSULATIONS

Blanket Insulation, Silvercote on Vapor Barrier Side—A popular building insulation available in various thicknesses and faced on the vapor barrier side of the blanket with Silvercote paper. Manufacturers of this type of reflective blanket apply an asphalt coating to the back of the Silvercote paper for the twofold purpose of bonding the insulation material to the Silvercote paper and to lend vapor resistant properties to the blanket at the room side of the insulation. The opposite side of this type of reflective blanket is a plain kraft liner perforated where necessary to allow compression packaging and also to assure ample water vapor permeability at the so-called breather side of the blanket.

Reflective Blanket, Silvercote on Breather Side—This product was developed for application in structures where only one air space adjacent to the cold side of the blanket is available. An air space faced on one side with a Silvercote surface will have approximately equal effectiveness whether formed on the vapor barrier or the breather side of the blanket. Since Silvercote paper is in itself not a vapor resistant material, it can readily be used on either side of blanket insulation.

Reflective Blanket, Silvercote on Both Sides.—A de luxe insulation material utilizing the maximum insulation value of blanket insulation, air spaces and two heat reflective surfaces. This double reflective blanket is manufactured with Silvercote Paper on the vapor barrier side and the breather side.

AVAILABLE UPON REQUEST

Silvercote's Handbook of "U" Values, 108 page illustrated booklet listing 12,852 "U" values of various walls, floors and ceilings—Silvercote's Handbook of "U" Values is unique in that it provides summer as well as winter "U" values. The Handbook's special listing of ceiling "U" values to indicate heat flow down characteristics will be of interest to those who are concerned with summer comfort as well as winter fuel savings.

United States Testing Company, Inc.

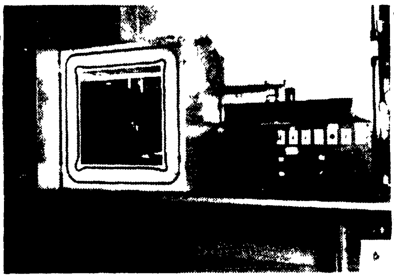
Est. 1880

Main Laboratories

1415 Park Ave., **Hoboken, N. J.**

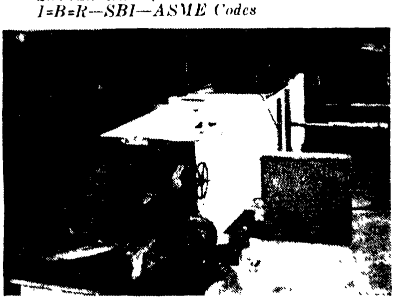
PHILADELPHIA, PA.—CHICAGO, ILL.—LOS ANGELES, CALIF.—DENVER, COLO.—NEW YORK, N. Y.—PROVIDENCE, R. I.—BOSTON, MASS.—MEMPHIS, TENN.—DALLAS, TEXAS

The United States Testing Company, Inc., is one of America's leading independent testing and research laboratories. Established in 1880, it is today an institution known and recognized from coast to coast—an institution whose scientific findings are accepted as impartial and authoritative by every branch of American industry.



Thermal Transmission Apparatus
For measuring thermal conductivity of rigid, semi-rigid or loose types of insulating materials.

Boiler Testing
We have all the necessary equipment to test and rate heating boilers in accordance with I=B=R—SBI—ASME Codes



Air Filter Duct Testing.

Our Engineering Laboratory is equipped with all the necessary apparatus to conduct tests on domestic heating boilers for determination of fuel consumption, draft-loss, stack temperature, steam quality, Btu output, flue gas analysis, and over-all efficiency. Rating tests are performed according to either the *SBI* or I=B=R codes, as well as under simulated operating conditions for development purposes.

Water heaters can be subjected to capacity tests under wide temperature requirements and varied heating loads. Refractories, fuel additives, valves, pumps, burners, and heating accessories of many types are evaluated scientifically in our laboratories.

Other facilities include all the necessary equipment for conducting exhaustive tests on air cleaning devices according to accepted impingement or the more recent discoloration methods. These tests may be performed with either artificial or natural dusts, under an almost infinite variety of flow rates and dust feed.

Diffusers, grilles, registers, and duct ventilators are regularly tested and rated.

SOUND MEASUREMENTS

Measurements of sound absorption and sound transmission coefficients are but two of the typical tests made on a great variety of building materials.

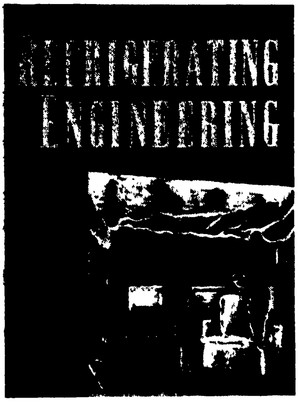
Field and laboratory tests are regularly conducted for determination of sound intensity in the audible range.

Our experts are available for court testimony in connection with any of the tests we perform, in support of our findings.

Write for Descriptive Literature and Price List.

American Society of Refrigerating Engineers

40 West 40th Street, **New York 18, N. Y.**



The most rapidly growing magazine in the refrigeration field

LONG acknowledged the most authoritative periodical in the field, *Refrigerating Engineering* has added steadily to the practical value of its contents, and its number of readers has grown in proportion. This magazine is a must for men who keep in touch with all that is new and important in refrigeration and air conditioning. The annual subscription price is $4.

ASRE DATA BOOKS

Published bi-annually since 1932, they now consist of two volumes, one issued each year. The Basic Volume is a standard reference work containing fundamental data, refrigerant tables, description of the different cycles, systems, and component parts—a gold mine of information—$7.50 a copy.
The Applications Volume is crammed full of how-it-is-done information on the use of refrigeration in 81 different applications. A must for the design or commercial engineer—$7.50 a copy.

REFRIGERATION ABSTRACTS

A Journal devoted to brief condensations, by specialists, of all articles of refrigerating interest appearing in hundreds of periodicals and reports published internationally, on refrigeration and allied subjects. Four issues and an index annually at $7.00.

APPLICATION DATA SECTIONS

The APPLICATION DATA Sections tell precisely how refrigeration is used in various fields, giving examples and specific information on the best practice known to date. A complete list of the subjects covered is available on request. Some 35 bulletins are available separately at reasonable prices for single copies or quantity orders and can also be had bound with a paper cover, the complete set for $5.00.

CODES AND STANDARDS

ASRE further contributes to refrigeration progress by establishing codes and standards in the industry. These standards cover approved methods for testing and rating various types of air conditioning and refrigerating equipment. Also included is the B-9 Safety Code for Mechanical Refrigeration. Sold separately, or a complete set for **$5.00**.

REFRIGERANT TABLES, CHARTS, AND CHARACTERISTICS

A handy 6¼ in. x 9 in. volume containing the thermodynamic properties of all refrigerants now in use, (reprinted from ASRE Data Book) forms a convenient reference for design and application engineers, contractors, professors and students. Cloth bound, $2.00, paper bound, $1.50.

MEMBERSHIP ACTIVITIES

IT is the policy of the ASRE to treat in its meetings current subjects touching upon all phases of the art of refrigeration. Membership is in several grades with dues from $10 to $18. Sections hold meetings in 31 principal cities. More detailed information will be sent on request.

To keep apace with progress in refrigeration and air conditioning, read the publications and follow the activities of the AMERICAN SOCIETY OF REFRIGERATING ENGINEERS, 40 West 40th St., New York 18, N. Y.

Coal-Heat

Published at

20 W. Jackson Blvd., **Chicago 4, Illinois**

Phone Wabash 2-9464 New York City, MUrray Hill 2-9192

Editorially

COAL-HEAT puts its emphasis on:

—ways and means of furthering the more satisfactory use and sale of coal.—sales and service—heating merchandising—the fundamentals

The 'Equipment Situation'—

Three times out of four the fuel dealer has two strikes against him because of the design, condition, and operation or the equipment in which much of his coal is used. For instance, findings of the Coal Heating Service inspection campaign in Minneapolis and St. Paul, Minn., covering thousands of individual homes show that:

1. Only one heating system in 10 is in good condition.
2. More than 2 out of 3 need cleaning.
3. More than 3 out of 7 have chimneys or smoke pipe that need attention or replacement.

In the larger field—among the nation's laundries, dairies, hotels, hospitals, apartments, schools, churches, medium and larger sized plants the reader can assume from detailed records of plant inspections of such fuel users in Akron, Canton, Cleveland, Middletown, Ohio; Detroit, Flint, and Lansing Mich ; Durham, N. C.; Altoona and Erie, Pa.; Milwaukee, Minneapolis and St. Paul, Morgantown W. Va., Toronto, Canada, and Wheeling, W. Va., that 'on the average' out of every 100 such plants:

—from 20 to 80% are mechanically defective;
—from 10 to 20% are obsolete;
—from 10 to 15% are overloaded;
—from 20 to 80% are improperly operated;

Obviously, therefore, in all too many cases

—far too much fuel is wasted;
—heating or power costs are excessive;
—heating and operating problems are far too frequent;
—the modernization and replacement market is all but unlimited.

So, if the reader recognizes this situation there can be no question as to **COAL-HEATS** place in the picture from the sales and service viewpoint.

Functionally, the 477 feature articles and regular departments 'indexed' in **COAL-HEAT** the past 12 months were divided as follows:

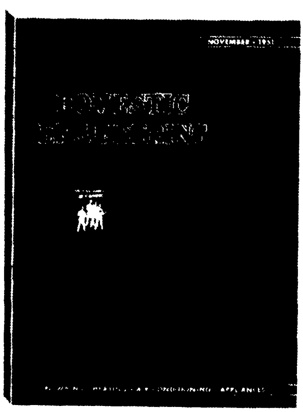
Association News	19	Operation & Maintenance	13
Business Trends	22	Personal Relations	8
Coal Handling	16	Personnel	22
Collections	13	Research	20
Costs	18	Sales	24
Credits		Salesmanship	33
Equipment	30	Sales Promotion	23
Fuels	32	Service	25
Fuel Engineering	13	Smoke Prevention	26
Heating	23	Training Activities	26
Housing	22		
Markets	21		
New Products	14		

each of these being covered as briefly and helpfully as possible.

Domestic Engineering Publications

1801 Prairie Avenue, **Chicago 16, Illinois**

EFFECTIVE 2-WAY COVERAGE OF THE HEATING, PLUMBING AND AIR CONDITIONING FIELD



DOMESTIC ENGINEERING MAGAZINE

DOMESTIC ENGINEERING means leadership! Leadership in advertising . . . leadership in paid circulation . . . leadership in editorial content!

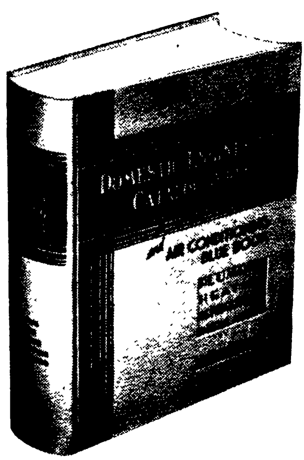
DOMESTIC ENGINEERING has won ten major national editorial achievement awards in ten years—a record unsurpassed by any business publication in any field. Editorial content is keyed high to attract and hold the reader loyalty of the top third who do eighty per cent of the business. These are inclusive contractor-dealers who sell and install broad lines of heating, air conditioning, plumbing and allied products. **DOMESTIC ENGINEERING** serves their total business interests—management, merchandising and technical.

Published monthly. Yearly subscription, U. S and Canada, $5.00. Foreign, $8 00.

DOMESTIC ENGINEERING CATALOG DIRECTORY

DOMESTIC ENGINEERING CATALOG DIRECTORY is one of the most comprehensive single sources of buying and specifying information for all types of products used in the heating, air conditioning, plumbing and allied products industry. Listing virtually every known product in the field, this volume supplies also names and addresses, trade names and complete catalogs of leading manufacturers.

Published annually. 1952 Edition, $7.50.



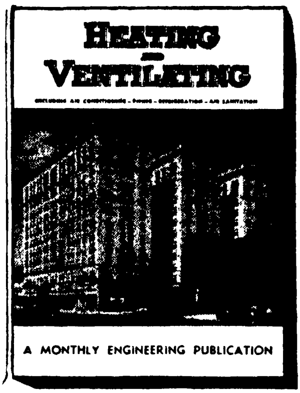
Together, **DOMESTIC ENGINEERING** and **DOMESTIC ENGINEERING CATALOG DIRECTORY** constitute the backbone of every well-conceived promotional program in this industry. Together, they afford you effective two-way coverage of the important buying factors that make up this field. **DOMESTIC ENGINEERING** supplements these leading publications with a secondary package of services including market and research information, manufacturers' agents assistance, complete mailing facilities and lists.

The Industrial Press

148 Lafayette Street, **New York 13, N. Y.**
Telephone: Canal 6-8120

Heating and Ventilating

Engineering magazine for the men who design systems and specify equipment



This monthly engineering magazine (established 1904) is read by the engineers—by whomever employed—who design, specify, install, and maintain systems for heating, ventilating, air conditioning, piping, plumbing, industrial refrigeration, process steam, and related services. These systems are installed in industrial plants, hospitals, office buildings, hotels, stores, schools, colleges, theaters, churches, institutions, government buildings, military installations, housing projects, etc. Readers include: Consulting engineers; engineers with architects, with large engineering-type contractors, with utilities; engineers with industrial plants and with large buildings; and others. To these men HEATING AND VENTILATING brings, month after month, a steady stream of crisply-written articles on the best current practice, boiled-down research results, mathematical short-cuts, handy tables and charts, popular "H & V Data Sheets." Result: High reader interest that carries over to the advertising pages. Informative 24-page booklet, "How Equipment Is Bought," describes the market and specifying practices.

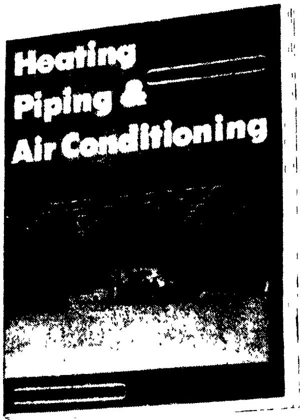
Subscription rates in U. S. and Canada; One year, $3; two years, $5; three years, $6. In all other countries, $5 per year.

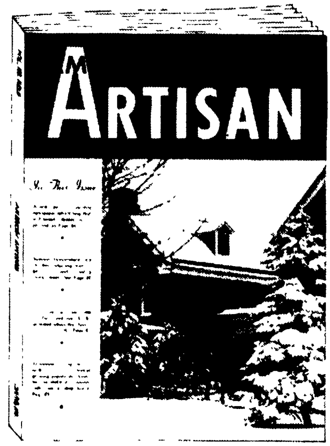
Heating & Plumbing Equipment News

The "new equipment" magazine serving Contractors and Wholesalers

This monthly "new equipment" magazine, published in tabloid format, reaches by controlled circulation 35,000 contractors, wholesalers and other groups concerned with the sale and installation of heating and plumbing equipment. Only publication that reaches *all* these categories: Radiator and warm-air heating contractors, oil-burner dealers, gas-burner dealers, plumbing contractors, and the wholesalers and distributors who supply all the foregoing. Editorial content consists of concise, illustrated articles describing new and improved equipment, materials, and tools placed on the market, also new catalogs and bulletins issued by manufacturers. First twelve issues brought more than 87,000 requests for information about specific products such as: Boilers, furnaces, gas-burners, oil-burners, toilets, sinks, pipe, fittings, valves, pumps, pipe-cutting and threading tools, hot water heaters, room coolers, water softeners, fans, blowers, controls, insulation, and many others. Sample copy and advertising facts upon request.

KEENEY PUBLISHING COMPANY
6 North Michigan Avenue, **Chicago 2, Ill.**





Heating, Piping and Air Conditioning carries the Journal of the A.S.H.V.E. as well as its own regular editorial section. Its field is that of industry and large buildings. It is devoted to the design, installation, operation, and maintenance of heating, piping and air conditioning systems in plants, commercial, institutional and public buildings.

Each January issue includes a complete directory of commercial and industrial heating, piping, and air conditioning equipment, which lists all products. their trade names, and the manufacturers' addresses. It is the established buying and specifying guide of the industry.

H. P. & A. C. is read by consulting engineers and architects . . . contractors . . . and engineers in charge of heating, piping and air conditioning in industrial plants, and other large buildings, federal, state, and city governments, school boards, and public utilities. All A.S.-H.V.E. members are subscribers.

Such coverage means, for the advertiser, consideration at all points in the selling of a heating, piping, or air conditioning product . . . consideration in its selection during the preparation of plans and specifications; in its actual purchase for installation; in its year-'round buying for operating and maintenance requirements. Without waste, the manufacturer of air conditioning products and equipment can reach through H. P. & A. C. those from whom he is seeking the necessary engineering acceptance.

Member—A.B.P.—A.B.C.
Subscription Prices—U. S. $3 per year.

AMERICAN ARTISAN covers the field of warm air heating, residential air conditioning, and sheet metal contracting. Its readers are warm air heating and sheet metal contractors, dealers, jobbers, manufacturers, and public utility companies.

Special features of each issue have been devoted to air conditioning since 1932, when it first became apparent that air conditioning for homes was to be along the lines of the central forced warm air heating system. As a result of the ready adaptability of this type of heating system to all air conditioning factors, hundreds of thousands of homes today have winter air conditioning—supplied through forced warm air heating with air cleaning and humidification. Cooling apparatus can be attached to these systems readily whenever year-'round air conditioning is desired.

Each January issue includes a complete directory of warm air heating, air conditioning, and sheet metal products and equipment, which lists all products, their trade names, and the manufacturers' addresses.

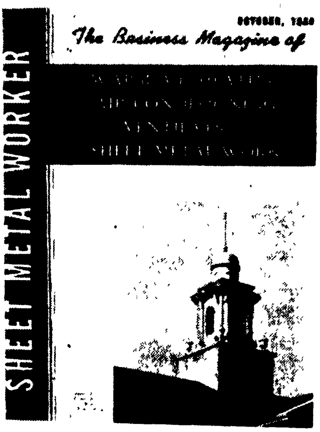
The key man in the residential air conditioning picture is the warm air heating and sheet metal contractor—the one man experienced in "treating air" at a central place and getting it properly distributed. And **American Artisan** is the key publication—because it reaches these key men with information that has made it the recognized authority on residential air conditioning practice.

Member—A.B.P.—A.B.C.
Subscription Prices—U.S. $3 per year.

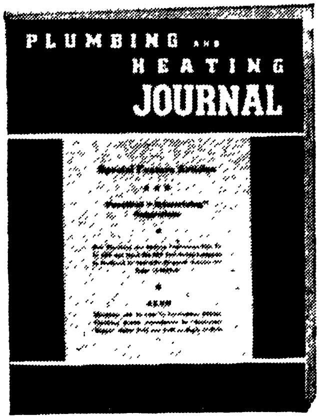
Canada, Central and So. America—$4.00 per year. Elsewhere $6 per year.

Sheet Metal Worker

Published by Edwin A. Scott Publishing Company
92 Martling Ave., **Tarrytown, N. Y.**



Subscription rates—$3.00 per year, U. S., Canada and Pan Amer. Foreign $4.00 Advertising rates on request



Subscription rates—$3.00 per year, U. S., Canada and Pan Amer. Foreign $4.00. Advertising rates on request.

THE January 1952 issue of SHEET METAL WORKER was its 78th Anniversary and Directory Number. It is the oldest publication in its field and is of vital importance to men interested in sheet metal work—air conditioning—warm-air heating and ventilation. Founded in 1874 and published to 1909 by David Williams Company; 1909 to 1920 by United Publishers Corp.; since 1920, by the Edwin A. Scott Publishing Co.

Subscribers are mainly merchandising contractors purchasing practically all products and equipment which they fabricate, erect or install. Manufacturers, jobbers and distributors also subscribe.

The market has three main divisions·

(1) Equipment for resale in connection with erection or installation work.
(2) Materials for fabrication.
(3) Shop equipment and supplies.

Circulation: SHEET METAL WORKER is a member of the Audit Bureau of Circulations, and Controlled Circulation Audit, Inc. Over 10,000 contractors are covered.

SHEET METAL WORKER also publishes books on heating, ventilating, sheet metal work, air conditioning, etc.

The Annual Issue published in January, contains a comprehensive and valuable Directory Section.

Plumbing and Heating Journal

Scott-Choate Publishing Co., Inc., *Publishers*
92 Martling Ave., **Tarrytown, N. Y.**

PLUMBING and Heating Journal is edited to furnish a well-rounded, efficient service to the men engaged in the plumbing, heating, ventilating and air conditioning fields. It covers both the technical and business phases of their work.

It gives free technical service through a staff of practical engineers; expert merchandising assistance, and its technical and business articles are by men of recognized competence.

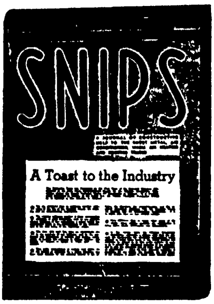
THE JOURNAL editorial department draws its news from scores of trained correspondents located at strategic points throughout the country.

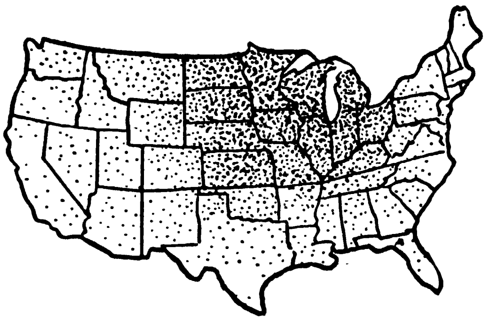
This combination of the technical, business, news and other aspects of the industry enables THE JOURNAL to achieve a finely balanced magazine that gives the reader the type of information he wants and needs, in brief, compact form.

A department "With the Water Systems," informs the trade of the latest developments in the rural plumbing field and its increasing potentialities for the plumbing—heating contractor, especially with the recent extensions of rural electric lines throughout the country.

Snips Magazine

5707 W. Lake St. **Chicago 44, Ill.**





While distributed in 48 States and widely in Canada, the shaded area of the map shows where SNIPS' circulation is most intense. It's the area where warm air heat predominates and where sheet metal work, in all its branches, has made its greatest strides.

READER INTEREST

SNIPS is packed each month, from cover to cover, with hundreds of live news stories and exclusive pictures of the trade it serves. No long contributed stories. It's all field gathered material, secured the hard way, rubbing shoulders with the readers. Such work gets for the periodical a reader interest seldom found in a trade publication. You'll find this feature the basis of the sensational inquiry pulling power and sales producing value of advertising space in SNIPS.

This live, friendly, close to the reader periodical "continues to go places" in reader interest, intense coverage and trade popularity.

An effective medium preferred by many Notable Industrial Advertisers year after year, and many Prominent Jobbers who know their market well.

We maintain our fine lists of known, reputable buyers, through constant cooperative work with the jobber trade of the industry.

Packed each month with practical applications of the latest advancements in the Sheet Metal, Heating and Engineering Field.

DON'T MISS OUR SPECIAL ISSUES

January Annual Issue—March Anniversary Spring Market Issue—September Fall Market Issue—December Holiday Greeting Issue—Further information gladly sent on request.

During the past year over 350 firms used space with us. These valued patrons see in our 13,000 readers, a carefully selected and substantial buying group, worthy of "talking to" month after month.

Rates for space have always been conservative. The carefully figured current rate with cream coverage available at the extremely low rate per thousand distribution is truly an exceptional value at this time, by any standard of comparison.

ABOUT SNIPS' MAILING LIST

It's truly a select group that gets SNIPS. They are the outstanding firms and individuals in the trade whom the better supply houses are selling or trying to sell. They are the contracting and installing concerns whom the principal jobbers and distributors of the industry—people who really know their territories best—consider worthy of cultivation by mail, promotion and salesmen's calls.

These advertising people will be alert to serve you. H. F. Hoy, Nick Carter, Jay Barton and Ed Carter in the Home Office. In the East, Snips Magazine, 501 - Fifth Avenue, New York 17, New York, Murray Hill 2-9192. In the West, Kimball-Menne Company, 1052 W. 6th Street, Los Angeles 17, California, Madison 6-9395, and Rm. 767, 681 Market St., San Francisco 5, Calif., Yukon 6-4588

ENGINEERS OF HUMAN COMFORT

The Heating, Ventilating and Air Conditioning Engineers through their work and research bring to our homes, schools, offices, factories, theaters, hospitals and other public buildings in both summer and winter, that climate best suited to our comfort and health. These men realize the basic importance of heating and ventilating as a primary element in the well-being of civilized mankind, living and working mostly indoors. They are truly *Engineers of Human Comfort.*

Started in 1894, by a small but progressive group, THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS now numbers over 8500 members, whose express purpose is to improve the Art through the interchange of ideas and the stimulation of scientific research and invention.

The Society membership now includes engineers, educators, scientists, physicians, architects, contractors, and leaders of industry. Membership consists of Charter, Honorary, Life, Presidential, Member, Associate, Junior, Affiliate and Student grades.

The management of the Society is entrusted to 4 Officers and a Council of 13 elected members. Continuity of policy is insured by electing 4 men annually for a 3-year term and retaining the retiring president on the Council for 1 year.

Two national meetings are held each year—the Annual Meeting during January or February, and the Semi-Annual Meeting usually in June or July.

The three major activities of the Society are: Membership service, Publication, and Research, the record of its accomplishments being permanently recorded in the annual TRANSACTIONS.

Headquarters of the Society are maintained at 62 Worth St., New York 13, N. Y., and its research laboratory, devoted to the study of fundamental principles of heating, ventilating and air conditioning, is located at 7218 Euclid Ave., Cleveland 3, Ohio.

In September, 1894, a little group of nationally known engineers, educators and manufacturers gathered in New York and agreed that the great art of heating and ventilating deserved and required recognition as an essential, distinctive and highly specialized division of modern engineering.

These keen, alert, progressive men knew that the methods and equipment of their day could be improved, even beyond their own vision, if all the personalities striving for such improvement could be welded into one organized cooperative group imbued with the same ideals and aiming for the same goal. They therefore formed themselves into the nucleus of such an organization and called it THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Foreseeing the need for research they made it one of their first acts to establish a Committee on Standards. That the Charter Members had great faith in their enterprise is evident, although little did they dream that progress would be so rapid in their profession.

During the intervening years, since that little group of 75 pioneers unfurled the banner of THE AMERICAN SOCIETY OF HEATING AND VENTILAT-

ING ENGINEERS, thousands of the real leaders of thought and action in heating, ventilating, and air conditioning have gathered about that standard and carried it proudly before them far along the way of outstanding accomplishment. They may be identified among engineering groups by the distinctive emblem which was adopted by the Charter Members.

The first Annual Meeting was held in New York, N. Y., January 22–24, 1895, and the organization was incorporated that same year, under the laws of the State.

A. S. H. V. E. RESEARCH LABORATORY

Since 1919 THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has maintained a permanent research staff and facilities devoted solely to the study of fundamental problems in the field of heating, ventilating and air conditioning. During the past quarter century much has been done to advance the art by establishing scientifically sound data which the engineer can apply in the design, operation, and maintenance of heating, ventilating, and air conditioning systems and equipment.

For twenty-five years the Society's Research Laboratory was located at the U. S. Bureau of Mines Building, Pittsburgh, Pa. The Laboratory was moved to Cleveland in 1944, and early in 1946 the Society purchased premises at 7218 Euclid Avenue. In addition to work at the Society's Laboratory, a substantial part of the research program has been carried on through the medium of cooperative agreements with leading educational institutions of the United States and Canada.

All research activities are planned and supervised by the Committee on Research of 15 elected members, assisted by various Technical Advisory Committees of the Society. The research activities are financed from Society funds, of which a portion comes from membership dues and from its publications, and these funds are amplified by contributions from friends in the industries engaged in the general field of heating, ventilating, and air conditioning.

The new Environment Laboratory is an important addition to the Society's facilities for fundamental research. It has been constructed for the development of authentic data for the design and installation of panel heating and cooling.

The room is the result of a conference in early 1947 of more than 100 representative engineers from consulting firms, technical societies, universities, trade associations and industry leaders. The group agreed that the Society should take the initiative in acquiring data for panel heating and cooling which were lacking at the time. In June 1947 the A.S.H.V.E. Committee on Research formed a Technical Advisory Committee on Panel Heating and Cooling to advise on technical details of this research. The work was divided into four spheres of activity:

1. Heat distribution within and behind the panel.
2. Heat transfer between the panel and the space in the room.
3. Comfort conditions.
4. Controls.

In order to accomplish parts 2 and 3 of the proposed program, it was decided to construct a full-sized room wherein the temperatures of all the room surfaces might be controlled independently. So that the room could also be used for a study of the relationship between human comfort and radiation, it was designed to permit division into two rooms when desired. After two years of planning, construction began in 1950 and was completed in November 1951.

The room itself is 25 ft by 12 ft with a ceiling adjustable in height from 7 to 12 ft. It can be divided into two separate rooms each 12 ft by 12 ft. The room is housed in the main A.S.H.V.E. Laboratory and is 3 ft above the main laboratory floor. The interior surfaces of the room are aluminum panels backed by ⅜ in. copper coils on 3 in. centers. Outside wall temperatures as low as 0 F can be simulated. Floor and ceiling surfaces can be kept at temperatures as high as 180 F.

The instrumentation provided, permits readings of surface temperatures within the room (400 thermocouples attached to back of panels), air temperatures, rate and temperature of infiltration air, and heat picked up or given off by all surfaces. It is intended to add more equipment which will permit maintaining air temperatures within the room, independent of surface temperatures. This will allow human comfort studies under varying internal dry bulb temperatures, relative humidities and air change rates.

It is expected that some of the practical results of the use of this room will be:

1. To measure the rate of the exchange of heat between surfaces of the room and air in the room as they are affected by temperature, humidity, ventilation, etc.
2. To determine the influence of radiation on human comfort, and using this to improve the A.S.H.V.E. Comfort Chart.
3. The use of the room to study the removal of heat by heat absorbing panels.
4. An evaluation of different types and colors of paint and wallpaper on panel performance.

In addition to these projects, the room also provides the A.S.H.V.E. Laboratory with a flexible research tool that should be extremely valuable for many years to come.

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