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THE ELEMENTS
OF
HEATING AND VENTILATION

A TEXT-BOOK
FOR
STUDENTS, ENGINEERS AND ARCHITECTS

BY

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FIRST THOUSAND



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PREFACE

THE aim of the author in preparing this book has been to bring together in logical order and in a small volume the necessary data from which to design the heating and ventilating systems of buildings. In doing this he has been guided by his own experience in the layout of such systems and by the previous works of many authors which he has used. He has consulted these works freely, as well as the valuable experimental data prepared by a number of the companies building apparatus for heating and ventilation. Where such work has been used credit has been given to the author or the company. These data, which in general are scattered among various books and pamphlets and must be used by the engineer in his work, have been brought together and placed in a logical position in a single book. The data are founded on careful experiment and may be used with confidence. In selecting these data, which are given in the forms of tables and curves, the endeavor has been made to include all that are necessary to solve any problem which may arise in connection with the warming of buildings or the supply and delivery of air. Many of the curves and tables are originally given, and in all cases these have been recomputed or redrawn for this book. Several methods given for the solution of problems are new.

The book is intended for the use of upper-class men in technical schools, for engineers, architects, and superintendents of buildings. All phases of the work are illustrated by actual problems for which the slide rule has been used. The use of the slide rule is recommended and advised for these problems. Certain architectural information has been added to aid those

who have not had experience with constructions of various forms.

The plan of the work has been for a continuous course of study without any omissions. Each chapter forms a unit in the subject of heating and ventilation. The tables and curves from commercial apparatus are given to aid in the layout of work, so that the engineer may have in one volume the necessary information for a possible solution. There are many different forms of apparatus which have not been mentioned in tabular form and these the engineer will probably have in his catalogue library. It is impossible to include all in a small volume. Those given have been selected because they represent common forms in use.

The author desires to thank his wife, Mary E. Lewis Greene, for the care she has taken in the preparation of copy and the reading of proof. He desires to thank those authors and manufacturers from whose works he has gained much as a student, or whom he has quoted in this book.

A. M. G., Jr.

SUNNYSLOPE, TROY, N. Y., November 26, 1912.

TABLE OF CONTENTS

CHAPTER I

	PAGE
METHODS OF HEATING AND VENTILATING BUILDINGS.....	I

CHAPTER II

AMOUNT AND CONDITION OF AIR FOR VENTILATION.....	20
--	----

CHAPTER III

LOSS AND GAIN OF HEAT.....	45
----------------------------	----

CHAPTER IV

RADIATORS, VALVES AND HEAT TRANSMISSION FROM RADIATORS.....	73
---	----

CHAPTER V

METHODS OF CALCULATING HEAT REQUIRED FOR ROOMS.....	118
---	-----

CHAPTER VI

DIRECT STEAM HEATING.....	127
---------------------------	-----

CHAPTER VII

HOT-WATER HEATING.....	153
------------------------	-----

CHAPTER VIII

INDIRECT HEATING.....	171
-----------------------	-----

TABLE OF CONTENTS

	PAGE
CHAPTER IX	
FURNACE HEATING.....	237
CHAPTER X	
FURNACES AND BOILERS.....	255
CHAPTER XI	
DISTRICT HEATING.....	279
CHAPTER XII	
TEMPERATURE CONTROL AND DRYING BY AIR.....	300

THE ELEMENTS OF HEATING AND VENTILATION

CHAPTER I

METHODS OF HEATING AND VENTILATING BUILDINGS

THERE are several methods of heating buildings in use to-day. For small buildings the **hot-air furnace** is quite common. In this, as shown in Fig. 1, the heat from the burning of coal is used to heat air on the opposite side of the **fire pot** *A* or **radiator** *B* through which the hot gases pass on their way to the chimney. The heated air from the **dome** or top of the heater is carried through the **leader pipes** *C* to **boots** *D* at the bottom of the **risers** or **heat stacks** *E*. It rises through the stacks to the various **register faces** *F*, where it enters the rooms to be heated. The air is forced to rise through the ducts by a difference in pressure due to the difference in the weight of the hot air inside and the cold air outside. The cold air enters at the bottom of the **heater casing** at *G*, being taken from the outside at *H* or the inside at *I*.

In some house work the foul air is removed by **vent stacks** *K* leading to the attic or to a chimney, while in most installations this removal is cared for by leakage of air from the windows or doors.

This method of heating is also a method of ventilation if the cold air or a part of it is taken from the outside.

Another common and excellent method of heating is that in which **radiators**, supplied with hot water or steam, are placed in the various rooms or spaces to be heated. As shown in Fig. 2

the steam is made or the water is heated in the boiler or heater *A* from which it is taken through the distributing main or flow

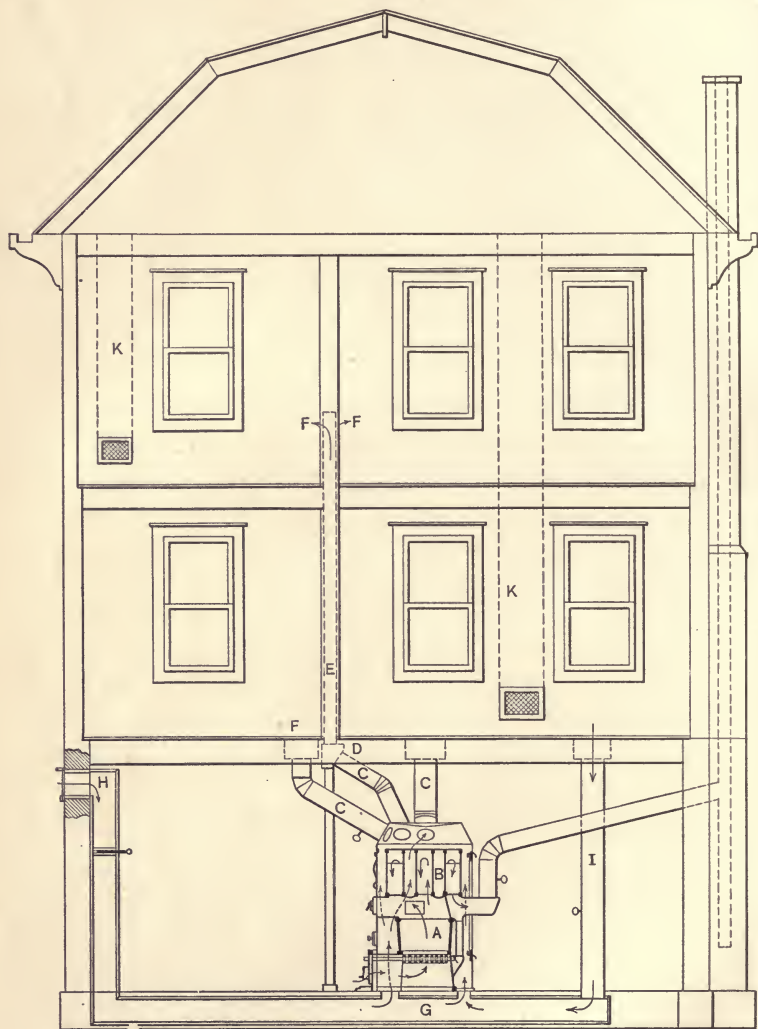


FIG. 1.—Hot-air Furnace Heating.

main *B* to the various risers *C* and thence to the radiators *D* by the connections *E*. The return water or condensed steam known

as the return is carried through branches *F*, return risers *G* and return main *H* to the boiler or heater *A*. The radiators are

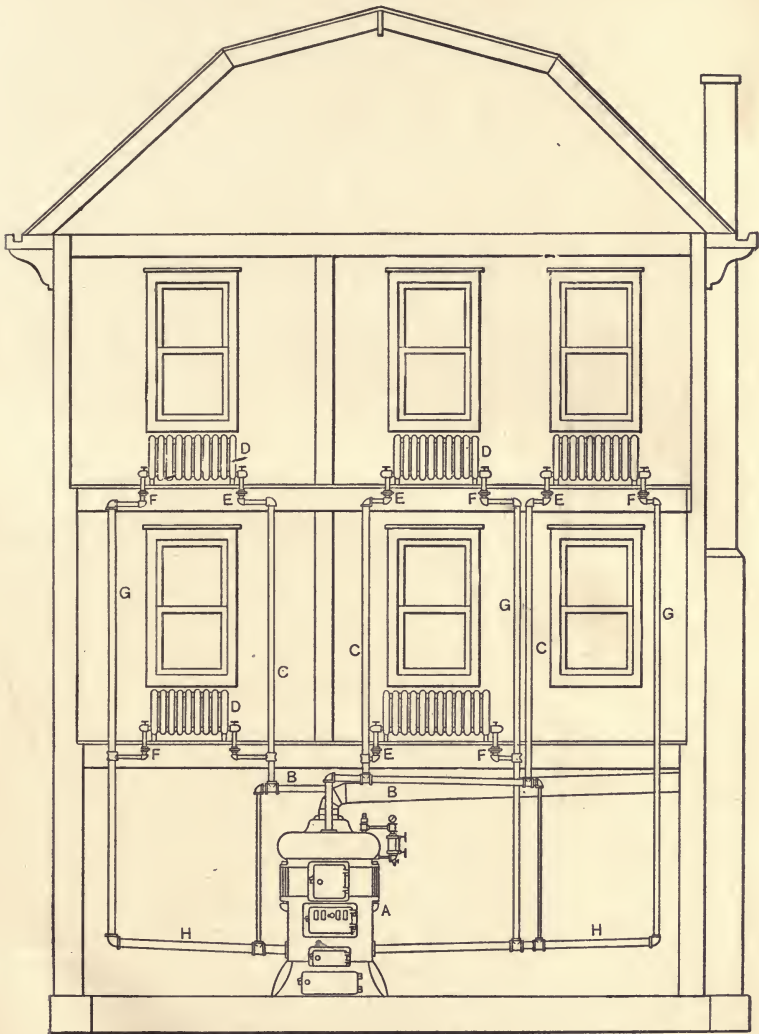


FIG. 2.—Direct Heating by Steam or Hot Water.

usually placed in front of windows where the up current of heated air will form a blanket warming the cold air which may

enter around the window and keep the same from producing cold currents or drafts. Each radiator is controlled by valves

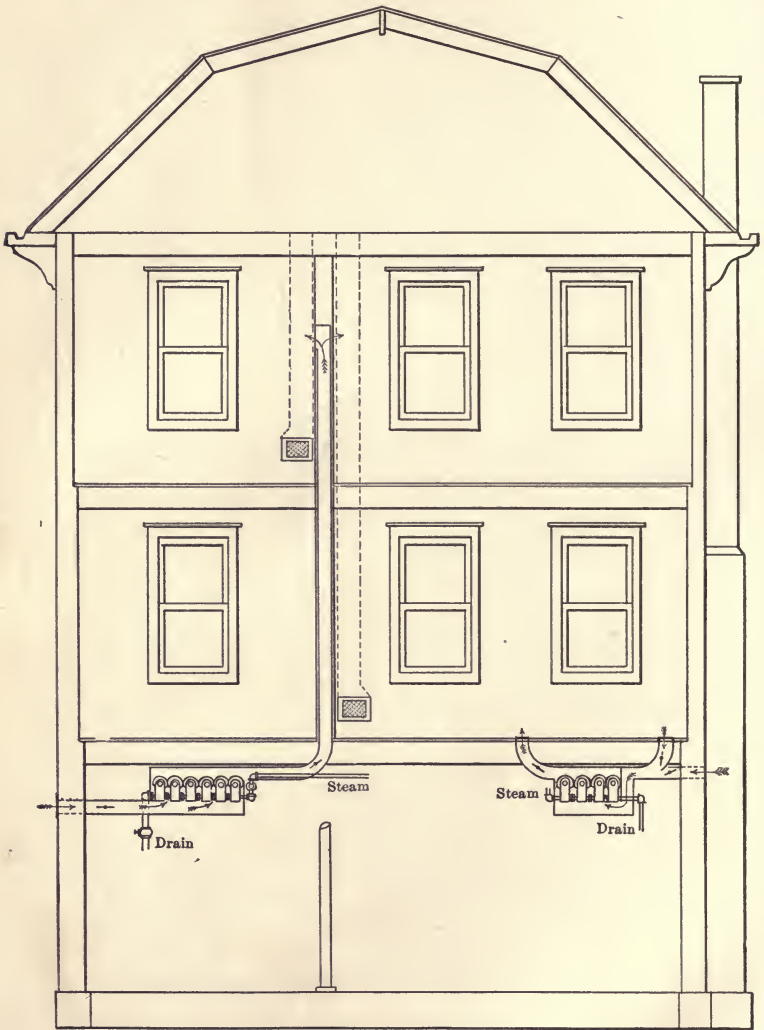


FIG. 3.—Indirect System of Heating with Individual Stacks.

on inlet and outlet side. This is known as the **direct radiation system**.

The **indirect system** of heating is that in which the radiators are used to heat air which is introduced into various rooms

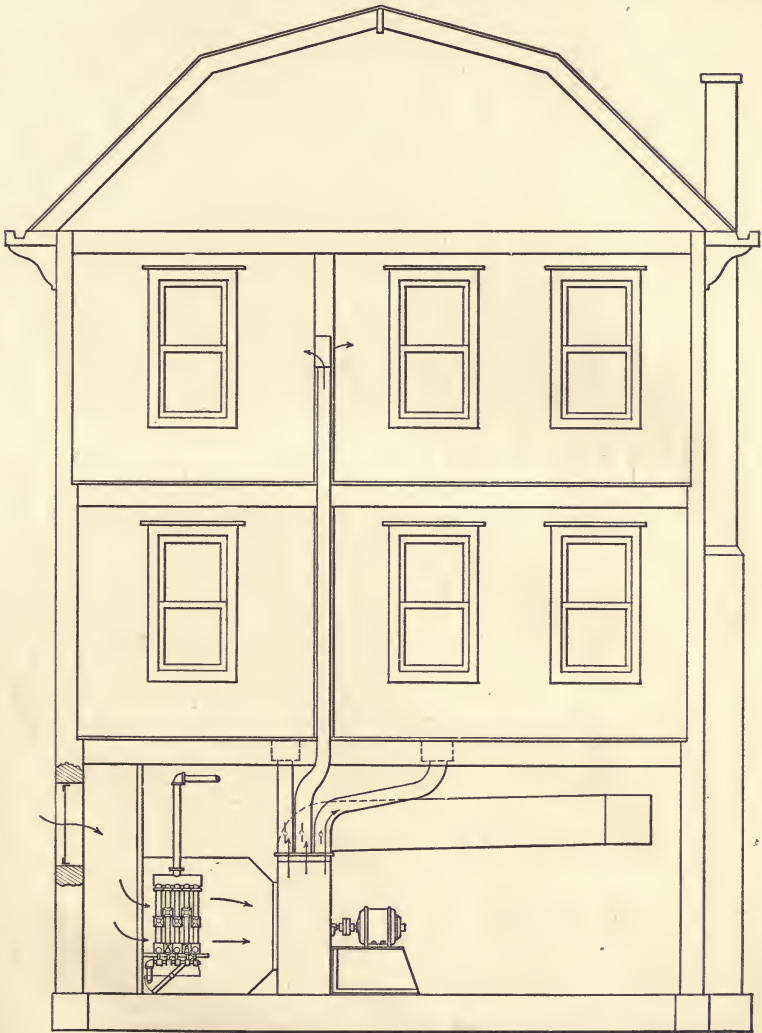


FIG. 4.—Plenum System of Indirect Heating with Single Coil.

by **riser stacks** as in the hot-air furnace method. Fig. 3 shows this method as applied to a residence where natural circula-

tion is depended on, while Fig. 4 shows one where a fan blower is used. This indirect system is a combined method of heating

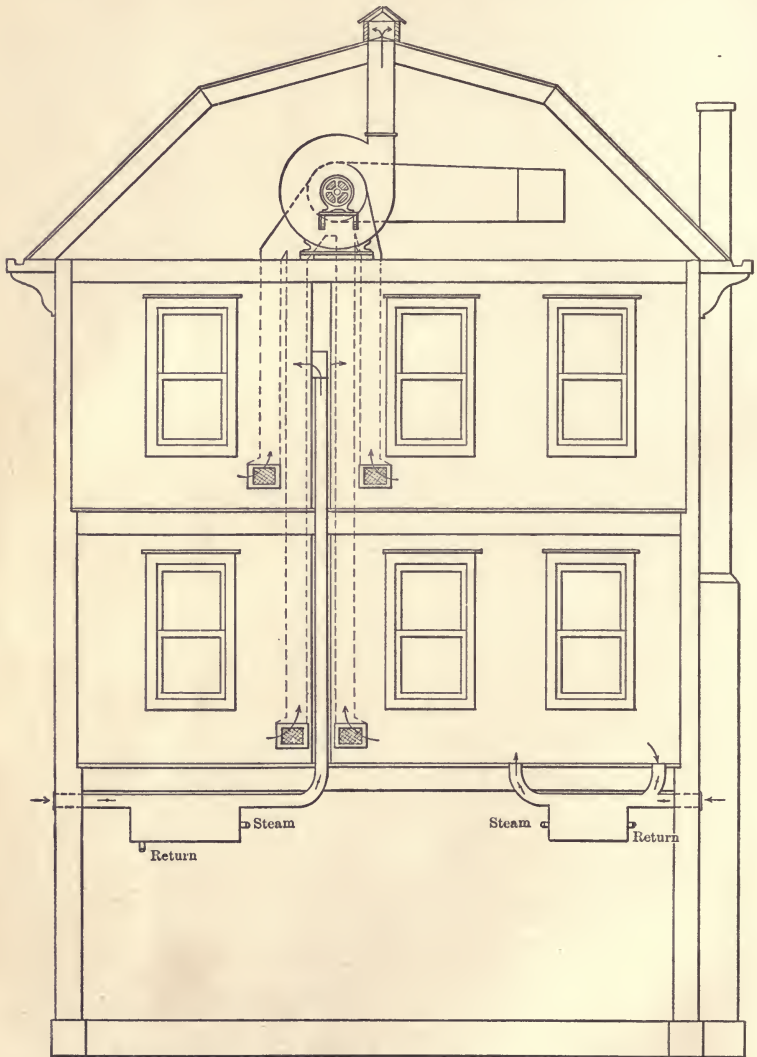


FIG. 5.—Vacuum System of Ventilation with Indirect Heating System.

and ventilation. It is used where definite ventilation is needed or where it is deemed advisable to have no radiators in the

room. The application of the method of steam heating of Fig. 3 is often found in buildings of such a size that hot-air furnace

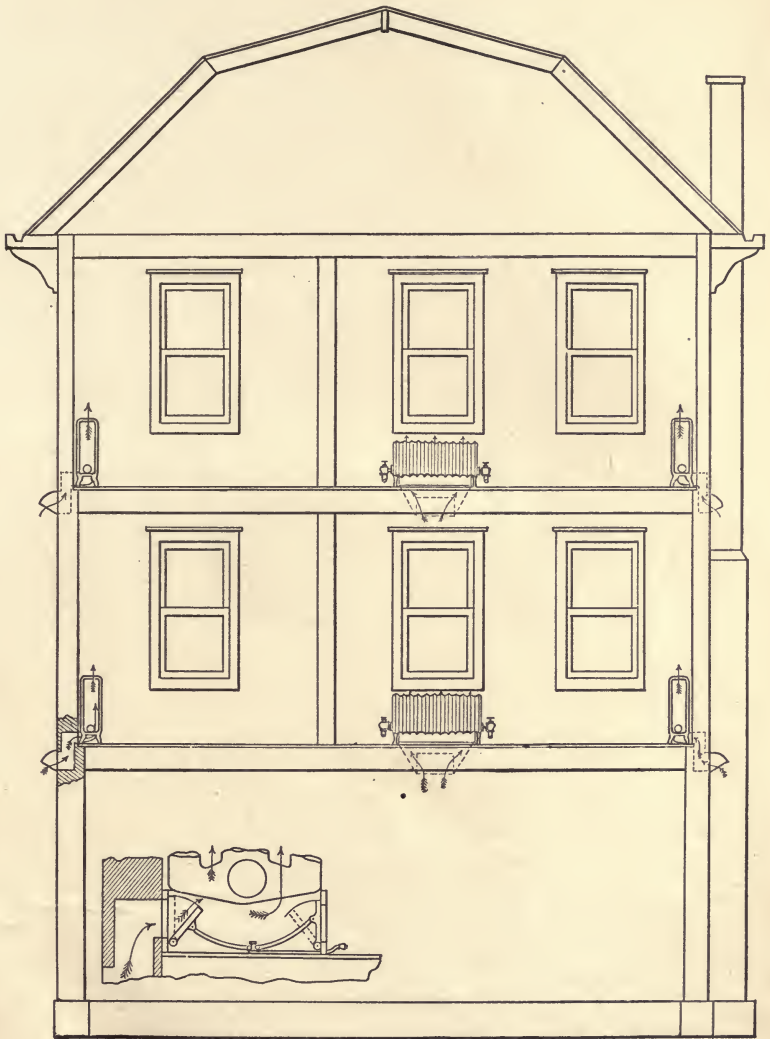


FIG. 6.—Direct Indirect System.

systems would not be possible and yet it is desired to bring the heat in by air with natural ventilation. The installation of an

indirect heating coil at the bottom of each **stack** or group of stacks makes this possible and the positive steam circulation furnishes the heat to each radiator. The indirect system of heating employing a **fan blower** is found in places where it is advisable to centralize most if not all of the heating surface in one set of coils and to make the flow of air positive to all parts of the building; a fan blower is used to produce a pressure difference. The fan may be employed to drive the air into the rooms as shown in Fig. 4, in which case the system is known as the **plenum system**, while in some cases the air is drawn through the heating coils and into the room by the suction of a fan which discharges the air from the room into the atmosphere. The air

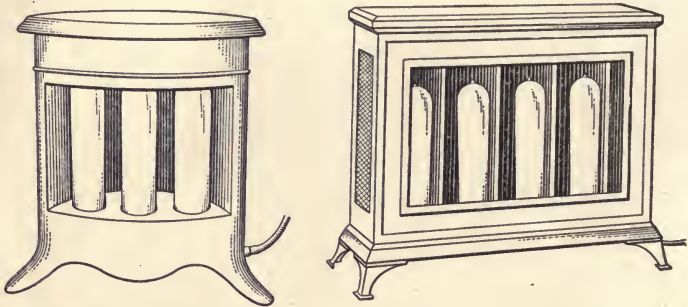


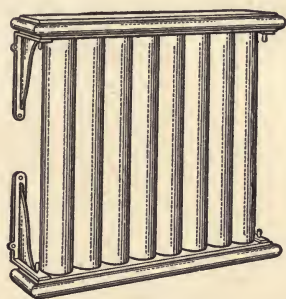
FIG. 7.—Lamp Radiators of G. E. Co.

in the room is under reduced pressure and for this reason the system is known as the **vacuum system** of ventilation, Fig. 5.

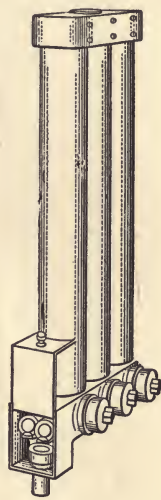
At times the ventilation of a room is accomplished by connecting the room to the outer atmosphere by a series of openings, each leading to a radiator. The radiator is so constructed that a series of flues is formed and as the heated air rises through these flues cold air enters from the outside. These radiators are known as **direct-indirect radiators** and as usually constructed, Fig. 6, they are provided with chambers at the base which can be used to cut off the external air when necessary and permit circulation from the inside.

The **electric method of heating** has flexibility, rapidity, and convenience in its favor, but its great expense confines its use to

special places and for specific aims. Fig. 7 illustrates a luminous form of **electric radiator** for use in removing the chill from bathrooms or dressing rooms where heat is needed on short notice and for a short time only. The one on the left is known as a **three-glower luminous radiator** and that on the right, a **four-glower**. Fig. 8 represents **coil-resistance radiators** or air heaters used on steamships, buildings or electric cars, when



Wall Type Tubular Electric
Air Heater.



Stateroom Type Tubular
Electric Air Heater.

FIG. 8.—Resistance Air Heaters of the G. E. Co.

the expense of operation and danger from individual hot water or hot-air heaters makes this form of heater practicable.

The glower luminous radiators are made of various capacities. Some of them are made small enough to be put on a lamp circuit. These are of 500 watts capacity and consist of two glowers. With four glowers there are two sizes built, one of 1000 watts and one of 2000 watts. The radiators are built for two sets of voltages, 95-125 and 200-225. The tubular air heaters shown in Fig. 8 are of the **wall type** and the **stateroom type**. They are made to dissipate 300 and 500 watts per tube. With

the first consumption there is no danger of reaching a scorching temperature. In the case of the stateroom heater the tubes are separately controlled and a junction box is shown to connect

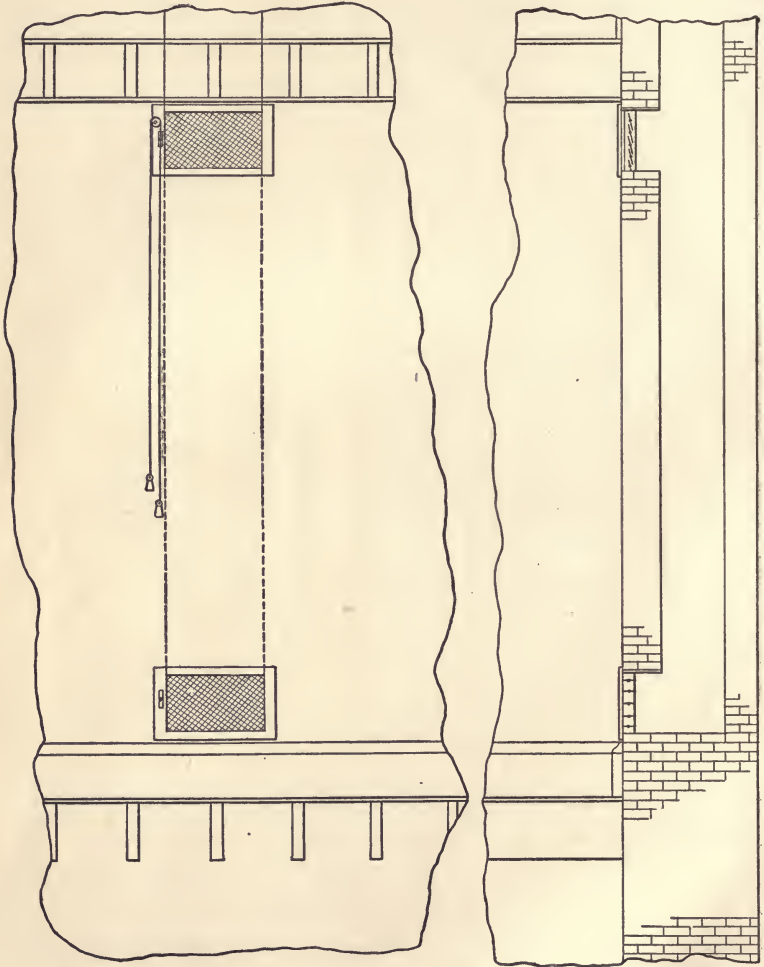


FIG. 9.—Floor and Ceiling Vent.

to the conduit system. The heating element in these air heaters is of special form originally intended for rheostat work; in it the wire of special composition is wound on a soft asbestos tube,

then fixed mechanically and the tube solidified by coating it with a fireproof cementing compound.

The manner of ventilating rooms with the various systems of heating described above has been examined, but there yet remain several systems for which the accompanying method of ventilation has not been described.

In many installations little or no provision is made for ventilation, the freshening of the air being produced by leakage around windows and doors. Such a method is not reliable and except for house or office installation in which there will be few occupants in large rooms it should not be employed.

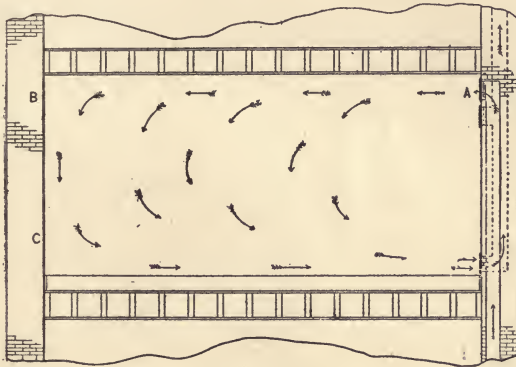


FIG. 10.—Method of Ventilation.

In some cases the fresh air is supposed to enter from the windows or doors while the foul air is taken out through ventilating ducts leading to the roof or attic. In this case, Fig. 9, there are usually two registers leading to the ventilating ducts, one at the floor and the other at the ceiling. The air is ordinarily taken from the floor of the room, as this will cause a better circulation. The upper register may be used when necessary to clear the room rapidly.

When fresh air is brought into the room from a hot-air furnace, fan and heater or from the heating box, this air should be delivered near the top of the room on its cold side as shown in Fig. 10. In this way the air is thoroughly mixed and there

is no chance for the air to short-circuit across the room. If air entered at *A* and was taken out at either *B* or *C* there would be a chance for ventilating air or in some cases for the heating air to be carried out before it had a chance to either heat or ventilate it and certain portions of the room would remain unprovided for. The supply on the cold wall makes it possible for down currents of cold air to pass out at the vent openings before mixing with the warmer currents.

When **vents** alone are placed in a room with direct heaters these should be placed on the exposed wall.

Although this method of bringing the heated air in through ducts on the exposed wall has certain advantages, the method of ducts in inner partitions is often employed because this location can be better arranged in the construction of the building and the heat loss from these can be used to warm the building, thus the heat will not escape directly to the atmosphere as may occur from the outer walls.

With direct radiation in steam, hot water or electric heating, air when introduced is for ventilation only and in such cases the air is heated to the temperature desired for the room and not above this temperature, as is the case when the air has to supply the heat to care for losses through the walls. The air in all of these cases should be introduced as described.

When the air is used for ventilation only it is brought to the rooms by ducts at 70° or at the temperature of the room, the air being introduced as shown in Fig. 4. In this case, however, it is customary to deliver the air into a main and to take branches to the various rooms. When, however, the rooms are heated as well as ventilated by the air it is then necessary to bring in the air at such a temperature that in giving up the heat to care for the heat losses it is cooled to the desired room temperature. Since the amount of heat for the heat losses varies in different rooms with an independent variation in the amount of ventilation it is necessary to bring air at different temperatures to the various rooms. This is accomplished by having all of the air **tempered** to about 70° F. and then to take a portion of this to a **reheater** and increase its temperature.

If now the **tempered air** and **heated air** are mixed in proper amounts any temperature from that of the tempered air to that of hot air can be obtained. If the hot air is not mixed with the tempered air, the highest temperature is reached while tempered air alone will give the lowest temperature. This is accomplished by separating the air into two portions at the reheater. If now the tempered air and the heated air are carried in pipes, flues or ducts throughout the building, a **two-duct** or

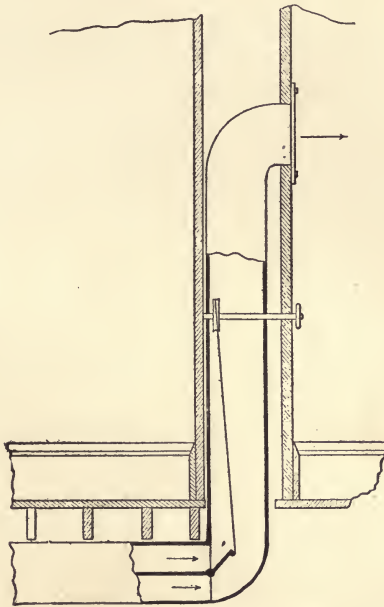


FIG. 11.—Hand Control of Mixing Damper.

double-duct system is the result. In this case two **branches** are carried to each **riser duct**. The **damper** at the bottom of this riser is hinged at the partition between the two ducts. By raising this damper by hand, Fig. 11, or by the **thermostatic motor**, Fig. 12, the air from the top duct is throttled while by lowering, the lower duct is cut off partially. In this way the temperature of the room into which the air discharges may be regulated.

At times this mixing is done just beyond the heater as in Fig.

13, and in this case separate lines of single pipes are run from this point to the riser for each line. This gives what is known as the **single-duct** system.

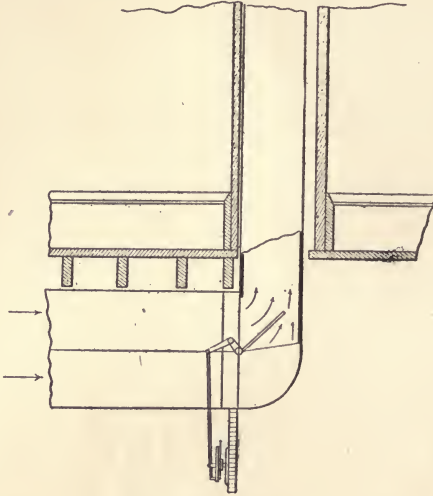


FIG. 12.—Motor Control of Mixing Damper.

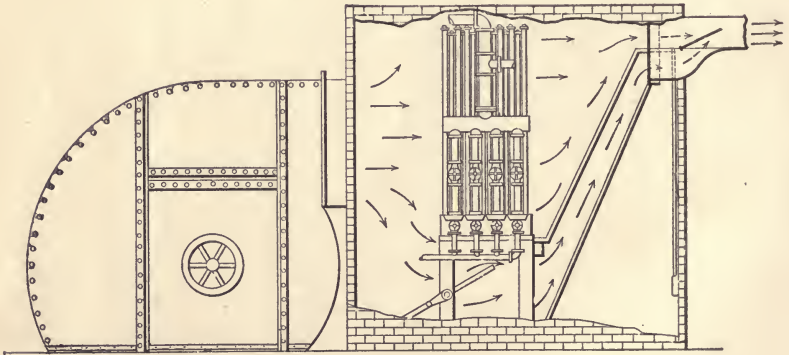


FIG. 13.—Single Duct System.

The use of the **ozonator** for the purpose of purifying the air must be mentioned at this point as well as the claims made by certain physiologists that the mere circulation of air will maintain its power of supporting respiration until the carbon dioxide

content is much higher than the amount usually allowed. As will be mentioned in the next chapter many claim that CO_2 is but an indicator of the presence of other impurities. If this be so then the circulation of air may cause some of these impurities to oxidize and in the case of the use of ozone, the breaking down of O_3 into O_2 and O give an active nascent atom which probably oxidizes the impure emanations from the breath and purifies the air. Experiments with ozonators have shown this to be the case.

Fig. 14 gives a view of the ozonator built by the General Electric Co. This, for alternating currents, consists of an electric motor and fan mounted on top of the case which contains a step-up transformer and six or more ozone generators. If direct current is supplied the motor is replaced by a rotary converter which not only drives the fan blower but converts the direct current into alternating current, so that this may be used in the transformer to get a sufficiently high voltage to produce a violet discharge but not so high that the discharge will be intense enough to produce nitrous oxide. The generating units consist of a number of glass tubes coated on the outside with a metallic coating and having on the inside an aluminum electrode composed of a series of aluminum cups mounted on a spindle. The diameter of the cups is smaller than that of the tube so that there is a small definite air-gap between the electrode and the tube. The exterior coatings are connected in multiple with one terminal of the high side of the transformer and the electrodes are connected in multiple with the other. The high alternating voltage induces charges on the inside of the tube and a violet discharge takes place across the gap, the energy of which is used in the production of ozone. The blower on top

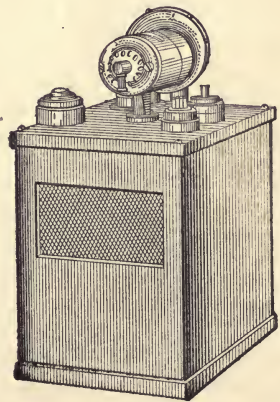


FIG. 14.—G. E. Alternating Current Ozonator.

forces 4000 cu.ft. of air per hour through the generators and this charges the air with six milligrams of ozone per cubic meter. To do this requires about 70 watts on A.C. supply and 87 watts on D.C. supply. There are two switches on the ozonator, one controls the whole apparatus, the other, a three-point switch, controls the voltage of the transformer, regulating the amount of ozone. The first point gives four milligrams per cubic meter; the second five, and the third six. The blower is always in operation when the first switch is turned on. The transformer may be cut out by the second switch at the off position. The ozonator requires $19\frac{3}{8}$ " in height and has a base $11\frac{1}{2}" \times 14"$.

The results obtained by the apparatus are remarkable as the following quotations from the General Electric Bulletin 4912, from which the above description is taken, will illustrate.

"Perhaps one of the most universal applications of ozone will be in the treatment of air for the destruction and removal of noxious odors, organisms and emanations. This subject has received some attention in America where the matter of ozone application is a new one, and much attention in Europe, particularly in France, where ozone has long been recognized as a valuable agent of sanitation.

"Ozone acts on the air as a bactericide as well as a powerful agency of deodorization. For the purpose of studying the power of ozone to destroy noxious odors, Scoutettin chose a ward of the hospital at Metz, having a magnitude of about 1100 cubic meters. In this hall he placed two piles of manure about 10 meters apart. These manure piles were permitted to remain 48 hours, during which period the room became filled with a pernicious odor indicating an advanced stage of putrefaction, as shown by the evidence of the ammonia evolved.

"When this had been accomplished, two vessels of 8 liters capacity were opened in the hall, permitting their contents of ozonized air to diffuse therein. The ammoniacal odor diminished considerably, though it did not disappear completely. The manure was then removed and the experiment repeated.

This time the odor disappeared completely and rapidly, the noxious gases, hydrogen sulphide, carbon bisulphide and ammonia having been destroyed.

“Experiments with cultures of the *tubercular bacilli* have shown that these grow with only one-fourth the rapidity of check cultures, when exposed to the action of ozonized air.

“These results show that where ozonized air comes in contact with the living colonies their development is impeded; but that when the bacterial colony grows deep within the culture medium, the action of ozone applied to the surface only is less marked, if not altogether imperceptible.

“This is what should be expected according to Ohlmüller, who has demonstrated that the bactericidal action of ozone is greatly interfered with in the case of colonies growing on organic matter; for the ozone oxidizes the organic medium, thus destroying itself, before it makes sensible its action on the bacteria. Ozone destroys itself in oxidizing organic matter and coagulates albuminous matter.

“It may be deduced from the foregoing that any extraneous organic matter found in air which it is desired to sterilize, will diminish the action of ozone by combining with it; and in consequence, the air should be first filtered whenever practicable. Many failures to produce sterilization in researches on ozonizing air have resulted from the presence of a relatively large amount of organic matter in the air.

“Ozone will find an application in the sterilization and deodorization of the air of hospitals, apartments, studios, schools, etc., wherever there is likely to be large crowds.

“In stables, chicken coops, toilets and factories, where there are evolved noxious emanations, ozone will greatly ameliorate the conditions. In particular, the shops for assorting rags, manufacture of fertilizers and factories which work gelatin, glue, hides, hair, fat, bones, horn and other slaughter-house by-products, and those which are a source of emanations dangerous to the public health, will find in ozone a powerful ally.

“Wherever pure sterile air is of value in the factory either before, during, or after the completion of the product, e.g., distilleries, breweries, wine houses, etc., the use of ozone should be resorted to.

“The Art Theater on State Street, Schenectady, a moving-picture show, had experienced difficulty with its ventilation. The theater consists of a hall about 30 by 100 feet, and the ventilation is provided by a suction blower capable of aspirating about ninety thousand cubic feet per hour. The management were very desirous of providing the best ventilation possible, as is evidenced by the elaborate and expensive system cited. It was found, however, that notwithstanding the magnitude of the blower, ‘crowd odors’ persisted in the room. The blower was as large as could be used, for anything larger would have produced obnoxious drafts.

“As a solution to the trouble, an ozonator was installed above the front entrance to the theater, in such a way as to permit the ozonized air to diffuse into the current of ventilating air drawn toward the aspirator. The instantaneous effect of this was remarkable. The theater has been entirely deodorized and even during the hottest weather of the past summer the air within the theater has been fresh, cool and odorless, excepting for the faint and rather pleasant smell of the slight excess of ozone.

“The next case which we may cite provides an even more remarkable instance of the efficacy of ozone in deodorizing obnoxious air, since this case relates to a factory in which, through the nature of the work carried on, emanations are evolved, which constitute a vehicle of certain volatilized diluents and solvents of the varnishes and adhesives used. In a workshop some 75 feet by 200 feet upwards of two hundred girls are employed in the preparation of various articles of pasted mica. It is easy to realize that the problem of providing clean air under such conditions will always be a difficult one, and in the present instance a considerable expenditure of money and ingenuity was incurred before the correct solution was found.

“ Finally two ozonators were installed, one near each end of the room, and the windows which had to be wide open to clear the air were now only slightly open. A number of other instances are recorded in this bulletin showing the usefulness of this apparatus.”

CHAPTER II

AMOUNT AND CONDITION OF AIR FOR VENTILATION

AIR is composed principally of **nitrogen** and **oxygen** with small quantities of **carbon dioxide**, water vapor and the rare element, **argon**.

The amounts of the various constituents are given below:

	By Weight.	By Volume.
Nitrogen.....	75.50%	78.06%
Oxygen.....	23.20	21.00
Argon.....	1.3	.94

Approximately this is

Nitrogen.....	77.	79.
Oxygen.....	23.	21.

Helium, **metargon**, **neon**, **krypton**, and **xenon** have been discovered in air in small but constant amounts.

Free air is found to contain from 3 to 4 parts of CO₂ in 10,000 parts by volume.

The amount of **water vapor** contained in air varies with the temperature and saturation. This vapor is not always sufficient to **saturate** the air with vapor. The ratio of the amount contained to that required to saturate the air at the temperature considered is called the **relative humidity** while the actual amount of vapor per cubic foot is known as the **absolute humidity**. Thus at 75° F. air may contain 0.00135 lb. or 9.5 grains of water vapor per cubic foot. This quantity will just saturate it. If the air is half saturated or the relative humidity is 50 per cent the quantity of water vapor is 4.75 grains to the cubic foot.

In addition to the above substances there may be impur-

ities given off from persons or processes and these are variable quantities depending on the particular place and time.

For general work, the air will be assumed to contain 79 parts of nitrogen by volume, 21 parts of oxygen, 4 parts in 10,000 of CO₂ and a variable amount of water vapor.

The amount of fresh air to be taken into a room depends on the permissible amount of CO₂ allowed. It is estimated that 20 cu.ins. of air is inhaled at each respiration and there are about 20 respirations per minute, making about a quarter of a cubic foot of air per minute. The CO₂ in this respired air amounts to about 4 per cent so that the respired air has its CO₂ content increased 100 fold from 4 parts per 10,000 up to 400 parts. Although the CO₂ is heavier than the air, having a molecular weight of 44 while the air has a weight of 28.9 it diffuses through the air and forms a mixture of air and CO₂. This foul air is then diluted by the air in the room and the condition of the air in the room is raised to x parts of CO₂ per 10,000 parts. If V is the volume of the air allowed per person per hour, the equation showing the value x would be

$$\frac{4V}{10000} + \frac{400 \times \frac{1}{4} \times 60}{10000} = \frac{xV}{10000}, \quad \dots \dots (1)$$

If the quantity of air respired per minute is changed from that assumed and if the quantity of CO₂ in the exhaled air is not 4 per cent for actual conditions, other values can be used in the equation. The equation reduces to

$$x = 4 + \frac{6000}{V}, \quad \dots \dots (2)$$

or

$$V = \frac{6000}{x - 4} \dots \dots (3)$$

Formula (3) gives the quantity of air per hour if the allowable quantity in vitiated air x is known. Formula (2) gives the quality of vitiated air if V is known. According to Pettenkofer

the amounts of CO₂ per hour developed by a strong workman at work is 1.275 cu.ft. while at rest it is only 0.825 cu.ft. These are higher than the amount assumed above. Scharling gives 0.635 cu.ft. as the amount of CO₂ per hour produced by men. This agrees with the assumption made above. The other values of CO₂ exhaled per person, given by Scharling are as follows:

Women.....	0.600 cu.ft. per hour
Young men.....	0.614 " " "
Young women.....	0.455 " " "
Boys.....	0.363 " " "
Girls.....	0.343 " " "

The presence of 11 parts of CO₂ per 10,000 in air when the increase has been due to respiration is found to be oppressive and harmful. In cases of gatherings of healthy persons this may reach 15 parts while with sick, it should be kept as low as 7 parts. Some claim that it is the other gases or organic matters which are exhaled which cause this air to be harmful. The lungs and the moistened mouth and nostrils must allow saliva and other fluids to be discharged in the forms of vapors and these with any bacteria exhaled contaminate the atmosphere. It may be that the CO₂ is merely an indicator of the other harmful constituents in the atmosphere of rooms which are poorly ventilated. When the CO₂ content is about 7 parts to 10,000 there is no evidence of discomfort and this may be taken as the limit of x for proper ventilation. Eq. (3) then gives

$$V = \frac{6000}{7-4} = 2000 \text{ cu.ft. per hr.}$$

This value is usually employed for the proper ventilation of buildings, 2000 cu.ft per person per hour. In cases where the number of persons is not definite it is customary to express the amount of air in terms of the number of times the volume of the rooms is changed per hour.

Rietschel recommends the following amounts:

TABLE OF CUBIC FEET OF AIR PER HOUR PER PERSON

	Max.	Min.
Hospitals for adults	2650	2650
Hospitals for children	1240	1240
Schools, children under 10 years	600	353
Schools, children over 10 years	885	530
Waiting-rooms with known number of people	1240	706
Auditoriums with unknown number of people	2 changes	1 change
House rooms in constant use	4 changes	3 changes
House rooms in occasional use	1 change	$\frac{1}{2}$ change
Kitchens and toilet rooms	5 changes	3 changes

When the number of persons is not known some designers use the following when provisions are made for changing the air:

- Residences: Halls, 3 changes
- Sitting rooms, and 1st floor rooms, 2 changes
- Sleeping rooms and 2d floor rooms, 1 change
- Stores: 1st floor, 2 to 3 changes, 2d, $1\frac{1}{2}$ to 2 changes
- Offices: 1st floor, 2 changes; 2d, $1\frac{1}{2}$ changes
- Churches and public assembly rooms, $\frac{3}{4}$ to 2 changes

When no provisions are made for ventilation, leakage will cause about $\frac{8}{10}$ of the above changes. Experiments have been made to show that this leakage actually does take place.

The following allowances are to be used:

FOR ADULTS PER HOUR PER PERSON

Hospitals	2400 cu.ft.
Auditoriums	2000 "
Workshops	2000 "
Waiting-rooms	1000 "

FOR CHILDREN

Hospitals	1500 cu.ft.
Schools	1500-2000 "
Auditoriums	1500 "
Workshops	1500 "

As gas flames give off CO₂, moisture and other gases, the use of gas burners in a room contaminates the air and requires additional air for ventilation. If a gas contains by volume 10 per cent carbon monoxide, 2 per cent carbon dioxide, 30 per cent methane, 4 per cent C₂H₄ and the remaining gases do not contain carbon, it will be found that the burning of one cubic

foot of gas will produce $\frac{1}{2}$ cu.ft. of CO_2 , if reduced to the initial temperature and pressure. The amount of CO_2 exhaled per hour per person equals 4 per cent of the air used which amounts to about 4 per cent of 15 cu.ft. or 0.6 cu.ft. The amount of contamination of the ordinary Welsbach mantle burner using

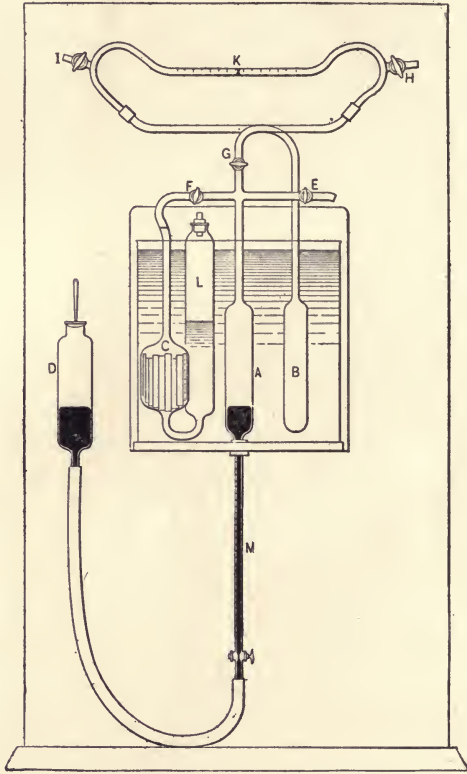


FIG. 15.—Petterson's CO_2 Apparatus.

3 cu.ft. equals that produced by two persons, while the fish-tail burner, using 5 cu.ft., is equivalent to four persons. Carpenter states this thus: The burning of 1 cu.ft. of gas per hour requires the amount of air equal to that required for one person.

The amount of CO_2 in the air in most cases is so slight that the ordinary gas apparatus cannot be used to determine it. A form due to **Petterson** is used. The tubes *A*, *B* and *C* are

immersed in a water bath to equalize the temperature. By lowering and raising the mercury bottle *D*, the caustic potash is sucked to the mark on the capillary tube below the cock *F* and then tube *A* is exhausted of air through the cock *E*. The air to be tested is then drawn into *A* to the small tube *M* at the bottom which is graduated. While this is being done *I* and *H* are open. After closing these and the cock *E*, and bringing the mercury in *D* and *A* to the same level, the cock *G* is open and the position of the liquid drop at *K* is noted. *G* is then shut off, *F* opened and the air is driven over into the burette *C* where it comes in contact with the KOH on the glass tubes, the

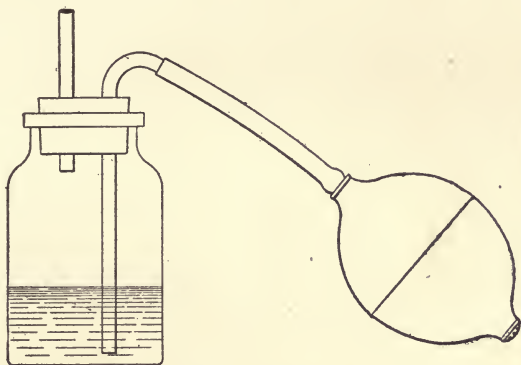


FIG. 16.—CO₂ Bottle.

KOH is driven into the vessel *L*. This is repeated several times, then the air is driven into *A* until the KOH fluid reaches the same mark below *F* when *F* is closed. *G* is opened and *D* is raised or lowered until the drop *K* is at the original scale reading. This means that the pressure on the gas in *A* is the same as before. The reading on the scale *M* then gives the diminution of volume or the amount of CO₂ absorbed. The small tube *M* permits one to determine small absorptions and the drop *K* gives an accurate method of getting the correct pressure.

A simple apparatus for getting relative results is shown in Fig. 16. The bottle is filled with a standard solution of sodium carbonate and phenolphthalein. If CO₂ is introduced into this until there is sufficient to change the sodium carbonate

into the bicarbonate, the pink color is destroyed and the solution becomes colorless. If now it takes A volumes of the bulb to destroy the color when atmospheric air is used and B volumes for the same amount of the same liquid when taking air from a room, the number of parts per 10,000 in the air of the room will be $\frac{4A}{B}$. Billings gives the strength of solution to be 5.3 grams of desiccated sodium carbonate, 1 gram of phenolphthalein and 1000 c.c. of distilled water which has been recently boiled and cooled. When ready to use, this is diluted still further with twenty-five times its volume of boiled distilled water.*

The amount of moisture in the air is another important item to consider in connection with the air for ventilating purposes. If the relative humidity is low, evaporation will take place from the surface of the body producing dry skin or dry mucous membrane in throat or nose and at the same time the temperature of the body will be lowered due to this evaporation. If on the other hand the air is saturated the body will feel damp and clammy. In either case the air is objectionable. The usual amount of moisture to make the room comfortable should be such as to give a relative humidity between 60 per cent and 80 per cent.

To determine the amount of moisture in air, **hygrometers** of some form are used. One method is to reduce the air or a sample of it to such a temperature that it will become saturated or deposit moisture. Fig. 17 illustrates one form of **dew-point apparatus** or hygrometer in which the aspiration of air through a volatile liquid reduces its temperature so that moisture begins to form on the silvered surface at the lowered end of the apparatus. The temperature at which this forms can be noted and again the temperature at which the moisture just disappears. The mean gives the temperature of saturation or the dew-point. The weight of moisture (or steam) per cubic foot at this temperature, from the steam tables gives the quantity of moisture present per cubic foot at actual temperature. This will be the

* Lunge to whom this method is due does not find the exact proportion and gives a proportional table of number of volumes of bulb and amount of CO_2 .

amount of moisture per cubic foot in the original air except for the slight reduction of volume due to the change of absolute temperature; hence if the weight of steam for the given temperature of the original air is divided into the actual weight the result is the relative humidity.

This form of apparatus is difficult to use and for that reason the **wet and dry-bulb hygrometer** is used. This consists of two thermometers on one of which a piece of wet wicking

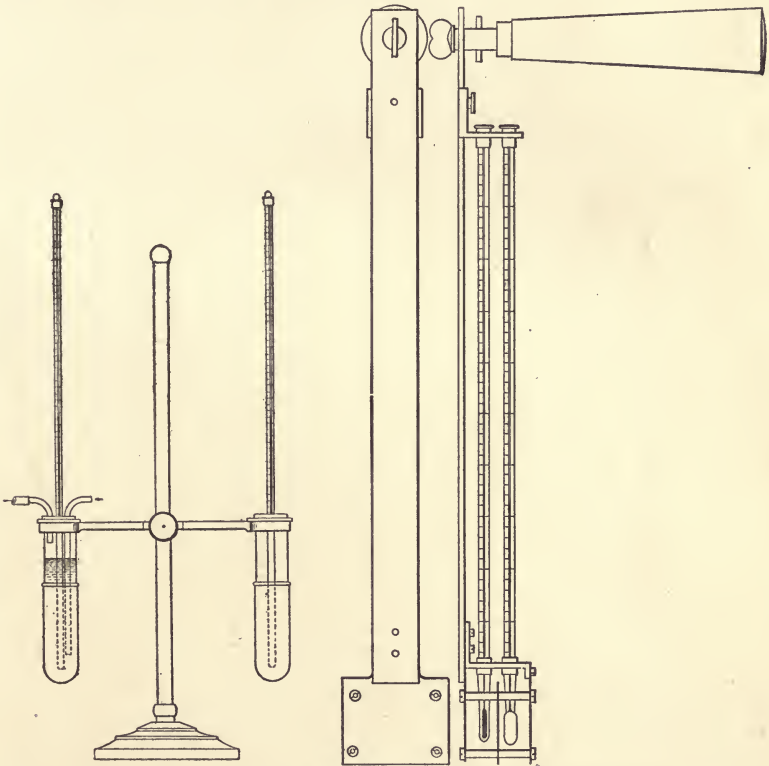


FIG. 17.—Dew Point Apparatus.

FIG. 18.—Sling Psychrometer.

encases the bulb. As water is evaporated from this wicking the temperature is lowered and the thermometer reads lower than the dry-bulb thermometer. This action is not regular if the thermometer or air is at rest and hence the U. S. Weather Bureau recommends whirling these thermometers.

Fig. 18 shows the appearance of this type of instrument. It is sometimes called a **sling psychrometer**.

Experiments have been made comparing readings of the dry bulb and difference between wet and dry bulb with the relative humidity readings obtained by a dew-point apparatus. From these Ferrel has reduced for the U. S. Weather Bureau the following formula:

$$p = p_1 - 0.000367B(t - t_1)[1 + 0.00064(t_1 - 32)]. \quad (4)$$

In this expression p is the pressure of the water vapor at the dew point, p_1 the pressure at the temperature of the wet bulb, B the barometric pressure, t the temperature of the dry bulb in degrees Fahrenheit and t_1 the temperature of the wet bulb in the same units, p , p_1 and B are measured in the same units, pounds per square inch, inches of mercury or by other unit. This formula contains the quantity B and hence all humidity charts and tables based on the formula are worked out for the standard atmospheric pressure of 29.92 ins. of mercury and corrections must be applied for other barometric pressures. If p is found by this formula, the weight of steam per cubic foot at this pressure and temperature may be found from the steam tables. The number of cubic feet of air for a given weight at the temperature t will be greater than that at the dew point t_a , and therefore the steam will occupy more space. If W_{sp} is the weight of a cubic foot of steam at the dew point, the weight in a cubic foot at the temperature t will be

$$\frac{W_{sp}(t_a + 459.6)}{t + 459.6} \quad (5)$$

In the above the air pressure is practically constant. If W_{st} is the amount of steam per cubic foot required to saturate the volume at the temperature t , the relative humidity will be given by

$$\rho = \frac{W_{sp}(t_a + 459.6)}{W_{st}(t + 459.6)} \quad (6)$$

Now since the pressure p is the **partial pressure** due to the

moisture present, the ratio of this to the **saturation pressure** will give the same value practically or

$$\text{relative humidity} = \frac{p}{p_i} \dots \dots \dots (7)$$

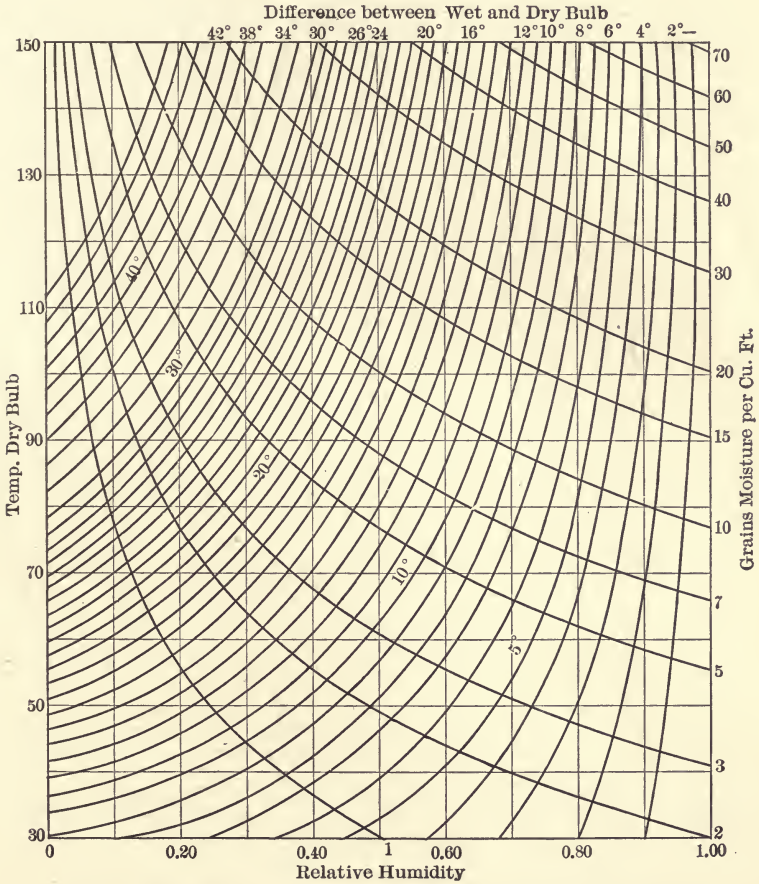


FIG. 19.—Relative Humidity and Moisture for 30.3 ins. Barometer.

Diagrams, Figs. 19 to 22, give relative humidities and weight of moisture per cubic foot of air in grains figured from this formula for barometric pressures of 29.1 ins., 29.5 ins., 29.9 ins., 30.3 ins. of mercury. Another instrument shown in Fig. 23,

known as the **hydrodeik** (as made by the Taylor Instrument Co.) employs a graphical chart by which relative humidity and other psychrometric data can be obtained.

The sliding pointer on the arm pivoted at the top is set

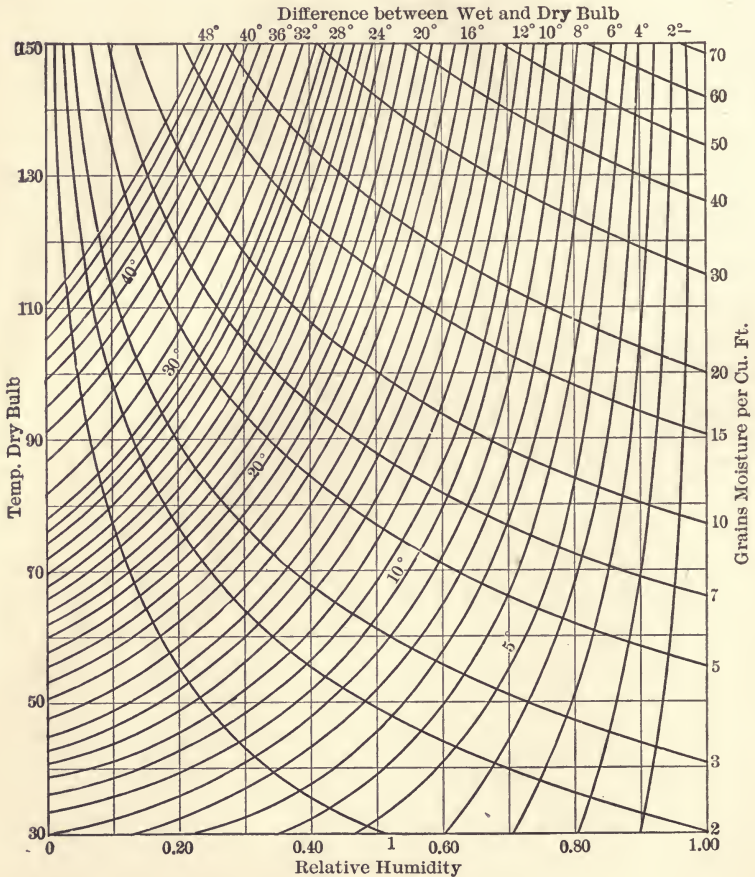


FIG. 20.—Relative Humidity and Moisture for 29.9 ins. Barometer.

opposite the reading of the wet bulb and is then swung over until it intersects the lines running down from the dry-bulb readings. These are in red on the apparatus. The reading at the bottom of the pointer will then be the relative humidity. The heavy black lines running up from the dry-bulb side give

the dew point if the one passing from the point of intersection of the pointer with the down line from the dry bulb is followed down to the dry side. At the upper end of these lines where they cut the down line from 120° F. will be found the number

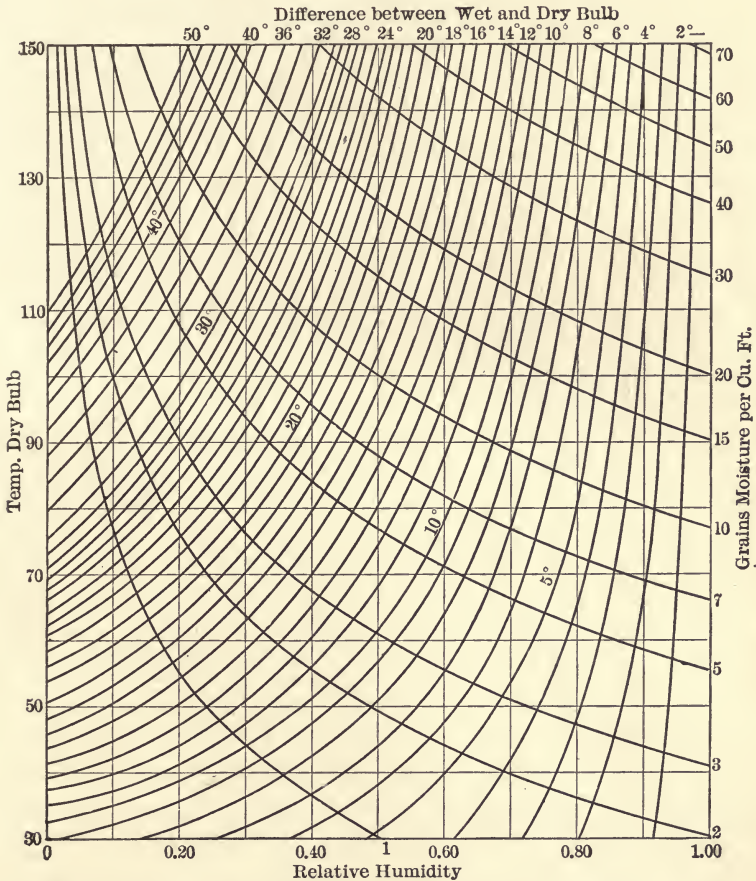


FIG. 21.—Relative Humidity and Moisture for 29.5 ins. Barometer.

of grains per cubic foot. The **Lambrecht Polymer** is an instrument consisting of a number of human hairs connected to a pointer which gives the humidity because the quantity of moisture which they will absorb and therefore the amount they will change in length is a function of the humidity of the medium

around them. This instrument should be checked constantly if used.

W. H. Carrier has discussed the matter of psychrometric formulae in the Journal of the American Society of Mechanical

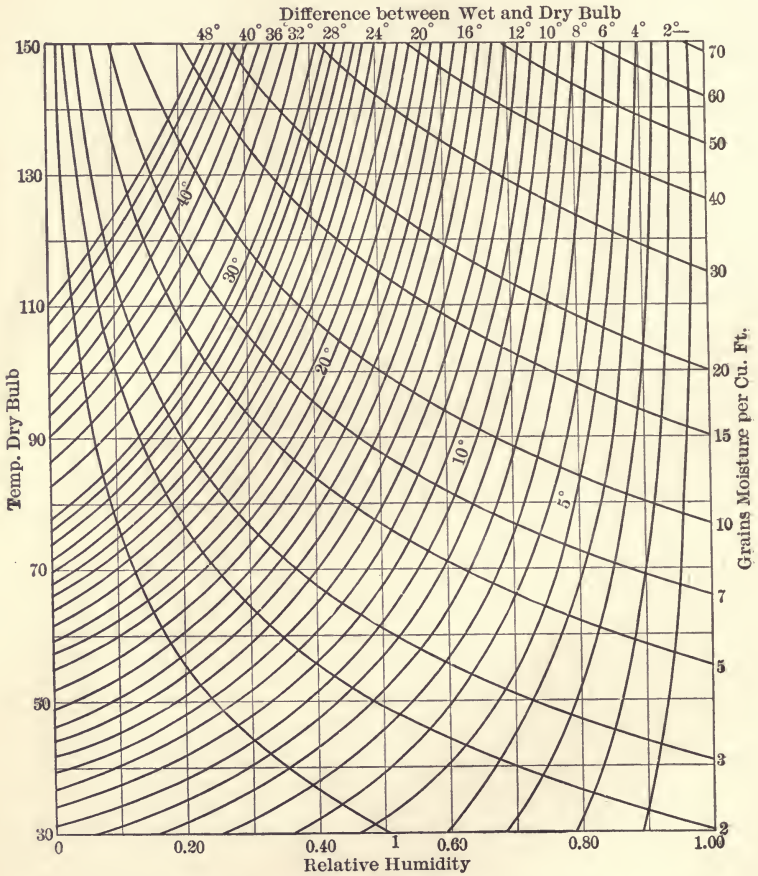


FIG. 22.—Relative Humidity and Moisture for 29.1 ins. Barometer.

Engineers for Nov., 1911 (p. 1311), and shows that the temperature of the wet bulb is such that the lowering of the air and water vapor in the atmosphere to this temperature will liberate sufficient heat to vaporize the necessary moisture to saturate the air at the wet-bulb temperature. This means that the

wet-bulb temperature will be above the dew point. Carrier states that this wet-bulb temperature depends on the total of the sensible and latent heat in the air and is independent of the relative proportions. Using this fact Carrier gives the following for 1 lb. of air, mixed with W lbs. of water:

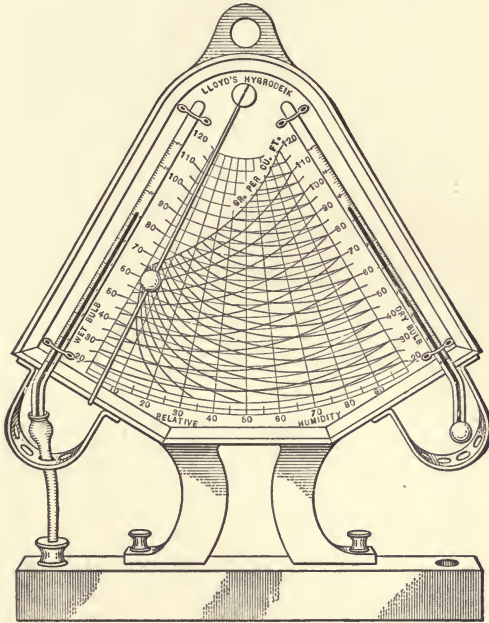


FIG. 23.—Hygrodeik.

Heat given up in cooling air and moisture

$$= (1 \times C_{pa} + C_{ps} \times W)(t - t'). \quad \dots \quad (8)$$

Heat received by water at temperature of wet bulb

$$= r'(W' - W). \quad \dots \quad (9)$$

C_{pa} = specific heat at constant pressure of air
 = $0.24112 + 0.000009t$;

C_{ps} = specific heat at constant pressure of steam
 = $0.4423 + 0.00018t$ (approximately);

- t = temperature of air or dry bulb;
 t' = temperature of wet bulb;
 W' = weight of moisture per pound of air to saturate air
 at temperature t' ;
 r' = heat of vaporization of steam at temperature t' .

These may be equated giving

$$(C_{pa} + C_{ps}W)(t - t') = r'(W' - W). \quad \dots \quad (10)$$

This equation is based on the theory that the heat content of the air remains constant and has the advantage of the empirical equation of Ferrel which does not hold for high temperatures which are obtained in kiln or drying sheds.

Solving the Eq. (10) for W gives

$$W = \frac{r'W' - C_{pa}(t - t')}{r' + C_{ps}(t - t')}. \quad \dots \quad (11)$$

This equation can be used to solve for W the moisture content when t and t' are known, for the other quantities can be found.

It is to be noted that

$$W' = \frac{53.35 \times (t' + 459.6)}{144(B - p')V_i'}. \quad \dots \quad (12)$$

V_i' = volume of 1 lb. of saturated steam at temperature t'
 $(B - p')$ = Partial pressure on air in pounds per square inch.

If W' and W are known the drop in temperature due to saturating the air is given by

$$(t - t') = \frac{r'(W' - W)}{C_{pa} + C_{ps}W}. \quad \dots \quad (13)$$

The amount of water to be absorbed is given by

$$W' - W = \frac{(C_{pa} + C_{ps}W)(t - t')}{r'}. \quad \dots \quad (14)$$

Carrier in addition to these derives the theoretical formula:

$$p = p' - \frac{(B - p')(t - t')}{2800 - 1.3t}, \quad \dots \quad (15)$$

where t and t' are the dry- and wet-bulb readings and p' is the pressure of the saturated steam for the temperature t' , B is the barometric reading.

On account of the error in the thermometer readings of the ordinary sling psychrometer, Carrier makes 1.6 per cent correction, giving the equation

$$p = p' - \frac{(B - p')(t - t')}{2755 - 1.28t'}, \quad \dots \quad (16)$$

to be applied with sling psychrometer. If p_t is the saturation pressure at the temperature t , the relative humidity is

$$\rho = \frac{p}{p_t} = \frac{p'}{p_t} - \frac{B - p'}{p_t} \frac{t - t'}{2755 - 1.28t'} \quad \dots \quad (17)$$

Either Ferrel's or Carrier's method can be used with agreement of 2 per cent or 3 per cent until high temperature and large temperature differences between wet- and dry-bulb readings are found.

The value of the hygrometer and the psychrometric formulae resulting from it is in the information it gives us of the amount of water vapor contained in the air and hence its condition for use in a ventilating system or for other service. If for instance air partially saturated is brought in contact with water finely divided, an evaporation will immediately take place, which will cool the air to practically the temperature of the wet bulb of the hygrometer. If, on the other hand, cold saturated air is warmed in a heater or by a steam coil, the moisture contained in the cold air is not sufficient to saturate the warm air and the unpleasant sensation of excessively dry air is produced unless the air is humidified in some way.

Air which is very warm may be cooled in some processes and

the capacity for water vapor may be much less than the actual quantity contained at the higher temperature and hence this reduction of temperature will cause a precipitation. By finding the number of pounds of water vapor per cubic foot from steam tables at a given temperature, the result may be compared with the capacity at another temperature and the relative humidity, quantity of vapor to be added or abstracted, or other property of the vapor at the new temperature found.

The pressure exerted by the water vapor is equal to the steam pressure at the temperature considered as given by any set of steam tables. This is usually known as **vapor tension**. If the air is not saturated, the pressure is equal to that of the saturated vapor multiplied by the relative humidity. Thus if p_s = pressure of saturation and ρ = the relative humidity, the vapor pressure p is given by

$$p = \rho p_s \dots \dots \dots (18)$$

A similar formula may be used to find the weight of water vapor γ_v in 1 cu.ft. of air, namely

$$\gamma_v = \rho \gamma_s \dots \dots \dots (19)$$

Now the volume of 1 lb. of air is given by the equation

$$v_a = \frac{53.35 \times (t + 459.6)}{144(B - p)}, \dots \dots \dots (20)$$

- where v_a = volume of 1 lb. of air in cubic feet;
- t = temperature of air in degrees F.;
- B = barometric pressure in pounds per square inch;
- p = partial pressure of moisture in pounds per square inch.

Hence the weight of moisture associated with 1 lb. of air is

$$w = \gamma_v v_a = \frac{53.35 \times (t + 459.6) \times \gamma_s}{144(B - p)}, \dots \dots \dots (21)$$

$$= \frac{53.35(t + 459.6) \times \rho \times \gamma_s}{144(B - \rho p_s)} \dots \dots \dots (22)$$

γ_s and p_s depend on t .

On account of γ being such a small quantity when expressed in pounds, this may be reduced to grains by multiplying by 7000.

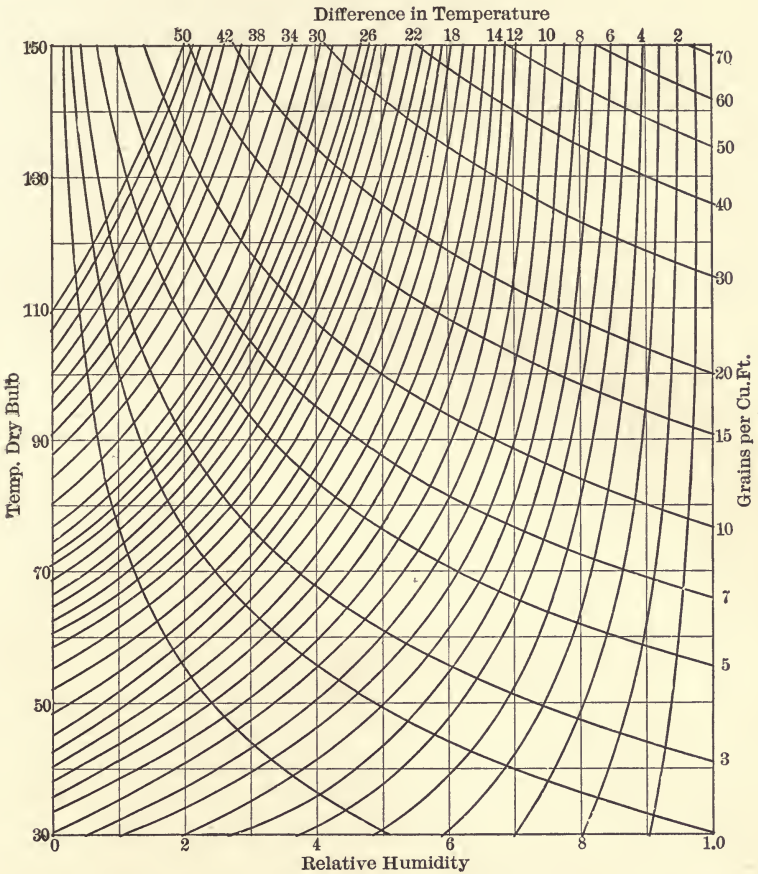


FIG. 24.—Relative Humidity and Moisture According to Carrier's Formula.

These formulae for the weight of moisture in the air in grains are =

$$W \text{ (per cu.ft.)} = 7000 \rho \gamma_s = 7000 \gamma_v. \quad (23)$$

$$W \text{ (per lb. dry air)} = \frac{7000 \times 53.35(t+459.6) \rho \gamma_s}{144(B - \rho p_s)}. \quad (24)$$

The relative humidity ρ and the partial pressure p may be found by the charts or the formulae and after these are known the weight of moisture may be found for any given condition.

To aid in computing problems relating to moisture, the curves, Figs. 19, 20, 21, 22 and 24 are given, the first four using the Weather Bureau formula, the last, from Carrier's article in the Journal of American Society of Mechanical Engineers. Table I is constructed to give certain of this information in tabular form.

TABLE I

PROPERTIES OF DRY AIR AND AIR SATURATED WITH MOISTURE

Temperature Air Degrees F.	Weight of 1 Cubic Foot of Dry Air.	Vapor Tension or Steam Pressure.*		Partial Pressure on Air.		Weight of Water Vapor in 1 Cubic Foot of Mixture.		Weight of Air in 1 Cubic Foot of Mixture in Pounds.	Pounds of Moisture Per Pound of Air.
		Pounds Per Square Inch.	Inches Mercury	Pounds Per Square Inch.	Inches Mercury	Pounds.	Grains.		
0	0.0863	0.019	0.038	14.678	29.873	0.00008	0.6	0.0862	0.0009
10	0.0845	0.031	0.063	14.666	29.858	0.00012	0.8	0.0843	0.0014
20	0.0827	0.050	0.103	14.647	29.818	0.00019	1.3	0.0825	0.0023
30	0.0810	0.081	0.164	14.616	29.757	0.00029	2.0	0.0805	0.0036
40	0.0794	0.122	0.248	14.575	29.673	0.00041	2.9	0.0788	0.0052
50	0.0779	0.178	0.362	14.519	29.559	0.00059	4.1	0.0762	0.0077
60	0.0765	0.256	0.521	14.441	29.400	0.00083	5.8	0.0752	0.0110
70	0.0749	0.363	0.739	14.334	29.182	0.00115	8.0	0.0731	0.0157
80	0.0735	0.506	1.030	14.191	28.891	0.00158	11.1	0.0710	0.0226
90	0.0722	0.696	1.417	14.001	28.504	0.00213	14.9	0.0688	0.0310
100	0.0709	0.946	1.926	13.751	27.995	0.00285	19.9	0.0663	0.0430
110	0.0697	1.271	2.588	13.426	27.333	0.00377	26.4	0.0636	0.0592
120	0.0685	1.689	3.439	13.008	26.482	0.00493	34.5	0.0606	0.0814
130	0.0673	2.220	4.520	12.477	25.401	0.00637	44.6	0.0571	0.1115
140	0.0662	2.885	5.873	11.812	24.048	0.00814	57.0	0.0532	0.1531
150	0.0651	3.715	7.563	10.982	22.358	0.01032	72.2	0.0486	0.2122
160	0.0640	4.738	9.646	9.959	20.275	0.01296	90.7	0.0434	0.2987
170	0.0630	5.990	12.195	8.707	17.726	0.01613	112.9	0.0373	0.4320
180	0.0620	7.510	15.289	7.187	14.632	0.01993	139.5	0.0303	0.6571
190	0.0611	9.339	19.013	5.358	10.908	0.02444	171.1	0.0223	1.0974
200	0.0602	11.528	23.469	3.169	6.452	0.02974	208.2	0.0130	2.2930

* From Peabody's Tables and Marvin's Results.

HUMIDIFIERS AND WASHERS

Another important consideration is that of the cleanliness of air for ventilation. Air from the atmosphere near a large city is laden with dust particles which will be introduced into buildings, where they collect on furniture, tapestry, walls or on goods in the process of manufacture. Moreover this air is not good for breathing and hence endeavor is made to cleanse it before discharging it into the rooms.

Several methods have been proposed for this purpose. Fig.

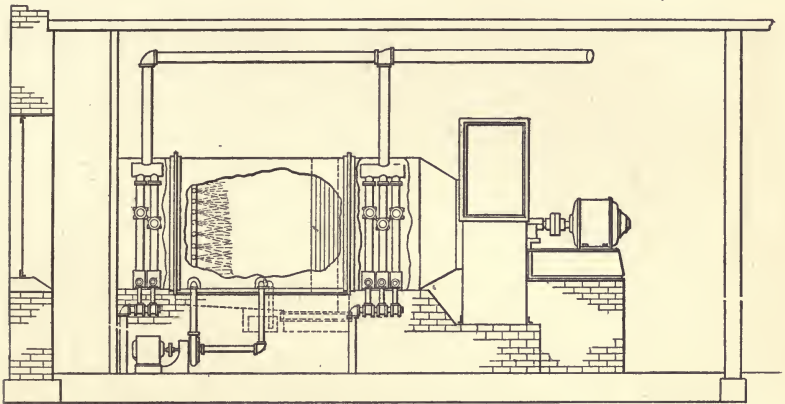


FIG. 25.—Carrier Air Washer and Humidifiers.

25 shows that of the Buffalo Forge Co., **The Carrier Air Washer and Humidifier**. After the air passes the tempering coils it enters a chamber in which are placed a number of nozzles from which on account of the rapid rotation of water within the nozzle a spray is formed which fills the chamber. As air is drawn through this spray, it collects the dust particles and part of them fall to the bottom of the chamber with the water. A large part of the moisture and dust particles are carried over to a series of vertical plates which are arranged so that the air has to take a zigzag course. In this way the moisture is collected and runs down the surface of the plates to a reservoir at the bottom from which it is taken from a settling

chamber through a strainer to a centrifugal pump by which it is again taken to the nozzles.

The **Eliminator** plates are so arranged with projections that the water cannot be carried along by the air. In this way the air leaves the eliminator without a trace of free moisture. The collection of dust and dirt may be washed out from the collecting chamber at intervals.

The degree to which this air is tempered before passing

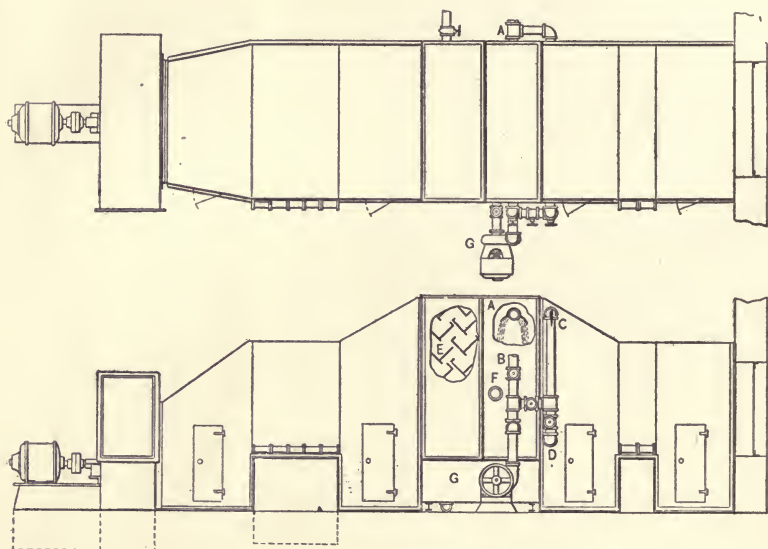


FIG. 26.—Warren Webster Washer and Humidifier.

through the washer and humidifier should be such that the moisture content is that desired for air to be used in the various rooms. If this air is too hot its moisture content will be so great that moisture will condense on being used at room temperature.

Fig. 26 shows the method used by Warren Webster & Co. in their air washer and humidifier. In this apparatus water is discharged from a pipe *A* at the top of the air passage just beyond the tempering coils. This pipe is perforated with holes arranged in a line but inclined to each other. The discharge from these holes, $\frac{3}{16}$ in. or larger in diameter, strikes against a copper

hood as shown. This makes the water fall in a sheet of crossing streams, the hood causing each individual jet to spread out lengthwise of the pipe. The pipe *A* is supplied from one end by the pipe *B* and at the other by the equalizer pipe *C*, which crosses over the apparatus. The valves on the pipes *B* and *C* are so arranged that water may be sent through the spray pipe from one end only, the water passing from the other end through the equalizing pipe to *D* and then to the sewer. In this way the dirt which may clog up the spray pipe can be cleaned out. The ends of the spray pipe or head are carried by deflecting plates which keep the water from coming in contact with the casing and so eroding it. After passing a short distance from the second sheet of water the air enters the eliminators or baffles *E*, the window *F* opening into the space in front of the baffles. The eliminator consists of two rows of V-shaped plates slightly inclined to the horizontal so that the water and dirt abstracted from the air may drain off at one end. These two rows are found sufficient to remove the excess moisture. The main portion of the dirt is removed by the sheet of spray and falls to the bottom where a metal tank is placed. From this point the water passes through a double strainer to the suction of the centrifugal pump and is again discharged through *A*. A float valve is used in the system to make up the loss of water to the air. An overflow and a drain are connected with the sewer for the purpose of removing the dirt at intervals. In some Warren Webster apparatus spray nozzles are used in addition to the spray head.

In most of this apparatus it is well to preheat the air to 40° or 50° so as to prevent the formation of ice in cold weather, and secondly to have such a temperature that the saturation of the air at that point will give the proper relative humidity at the temperature of the building. The Warren Webster purifier has also a device attached to it such that water is heated to about the temperature of the tempered air, thus making the air practically saturated at the temperature fixed by the tempering coils. This temperature is controlled by a thermostat placed in the space just in front of the reheating coil. By finding the temperature of the air saturated with vapor

at which the amount of water vapor is equal to that desired in the heated air the thermostat is set at this temperature and then steam will be admitted to heat the air and water until this temperature is obtained.

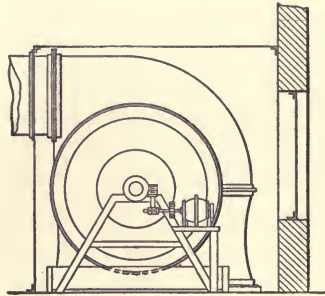
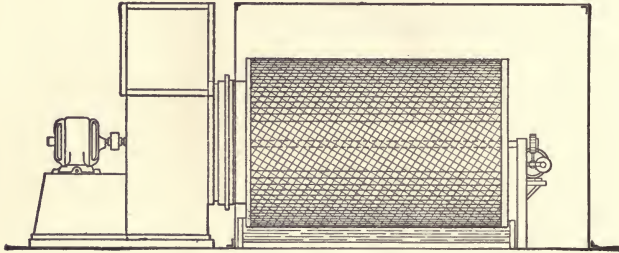


FIG. 27.—Cloth Filter.

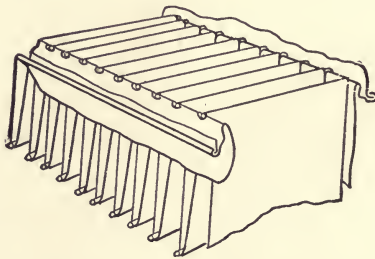


FIG. 28.—Whitley Patented Air Filter.

In Figs. 27 and 28 simple methods of cleaning the air are shown. A cheese cloth fabric is placed around a wire cylinder, closing off the inlet of a fan blower. The air is drawn through this and deposits its dust or dirt. The dirt is washed from the

cloth by revolving the cylinder in a trough of water by which it is washed clean. In Fig. 28 the cloth is folded over sticks and is hung in folds between the edges of a box. The air in passing through the cloth gives up the dirt. In both of these the endeavor has been to make the cloth surface as large as possible to cut down the velocity.

The air washer becomes an apparatus for cooling the air in the summer time if the humidity of the atmosphere is low, as the evaporation of the water vapor will cool the air. It practically reduces the air to the temperature of the wet bulb of a hygrometer. If the air is saturated with moisture, there

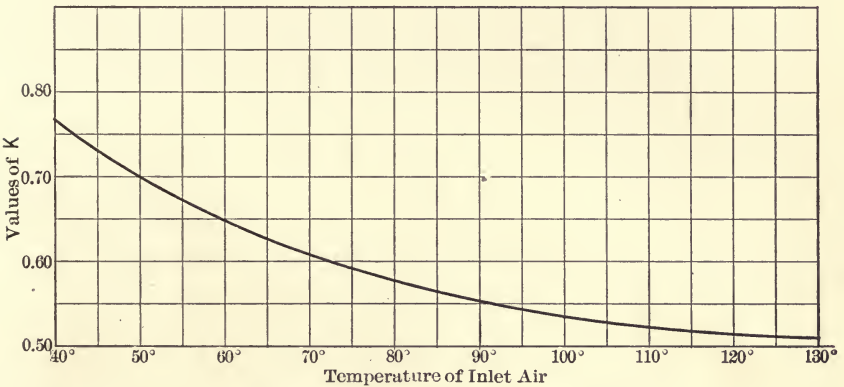


FIG. 29.—Values of K for Various Temperatures.

will be no decrease in temperature unless the water used in the washer is low. In such a case fresh cold water from the city supply or a deep well is passed through the washer and discharged directly into the sewer after falling to the bottom. In this way the water is not heated as would occur if this were used repeatedly. This method of cooling is an important one for summer ventilation. Warren Webster & Co. have experimented with their apparatus and give a series of curves showing the amount of cooling in air of various temperatures and various relative humidities when using the water repeatedly and when using cold water at various temperatures, allowing the water from the apparatus to waste to the sewer. The author has shown the

results of their work as an approximate formula. When the water is recirculated the drop in temperature is given by

$$\Delta t_r = K[t_a - t_{wb}], \dots \dots \dots (25)$$

where t_a = temperature of the air in degrees F.;
 t_{wb} = " " wet bulb in degrees F.

K is a constant depending on the temperature of the air and is given by the curve Fig. 29 Δt_r = drop in temperature of air. When cold water is used and then wasted, the drop is given by

$$\Delta t_c = \frac{I}{24} \left[t_a - t_w + \frac{\Delta t_r}{2} \right], \dots \dots \dots (26)$$

t_w = temperature of water in degrees F.

CHAPTER III

LOSS AND GAIN OF HEAT

HEAT being a form of energy, may be measured in any unit of energy. In the United States and other English-speaking countries it is customary to measure it in **British Thermal Units. (B.t.u.)** A British thermal unit is the amount of heat required to heat 1 lb. of water from 62° F. to 63° F. Of recent years some authors prefer to use the mean B.t.u., which is $\frac{1}{180}$ of the heat required to raise the temperature of 1 lb. of water from 32° F. to 212° F. The second or mean B.t.u. is equal to 1.003 B.t.u.

The French use the **Calorie**, the amount to raise 1 kg. of water from 15° C. to 16° C. or $\frac{1}{100}$ of the amount to raise 1 kg. from 0° C. to 100° C.

By experiment the relation of these units with the other units of energy have been determined and are given below with certain other transformation constants:

$$1 \text{ B.t.u.} = 778 \text{ ft.-lbs.}$$

$$1 \text{ B.t.u.} = 0.998 \text{ mean B.t.u.}$$

$$1 \text{ B.t.u.} = 0.252 \text{ calories}$$

$$1 \text{ calorie} = 426.6 \text{ kg.m.}$$

$$1 \text{ calorie} = 3.968 \text{ B.t.u.}$$

$$1 \text{ ft.-lb.} = 0.1383 \text{ kg.m.}$$

$$1 \text{ kg.m.} = 7.2330 \text{ ft.-lbs.}$$

$$1 \text{ kg.} = 2.2046 \text{ lbs.}$$

$$1 \text{ lb.} = 0.4536 \text{ kg.}$$

$$1 \text{ m.} = 39.37 \text{ in.}$$

$$1 \text{ m.} = 3.2808 \text{ ft.}$$

$$1 \text{ ft.} = 0.3048 \text{ m.}$$

$$1 \text{ sq.ft.} = 0.09290 \text{ sq.m.}$$

1 sq.m. = 10.7639 sq.ft.

1 cu.ft. = 0.0283 cu.m.

1 cu.m. = 35.31 cu.ft.

1 atmosphere = 14.696 lbs. per sq.in.

= 29.921 ins. mercury

= 760 mm. mercury

= 10,333 kg. per sq. meter

1 kg. per sq.cm. = 14.22 lbs. per sq.in.

1 watt hour = 3.41 B.t.u.

= 2652 ft.-lbs.

1 cu. liter of mercury = 13.5959 kg.

1 horse-power = 550 ft.-lbs. per sec.

= 33,000 ft.-lbs. per min.

= 746 watts

1 horse-power hour = 2546 B.t.u.

1 inch of mercury = 0.4912 lbs. per sq.in.

1 inch of water = 0.036 lbs. per sq.in.

= 0.58 oz. per sq.in.

1 oz. per sq.in. = 1.72 inches of water

1 U. S. gallon = 231 cu.in.

The heat loss from rooms is made up of several parts. There are **radiation** and **conduction** from walls, windows and doors and **convection** losses due to warming of the **leakage air** or the air for ventilation. The gain of heat is derived from persons or apparatus used in the room or from sources of light of various kinds.

The loss of heat through walls partakes of the nature of radiation and conduction. The principal loss is made up of transmission which is found to depend on the difference of temperature and therefore it is similar to conduction rather than radiation which depends on a higher power of the temperatures. The general form in which this heat loss is given is

$$H = KA(t_1 - t_0), \quad (27)$$

where A = area in square feet;

K = Heat transmitted per square foot per hour per degree difference of temperature in B.t.u.;

- t_1 = room temperature in degrees F.;
- t_o = outside temperature in degrees F.;
- H = B.t.u. transmitted per hour.

The value of K depends on several factors: the surface, thickness, and kind of material, air spaces and condition of air at surface. The following German method from H. Rietschel's *Leitfaden zum Berechnen und Entwerfen von Lüftungs- und Heizungs-Anlagen* is useful for future reference for cases which have not been calculated in the text.

The rate of transmission of heat through any substance depends on the thickness and on the difference of temperature. If for instance the wall shown in Fig. 30 is made up of several thicknesses and the temperatures are those marked, the equations for the transmission of heat through each section must each give the quantity of heat transmitted by the wall and these, therefore, must be equal to each other.

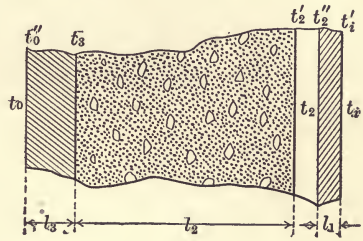


FIG. 30.—Wall Section.

The amount of heat conducted by any material per square foot of cross-section varies directly with the temperature difference and inversely with the length. This gives

$$H = \frac{C}{l}(t_1 - t_2), \dots \dots \dots (28)$$

where C is the constant of conduction for 1 foot thickness in B.t.u. per square foot per degree, l is the thickness in feet and $t_1 - t_2$ is the difference of temperature. Using this for the wall shown in Fig. 30 the following results:

$$H = \frac{C_1}{l_1}(t_4' - t_2'') = \frac{C_2}{l_2}(t_2' - t_3) = \frac{C_3}{l_3}(t_3 - t_o''), \dots \dots (29)$$

At the surface of any material there is to be found a temperature different from that of the space around and it is this difference which determines the flow of heat at the surface. If a is the **coefficient of transmission** per square foot per hour per degree across the surface this becomes at different surfaces:

$$\left. \begin{aligned} H &= a_1(t_4 - t_4') \\ &= a_2(t_2'' - t_2) \\ &= a_3(t_2 - t_2') \\ &= a_4(t_0'' - t_0) \end{aligned} \right\} \dots \dots \dots (30)$$

The values of H in the sets above are all the same, hence solving for temperature differences and adding, the following results:

$$\left. \begin{aligned} \frac{H}{a_1} + \frac{Hl_1}{C_1} + \frac{H}{a_2} + \frac{H}{a_3} + \frac{Hl_2}{C_2} + \frac{Hl_3}{C_3} + \frac{H}{a_4} &= t_4 - t_4' + t_4' - t_2'' + t_2'' \\ &\quad - t_2 + t_2 - t_2' + t_2' - t_3 + t_3 - t_0'' + t_0'' - t_0 \end{aligned} \right\} \dots (31)$$

$$H \left[\frac{1}{a_1} + \frac{l_1}{C_1} + \frac{1}{a_2} + \frac{1}{a_3} + \frac{l_2}{C_2} + \frac{l_3}{C_3} + \frac{1}{a_4} \right] = t_4 - t_0$$

Now

$$K = \frac{H}{t_4 - t_0},$$

hence

$$K = \frac{1}{\frac{1}{a_1} + \frac{1}{a_2} + \frac{1}{a_3} + \frac{1}{a_4} + \frac{l_1}{C_1} + \frac{l_2}{C_2} + \frac{l_3}{C_3}} \dots \dots (32)$$

The values of C_1, C_2, C_3 are found for different substances. They are the amounts of heat transmitted for unit thickness of material per hour per square foot of surface per degree. The experimental values given by Rietschel are quoted below in B.t.u.

VALUES OF C.

Air, still.....	0.03
Brass.....	61.00
Brickwork.....	0.46
Building paper.....	0.08
Cement.....	0.40
Copper.....	202.00
Cork.....	0.17
Cotton.....	0.03
Felt.....	0.02
Glass.....	0.54
Lead.....	20.00
Limestone.....	1.35
Marble (fine).....	1.88
Mortar and plaster.....	0.46
Oak.....	0.13
Pine (along the grain).....	0.11
Pine (across the grain).....	0.06
Plaster of Paris.....	0.34
Sandstone.....	0.87
Sawdust.....	0.03
Slate.....	0.19
Terra Cotta.....	0.54
Tin.....	35.60
Zinc.....	74.00

The values of the quantities a_1 are of the form as given from Grashof and Rietschel,

$$a = d + e + \frac{(42d + 31e)T}{10000} \quad \dots \quad (33)$$

d and e are constants, d depends on the condition of the air around surface and e depends on the material. T is the temperature difference between the air and the wall at any point.

To determine the quantity T a method of approximation is

used until by practice one knows what to expect. The value of the term involving T ,

$$\frac{42d+31e}{10000}T,$$

is small, hence for a first approximation this term may be neglected and the value of the various a 's may be found. These then may be used to find K ,

$$K = \frac{1}{\Sigma \frac{1}{a_1} + \Sigma \frac{l}{C}} \quad \dots \dots \dots (34)$$

After this is known, the following results:

$$\left. \begin{aligned} K(t_1 - t_0) &= a_1(t_1 - t_1') = a_2(t_2 - t_2') \\ &= a_1 T_1 = a_2 T_0 = \text{etc.} \end{aligned} \right\} \dots \dots (35)$$

since

$$T = t_1 - t_1' \text{ or } t_0'' - t_0.$$

These equations give the first approximation for T .

In this way after T is found as a first approximation the value may be used to determine a second value of a and then a new value of T . In this way two or three trials will lead to the correct result.

In any case the value of T is small and this particularly is true for thick walls or cases in which $t_1 - t_0$ is a small quantity.

Rietschel gives results used in practice for the value of T for masonry walls. These may be put into the form of an equation.

$$T = 16.2 - 4.00l. \quad \dots \dots \dots (36)$$

This may be used for masonry walls with air spaces where l is the sum of the various thicknesses, although this result is slightly too large in this case as the quantity $K(t_1 - t_0)$ is smaller than for a solid wall of the combined thickness.

For a single glass T is taken as $\frac{1}{2}(t_1 - t_0)$ while for double windows $\frac{1}{4}(t_1 - t_0)$ is taken at each surface. Since glass is so

thin there is practically no temperature drop in it. This will be seen later.

The value of T for wooden floors is given as $T = 1.8^\circ \text{ F}$. The values of d as given from Grashof are as follows:

VALUES OF d .

Air at rest as in rooms or channels	0.82
Air with slow motion as over windows	1.03
Air with quick motion as outside of building	1.23

The values of the coefficient e are determined by Rietschel as follows:

VALUES OF e .

Brass, polished	0.05
Brickwork and masonry	0.74
Cast iron, new	0.65
Cotton	0.75
Charcoal	0.71
Copper	0.03
Glass	0.60
Mortar and lime mortar	0.74
Paper	0.78
Plaster of Paris	0.74
Polished sheet iron	0.092
Rusted iron	0.69
Sawdust	0.72
Sheet iron	0.57
Silk	0.76
Tin	0.045
Water	1.07
Wet glass	1.09
Wool	0.76
Zinc	0.049
Wood	0.74

To explain the application of the above, the wall given in

Fig. 31 will be investigated. The wall is composed of 4 ins. of sandstone, 18 ins. of brick work, a 2-in. air space, 8 ins. of brick and 1 in. of plaster. Where sections of the wall actually come in contact, there is no surface resistance and the wall may be considered as solid except for differences in the values of C for the various materials. To find a the

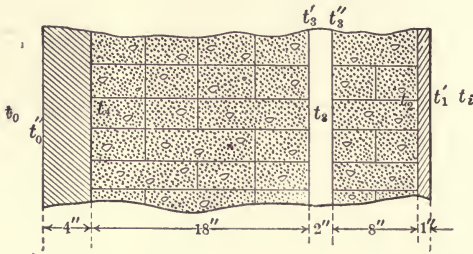


FIG. 31.—Wall Section.

various values of T must be known; now T is given by the following:

$$\begin{aligned} T_1 &= t_1 - t'_1; \\ T_3 &= t_3'' - t_3; \\ T_3' &= t_3 - t_3'; \\ T_c &= t_0'' - t_0. \end{aligned}$$

These quantities vary inversely with the different values of a , since

$$a_1 T_1 = a_2 T_3 = a_2' T_3' = a_4 T_c.$$

As the quantities a do not differ by great amounts these various values of T are considered as equal quantities in computing a .

T may then be found from the equation,

$$T = 16.2 - 4.00l.$$

In this case the total thickness is 31 ins. and

$$T = 16.2 - 4 \times \frac{31}{12} = 6^\circ.$$

$$a_4 = 1.23 + 0.74 + \frac{(42 \times 1.23 + 31 \times 0.74)6^\circ}{10000};$$

$$a_3 = 0.82 + 0.74 + \frac{(42 \times 0.82 + 31 \times 0.74)6^\circ}{10000};$$

$$a_2 = 0.82 + 0.74 + \frac{(42 \times 0.82 + 31 \times 0.74)6^\circ}{10000};$$

$$a_1 = 0.82 + 0.74 + \frac{(42 \times 0.82 + 31 \times 0.74)6^\circ}{10000};$$

$$a_4 = 2.01;$$

$$a_3 = 1.59 = a_2 = a_1.$$

K is then found as follows:

$$K = \frac{1}{\frac{1}{2.01} + \frac{1}{1.58} + \frac{1}{1.58} + \frac{1}{1.58} + \frac{0.33}{.87} + \frac{1.5}{.46} + \frac{.017}{.03} + \frac{0.66}{0.46} + \frac{.007}{.46}}$$

$$= \frac{1}{0.497 + 0.633 + 0.633 + 0.633 + 0.718 + 3.26 + 0.56 + 1.435 + 0.037}$$

$$= \frac{1}{8.066} = .124.$$

For a floor or ceiling as shown in *A*, Fig. 32, the method is quite the same. When the high temperature is at the top, however, there is no circulation in the air space between the plaster and the floor and the air acts as an insulating material.

When the high temperature is below or if the air space is in a vertical position the circulation of the air transmits heat by convection and the air does not act as an insulating material as was the case with the wall just considered. In any case, however, there is a resistance at the surface between the air and the partition due to the drop T .

When the same constant does not hold over a complete wall or floor owing to a change in the construction as occurs at studs in a partition or at joists in a floor, the value of K for the whole surface is found thus:

$$K(A_1 + A_2)(t_1 - t_0) = K_1A_1(t_1 - t_0) + K_2A_2(t_1 - t_0),$$

or

$$K = \frac{K_1A_1 + K_2A_2}{A_1 + A_2} = \frac{\Sigma KA}{\Sigma A} \dots \dots \dots (37)$$

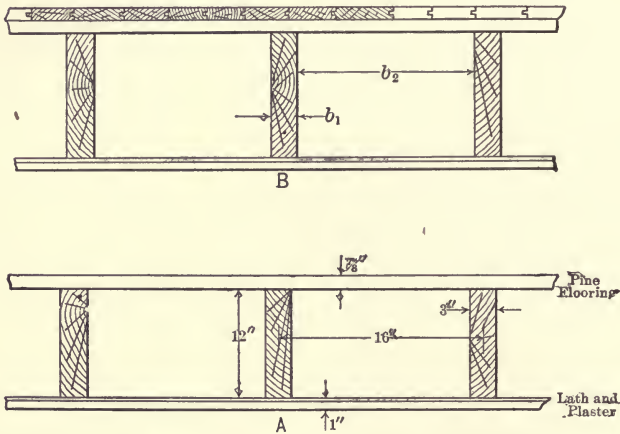


FIG. 32.—Wood Floor Construction.

In most cases the areas A have a common dimension so that the areas are proportional to the widths. If these are b_1 and b_2 there results (Fig. 32),

$$K = \frac{K_1b_1 + K_2b_2}{b_1 + b_2} \dots \dots \dots (38)$$

The mean constant is not usually found for a wall in terms of the glass and wall coefficient as these are kept separate, but there is no reason why this could not be done as is the case with the coefficient for partitions with partition studs in the cases below.

With the high temperature above the air acts as an insulating substance and the following results for the floor, Fig. 32, A:

$$a_u = 0.82 + 0.74 + \frac{42 \times 0.82 + 31 \times 0.74}{10000} 1.8 = 1.57,$$

at joists,

$$K_j = \frac{1}{\frac{1}{1.57} + \frac{13.25}{12 \times 0.06} + \frac{5}{8 \times 12 \times 0.46} + \frac{1}{1.57}} = 0.05,$$

at space between joists,

$$K_a = \frac{1}{\frac{1}{1.57} + \frac{7}{12 \times 8 \times 0.06} + \frac{2}{1.57} + \frac{12}{12 \times 0.03} + \frac{3}{8 \times 12 \times 0.06} + \frac{5}{8 \times 12 \times 0.46} + \frac{1}{1.57}} = 0.027.$$

Combined

$$K = \frac{3 \times 0.05 + 13 \times 0.027}{16} = 0.031.$$

With the high temperature below on account of the convection currents, the air does not act as an insulating substance and the following results:

$$a = 1.57;$$

$$K_j = 0.05;$$

$$K_a = \frac{1}{\frac{4}{1.57} + \frac{1.25}{12 \times 0.06} + \frac{5}{8 \times 12 \times 0.46}} = 0.22;$$

$$K = \frac{3 \times 0.05 + 13 \times 0.22}{16} = 0.19.$$

This method may be used for various kinds of walls and partitions. The following values have been computed by the author and these values compared with those given by Kinealy, Rietschel and others.

VALUES OF a .

For brick and plaster or masonry:

$$\begin{aligned} \text{Outside} \quad a &= 1.23 + 0.74 + \frac{43 \times 1.23 + 31 \times 0.74}{10000} T \\ &= 1.97 + 0.0075T \\ &= 2.09 - 0.03l, \end{aligned}$$

since

$$T = 16.2 - 4l.$$

$$\begin{aligned} \text{Inside,} \quad a &= 1.56 + 0.0057T \\ &= 1.65 - 0.023l. \end{aligned}$$

For wood and approximately for paper, cotton, wool, sawdust, charcoal:

$$\text{Outside,} \quad a = 1.97 + 0.0075T = 1.98.$$

$$\text{Inside,} \quad a = 1.56 + 0.0057T = 1.57.$$

For glass:

$$\begin{aligned} \text{Outside,} \quad a &= 1.83 + 0.007T \\ &= 2.07 \left(T = \frac{t_i - t_o}{2} = 35^\circ \right). \end{aligned}$$

Inside with motion.

$$\begin{aligned} a &= 1.63 + 0.006T \\ &= 1.83 \left(T = 35 \right). \end{aligned}$$

Inside without motion:

$$\begin{aligned} a &= 1.42 + 0.005T \\ &= 1.59 \left(T = 35 \right). \end{aligned}$$

Inside with motion and wet from condensation:

$$\begin{aligned} a &= 2.11 + 0.008T \\ &= 2.39. \end{aligned}$$

For double windows:

$$\text{Outside,} \quad a = 1.95 \left(T = \frac{1}{4} \times 70 \right)$$

$$\text{Center,} \quad a = 1.51.$$

$$\text{Inside dry,} \quad a = 1.74.$$

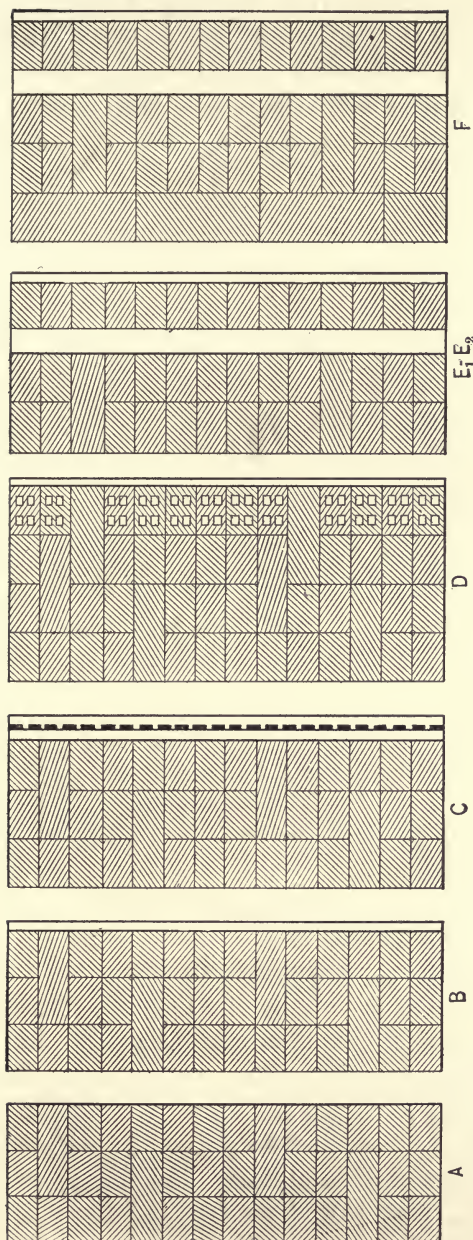


FIG. 33.—Wall Sections.

Before proceeding with the tabular values an explanation will be made of the various common forms of building construction which are to be found in the following figures.

Walls. Walls are built in many cases of brick. Bricks vary in size in different localities. In some places the dimensions are $8\frac{1}{8} \times 3\frac{1}{2} \times 2\frac{1}{4}$. A standard $8\frac{1}{4} \times 4 \times 2\frac{1}{4}$ was adopted by the national Brickmakers' Association. These are built in various thicknesses. At times an **air space** is made in the wall as shown in *E*, Fig. 33, so that the interior of the wall will be dry and also that the wall will be a better non-conductor of heat. When a solid wall is built it is not advisable to put the plaster directly on the wall as in *B*, Fig. 33, because the water soaking through the wall will produce a damp surface. To prevent this the wall, *C*, Fig. 33, is **furred** before the plaster is put on. This consists in placing wood or iron **furring strips** on the wall and attaching wood or metal **lath** to these. The wood strips are usually $\frac{3}{4}$ by 2 ins. so that the laths are removed about $\frac{3}{4}$ in. from the wall, thus giving an air space. The furring strips are attached to wooden wedges or plugs driven into the joints in the walls. The same effect is obtained by the use of **hollow bricks**, known in some localities as "**Haverstraw bricks**," as the lining of the wall. These are shown at *D*, Fig. 33. Walls are sometimes faced with stone as shown at *F*, Fig. 33. A recent method of wall construction for residences and small buildings is to use **hollow partition tile** and face them with brick or plaster as shown in *C*, Fig. 40.

Stone and concrete walls are used for buildings and constants have been computed for these.

Wooden walls or frame constructions are used at certain times. In this, **studs**, usually of 2 by 4 in., 3 by 4 in., or 4 by 6 in. timbers, are placed upright at 16-in **centers**, and on these are nailed **sheathing boards**, which may be planed boards, although at times **shiplap** as shown in *D*, Fig. 34, or **tongued and grooved boards**, *E*, Fig. 34, are used to make tight joints for the purpose of keeping down the loss of heat. Building paper of one or two thicknesses is then placed on top and the **clapboards** *F*, or **shingles** *G*, are placed on top of this. On account of the frequent

vertical joints in the shingles these are usually placed so that not over one-third of the length of the shingle is exposed to the weather. *A* represents a wall without plaster while *B* and *C* are plastered on the interior.

Floors. Floors for dwellings and many small buildings

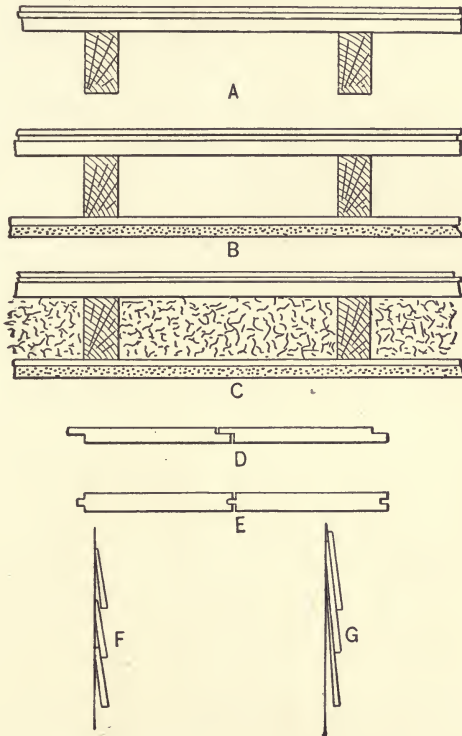


FIG. 34.—Wooden Walls.

are built as shown in Fig. 32. In this form wooden joists are placed on 16-in. centers and on top of them a floor is built composed of one or two layers of tongued and grooved floor boards or the lower layer may be of shiplap or square edged-boards. In any case at least one side of the boards must be planed to bring the boards to a uniform thickness. To the lower side of the joist, laths are attached and these carry the

plaster. **Mill or slow-burning construction**, Fig. 35, consists in using heavy wooden girders (12 by 12 ins.) about 8 ft. apart and on these is a floor of 3-in. planks. The planks are usually of yellow pine above which is placed a hardwood floor covering,

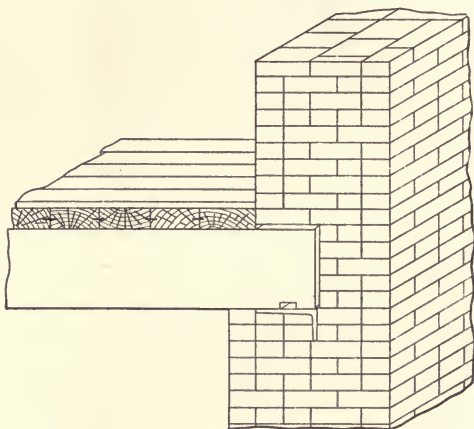


FIG. 35.—Mill or Slow-burning Construction.

separated by building paper of some kind. Fig. 36 shows a **fireproof-floor construction** using reinforced cinder concrete between I-beams. The cinder concrete encases the beams. This is reinforced by metal, resting on the beams. The ceiling below



FIG. 36.—Reinforced Concrete Floor.

is carried on metal laths attached to small channel irons running from beam to beam when a **flush ceiling** is desired, while at times the plaster is attached directly to the concrete, when a **panel effect** is desired. On top of the reinforced concrete a **lean cinder concrete fill** is made between the **dovetailed sleepers**

to which the floor boards are attached. The sleeper usually runs at right angles to the beams. Two thicknesses of floor

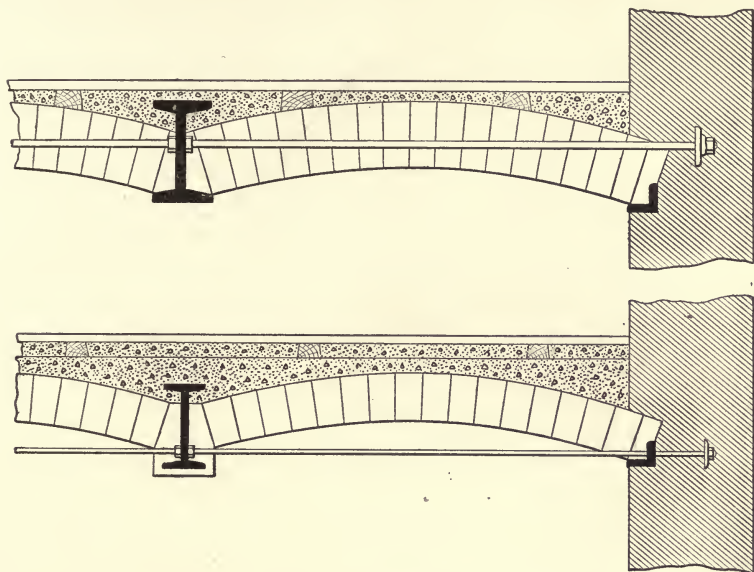


FIG. 37.—Brick Arches.

boards are used, the upper surface being of maple when lasting qualities are required.

Fig. 37 illustrates methods of using **arch bricks** with concrete

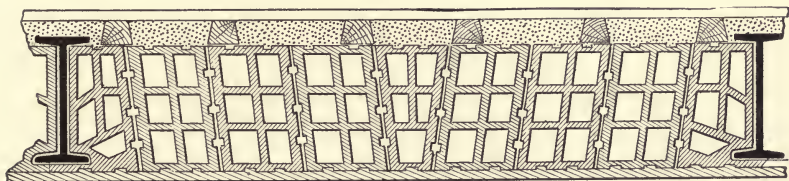


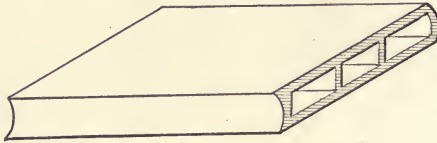
FIG. 38.—Hollow Tile Flat Arch.

filling. The end **voussoirs** or **skew backs** are specially made or cut. The brick **voussoirs** are spoken of as **rowlocks** when used this way.

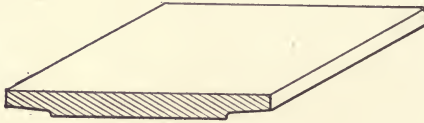
Fig. 38 exemplifies the method used with hollow **fireproofing**

tile. The tiles are made to form a **flat arch**. The cement mortar is arranged in grooves to increase the bonding of the tiles.

A metal lath and plaster ceiling hung from a reinforced concrete roof is shown in Fig. 39.



A—Book Tile.



B—Roof Tile.

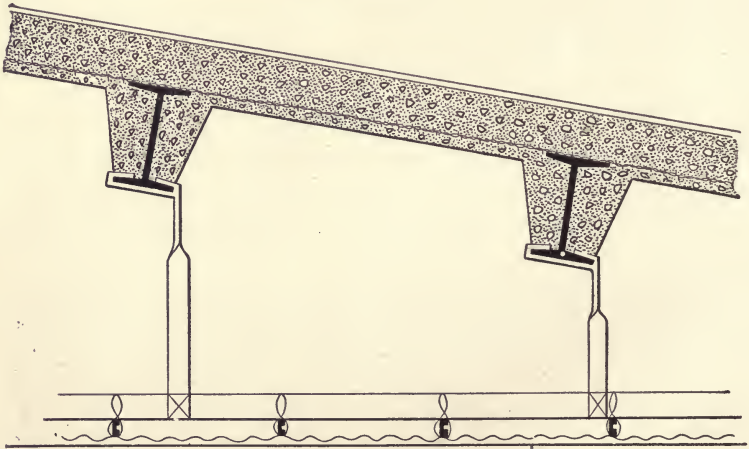


FIG. 39.—Roof and Roof Ceiling Tile.

Fig. 39, *A*, illustrates a **book tile** and *B*, a **government tile**, used in forming fireproof roofs. The book tiles are placed so that the rounded projecting edges fit into the hollowed edges of the adjacent tiles, the square corners resting on the flanges of

T-irons which are supported by or form the purlins of the roof. The tiles are 3 by 12 by $17\frac{1}{2}$ ins. long. The **government roofing tiles** $2\frac{1}{2}$ by 12 by $15\frac{1}{2}$ ins. rest in the T-irons, the depressed edges being so arranged that the bottom of the tile is flush with the bottom of the irons.

Partitions. The horizontal section of the ordinary forms of partitions are shown in Fig. 40. *A* represents the standard form of wood partition made of 2 by 4 ins. or by 3 by 4 ins. studs of spruce or hemlock placed on 16-in. centers to which

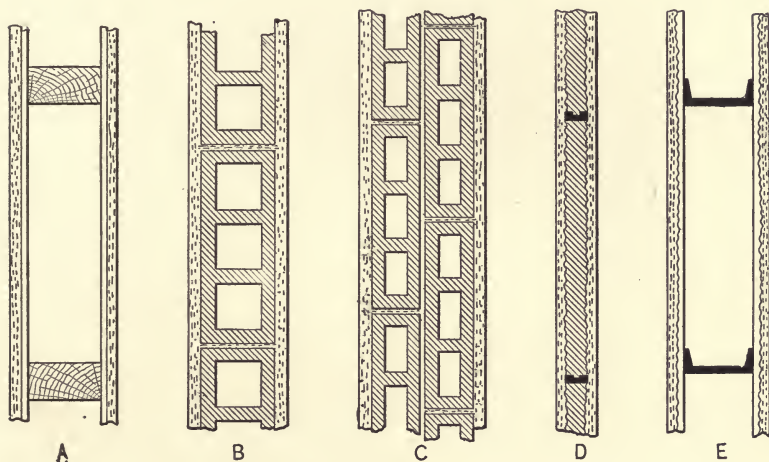


FIG. 40.—Partitions.

are fastened wooden or metal laths or a patented sheet substance known as plaster board, over which plaster is placed.

Form *B* consists of a single row of fireproof tile with plaster on each side and *C* shows two sets of tiles with an air space between them. *E* gives a partition formed of wire lath attached to channel irons and *D* gives one in which expanded metal is used to support the plaster while the space on the inside is filled with asbestos to deaden the sound, as well as to cut down air currents and thus cutting down heat transmission. The plaster coat in most cases takes up $\frac{5}{8}$ in. and this may be used in figuring thicknesses. The above figures represent

various typical methods of building construction, and the transmission coefficients which are computed for them will serve as guides for other constructions if time is not available for computing K .

VALUES OF K .

WALLS AND PARTITIONS

(See Fig. 33)

Combined Thickness of Brickwork.	A.	B.	C.	D.	E ₁ .	E ₂ .	F.
4".....	0.55	0.51	0.28				
8".....	0.39	0.37	0.24	0.29	0.27	0.26	0.32
12".....	0.31	0.29	0.20	0.24	0.22	0.22	0.20
16".....	0.25	0.24	0.18	0.20	0.19	0.19	0.17
20".....	0.21	0.21	0.16	0.18	0.17	0.17	0.15
24".....	0.18	0.18	0.14	0.16	0.15	0.15	0.14
28".....	0.16	0.16	0.13	0.14	0.13	0.13	0.12
32".....	0.15	0.14	0.12	0.13	0.12	0.12	0.12

MASONRY

(Fig. 33)

Masonry Thickness.	A.	B.	C.
12".....	0.54	0.51	0.28
18".....	0.45	0.43	0.25
24".....	0.39	0.37	0.23
30".....	0.34	0.32	0.21
36".....	0.30	0.28	0.19
48".....	0.24	0.23	0.17

CONCRETE

(Fig. 33)

Concrete Thickness.	A.	C.
12".....	0.48	0.26
18".....	0.38	0.24
24".....	0.31	0.21
30".....	0.27	0.19
36".....	0.24	0.17
48".....	0.19	0.15

WOODEN WALLS Fig. 34.	}	Fig. A, 0.31
		Fig. B, 0.20
		Fig. C, 0.08

PARTITIONS

(Fig. 40)

Fig. A. Plaster, one side	0.49
Fig. A. Plaster, two sides	0.36
Fig. B	0.30
Fig. C	0.21
Fig. D	0.21
Fig. E	0.34

FLOORS AND CEILINGS AND ROOFS

	Heated Room Above.	Heated Room Below.			
Fig. 32 {	A {	With plaster	0.031	0.19	
		Without plaster	0.33	0.33	
		B {	With plaster	0.030	0.16
			Without plaster	0.23	0.23
Fig. 35	0.16	0.16			
Fig. 36	0.025	0.15			
Fig. 37	0.25	0.25			
Fig. 38	0.029	0.14			
Fig. 39. With ceiling		0.22			
Fig. 39. Without ceiling		0.42			
Book tile with wood sheathing		0.25			
Fig. 39. Government tile with wood sheathing		0.35			
Shingle roof, with no sheathing		0.42			
Shingle roof, on sheathing		0.31			
Glass					
Single window ($\frac{1}{8}$ ") {	Dry	0.96			
		Wet	1.10		
Double window ($\frac{1}{8}$ ")	0.41				
Single skylight ($\frac{1}{4}$ ")	1.06				
Double skylight ($\frac{1}{4}$ ")	0.51				
Doors.					
$\frac{3}{4}$ " doors	0.55				
1" doors	0.48				
1 $\frac{1}{2}$ " doors	0.40				
2" doors	0.34				

The area *A* of the conducting and radiating walls or partitions is found from the plans of the buildings. The total area of each side of the room is first found and these are kept separate

and called north wall, east wall, south wall and west wall. The glass and door area in each of these walls is found and the results added together. This sum is subtracted from the wall area and then the difference is known as net wall area. The door area is considered to be equal to a window in radiating value because of its repeated opening and closing. The author usually takes the full window frame area as window area in making these computations. This allows for the loss of heat due to leakage around the frames. The floor and ceiling areas are also computed at this point for the determination of heat losses.

Temperatures of Rooms and Atmosphere. In figuring t_i and t_o , the temperatures on the two sides of a wall, it is well to remember that the use for which the room is intended fixes one of these and if the wall is an outside one the other, t_o , is fixed by the lowest temperature which may continue for several days or for a week. In the latitude of northern New York t_o may be taken as 0° F. or it may be -10° F., while in New York City, 10° F. or 20° F. may be used. In Washington 20° F. or 30° F. might be used. In most contracts the specifications call for sufficient installation to heat the building to a desired temperature in zero weather, but it seems that this should not be used in design unless zero weather is found for several consecutive days in the locality considered.

The temperature t_i depends on the use of the room. In general living rooms, offices, schoolrooms and other places where persons may be seated in small numbers with wraps removed, the temperature is taken as 70° F., while for halls a lower temperature is used and for churches where street clothes are not removed a temperature less than 70° is used. This also applies to places where muscular exercise takes place. The following temperatures, averaged from several authors, are recommended:

Warm air baths.....	122° F.
Steam baths.....	113° F.
Massage rooms.....	77° F.
Hothouses.....	77° F.
Bathrooms.....	72° F.

Hospital rooms	72° F.
Houses, offices, schools	70° F.
Sewing rooms	70° F.
Laboratories where observers are seated and physically at rest	70° F.
Lecture halls, auditoriums	66° F.
Prisons	65° F.
Shops for light work	64° F.
Churches	64° F.
Sleeping rooms	60° F.
Entrances, corridors	60° F.
Laboratories with engines	55° F.
Gymnasiums, workshops	55° F.

When rooms are not heated the radiation from other rooms gives heat sufficient to bring them to a temperature above the surrounding air. The following are quoted from Kinealy as used by German engineers:

Cellars and rooms kept closed	32° F.
Rooms often open to outside as vestibules	23° F.
Attics under metal or slate roofs	14° F.
Attics under tile, cement or tar and gravel roofs	23° F.

The temperature of garrets may be computed better by equating the heat loss through the roof to the heat gain from the floor to the attic. This gives

$$K_c A_c (t_i - t_a) = K_r A_r (t_a - t_o). \quad \therefore \quad (39)$$

The only unknown is t_a , and this may be readily computed. The author would recommend the following:

Cellars	36° F.
Vestibules	20° F.
Attic under slate roof	25° F.
Attic under booktile and metal roof	40° F.
Attic under wood and metal roof	32° F.

The above temperatures of rooms for various purposes are the temperatures found about 5 ft. from the floor and may be used as the average t_4 for rooms which are not over 10 ft. high. When the room is higher than this the average temperature t_4 is higher than the values in the table. According to Rietschel the values t are those desired at head height, and the mean temperature t_4 must be such that

$$t_4 = t + 0.017(h - 10)t. \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

$$t_4 \text{ never exceeds } 1.15t;$$

t = desired temperature head high;

h = height in room;

t_4 = mean temperature.

If

t_c = temperature at ceiling,

$$t_c = t + 0.035(h - 10)t. \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

Having found the mean temperatures of each room or space in a building as well as the floor and ceiling temperatures of high rooms, these temperatures should be marked on the plans in each room.

Effect of Exposure and Intermittent Heat. The constants of transmission are those found in rooms which are not exposed to violent winds and are heated continuously. When the room is exposed a percentage is added to the quantity found and if the building is heated intermittently a similar method is used so that the building may be heated in a reasonable time. The following averages of the allowances made by various authors are suggested:

For north walls.....	add	15 per cent
For east or west walls.....	"	10 "
For corner walls.....	"	10 "
For rooms exposed to winds.....	"	15 "
For heating during day and building closed at night.....	"	10 "
For heating during day and open at night.....	"	30 "
For heating occurring at long intervals.....	"	50 "

These allowances for occasional heating may be reduced if time is available for heating the building, say 10 to 18 hours.

Heat Loss through Walls. The various factors K , A , and $(t_1 - t_0)$ now being known; the heat loss from the four walls, floor and ceiling of any room may be found by the equation (27).

A very simple equation for rapid calculation is one due to Professor R. C. Carpenter and is often spoken of as Carpenter's Rule. This is

$$H = (G + \frac{1}{4}W + 0.02V)(t_1 - t_0). \quad (42)$$

H = B.t.u.'s to be supplied per hour;

G = glass surface in sq.ft.

W = net exposed wall area in sq.ft.;

V = volume of air supplied per hour

= nv where n = No. of times air in room of volume v cu.ft. is changed per hour.

This gives a rapid method of estimating the amount of heat to be supplied.

Heat for Ventilation. If V is the number of cubic feet of air per hour introduced into the room from a temperature t_0 and raised to a temperature t_1 , the amount of heat required for this will be

$$H_v = \frac{pV}{RT_1} c_p (t_1 - t_0). \quad (43)$$

This becomes under general conditions,

$$H_v = \frac{14.7 \times 144 \times .24}{53.34 \times (70 + 460)} V(t_1 - t_0)$$

$$= 0.018V(t_1 - t_0). \quad (44)$$

In this 0.018 is therefore the amount of heat to raise 1 cu.ft. of air at 70° F., 1° F. at constant pressure. This quantity for rough calculations may be considered as 0.02. The quantity

H_v is the amount of heat required to warm the air to room temperature and represents the amount of heat necessary to warm the air which may enter by leakage; or if the air escapes at room temperature it represents the amount carried out by that air above the heat in the air at the temperature t_o .

It is not necessary in finding the heat required to warm the ventilating or leakage air, to consider the air as having a higher temperature than t_i , because the air is supposed to leave at this temperature, the higher temperature at which the air entered the room being decreased to that of the room by the heat loss from the walls, and the amount for the change of temperature to that of the room is equal to the heat loss from the walls. If the amount of heat to warm V cu.ft. of air from t_2 to t_3 is required, this is approximately given by

$$H_a = 0.018V(t_3 - t_2). \quad (45)$$

If the V is measured at $t_3 = 120^\circ$ F. instead of 70° F. the constant 0.018 would truly be 0.0165. In using this value, 0.018, it is well to remember that it applies only when V is the volume considered at 70° F.

Heat Given Out by Persons, Lights, Motors, etc. The amount of heat given out by persons working or sitting in a room, although not so important when only a few are present, is of genuine importance when a number are to be considered. The following average values may be used:

Adult at rest	380	B.t.u. per hour
Adult at work	470	“ “
Adult at hard work	550	“ “
Adult, in old age	360	“ “
Infant	63	“ “
Child	240	“ “

For electric lighting:

$$1 \text{ Watt hour} \frac{33000 \times 60}{746 \times 778} = 3.41 \text{ B.t.u.}$$

For gas lighting:

1 cu.ft. natural gas	1000	B.t.u.
1 cu.ft. illuminating gas	700	“
1 cu.ft producer gas	150	“
1 Welsbach burner uses 3 cu.ft. per hour.		
1 Fish-tail burner uses 5 cu.ft. per hour.		

For motors:

Motors used in a room for tools or apparatus of various kinds turn eventually all of the power input into heat which remains in the room, hence,

$$\begin{aligned}
 1 \text{ K.W. hour supply} &= 3410 \text{ B.t.u.} \\
 1 \text{ H.P. " " " " } &= 2546 \text{ " " " " }
 \end{aligned}$$

It is important to realize that the power developed to drive all machines is turned into heat provided that all of the energy is used in a room. This heat may be an important item in heating a shop containing a large number of tools.

Equivalent Temperature. When a given temperature T_b may be obtained in the room of a building, if the atmospheric temperature is T_a , it is often necessary to know if this is equivalent to a temperature of T_θ for the rooms when the air temperature is $T_{a\theta}$ these being the guaranteed conditions. Now for direct radiation, the loss equals the heat given by the radiators. This means,

$$(A_w K_w + A_\theta K_\theta + 0.02V)(T_b - T_a) = K A_r (T_s - T_b) \quad . \quad (46)$$

and under guaranteed conditions,

$$(A_w K_w + A_\theta K_\theta + 0.02V)(T_\theta - T_{a\theta}) = K_1 A_r (T_s - T_\theta) \quad . \quad (47)$$

Dividing these the following result:

$$\frac{T_b - T_a}{T_\theta - T_{a\theta}} = \frac{K}{K_1} \frac{T_s - T_b}{T_s - T_\theta} \quad . \quad . \quad . \quad . \quad (48)$$

Now K_1 is practically the same as K , and if it is so considered the following is found:

$$T_b = \frac{T_s[T_g + T_a - T_{ag}] - T_a T_g}{T_s - T_{ag}} \dots \dots (49)$$

For an indirect system the equation becomes

$$\frac{T_b - T_a}{T_g - T_{ag}} = \frac{K \left[T_s - \frac{T_a + T_b}{2} \right]}{K' \left[T_s - \frac{T_{ag} + T_g}{2} \right]} \dots \dots (50)$$

In this case K' and K will differ more than in the previous case, but if the actual conditions and guaranteed conditions are not far different these two quantities may be considered equal, and the expression for T_b is the same as given above.

This is given in a table below for zero outside weather in the guarantee and steam at 5 lbs. gage pressure.

EQUIVALENT INSIDE TEMPERATURE

Actual Outside Temperature.	Guaranteed inside Temperature.			
	50	60	70	80
-10	42.2	52.6	63.1	73.5
0	50.0	60.0	70.0	80.0
10	57.8	67.4	76.9	86.5
20	65.6	74.8	83.8	93.0
30	73.4	82.2	90.7	99.5

CHAPTER IV

RADIATORS, VALVES AND HEAT TRANSMISSION FROM RADIATORS

Radiators are of various forms. Some are made of cast iron or pressed steel and some are made of wrought-iron pipe.

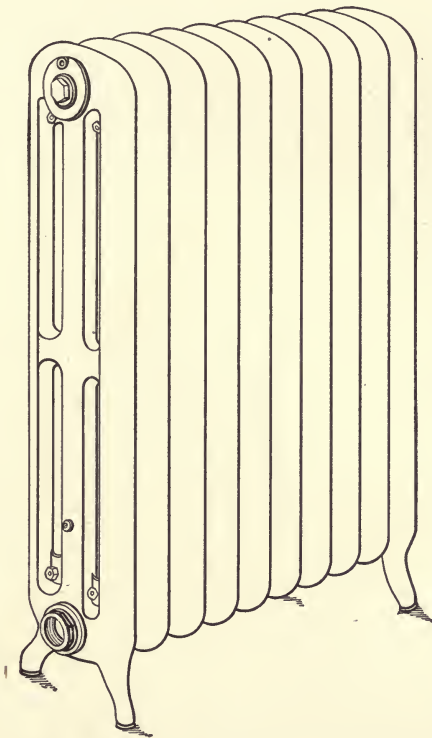


FIG. 41.—Peerless Three-column Radiator.

The common form of cast-iron radiator, Fig. 41, is made by connecting several cast-iron sections together by close threaded **right and left nipples** as shown in Fig. 42, or by using **conical**

ended thimbles which are held in place in the sections by means of the bolts. The nipples are made with a projection on the inside so that they may be turned by a bar which forms a plug wrench. The sections are made of various heights from 13 to 45 ins., and in order to give varying amounts of heating

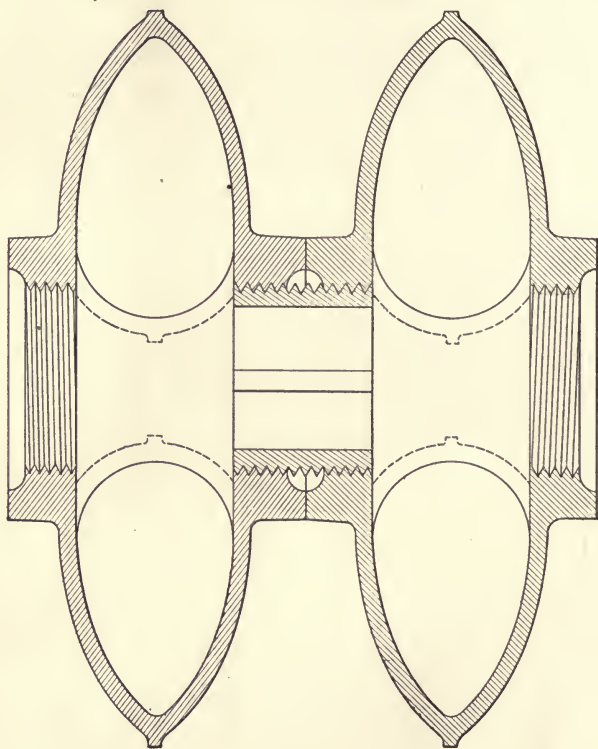


FIG. 42.—Section of Loops Showing Nipple.

surface they are made of various widths. The widths are changed by increasing the number of tubes forming **single column, two column, three column, or four column radiators**. The design of the exterior of these radiators is varied to suit different architectural requirements. Figs. 41, 43, 44 and 45 illustrate different styles of various typical radiators made by the American Radiator Company.

These radiators are connected at the lower end of each section for steam work, while for hot-water systems the sections are connected at the top and bottom as shown in Fig. 46. The purpose of this is to aid in the circulation of the water by bringing it in at the top and distributing it to the various sections

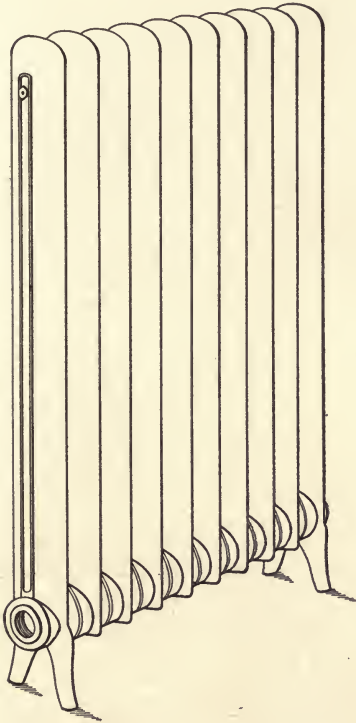


FIG. 43.—Peerless Single Column Radiator.

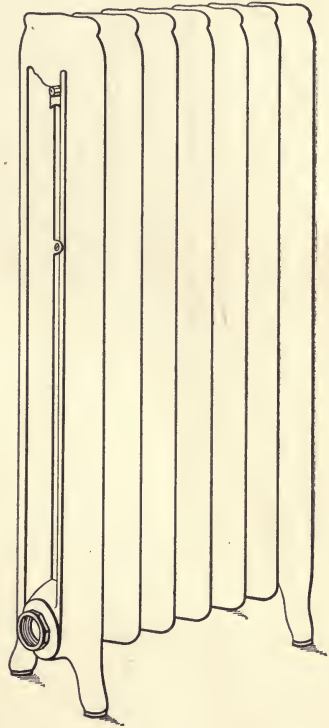


FIG. 44.—Rococo Two-column Radiator.

which act as **down-takes**. This figure illustrates the method of constructing these radiators when it is desired to have the floor line clear. The **leg sections** are not used in forming the radiator and the radiator is held by a **bottom support and top guide**, Fig. 47.

At times the sections are so constructed that they form, by projecting fins or webs, a series of closed passages between

sections. Such radiators, Fig. 48, are known as **flue** or **box radiators**. This form is used in the **direct-indirect** system of heat-

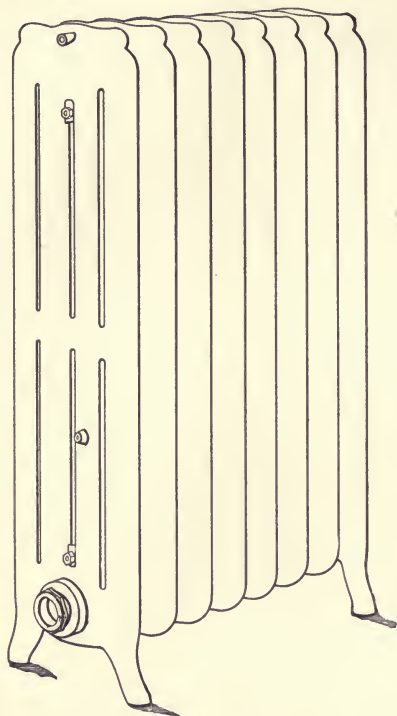


FIG. 45.—Rococo Four-column Radiator.

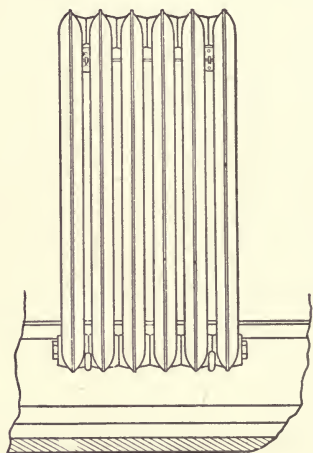
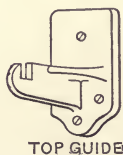
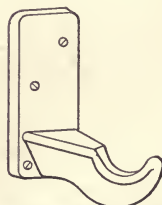


FIG. 46.—Hot Water Radiator on Brackets.



TOP GUIDE



BOTTOM SUPPORT

FIG. 47.—Brackets for Wall Radiators.

ing, shown in Fig. 6, in which it is desired to draw in a certain amount of air for ventilation.

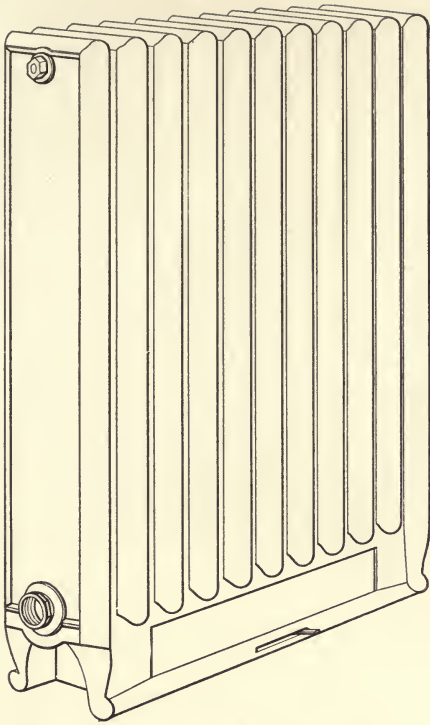


FIG. 48.—Flue Box Base Radiator.

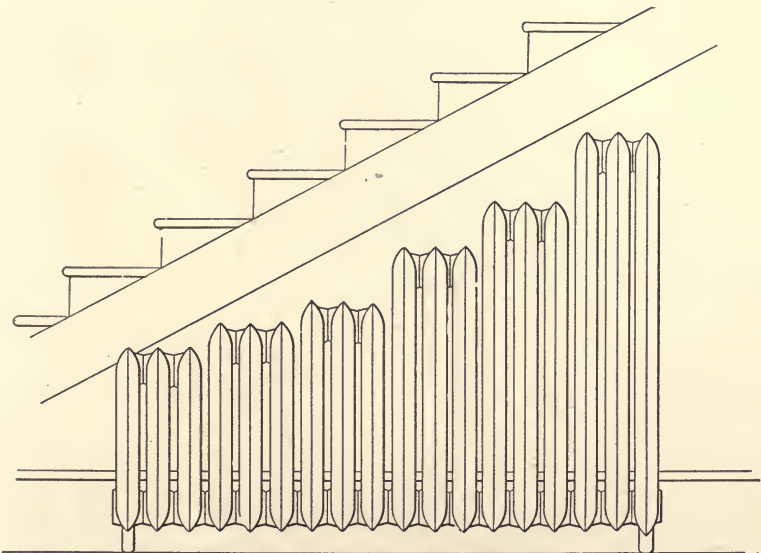


FIG. 49.—Stairway Radiator.

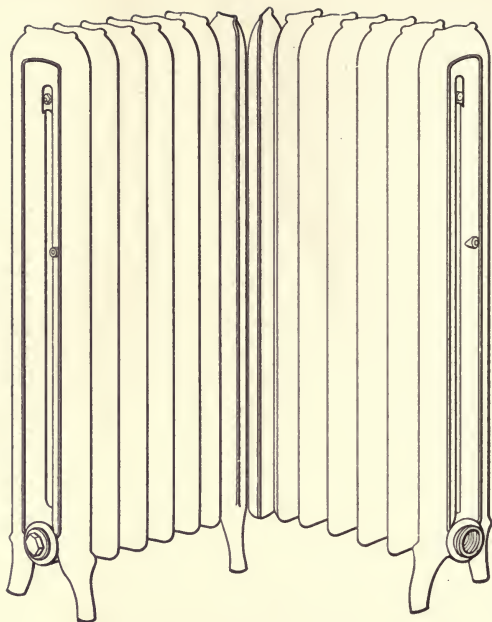


FIG. 50.—Corner Radiator.

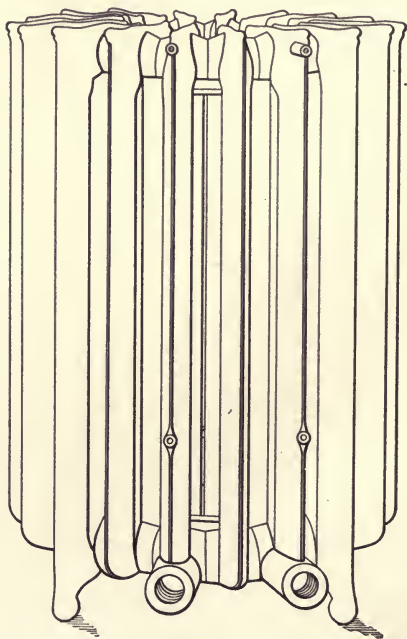
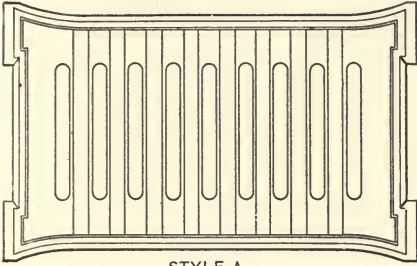
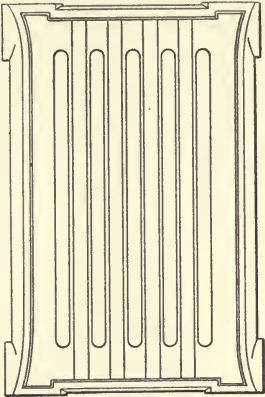


FIG. 51.—Circular Radiators.



STYLE A



STYLE B

FIG. 52.—Rococo Wall Radiators.

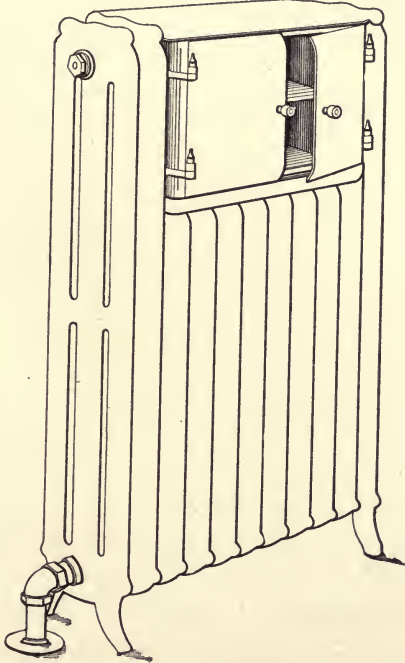


FIG. 53.—Dining Room Radiator.

The heights of the radiators are selected to fit the particular positions in which they are to be placed. Usually it is well to have them lower than a window sill in front of which they are to stand. At times they are put beside the jib panel of a stairway, and in such a case a number of different sizes may be united to suit the steps. Fig. 49 is one of this form. **Low**

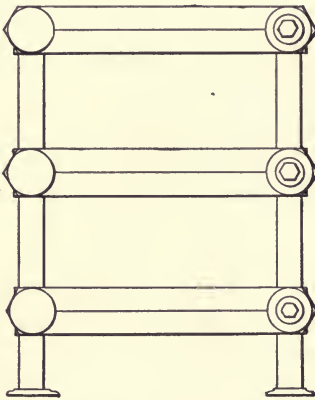
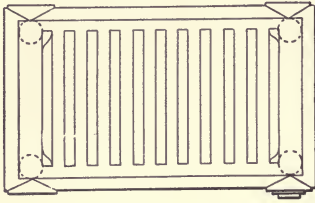


FIG. 54.—Plate Warmer or Pantry Radiator.



FIG. 55.—Detachable High Leg Section.

radiators are made of such dimensions that they may be placed under window seats, and in these cases they are known as **window radiators**.

When a **corner radiator** is to be placed in a room, Fig. 50, or is to be made **circular** to fit around a column, Fig. 51, special sections are made and joined together to form these. These may be placed where they are needed for architectural effects;

they are not often used. In most cases the circular radiator is made in halves.

The radiator shown in Fig. 52 is known as the **Rococo wall radiator**, and this type is employed when little space is available. The sections are made so that they may be joined at the various corners and by making various combinations of elements, dif-

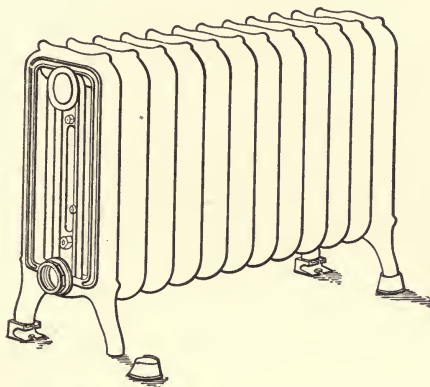


FIG. 56.—Radiator Foot Ups.

ferent shaped spaces may be filled. These radiators are made of the following dimensions:

Section No.	Length.	Width.	Thickness.	Area.
5A	$16\frac{5}{8}$ ins.	$13\frac{5}{16}$ ins.	$2\frac{7}{8}$ ins.	5 sq.ft.
7A and 7B	$21\frac{7}{8}$ "	$13\frac{5}{16}$ "	$2\frac{7}{8}$ "	7 "
9A and 9B	$29\frac{1}{16}$ "	$13\frac{5}{16}$ "	$2\frac{7}{8}$ "	9 "

These are the over all dimensions of the section, and various sections may be so connected by nipples that they come together, iron to iron.

Radiators may be joined together forming a sort of cupboard, giving a so-called **dining-room** or **pantry radiator**, Fig. 53. **Plate warmers**, Fig. 54, may be constructed by using wall radiators.

The **leg loops** of radiators are sometimes made with a **detachable leg**, Fig. 55, so that in moving carpets the leg at

one end may be removed and then at the other. The radiator

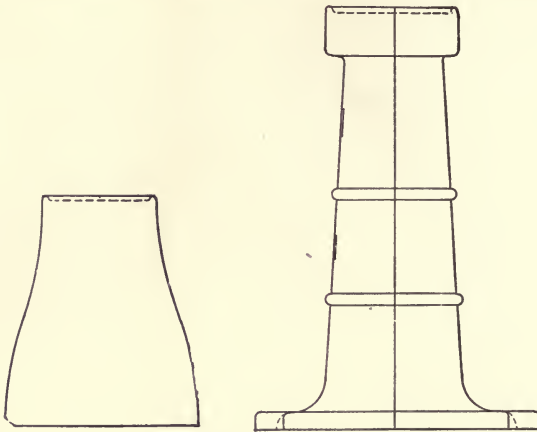


FIG. 57.—Pedestals.

foot ups, Fig. 56, may be used for the same purposes. **Radiator pedestals**, Fig. 57, may be used to raise the radiator from the floor an additional distance when necessary.

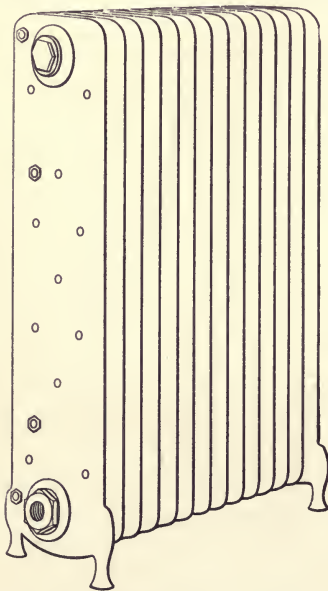
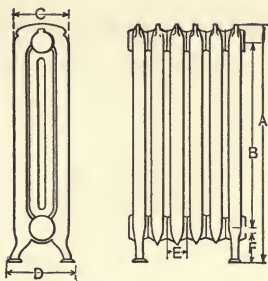


FIG. 58.—Pressed Steel Radiator.

Fig. 58 gives the appearance of the Kinnear **pressed steel radiator**. The radiators are light and will not crack. They are giving good satisfaction. The following tables give data in regard to the size of the various radiators, where F_1 refers to height to center of steam supply, or supply and return tapping for water, while F_2 refers to tapping for single pipe steam or return in two pipe steam.

Pipe coil radiators or **pipe coils** are very satisfactory. They are made by joining pipes together



MEASUREMENTS OF AMERICAN RADIATORS

Pattern.	A	B	C	D	E	F ₁	F ₂	Heating surface sq.ft.
Rococo, Peerless one column.	38	31 ³ / ₃₂	4 ¹ / ₂	5 ¹ / ₂	2 ¹ / ₂	4 ¹ / ₂	4	3
	32	25 ³ / ₆₄	4 ¹ / ₂	5 ¹ / ₂	2 ¹ / ₂	4 ¹ / ₂	4	2 ¹ / ₃
	26	19 ³ / ₃₂	4 ¹ / ₂	5 ¹ / ₂	2 ¹ / ₂	4 ¹ / ₂	4	2
	23	16 ⁹ / ₁₆	4 ¹ / ₂	5 ¹ / ₂	2 ¹ / ₂	4 ¹ / ₂	4	1 ² / ₃
	20	13 ¹ / ₆₄	4 ¹ / ₂	5 ¹ / ₂	2 ¹ / ₂	4 ¹ / ₂	4	1 ¹ / ₂
Two-column (Peerless Hospital radiators same as this, except E is 3 ins. in place of 2 ¹ / ₂)	45	38 ²⁵ / ₃₂	7 ³ / ₈	8 ¹ / ₂	2 ¹ / ₂	4 ¹ / ₂	4	5
	38	31 ³ / ₃₂	7 ³ / ₈	8 ¹ / ₂	2 ¹ / ₂	4 ¹ / ₂	4	4
	32	25 ³ / ₆₄	7 ³ / ₈	8 ¹ / ₂	2 ¹ / ₂	4 ¹ / ₂	4	3 ¹ / ₃
	26	19 ¹⁵ / ₃₂	7 ³ / ₈	8 ¹ / ₂	2 ¹ / ₂	4 ¹ / ₂	4	2 ² / ₃
	23	16 ⁹ / ₁₆	7 ³ / ₈	8 ¹ / ₂	2 ¹ / ₂	4 ¹ / ₂	4	2 ¹ / ₃
20	13 ¹ / ₆₄	7 ³ / ₈	8 ¹ / ₂	2 ¹ / ₂	4 ¹ / ₂	4	2	
Peerless only	15	7 ³ / ₈	8 ¹ / ₂	2 ¹ / ₂	4 ¹ / ₂	4	1 ¹ / ₂
Three column	45	38 ²⁵ / ₃₂	Peerless	10	2 ¹ / ₂	4 ¹ / ₂	4	6
	38	31 ³ / ₃₂	9 ins. Rococo	10	2 ¹ / ₂	4 ¹ / ₂	4	5
	32	25 ³ / ₆₄	9 ¹ / ₄ ins. Rococo	10	2 ¹ / ₂	4 ¹ / ₂	4	4 ¹ / ₂
	26	19 ¹⁵ / ₃₂		10	2 ¹ / ₂	4 ¹ / ₂	4	3 ³ / ₄
	22	15 ⁷ / ₃₂		10	2 ¹ / ₂	4 ¹ / ₂	4	3
	18	11 ³ / ₁₆		10	2 ¹ / ₂	4 ¹ / ₂	4	2 ¹ / ₄
Four column	45	38 ²⁵ / ₃₂	10 ¹ / ₂	11 ¹ / ₄	3	4 ¹ / ₂	4 ¹ / ₂	10
	38	31 ³ / ₃₂	10 ¹ / ₂	11 ¹ / ₄	3	4 ¹ / ₂	4 ¹ / ₂	8
	32	25 ¹³ / ₆₄	10 ¹ / ₂	11 ¹ / ₄	3	4 ¹ / ₂	4 ¹ / ₂	6 ¹ / ₂
	26	19 ¹⁵ / ₃₂	10 ¹ / ₂	11 ¹ / ₄	3	4 ¹ / ₂	4 ¹ / ₂	5
	22	15 ⁷ / ₃₂	10 ¹ / ₂	11 ¹ / ₄	3	4 ¹ / ₂	4 ¹ / ₂	4
	18	11 ³ / ₁₆	10 ¹ / ₂	11 ¹ / ₄	3	4 ¹ / ₂	4 ¹ / ₂	3
Rococo window	20	15 ¹ / ₁₆	12 ¹ / ₂	12 ¹ / ₂	3	3	3	5
	16	11 ¹ / ₃₂	12 ¹ / ₂	12 ¹ / ₂	3	3	3	3 ³ / ₄
	13	8 ¹ / ₁₆	12 ¹ / ₂	12 ¹ / ₂	3	3	3	3
Italian flue	38	31 ⁵ / ₁₆	8 ¹ / ₂	8 ¹ / ₂	3	4 ¹ / ₂	4	7
	32	25 ⁷ / ₁₆	8 ¹ / ₂	8 ¹ / ₂	3	4 ¹ / ₂	4	5 ³ / ₄
	26	19 ¹ / ₂	8 ¹ / ₂	8 ¹ / ₂	3	4 ¹ / ₂	4	4 ¹ / ₂
	20	13 ⁹ / ₁₆	8 ¹ / ₂	8 ¹ / ₂	3	4 ¹ / ₂	4	3 ¹ / ₄
Italian flue Dir.-ind.	39 ¹ / ₂	31 ⁵ / ₁₆	8 ¹ / ₂	8 ¹ / ₂	3	6	5 ¹ / ₂	7
	33 ¹ / ₂	25 ⁷ / ₁₆	8 ¹ / ₂	8 ¹ / ₂	3	6	5 ¹ / ₂	5 ³ / ₄
	27 ¹ / ₂	19 ¹ / ₂	8 ¹ / ₂	8 ¹ / ₂	3	6	5 ¹ / ₂	4 ¹ / ₂
	21 ¹ / ₂	13 ⁹ / ₁₆	8 ¹ / ₂	8 ¹ / ₂	3	6	5 ¹ / ₂	3 ¹ / ₄

by **return bends**, Fig. 59, or by **branch tees** or **manifolds**, Fig. 60. In all cases, it must be remembered that some of the pipes are



FIG. 59.—Pipe Coil with Return Bends.

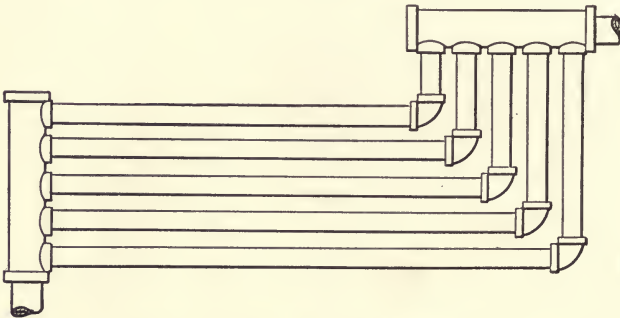


FIG. 60.—Pipe Coil with Manifolds.

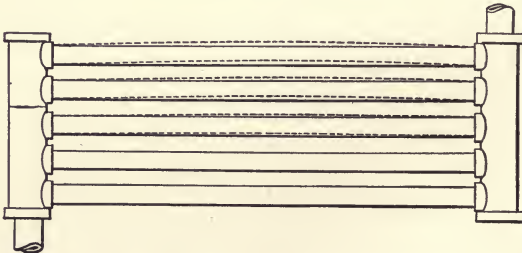


FIG. 61.—Improper Form of Pipe Coil.

apt to heat first, so that such a construction as shown in Fig. 61 is improper, as the expansion of the top pipe before the other pipes are heated is apt to cause the rupture of the branch tees

The arrangement shown in Figs. 59 and 60 permits the expansion of any individual pipe without straining the system. In all coil construction one must remember that any one pipe

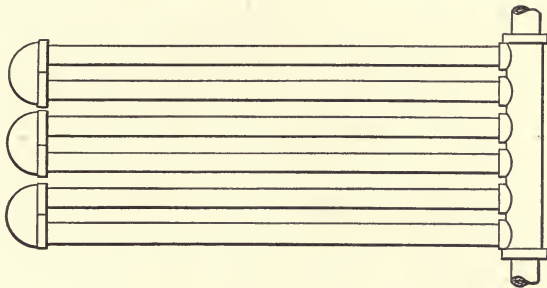


FIG. 62.—Coil with Bends and Manifold.

may expand more than any other. Fig. 62 is a fairly good construction, although here there may be unequal expansion of one pipe of any set which will bring a strain on the return bend.

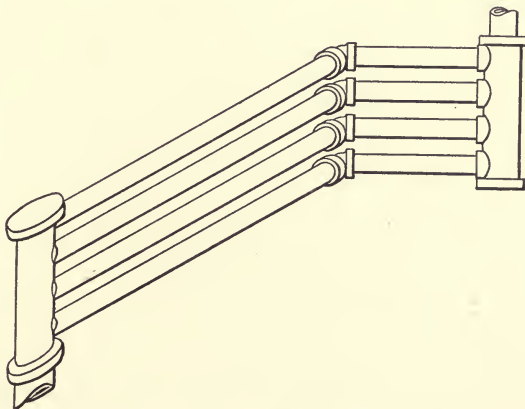


FIG. 63.—Corner Coil.

The fact that air may collect in any set is another objection to this system. Fig. 63 is a coil of good construction to be used at the corner of a room. In this form expansion may occur in any pipe.

For the purpose of giving data to be used for coil construction, the following table of certain sizes is added.

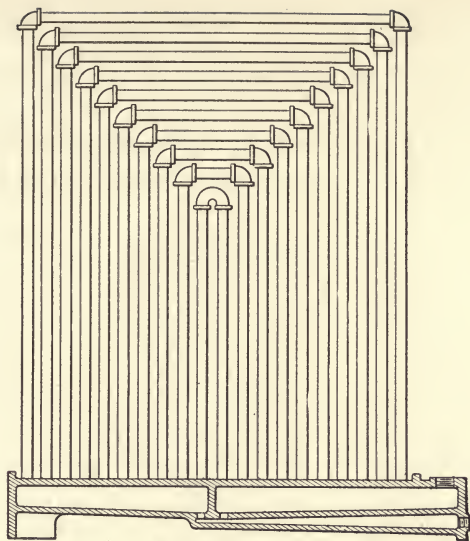
DATA FROM WROUGHT IRON PIPE

Nom. Diam.	Actual Inside Diam.	Actual Outside Diam.	Internal Area.		External Area.		Internal surface per ft. of length, sq.ft.	External surface per ft. of length, sq.ft.	Weight per ft. lbs.	Cu.ft. of Contents per ft. of length.	Threads per Inch.	Length of Thread.	List Price.
			sq.in.	sq.ft.	sq.in.	sq.ft.							
1	0.36	0.54	0.104	0.0007	0.229	0.0016	0.095	0.14	0.42	18	0.29	0.05
1	0.62	0.84	0.304	0.0021	0.554	0.0038	0.163	0.22	0.84	14	0.39	0.08
1	0.82	1.05	0.533	0.0037	0.866	0.0060	0.216	0.28	1.12	14	0.40	0.11
1	1.05	1.32	0.861	0.0060	1.358	0.0094	0.274	0.34	1.67	0.006	11	0.51	0.16
1 1/2	1.38	1.66	1.50	0.0104	2.164	0.0150	0.361	0.43	2.24	0.010	11	0.54	0.22
1 1/2	1.61	1.90	2.036	0.0141	2.835	0.0197	0.443	0.50	2.68	0.014	11	0.55	0.27
2	2.07	2.38	3.356	0.0233	4.430	0.0308	0.541	0.62	3.61	0.023	11	0.58	0.36
2	2.47	2.87	4.78	0.0332	6.492	0.0451	0.646	0.75	5.74	0.03	8	0.89	0.57
3	3.07	3.50	7.38	0.0513	9.621	0.0668	0.744	0.92	7.54	0.05	8	0.95	0.75
4	4.03	4.50	12.73	0.0884	15.904	0.1104	1.05	1.18	10.66	0.09	8	1.05	1.08
5	5.05	5.56	19.98	0.1388	24.301	0.1688	1.32	1.45	14.50	0.14	8	1.16	1.45
6	6.06	6.63	28.89	0.2006	34.47	0.2394	1.59	1.73	18.76	0.2	8	1.26	1.88
8	7.98	8.62	50.03	0.3474	58.43	0.4057	2.09	2.25	28.18	0.34	8	1.46	2.82
10	10.02	10.75	78.83	0.5474	90.76	0.6303	2.62	2.81	40.06	0.54	8	1.68	3.20
12	12.00	12.75	113.10	0.7854	127.68	0.8867	3.14	3.33	49.00	0.78	8	1.88	4.50

For indirect heating the radiators are so arranged that the air passing the surface may abstract heat readily. The coil form of radiator or heater is arranged usually with four staggered rows of 1-in. pipe. The pipes, Fig. 64, are attached at two places to a cast-iron base. In some forms, as those of the B. F. Sturtevant Co. and the Buffalo Forge Co., the division between the supply and return is made at the center of the base, while the Massachusetts Fan Company divides its base by a longitudinal partition.

The pipes of Fig. 64 are constructed with one piece of each section with right and left ends. The steam is supplied through the upper section of the base and is discharged into the lower section. The small hole relieves the upper part of water. The figure shows the forms of base used by the two companies.

The type shown in Fig. 65 is that used generally by the Massachusetts Fan Company, although it is also used by others. The base is divided by a vertical partition with a drain hole



BUFFALO BASE



STURTEVANT BASE

FIG. 64.—Coil Heater Section.

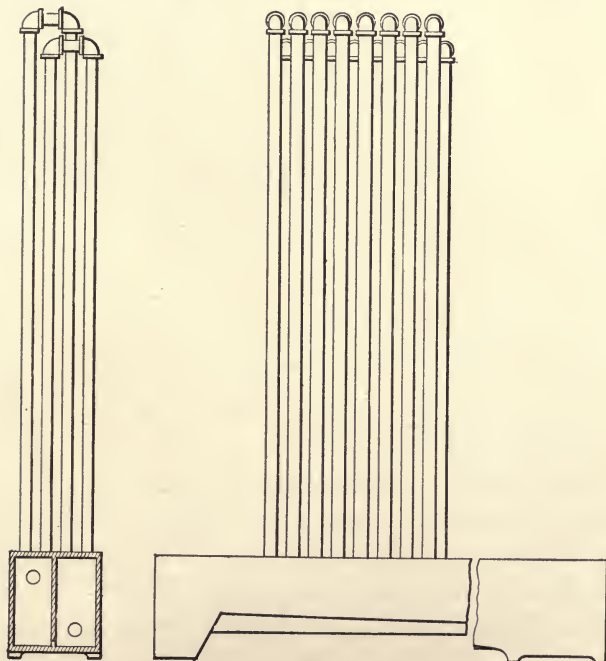


FIG. 65.—Massachusetts Coil.

at one point. The sections are of varying length for simplicity in construction.

Fig. 66 gives the form of the "positivflo" heater of the Green Fuel Economizer Co. In this heater the horizontal tubes are

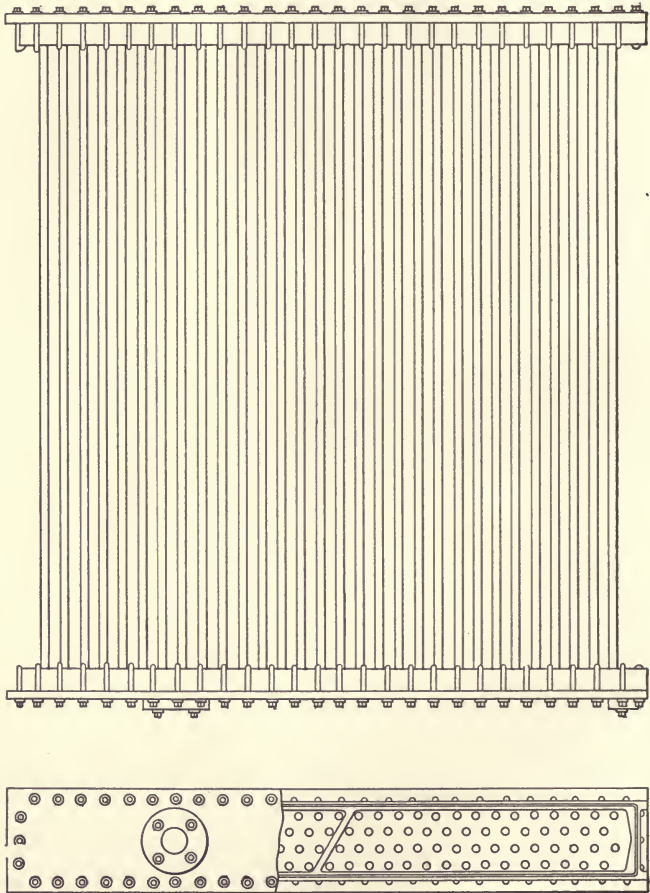


FIG. 66.—Positivflo of the Green Co.

attached at each end to a header box, the front one of which is separated into two parts by a partition, so that the steam flows in one direction through one half of the tubes, returning to the front by the other half.

Cast forms of indirect radiators have been used for many

years; the advantage claimed for them is a small number of joints and, if properly designed, a more compact radiator. A recent form introduced by the American Radiator Company is the **Vento Cast Iron Hot Blast Heater**, Fig. 67. In the figure

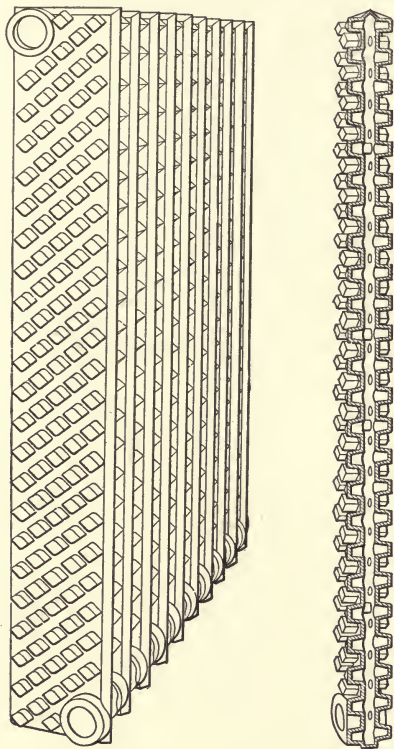


FIG. 67.—Vento Heater.

a section through the projections of the heater is shown from which it is seen that there is ample space for steam or water to reach all parts of the section. Occasional **ties** are cast in the interior to support the flat plate. These sections are united by **right and left hexagonal nipples** which

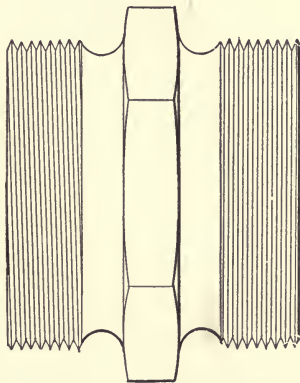


FIG. 68.—Right and Left Nipples.

are screwed in when the heaters are made up into stacks. The number of sections in a stack varies with the requirements of an installation. Twenty-four is the largest number that is recommended to be put together, and when supply and return are to be placed at the same end of a stack eighteen sections is the limit.

These are made of two forms, regular and narrow, in three lengths, 40 ins., containing 10.75 sq.ft. (reg.) or 7.5 sq.ft. (narrow),

50 ins. containing 13.5 sq.ft. (reg.) or 9.5 sq.ft. (narrow), and 60 ins. containing 16 sq.ft. (reg.) or 11 sq.ft. (narrow).

The nipples are made of such length that the sections may be placed $4\frac{5}{8}$, 5 or $5\frac{3}{8}$ ins. on centers. These give the following areas between sections for the passage of air.

NET AIR SPACE IN SQUARE FEET

Sections	Centers		
	$4\frac{5}{8}$ in.	5 in.	$5\frac{3}{8}$ in.
40	0.52	0.62	0.72
50	0.65	0.77	0.91
60	0.78	0.92	1.08

The widths of the regular sections are $9\frac{1}{8}$ ins. while the narrow sections are $6\frac{3}{4}$ ins.

Fig. 69 gives the dimensions of the vento sections. Fig.

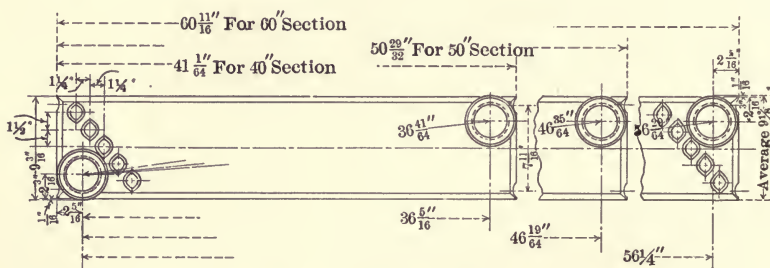


FIG. 69.—Measurements of Regular Vento Heaters.

70 illustrates an **Excelsior Indirect Radiator** built by the American Radiator Co. The detail of the end shows how the flow is directed through the upper pipe and down to the bottom pipe to the next radiator. The fins on the outside increase the heating surface. These radiators are usually employed where individual heat stacks are placed beneath a room or set of rooms to be heated.

The sections are $23\frac{3}{4}$ ins. or 36 by 8 ins. and the thickness or width is $3\frac{3}{8}$ ins. These contain 8 or 12 sq.ft. of radiating

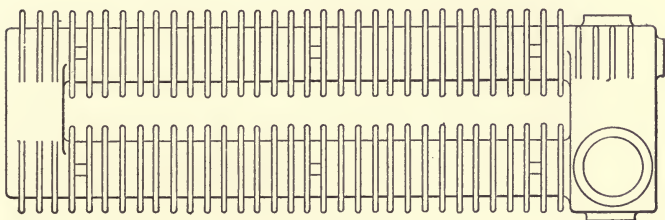
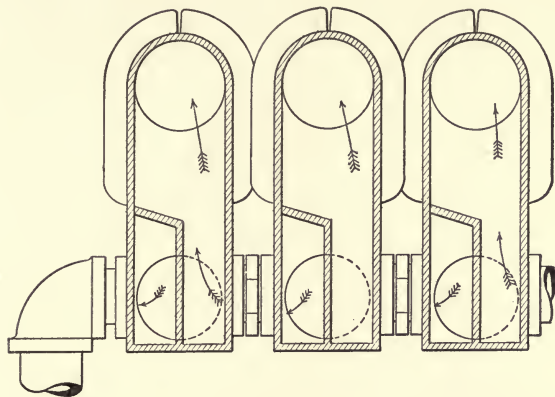


FIG. 70.—Excelsior Indirect Radiator.

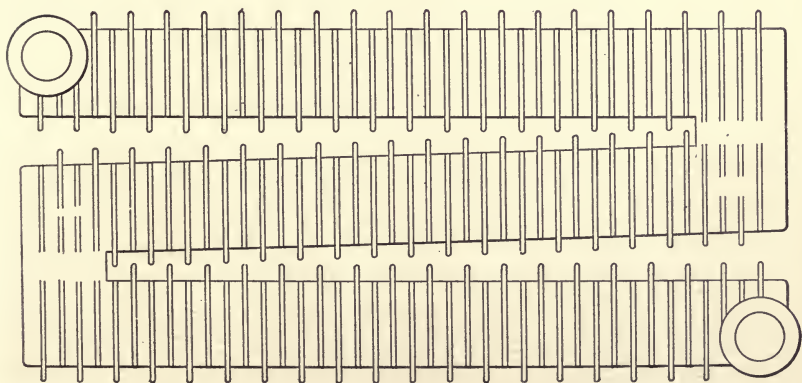


FIG. 71.—Sterling Indirect Radiator.

surface. Fig. 71 represents their **Sterling Indirect Radiator**, which is somewhat similar in form. Each section contains 20 sq.ft. of surface and requires a space of $36\frac{3}{4}$ by $15\frac{3}{4}$ by $3\frac{1}{2}$ ins.

The **Perfection Pin Indirect Radiator**, Fig. 72, may be used for the same purpose. The American Radiator Company build these in different forms, containing 10 sq.ft. and 15 sq.ft. of heating surface. The length is $36\frac{1}{4}$ ins. and the height is $7\frac{1}{2}$ or $11\frac{1}{2}$ ins. on the square part, while the projecting points increase the height $2\frac{1}{2}$ ins. The width of the sections is about $2\frac{7}{8}$ ins. The tappings for this radiator are made at different points on the projecting ends to suit given conditions.

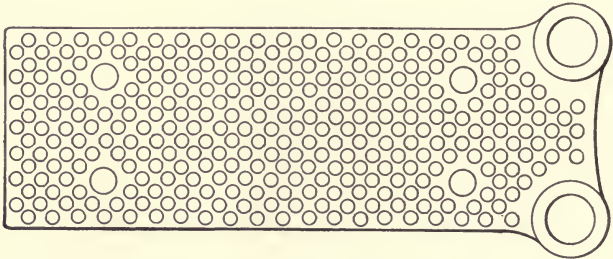


FIG. 72.—Perfection Pin Radiator.

All of the last forms are intended for small individual stacks with natural circulation.

The coil radiators are sometimes formed as **miter coils**, Fig. 73, in which steam is admitted at the upper end and the condensed steam is discharged at the lower end. The arrangement permits expansion to occur.

Valves. The valves used on radiators are usually of the **angle** form and made with a **union** on one side, although any form of valve could be used. The reason for this peculiar form is that it is desirable to have close connections from the radiator, as it is objectionable to have the pipes extend far beyond the radiator, obstructing space and forming a barrier over which persons might trip.

The **Norwall Packless Radiator valve**, Fig. 74, illustrates the form of **angle valve** with the union on the left-hand side.

In the valve shown the valve disc has a piston *A* which prevents steam from coming in contact with the spindle. In addition there is a packing around the spindle at *C* held tight

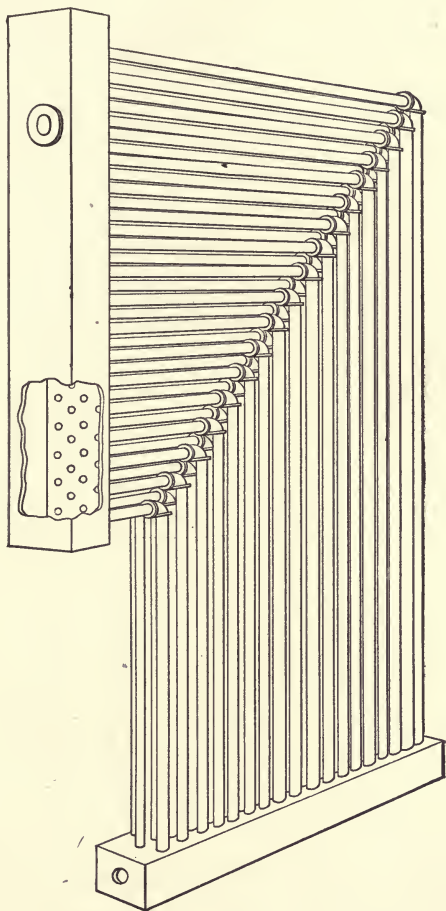


FIG. 73.—Miter Coil.

by the spring *D*. The spring disc at *B* is intended to make up for the contraction of the spindle as the valve cools off, thus keeping the disc against its seat. The arrangement of threads makes this a quick opening valve.

The ordinary form of radiator valve as made by Jenkins Bros. or Crane is similar to this, with the exception that the spindle is attached by a shoulder to the disc. The spindle thread works in the bonnet of the valve and the stem is packed by a stuffing box.

The American Radiator Company has adapted their cor-

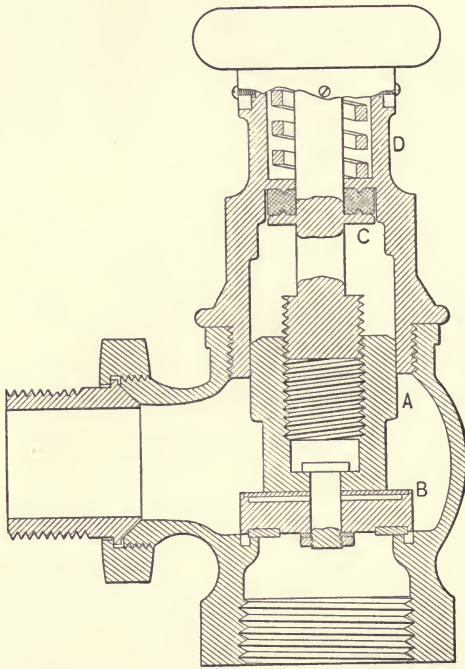


FIG. 74.—Norwall Packless Valve.

rugated metallic bellows or **Sylphon** to the construction of the **Sylphon Packless Radiator Valve**, Fig. 75. In this valve the bellows *A* is attached to a cup *B* on the valve disc and to the bonnet at the top. This encloses the spindle in a space which is free from steam. The extension or contraction of the bellows occurs as the valve is moved in or out. The sylphon is made of steam brass and will stand repeated extensions without rupture.

The **Q. O. Water Radiator Valve** (quick opening) is shown in Fig. 76. Most water radiator valves are made in the form of cocks, so that a quarter turn will open them. The valve shown is so made that the conical body comes in contact with the body of the valve at top, bottom and small vertical strips on each side, which form a partition between the outlet and inlet

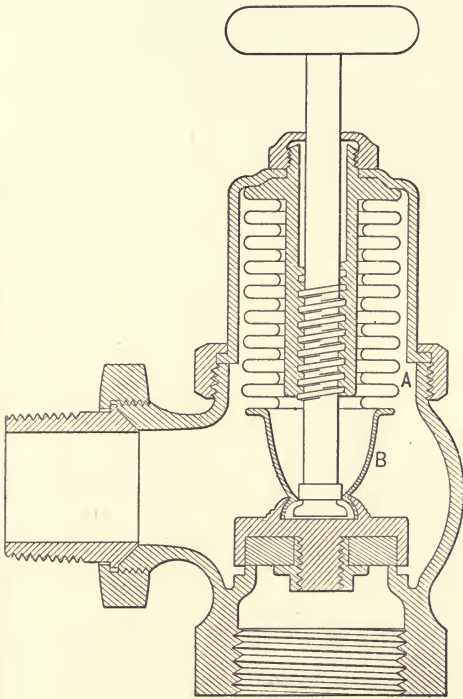


FIG. 75.—Sylphon Packless Valve.

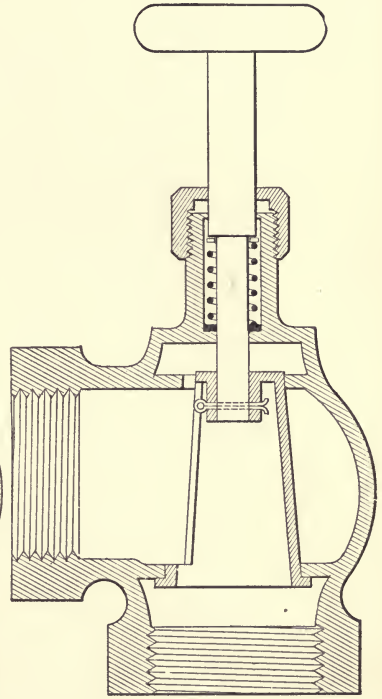


FIG. 76.—Q. O. Water Radiator Valve.

sides of the valve. The taper permits of taking up wear, the spring at the top pressing the valve against its seat. This spring presses against the washer around the spindle of the valve.

In some cases a small hole is made in the valve allowing sufficient circulation in radiators which are shut off, to prevent freezing in cold weather.

A **corner valve**, Fig. 77, of either right or left-hand construction is necessary at times on account of the position of the piping. This does away with elbows and shortens up connections.

When it is necessary to turn steam or water on a radiator and have the same beyond the control of the occupants of the room, a **lock and shield radiator valve**, Fig. 78, is used. The spindle of the valve is surrounded by a shield, and a key

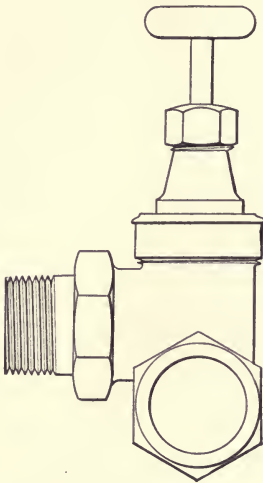


FIG. 77.—Corner Valve.

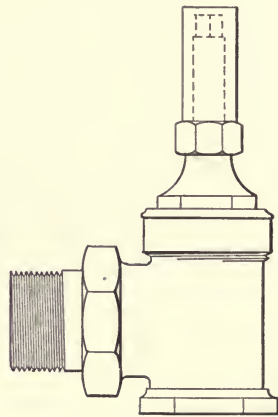


FIG. 78.—Lock and Shield Valve.

or extension handle fitting over the spindle is used to operate the valve.

Special valves are used on radiators to cut down excessive use of steam and to reduce the pressure in the return pipes. Figs. 79 and 80 illustrate the **Thermograde System** of valves for this purpose. There are two valves used, the **modulation or control valve A** and the **auto-valve B**. The **control valve A** has a handle *C* with a pointer moving over a graduated scale, and on the handle is an adjustable pin *D*, moving over a cam *E*, so that when the pointer is at $\frac{1}{4}$ the valve *F* is raised from its seat an amount which will just admit enough steam to heat $\frac{1}{4}$ of the radiator. When the handle is moved to "full" the rapid

rise in the cam opens the valve a large amount. The pin *G*

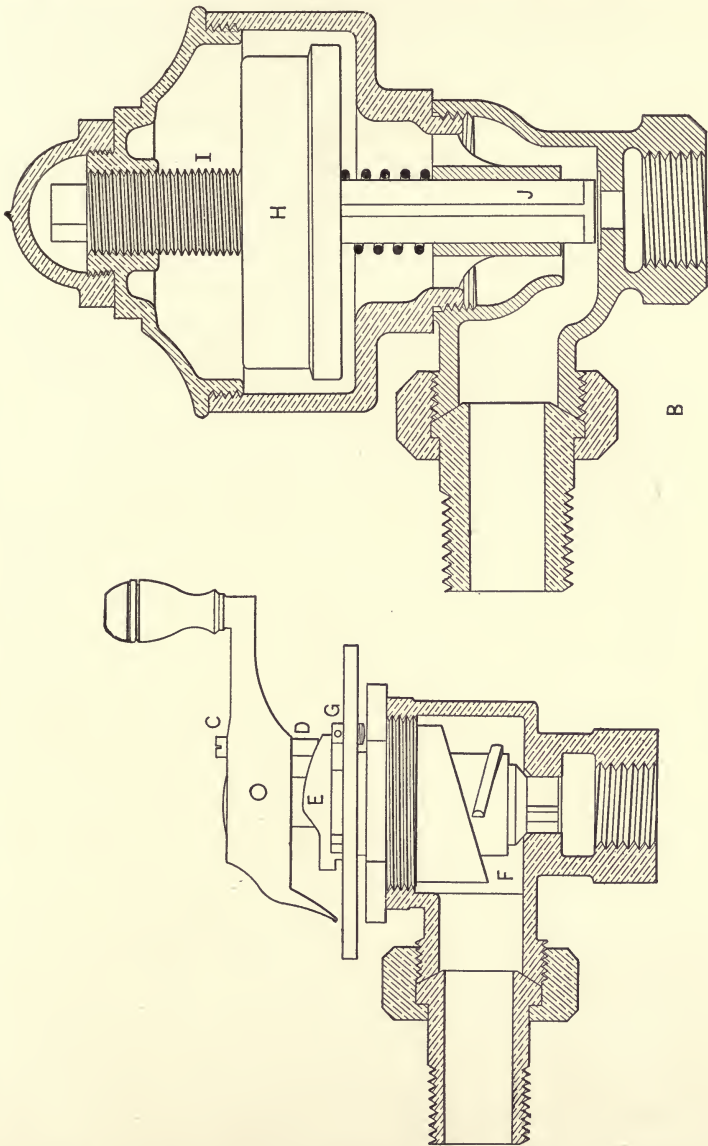


FIG. 79.—Thermograde Valves.

is used to adjust the cam which is attached on one part of its circumference, the remaining part being split.

The **auto-valve** is in reality a trap which will not allow steam to pass. It consists of a brass vessel *H* with a copper bottom which contains a liquid hydrocarbon such as gasoline. When this is cold the spring pressure lifts the valve, pressing in the copper bottom, which may be corrugated. The screw

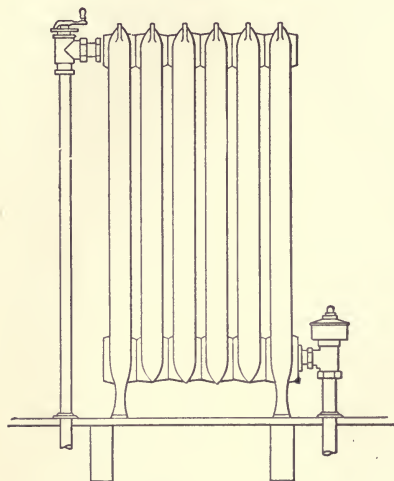


FIG. 80.—Thermograde Valves.

I adjusts the position of this so that *J* is lifted from its seat when this condition occurs, but when steam reaches the discharge, the heat volatilizes part of the liquid in *H* and produces sufficient pressure to push the valve stem down against the spring pressure closing off the valve. *I* may be adjusted so that this occurs at some definite temperature.

The two valves are shown in their position on a radiator in Fig. 80. In this installation hot-water radiators are used, as it is desired to have the steam supply the tops of all coils and blow out the air contained.

There is no necessity for air valves in such a system as this, and with proper adjustment of the modulation valve the admission of steam will heat about the amount of surface marked on the dial.

Fig. 81 illustrates the method used by Warren Webster & Co. to accomplish the same result.

The **modulation valve** *A* consists of a valve casing in which a disc *D* having four holes in it passes beneath a diaphragm *E* having one hole. The disc is moved by the handle *F* on the spindle. When the pointer on the handle is over the mark "1" on the dial a small hole in the disc registers with the hole in the diaphragm. This admits enough heat to warm part of the radiator surface. When the handle is placed at the other

marks, "2," "3" or "open," holes of the correct size come into the proper positions to admit more steam.

The motor discharge valve is placed on the return end of the

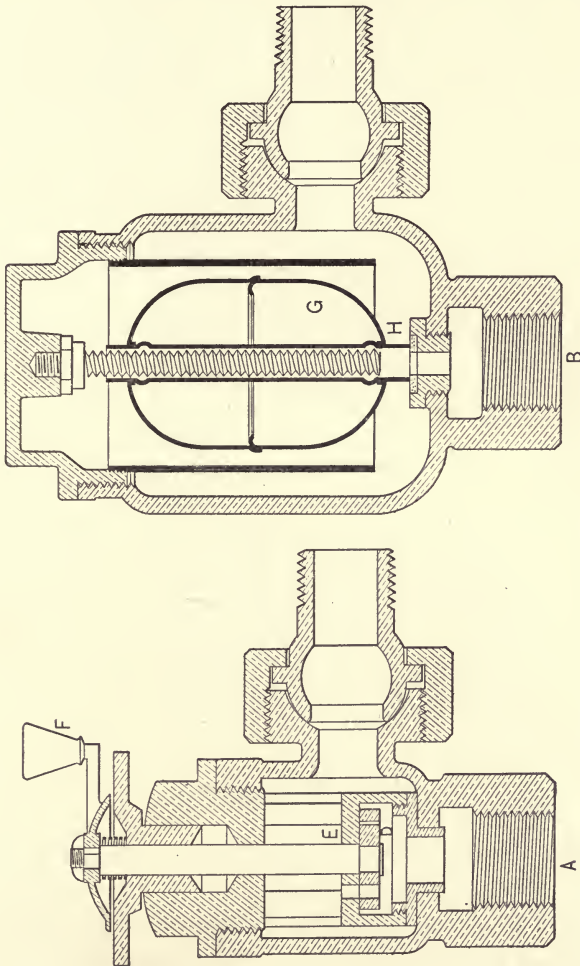


FIG. 81.—Warren Webster Valves.

radiator. Water collecting will cause the ball float *G* to rise. This float acts as a valve and the water escapes over the composition seat *H*. When the ball is seated air may be drawn from the radiator through the space between the tube in the ball

G and the threaded rod. As soon as steam begins to go over, this condenses in the threads and the passage of this condensate over the threads is very slow.

The Monash valve to be placed on the drip of the radiator is somewhat similar in action to the above valve.

As shown in Fig. 82 there is a water seal in this valve, and yet the air can be drawn around the center spindle by the

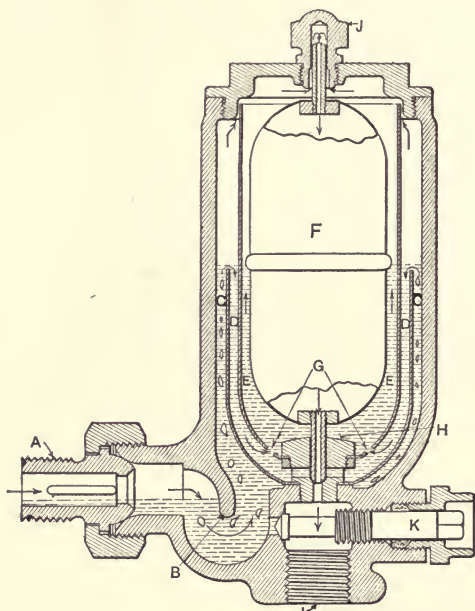


FIG. 82.—Monash Radifier.

vacuum pump. There are by-passes to drain and clean the valve. This valve is called a **radifier**. The advantage of these three return valves is that there is no pressure in the return pipe.

The fact that air may collect in most radiators, preventing the proper heating of the radiator and producing noise, makes it necessary to use **air valves** on many forms of radiators. These may be **automatic** or **hand controlled**. In most cases there is considerable trouble with the automatic form, as they are liable to get out of adjustment. Fig. 83 illustrates two forms of

compression air valves, one operated by a removable key, the other by a handle. These are nothing but small conical valves.

In Fig. 84, an expansion form of automatic air valve is shown.

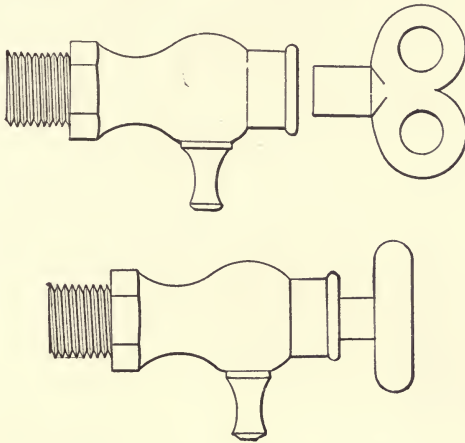


FIG. 83.—Compression Air Valves.

In this valve the core or cylinder has a coefficient of expansion so different from that of the body of the valve that when adjusted by the slot in its end so that air will just discharge when cold,

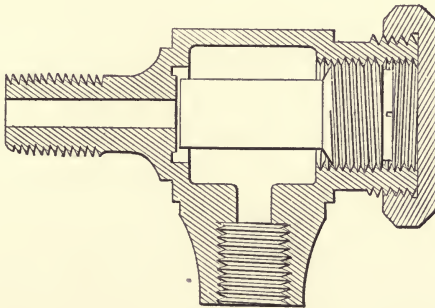
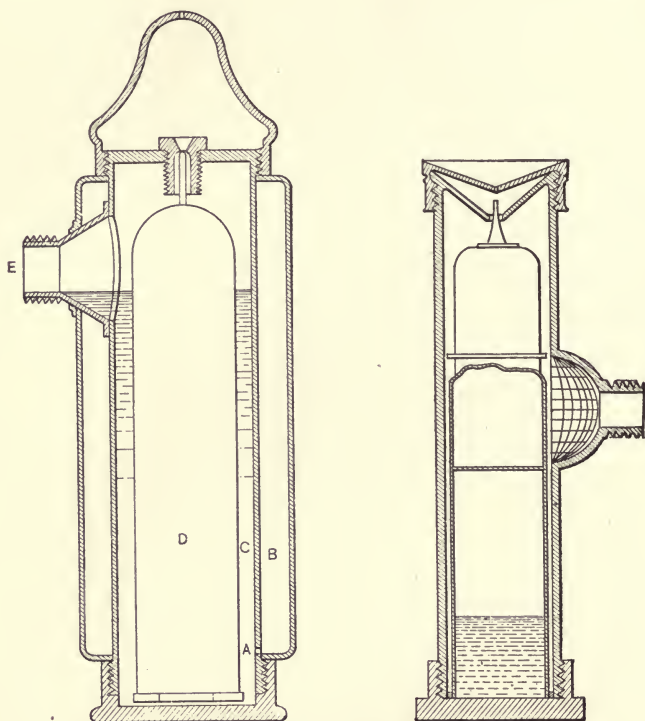


FIG. 84.—Expansion Air Valve.

the valve will be shut off as soon as hot water or steam strikes the valve. This form has been used for a considerable time.

The **Allen Automatic Air Valve**, Fig. 85, is formed of two chambers, *C* and *B*, connected by a hole *A* at the bottom.

The cylinder *D* is closed. When there is no water in *C* the cylinder *D* rests on the bottom of *C* and the pin at its upper end does not fill the hole. Air will then escape from the radiator which is connected at *E*. When steam reaches the valve, the condensation of it fills the bottom and floats the cylinder *D*,



Allen Air Valve.

Libra Automatic Air Valve.

FIG. 85.—Air Valves.

closing the hole. After this cools off the contraction of the air in *B* draws some of the water from *C* and allows *D* to fall, thus opening the air vent. The valve is kept hot by conduction from the radiator as long as the steam is turned on.

The valve shown in Fig. 85 is the **Libra Automatic Air Valve**. The inner cylinder is open at the bottom and when steam enters the valve, the air contained within the cylinder, expanding

from the heat, forces the water out and floats the cylinder, closing the valve. This water collects when the valve is first attached to the radiator.

The above valves permit air to enter the radiator after steam is shut off. This of course means that when the radiator is opened again, a certain time must elapse before the air is all removed and the radiator is completely heated. To prevent this action the **Norwall Automatic Syphon Air and Vacuum Valve** is used. This valve is similar

to those described above with the addition of a syphon *A* and the vacuum head *B*. The syphon *A* extends into the radiator and fills the lower part of the valve with water, if the radiator has liquid in it, before this reaches the level of the outlet. This water lifts the float and closes off the discharge until the water is drawn from the radiator. The vacuum head *B* contains a diaphragm of bronze containing a port which may be closed by a ball on the end of a small rod. The rod is supported by a yoke and two adjusting nuts. The length of the rod is such that

when the pressure below the diaphragm is slightly above atmospheric pressure the disc is raised, allowing air or vapor to pass out through holes in the head. As soon as the pressure is atmospheric the diaphragm rests on the ball, closing off the opening around the rod. The greater the vacuum below the diaphragm, the tighter the opening is closed.

The **syphon bellows** is applied to an air valve in Fig. 87*A*. The bellows has a volatile liquid within it which vaporizes when subject to heat from steam and closes the air valve. The float

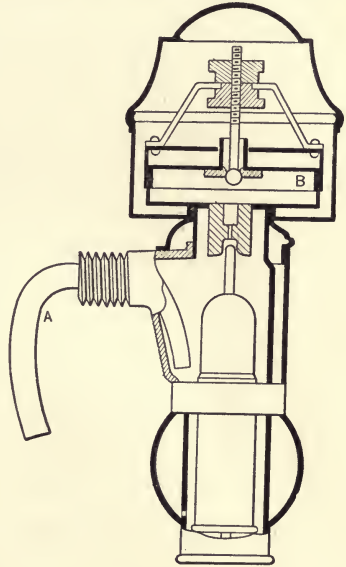


FIG. 86.—Combined Air and Vacuum Valve.

above is intended to close the vent if the radiator is flooded with water. The vent pin is self-guiding.

The **Sylphon Vacuum and Air Valve**. Fig. 87*B* has the closed bellows with the volatile liquid at the bottom and a float *D*, while above this, carrying the valve seat, is an empty bellows so made that the valve seat rests against the valve. When pressure from the radiator is slightly in excess of the atmosphere, the bellows moves upward from the pressure and the valve opens so that air can escape, the valve closing by the lower bellows or float as soon as this is heated by steam or lifted by

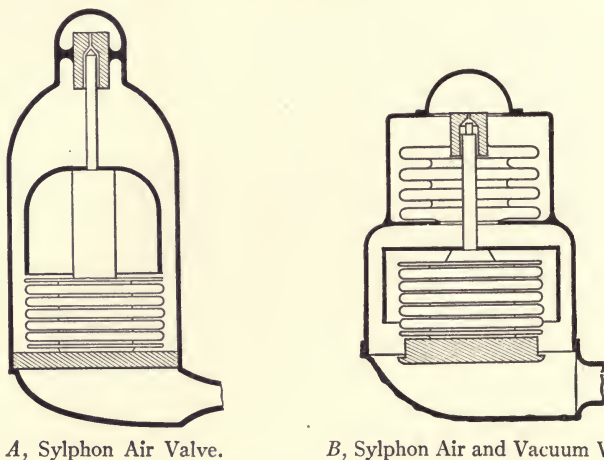


FIG. 87.—Sylphon Air Valves.

water. When a vacuum is formed in the radiator the valve is held closed by the excess of pressure on the outer part of the upper bellows.

For venting the whole system at the end of a return line the **Sylphon Vent Valve**, Fig. 88, is used. In this the inner sylphon is closed and filled with a fluid which operates to close the upper valve by pushing it upward against the seat and the outer sylphon operates to close the vent as soon as the pressure in the pipe line becomes equal or less than atmospheric pressure. The cylinder within the lower sylphon is to keep this from closing too much when cold.

Heat Transmission through Radiators. Radiators transmit heat by conduction to the outer surface of the wall; from this point it is transmitted by radiation to objects around and by convection to the air. The radiant energy emanates in all directions, and hence for the ordinary radiator the large part of the surface cannot radiate heat to the outside objects, as this heat

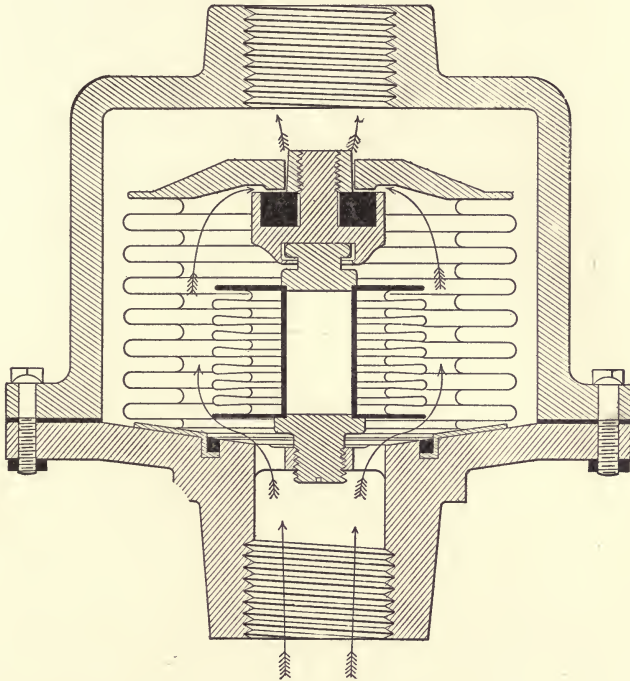


FIG. 88.—Sylphon Air Vent Valve.

is intercepted by the other sections of the radiator. Newton was one of the first to state a law for radiant energy, and he was followed by Dulong and Petit, Rosetti, Stefan, Weber, Bottomley, Paschen and Petavel, extending from 1690-1898. Most of these give radiation in the form $H = K(T_1^\alpha - T_2^\alpha)$ where K is the constant of radiation and T_1 and T_2 are the absolute temperatures of the hot body and the surrounding cold bodies respect-

ively. The values of α vary according to the different investigations,

- $\alpha = 1$, Newton;
- $\alpha = 3$, Rosetti;
- $\alpha = 4$, Stefan;
- $\alpha = 5.7$, Bottomley, Paschen;
- $\alpha = 5$, Petavel.

According to Dulong and Petit the expression is

$$H = KC^T,$$

and to Weber,

$$H = KTC^T.$$

Stefan, from experimental data, and Boltzmann from thermodynamic reasoning independently determined that the energy radiated from a black body is proportional to the fourth power of the absolute temperature, or

$$E = K(T^4 - T_o^4);$$

E = radiated energy per square foot per hour;

K = a constant;

T = absolute temperature of radiating body;

T_o = absolute temperature of receiving body.

In English measures $K = (16 \times 10^{-10})$.

This is known as the **Stefan-Boltzman radiation law**, and refers truly to a black body; but since the heat reflected and radiated from a body is equal (by the Stewart-Kirchhoff law) to that emitted from a black body, this rule may be applied if necessary to any body. It represents the best law of radiation.

Carpenter states that the larger part of the heat transmitted from a radiator is due to convection, and this is reasonable as pointed out before on account of interference. From experiments of Ser and others as mentioned by Dalby in London Engineering, Oct. 22, 1910, and from the discussion of this paper by Nicholson, as well as by his experiments and those performed by the makers of indirect heaters, it is evident that

the amount of heat removed from a radiator depends on the velocity of the air passing it.

The air seems to wipe the heat from the surface. There is apparently a film of air in contact with the heating surface which prevents the transmission of heat, as its conductivity is low, and so long as it remains in contact with the surface it cuts down the temperature difference between the air to be warmed and the conducting substance, which in this case is the air film. By causing the air to pass rapidly over the metal surface or to impinge on it, the thickness of the film is decreased, the true temperature difference increased, and the amount of heat transference is thus increased.

The data seem to show that the heat transference at any place is given by the forms:

$$H = f\sqrt{v}(t_s - t_a)A \quad (51)$$

or

$$H = (B + C\rho v)(t_s - t_a)A \quad (52)$$

- v = velocity in feet per second;
- t_s = temperature of steam or hot water in deg. F.;
- t_a = temperature of air in deg. F.;
- A = area in square feet;
- f, C, B , constants;
- ρ = density.

For a given velocity v , the quantity of heat given off by the area dA , is

$$dH = K_1(t_s - t)dA, \quad (53)$$

where K_1 is a constant, although by some it is considered to depend on t .

The quantity t will vary between the two limits t_1 and t_2 as the area varies between 0 and A .

$$H = K_1(\text{mean } \Delta t)A \quad (54)$$

Now

$$dH = Mc_p dt;$$

M = mass of air heated per hour;

c_p = specific heat of air;

$$H = Mc_p(t_2 - t_1)$$

(Variation in c_p not appreciable over this range.)

t_2 = temperature of air leaving radiator;

t_1 = temperature of air entering radiator.

Hence

$$Mc_p dt = K_1(t_s - t) dA. \quad \dots \quad (55)$$

$$Mc_p \frac{dt}{t_s - t} = K_1 dA$$

$$Mc_p \log_e \frac{t_s - t_1}{t_s - t_2} = K_1 A. \quad \dots \quad (56)$$

Now

$$Mc_p = \frac{H}{t_2 - t_1}.$$

Hence

$$K_1 A = \frac{H \log_e \frac{t_s - t_1}{t_s - t_2}}{t_2 - t_1}$$

and substituting in (54)

$$H = \frac{H \log_e \frac{t_s - t_1}{t_s - t_2}}{t_2 - t_1} \Delta t,$$

$$\Delta t = \frac{t_2 - t_1}{\log_e \frac{t_s - t_1}{t_s - t_2}} \dots \dots \dots (57)$$

or

$$H = K_1 \frac{(t_2 - t_1)}{\log_e \frac{t_s - t_1}{t_s - t_2}} A. \quad \dots \quad (58)$$

If the term involving velocity is introduced this becomes

$$H = K_1 \sqrt{v} \frac{t_2 - t_1}{\log_e \frac{t_s - t_1}{t_s - t_2}} A^* \dots \dots \dots (59)$$

Now in general the formula used for heat transmission is

$$H = K(t_s - t_r)A. \dots \dots \dots (60)$$

where t_r = temperature of the room or the mean temperature of the air in an indirect heater. Comparing this with (54) it is seen that K must be of the form

$$K = K_1 \sqrt{v} \frac{t_2 - t_1}{t_s - t_r} \frac{1}{\log_e \frac{t_s - t_1}{t_s - t_2}} \dots \dots \dots (61)$$

It is seen that K will vary with v , t_s , t_2 , t_1 and t_r . In case of direct radiation $t_r = t_1$, while in indirect radiation

$$t_r = \frac{t_2 + t_1}{2}.$$

* If the transmission depends on a power of the temperature difference the following results:

$$dH = \frac{K_1}{(t_s - t)^n} (t_s - t) dA = M c_p dt.$$

$$K_1 dA = M c_p (t_s - t)^{n-1} dt.$$

$$K_1 A = \frac{M c_p}{n} \left[(t_s - t_1)^n - (t_s - t_2)^n \right].$$

Now $H = M c_p (t_2 - t_1).$

Hence $H = \frac{K_1 A n (t_2 - t_1)}{(t_s - t_1)^n - (t_s - t_2)^n}.$

Now $H = \frac{K_1}{(\Delta t)^n} (\Delta t) A.$

Hence $\Delta t = \left[\frac{n(t_2 - t_1)}{(t_s - t_1)^n - (t_s - t_2)^n} \right]^{\frac{1}{1-n}}$

This holds for all values of n except for $n = 0$. This was the value used above.

Because of the complication of this formula, experimental determination of the quantity K is made without finding K_1 and computing K .

In performing the experiment the radiator is supplied with steam, the quality and pressure of which are determined by instruments, and the temperature and weight of the condensate is found. From the data H is given by

$$H = M(q + xr - q_o) = M(i - q_o),$$

where

M = weight of steam condensed per hour;

i = heat content of entering steam;

q_o = heat content of leaving steam.

Knowing A_1 , t_s and t_a , the quantity K may be found.

In direct radiators the velocity v is about the same in radiators of the same type, and so there will not be much difference in K for different conditions. But on account of the height and position of the radiating surface causing a difference in this quantity a change is expected in the value of K for differing radiators. The kind of surface would affect K , so that wrought iron, cast iron, painted radiators and bronzed radiators are all expected to give varying values of K .

The effect of radiation is present principally on the edges of the center sections and from the faces of the end sections, and as a result of this the value of K is greater for radiators of a few sections or coils than for those having a number of sections. The formula shows clearly that t_s , t_2 , t_1 and t_a affect K , and so in reality there should be a variation in this quantity for different steam, water and air conditions.

Carpenter quotes tests giving K from 1.23 to 1.97, while Rietschel gives values in his tables as low as 0.51 and as high as 2.65 with steam in a single pipe.

With a single section Denton and Jacobus found a constant as high as 2.39, while with a series of sections the value 1.97 was obtained. In another case 2.24 for a single section was reduced to 1.30 when 9 were used.

The condition of the surface of the radiator effects the amount of transmission. Experiments have been made by Carpenter and by Allen, and these seem to show that bright paints would decrease the efficiency of transmission while dark paints or even white lead increased the efficiency. In regard to bronzing, their results differed: Carpenter found an increase while Allen reports a decrease in efficiency. The number of coats of paint did not seem to effect this result, the outer coat being the determining factor.

The results of experiments show that the radiator of large surface, long or high, will give a smaller rate of transmission, K , than a small radiator.

Considering all results it will be well to take K as 1.75 *

Then $H = 1.75(222 - 70) = 265$ for low pressure steam;

$$H = 1.75 \left(\frac{170 + 150}{2} - 70 \right) = 160 \text{ for hot water.}$$

In the case of low pressure steam the pressure is assumed to be 3 lbs. gage, and for the hot water, the water is supposed to enter at 170° F. and leave at 150° F.

With indirect heaters the effect varies with the velocity. The usual method is to use the formula,

$$H = K \left(t_s - \frac{t_i + t_o}{2} \right).$$

According to Carpenter,

$$K = 2 + 1.3 \sqrt{v};$$

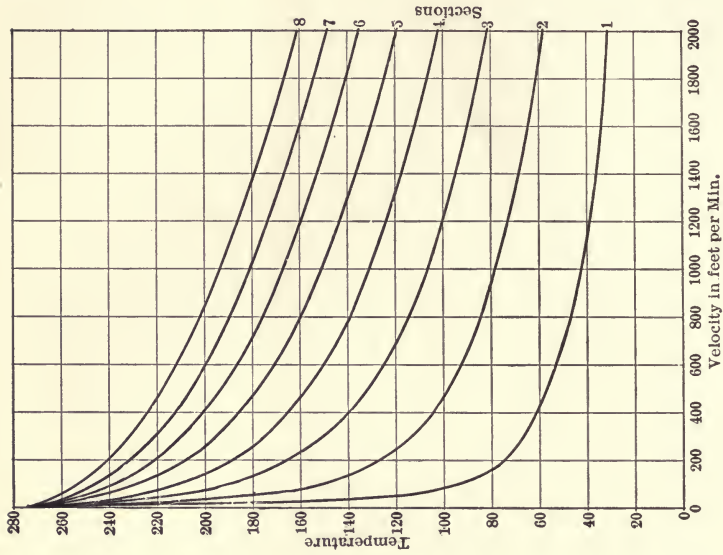
v = average velocity over coils in feet per second;

t_s = steam temperature;

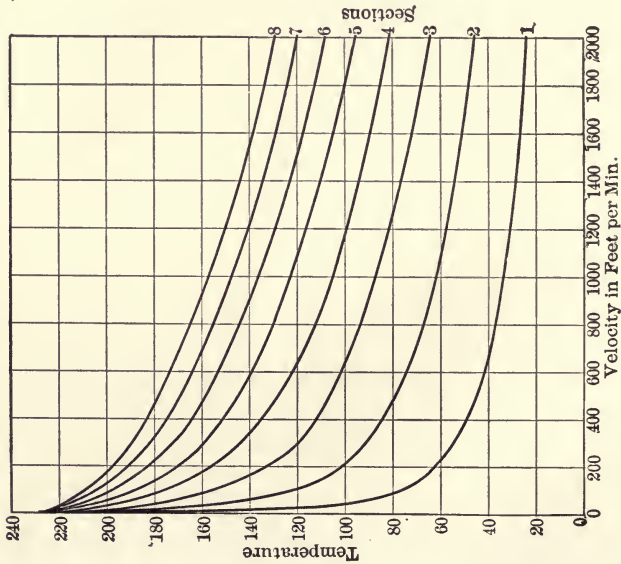
t_i = inlet temperature;

t_o = outlet temperature.

* Rietschel gives the following values of K , according to Kinealy: gas to air through clay plate 1.09, gas to air through cast or sheet iron 1.4 to 2.0, gas to water or reverse through cast or sheet iron 2.6 to 4.0, steam to air through cast or wrought iron 2.2 to 3.6, steam to water through metal 166 to 200.

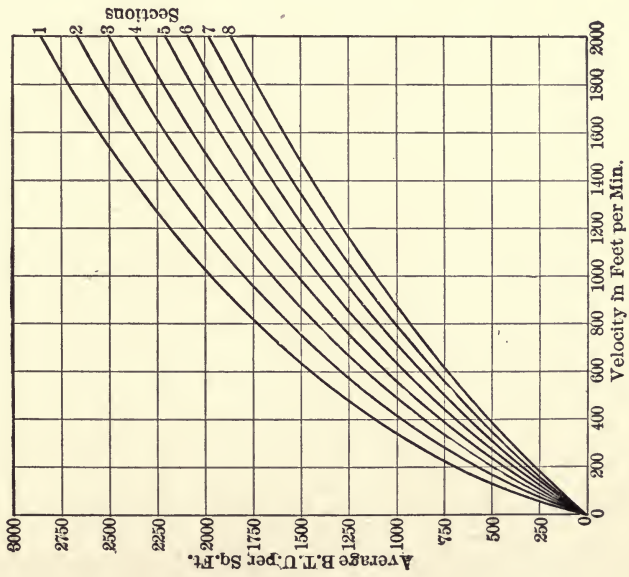
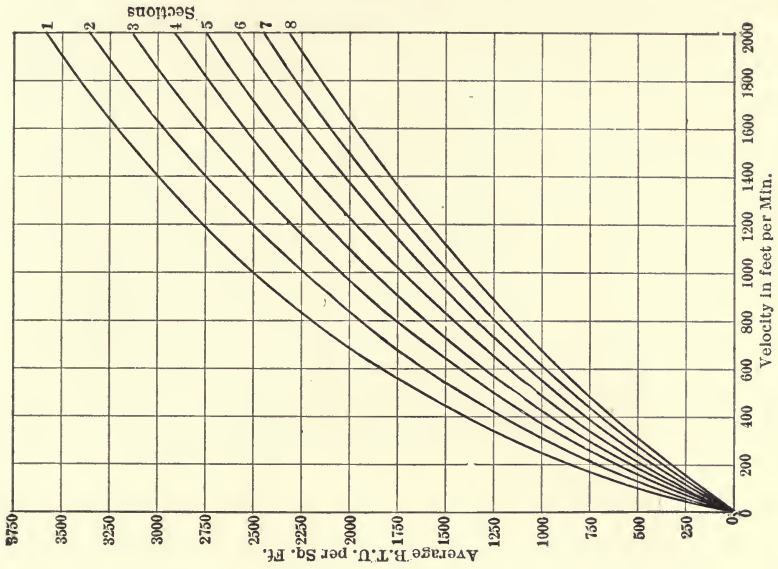


(a) Steam at 5 lbs. Gage Pressure.

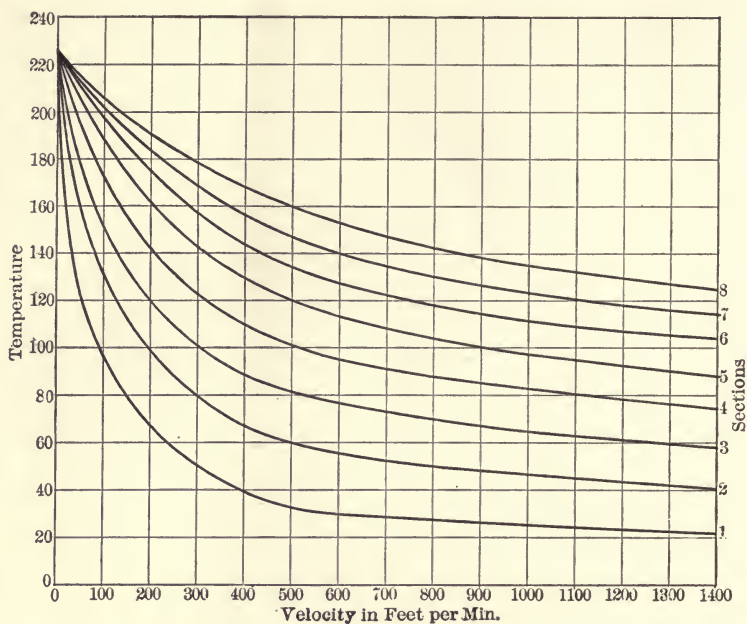


(b) Steam at 30 lbs. Steam Pressure.

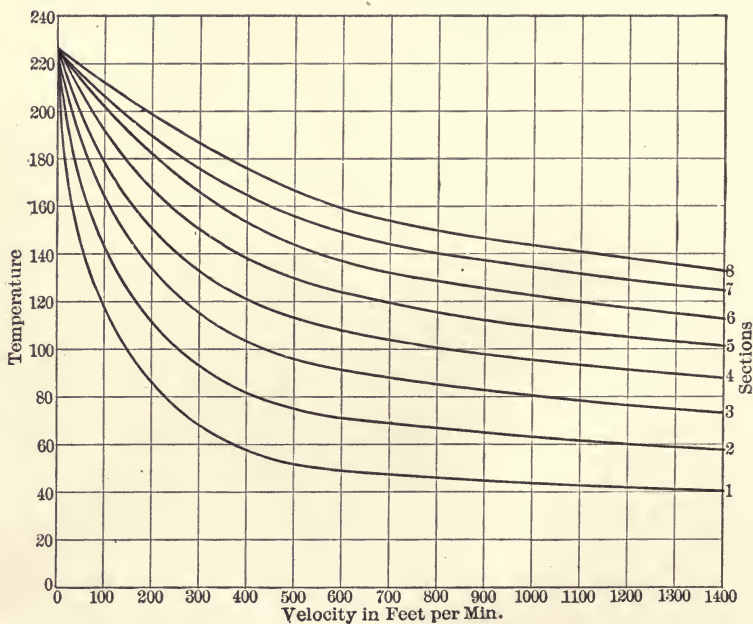
FIG. 89.—Temperature of Air Leaving Vento Heaters with Entering Air at 0° F. Regular Sections with 5" Centers. Velocity at 70°.



(a) Steam at 5 lbs. Gage Pressure. (b) Steam at 30 lbs. Gage Pressure.
 Fig. 90.—Heat Transmitted per Square Foot of Vento Heating Surface per Hour with Air Entering at 0° F. Regular Sections with 2" Centers. Velocity at 70° F.

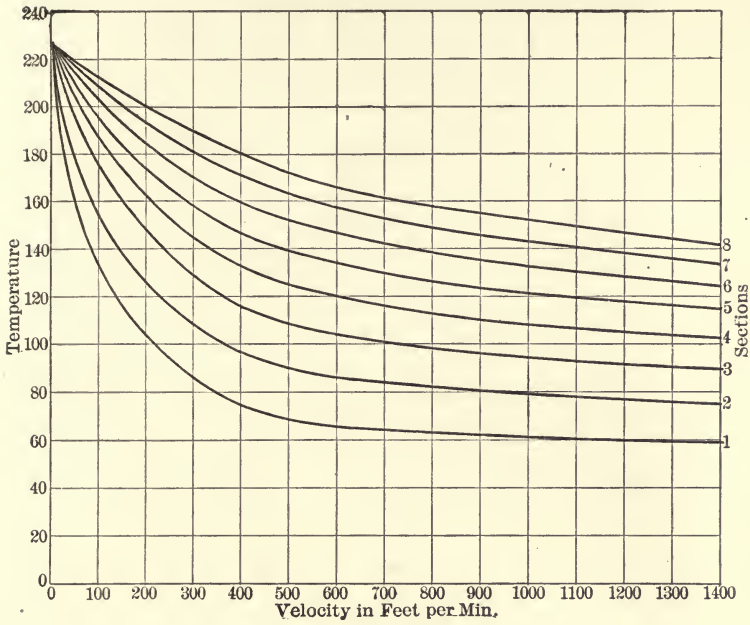


(a) Entering Air at 0° F.



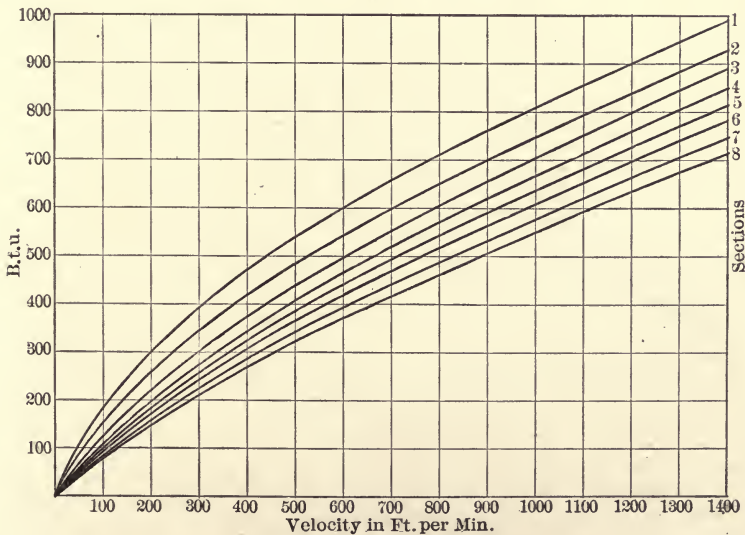
(b) Air Entering at 20° F.

FIG. 91.—Outlet Temperature from Buffalo Forge Heaters of Four Coils to the Section with Steam at 5 lbs. Gage Pressure.



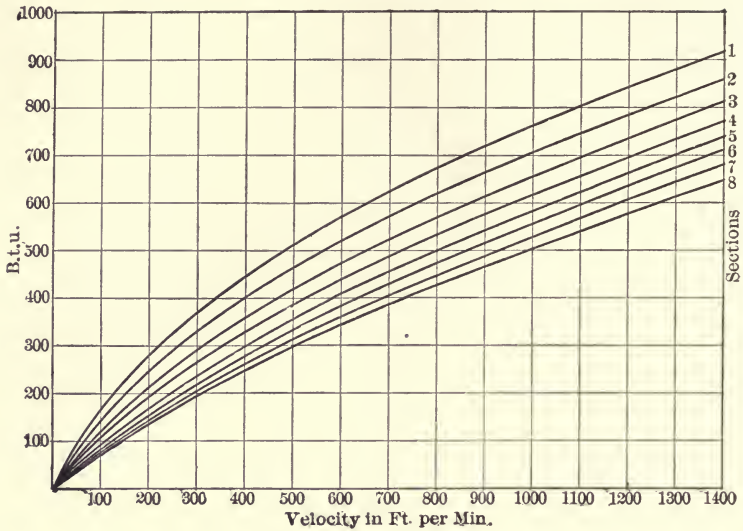
(c) Entering Air at 40° F.

FIG. 91.—Outlet Temperature from Buffalo Forge Heaters of Four Coils to the Section with Steam at 5 lbs. Gage Pressure.

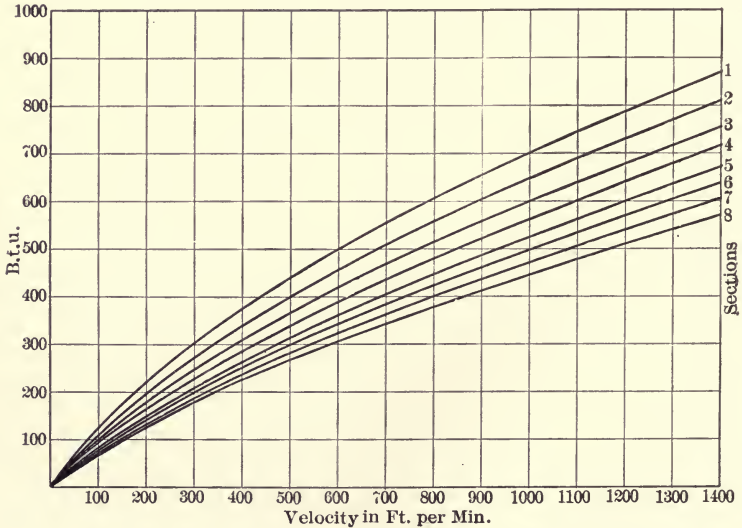


(a) Entering Air at 0° F.

FIG. 92.—Heat Transmitted per Lineal Foot of 1 Inch Pipe at Different Velocities with Various Sections of 4 rows of Pipe each, Steam being at 5 lbs. Gage pressure and Entering Air at 0° F., 20° F. and 40° F. Velocity at 70° F.



(b) Entering Air at 20° F.



(c) Entering Air at 40° F.

FIG. 92.—Heat Transmitted per Lineal Foot of 1 Inch Pipe at Different Velocities with Various Sections of 4 rows of Pipe each, Steam being at 5 lbs. Gage pressure and Entering Air at 0° F., 20° F., and 40° F. Velocity at 70° F. (Curves constructed from data published by Buffalo Forge Co.)

The Buffalo Forge Company give tables to be used with their heating coils of pipe, and the American Radiator Company give results of tests with their vento heater sections. These curves give the temperature increase for different velocities. These have been plotted as curves shown in Figs. 89, 90, 91, 92, which may be used for computing areas required for indirect radiators. These curves and the formula of Carpenter give for values of K the following:

Velocity in Ft. per M.	Carpenter K .	1 Section of 4 Coils.	3 Sections of 12 Coils.	1 Vento.	3 Vento.
600	6.1	7.1	7.7	7.1	6.6
800	6.7	9.6	9.2	8.3	7.9
1000	7.3	11.2	10.7	9.3	9.0
1200	7.8	12.4	11.8	10.4	10.0
1400	8.3	13.2	12.5	11.2	10.8
1600	8.6	13.7	13.0	11.8	11.5

These results indicate that a formula such as that of Carpenter does not hold as K is a function of the temperature drop along a set of surfaces, hence it is best to use a set of charts similar to those given.

CHAPTER V

METHODS OF CALCULATING HEAT REQUIRED FOR ROOMS

The methods used in practice for finding the heat loss in rooms varies according to the engineer. In any case a system

No.									
BDG.		ROOM			LOCATION		CEILING	FT.	
WALL	KIND	THICK	AREA	ΔT	C	EX. C.	HEAT LOSS		
CEIL.									
FLOOR									
W	SKETCH	N		TOTAL					
	E			VENTILATION					
				OCCUPANTS NO. HT.					
				AIR SUPPLY CU. FT.					
S	T. CHANGE HT.								
	FLUES								
	HEATING SURFACE								
	KIND	MAKE	SQ. FT.	ST.	D				
VOLUME									

FIG. 93.—Data Sheet.

of calculation should be used which will give results in a direct manner and one in which these may be readily checked. One method is to have ruled pages in a book with entries, as shown in Plates I, II. A certain part of the page is used for one room,

and the data for the room is taken from plans and entered. Another form used by the late Prof. H. W. Spangler is shown in Fig. 93. To fill out sheets in either method the plans of the building to be heated are first studied and the rooms are given **designating numbers**, and beside the numbers the desired or assumed mean temperature of the room is marked. The points

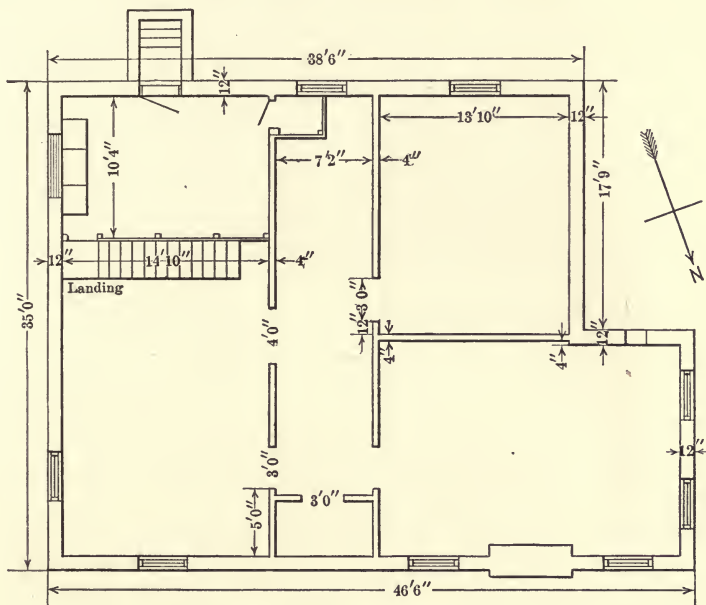


FIG. 94.—Cellar Plan.

of the compass are marked and then the data required on the sheets are filled out.

To illustrate the method of work the plans of a two-storied house, with the first story of brick and the second story of shingles, shown in Figs. 94 to 98, will be considered.

The temperature of the attic will be taken as 26° F. and of the cellar as 36° F. The ceilings as shown by the section are not high, and no allowance must be added to the temperature at head level to allow for excessive height of story.

The slanting roof is of shingles without **sheathing** or **paper** and the **deck roof** is sheathed with wood on which is a metal

PLATE I.

Heating for the Residence of L. Q. Smith, Rydal, N. Y.

1st Floor

No.	Sketch	Height Volume	Chgs. Occ.	Area		K		Factor	Heat		Rad.	Remarks	
				Total	Glass	Glass	Wall		Glass	Wall			Total
1		9'-0" 3110	2	N	198	28	170	0	0.96	0.24	1.15	2160	3280
				E	63	28	35	70	0.96	0.24	1.00	1880	590
				S	141	65	76	70	0.96	0.24	1.10	4800	1400
				F	345	34	34	0	0.96	0.23	1.00	2700	2700
				V	6200	0	0	70	0.018	0	1.00	16810	7810
2		9'-0" 2160	2	N	154	17	137	0	0.96	0.36	1.00	33	99
				E	126	34	92	70	0.96	0.24	1.00	2280	1540
				S	154	36	96	70	0.96	0.24	1.10	4140	1820
				F	234	34	34	0	0.96	0.23	1.00	1880	1880
				V	4300	0	0	70	0.018	0	1.00	11792	5410
3		9'-0" 400	2	N	63	12	51	0	0.96	0.36	1.00	860	860
				E	63	17	43	70	0.40	0.36	1.00	-77	300
				S	60	17	43	29	0.40	0.23	1.00	1745	1010
				F	45	0	0	0	0.018	0	1.00	1010	1010
				V	800	0	0	70	0.018	0	1.00	1745	1010
4		9'-0" 1390	2	N	92	20	72	0	0.96	0.24	1.00	1340	1210
				E	135	15	120	70	0.96	0.24	1.10	1100	2210
				S	154	23	85	70	0.96	0.24	1.10	1930	2120
				F	180	34	34	0	0.96	0.23	1.00	1090	1580
				V	2800	0	0	70	0.018	0	1.00	5890	3530
5		9'-0" 1620	2	N	135	25	110	70	0.96	0.24	1.15	1930	2120
				E	108	23	85	70	0.96	0.24	1.10	1700	1580
				S	180	34	34	0	0.96	0.23	1.00	1410	1410
				F	180	0	0	0	0.018	0	1.00	8740	4030
				V	3200	0	0	70	0.018	0	1.00	8740	4030
6		9'-0" 2110	2	N	63	26	37	70	0.96	0.24	1.15	2010	715
				E	126	52	41	70	0.96	0.24	1.10	3850	820
				S	234	34	34	0	0.96	0.23	1.00	1840	1400
				F	168	34	34	0	0.96	0.22	1.00	10635	7800
				V	6200	0	0	70	0.018	0	1.00	10635	7800

cover. The first floor is composed of ship lap over which **building paper** is placed and on top of this is a layer of tongued and grooved flooring. There is no plaster below. The second floor is similar to this with plaster on the lower side of the joists, while the attic floor is composed of **ship lap** only with plaster

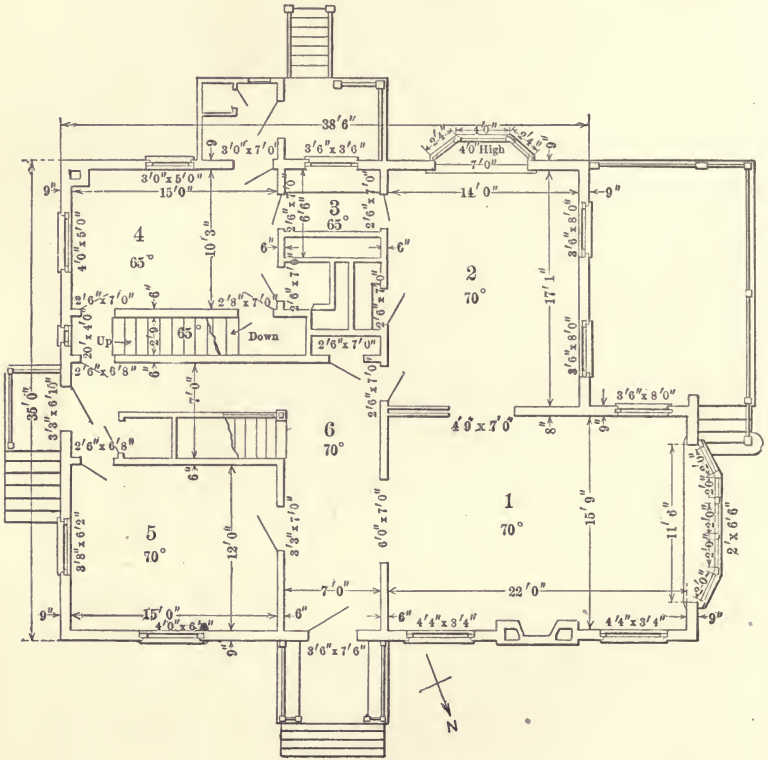


FIG. 95.—First Floor Plan.

below the joists. The values of K taken for these are 0.23 for the first floor and 0.22 for the attic floor or ceiling of the second floor, as the heated room is below, see page 65. The second floor is not considered as the temperature is the same on both sides; but had the second floor been 15° lower in temperature, the value of K used here for the ceiling of the first floor and floor of the second story would have been 0.16. If the second story

were 15° higher in temperature than the first floor, then the value of K would have been 0.03, as the hot room would be above.

The first floor walls are made of 8 ins. of brick work, a $\frac{5}{8}$ in. air space and 1 in. of lath and plaster, hence, from page 64, the value of K is 0.24. The second story is sheathed with ship lap on which shingles are placed over two layers of paper; the

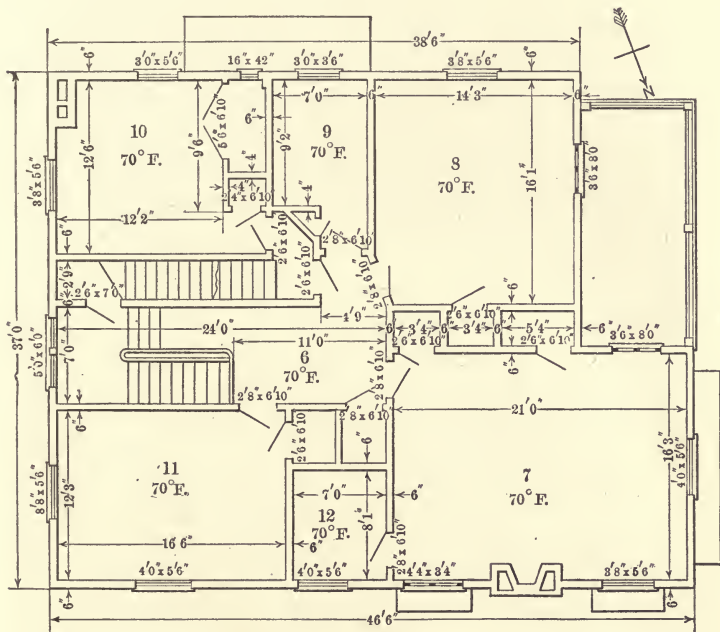


FIG. 96.—Second Floor Plan.

value of K is therefore 0.20. The glass has a value of K of 0.96. The doors on the outside and the **French windows** are considered as having the same values as glass.

The temperature of the attic could have been worked out by the method given by Eq. (39),

$$0.22(70 - t_a) = 0.4(t_a - 0^{\circ}).$$

In the equation above the value of K for the roof is taken

as 0.4; t_a is the temperature of the attic. In this case, the solution of the equation gives

$$t_a = 25^\circ \text{ F.}$$

With a thicker roof or one sheathed and with paper beneath



FIG. 97.—Section.

the shingles, the value of K would equal 0.3, and then the equation would give

$$t_a = 30^\circ \text{ F.}$$

The drawings of the building are now taken and the various rooms are marked with distinguishing numbers, and beneath these numbers the temperatures to be expected and the changes per hour if the ventilation is figured in this way. In numbering rooms in large buildings it is well to use combinations of numbers to designate floors. Thus all numbers beginning with 100,

as 105, 120, 132, etc., are found on the first floor, while 320, 318, 306 are found on the third floor.

The table is now filled in, the number of the room, then a rough sketch of the same, showing the form of the room, then

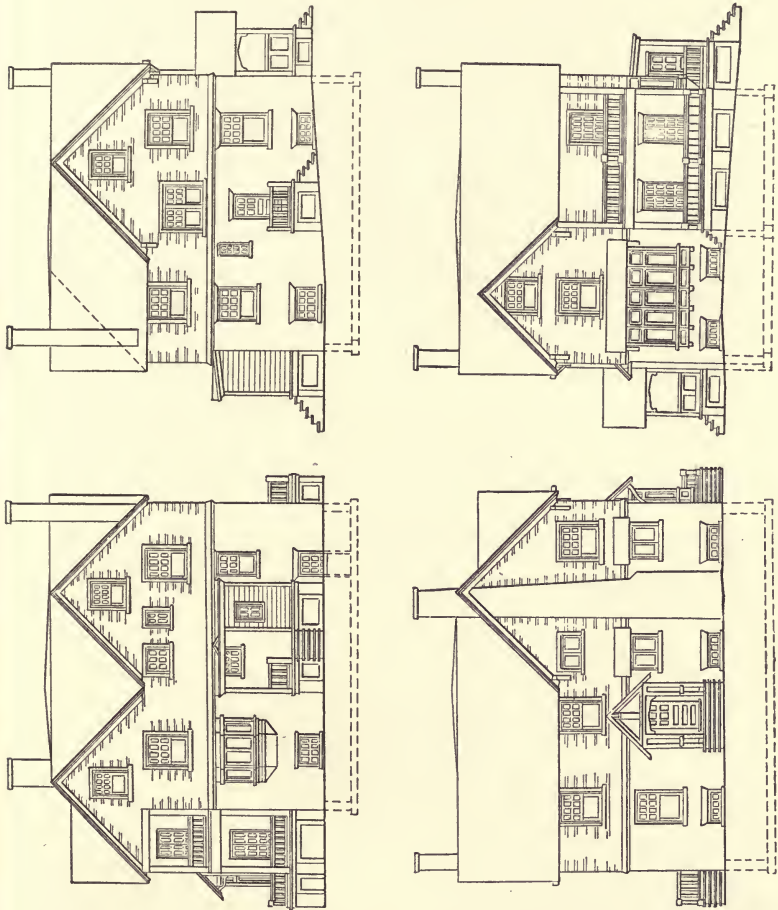


FIG. 98.—Elevations of Residence.

a column for heights and volumes and then a column giving the occupants or the number of changes per hour. The sketch is made with the north side to the top and the windows and doors designated by heavier lines. It is also well to mark temperatures on the various sides of the room.

The losses for the various sides of the room are now filled out with the floor and ceiling, giving the total area, glass or door area, net wall area, difference in temperature, value of the coefficient and finally the factor which is to care for exposure.

For high rooms the effect of increased temperature required would be shown in a greater temperature difference. The fact that increased radiation might be needed for intermittent heating would be shown in column headed, "Remarks."

By multiplying together the area, constant, temperature difference and factor, the amount of heat required for each side is found and entered in the heat column. The sum of all heats excepting that for the ventilating air is placed in the total heat column, with the ventilating heat as a separate item. These various columns may be added together by floors giving totals which are used in making checks.

In applying this method, although outside doors are taken as equal to windows, they in reality have a much lower constant, and for that reason on interior work or in any place not exposed to the wind, their true values, given in Chapter III, may be used.

The application of the table described above will vary with various systems and will be considered in the succeeding chapters.

CHAPTER VI

DIRECT STEAM HEATING

IN direct steam heating the steam is carried to the radiators on the different floors in several ways.

Fig. 99 illustrates the **single pipe system**. In this method of arranging piping, the steam flows from the boiler through

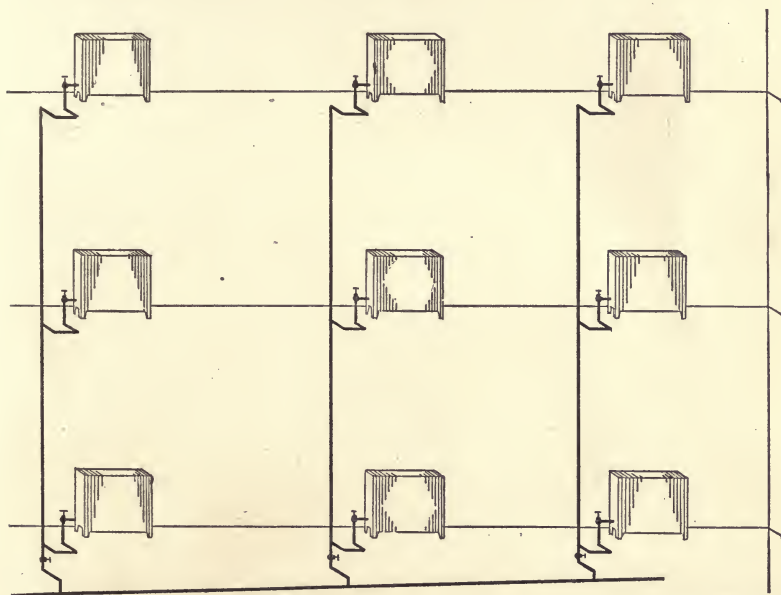


FIG. 99.—Single Pipe System.

the **main** and **risers** to the various radiators. After condensation the water falls back through the radiators and rises into the main. The main is carried around the basement pitching in the direction of flow until it drops below the **water line** of the boiler. In this system the radiators are each provided

with an **air valve** on the opposite side from the inlet. The system is not as positive as the **two-pipe system** of Fig. 100, but the pipes are fewer in number. Although larger pipes are required, the cost is less than the two-pipe system. This system, unless very carefully installed, is a noisy one, owing to the water hammer.

In the **double-pipe system** of Fig. 100 separate steam risers and **return risers** are connected in the basement to the steam main and the **return main** respectively.

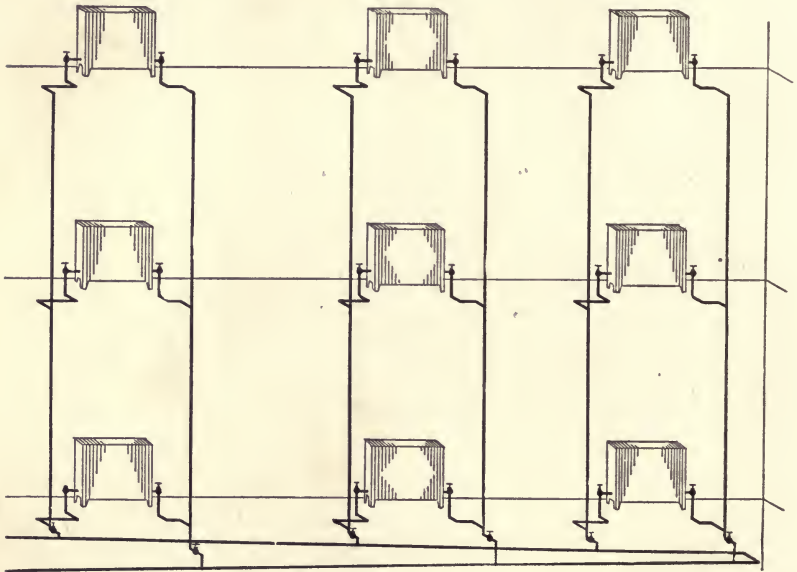


FIG. 100.—Two-pipe System.

The connections to the risers are made so that no **pockets** will form and each riser is controlled by a valve.

This system has the advantage that the condensation is cared for by a separate line and does not travel in the same pipe as the steam. This double flow in the same pipe may cause hammering when the system is started. Air valves are required to relieve the air which may be held in the top of a radiator, as, after a very short time, the supply and return pipes are filled with steam. This is the objection to this system

with ordinary valves. The steam in the return pipe will feed a radiator even after the supply valve is closed unless the return valve is shut also. The pressure in the return pipe causes a sluggish circulation at times. To overcome this such valves as the **auto valve** of the Thermograde Co., the **seal valve** of Warren Webster & Co., the **seal valve** of the Monash Co., or a similar one is used on the return. In this way there is no pressure in the return and the circulation is positive. Warren Webster & Co. use a **vacuum pump** on their return pipe in their system, thus carrying so low a pressure that the steam in the supply pipe is practically at atmospheric pressure. This is of advantage when the exhaust from engines is used, as the back pressure on the engine is not great and there is little pitch to the return pipe. Unless a valve of the type of these return valves were used it would not be possible to keep the pressure at such a low point in the return. This system is known as the **vacuum system**.

Another system of distributing the steam is that of the **overhead distributing main** or **Mills' system**, Fig. 101. In this the steam is carried through a riser *A* to the **distributing main** *B* in an attic or upper floor of a building. From this the risers *C* are taken off. The return risers *D* are carried downward and connect to the return main *F*. The condensation in the steam risers *C* is collected in the **drip line** *E*, although at times these drips are put into the return main or the condensation is cared for by the lowest radiator. These two latter methods are not good and that shown in Fig. 101 should be used in caring for the drip. In such a case the drip line is connected to the return through a steam trap.

For high buildings this system is often employed with the distributing main near the middle of the height of the building, steam being distributed up and down from that point.

The same system is used with a single riser system at times, the riser being connected to the distributing main above and the return main below. The drip is eliminated in this case and so are the return risers. This is sometimes called the **complete circuit system** and the **one pipe system**.

In many small steam installations the complete heating

system is connected to a boiler without the use of a pump, as shown in Fig. 2, and in such a case the drip of Fig. 101 is not necessary. The method just described, of a single riser system with an overhead distribution, is quite good, as the circulation is positive in the supplies, especially if the return is sealed by

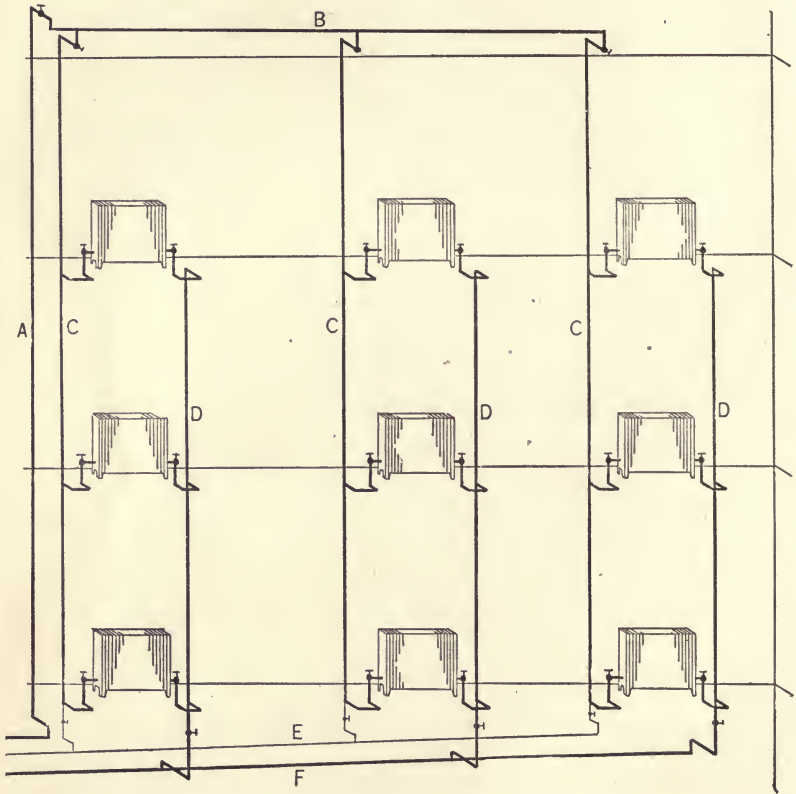


FIG. 101.—Mills System.

dropping the pipe below the water line by a **vertical leg**. In cases of central station heating, where returns from several buildings are delivered into a common return pipe, it is well to allow each building to discharge through a trap into the **main return pipe**. If this is not done one building may interfere with the proper heating of another. The return from one

building may prevent the air from being driven out of another one.

To find the size of radiators required for any room, the heat found in Plates I or II is divided by the heat transmission per square foot of heating surface as found in Chapter IV. A simple number to remember for low pressure steam heating is 250 B.t.u. per square foot per hour. Dividing the amount of heat in the case mentioned above by this, the amount of heating surface is found.

To check the results of computation it is well to divide the cubic capacity of the various rooms by the square feet of radiation in them, obtaining the cubic feet of space heated by one square foot of heating surface. This quantity varies with different kinds of construction and amount of window space, but the numbers serve to check any large errors after the designer has had experience. To guide the student the following table has been prepared from various sources:

CUBIC FEET OF VOLUME PER SQUARE FOOT OF
DIRECT RADIATION

Residences:

Living rooms.....	35 to 60
Sleeping rooms.....	50 to 80
Offices.....	40 to 80
Schools.....	40 to 80
Factories.....	75 to 100
Assembly halls.....	75 to 100
Hotels.....	75 to 100
Stores.....	75 to 100
Churches.....	125 to 200
Auditoriums.....	125 to 200
Gymnasiums.....	100
Work shops.....	150

From the table of Chapter IV, the numbers of sections of different forms of radiators required for these amounts of heating surface are found. The height of the radiator is fixed by the

condition in the building, such as heights of window sills, etc. The selections for the building computed in Chapter V are shown in the tables of that chapter. Radiators are usually placed in front of windows to cause a blanket of hot air in front of the window and to heat the leakage air. They should not extend above the sill.

Some authorities prefer to put them on a wall near a window, but not in front of it, claiming that in this way there is no down current of cold air to interfere with the up current of hot air. The cold air drops from the window, passes to the radiator, and there is a strong rising current aiding in the heat transference. There are two objections against this: valuable wall space for furniture is taken, and there will be cold drafts from the window.

The radiators being selected, the size of **outlets** should be fixed. These are standardized by the radiator companies and are given in the table below:

TAPPING FOR RADIATORS

Single Pipe.		Two Pipe Steam.			Hot Water.		
Sq.ft. Area.	Tapping.	Sq.ft. Area.	Supply.	Return.	Sq.ft. Area.	Supply.	Return.
24	1 in.	48	1	$\frac{3}{4}$	40	1	1
24-60	$1\frac{1}{4}$ "	48-96	$1\frac{1}{4}$	1	40-72	$1\frac{1}{4}$	$1\frac{1}{4}$
60-100	$1\frac{1}{2}$ "	96	$1\frac{1}{2}$	$1\frac{1}{4}$	72	$1\frac{1}{2}$	$1\frac{1}{2}$
100	2 "						

The **connections** from the radiator to the risers should be arranged so that as the riser expands and contracts, the radiator will not be lifted from the floor. The best way to arrange these branches is shown in Fig. 102, an **isometric drawing**. The isometric method of representing piping is very useful. In this the vertical lines represent vertical pipes; horizontal lines, those parallel to the plane of the drawing; and lines at 30° , lines perpendicular to the plane of the paper. Single lines are used to represent the pipes, and these may be solid or dot and dash lines. Thus, Fig. 104, represents four ways of connecting radi-

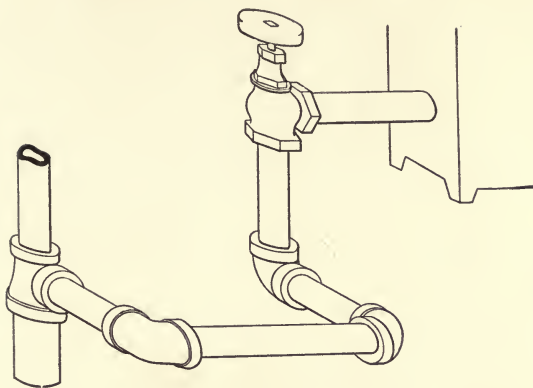


FIG. 102.—Branch Connection.

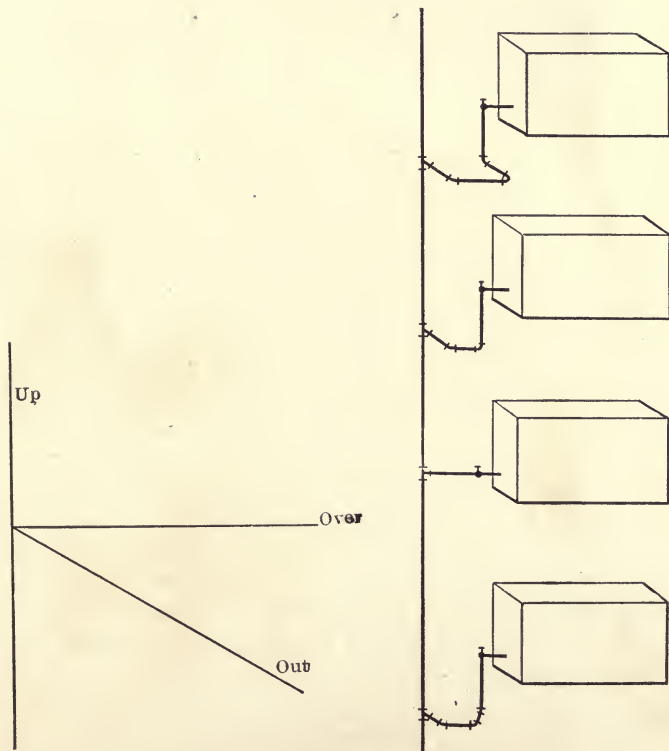


FIG. 103.—Isometric Directions.

FIG. 104.—Radiator Connections.

ators. The third method should never be used, as the expansion of the riser will lift the radiator and this tends to break the fittings at radiator if it does not actually break them. The second method is satisfactory if the branch is long enough to have some spring, and in the first or best method there is a chance for the branch to swivel on the ells and care for the expansion.



FIG. 105.—Development of House.

Risers are usually arranged so that a number of radiators come on the same line. This cuts down the number of risers and gives a cheaper job. To study the distribution properly and to give the heating contractor a knowledge of what is to be done it is well to make a small scale drawing, usually $\frac{1}{16}$ in. to the foot, showing the **development** of the walls of the building, with windows and partitions marked on it, and on this lay out

the radiators and risers. These views are all from the inside of the building, showing the wall as seen from the room. Fig. 105 shows such a development for the house figured in Chapter V. On this view the radiators are placed and each radiator and riser is given a designating mark. The radiators are marked with the number of the room followed by small letters *a*, *b*, *c*, *d* and *e*, if there are more than one.

Thus in room 5 there are two radiators, 5*a* and 5*b*. The risers are marked *A*, *B*, *C*, *D* and *E*, or by some other method, as I, II, III, IV, etc. The radiators and risers are also marked on the plans. Care must be taken in placing these to see that there is no interference with the placing of the furniture. Where possible the heating engineer should consult with his client about the placing of radiators.

The risers are usually proportioned by the number of square feet of radiation on them, and the table below gives the size for various amounts of radiation as recommended by Carpenter and others.

RISERS FOR VARIOUS HEIGHTS OF BUILDING

Radiator Surfaces in Sq. Ft.	Riser Size			Radiator Surfaces in Sq. Ft.	Riser Size		
	Low.	Medium.	High.		Low.	Medium.	High.
20	1	1 $\frac{1}{4}$	1 $\frac{1}{4}$	400	2	2 $\frac{1}{2}$	3
60	1 $\frac{1}{4}$	1 $\frac{1}{4}$	1 $\frac{1}{4}$	500	2 $\frac{1}{2}$	3	3
100	1 $\frac{1}{2}$	1 $\frac{1}{2}$	1 $\frac{1}{2}$	600	2 $\frac{1}{2}$	3	3 $\frac{1}{2}$
200	1 $\frac{1}{2}$	2	2	700	3	3	3 $\frac{1}{2}$
300	2	2	2	800	3	3 $\frac{1}{2}$	4

The risers should be controlled by valves, so that any radiator or branch may be fixed without shutting down the whole plant. The valves on the risers leading to a given radiator are closed when necessary for the repair of the radiator or connection. This can be quickly done and there is no interference with the radiators on other lines.

Where risers and vertical pipes of branches pass through floors or where horizontal pipes pass through partitions, floor and ceiling plates, Fig. 106, are placed around the pipes to close

the end of the hole in the plaster and give a neat finish. These are sometimes attached to the ends of the **sleeves** which are used to surround pipes which pass floors and partitions. The

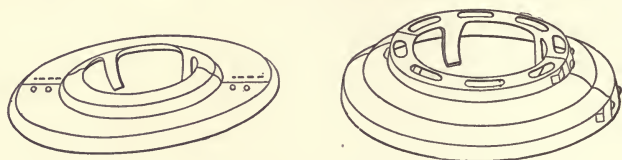


FIG. 106.—Floor and Ceiling Plates.

sleeves, Fig. 107, are made of cast iron or galvanized iron. They form an air space around the pipe and also ensure the pipe having a chance to expand and contract without breaking the plaster. By the use of an **air space** around the pipe the chance

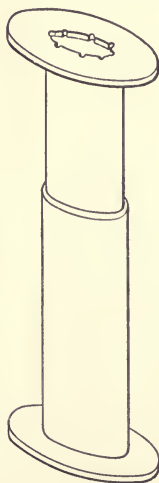


FIG. 107.—Floor Sleeve.

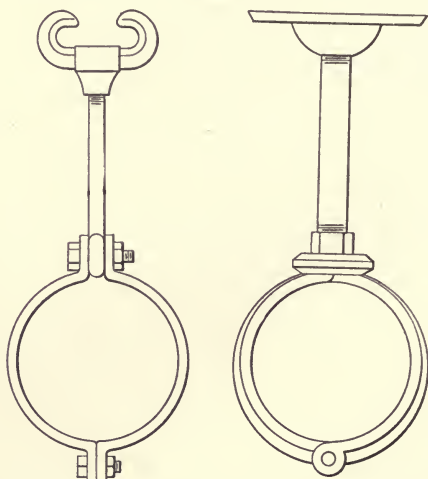


FIG. 108.—Pipe Hangers.

for charring timber construction, and the resulting danger of fire is removed.

The risers are supported by **pipe hangers**, Fig. 108, attached to a top horizontal branch of the riser or by **band anchors**, Fig. 109, attached around the vertical feeder. The anchor type is the better one to employ, as this permits one to support the riser

near the base or at the middle of its length. If at the middle the expansion occurs in each direction. If the end of a riser has any expansion the connection to the feed line or return must be made to allow for this. Of course, it is possible for pipes to bend sufficiently to permit of expansion if long enough, but there is danger of the fitting breaking, consequently it is well to arrange the connection from main to branch as in *A*, Fig. 110. If the end of the pipe is anchored the connection may be made as at *B*, Fig. 110. The connections *A* and *B*, Fig. 110, permit the main to move without danger of rupture, as the vertical elbows will allow the branch to swivel. At times 45°

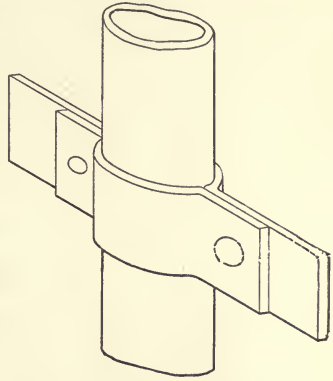


FIG. 109.—Clamp or Anchor.

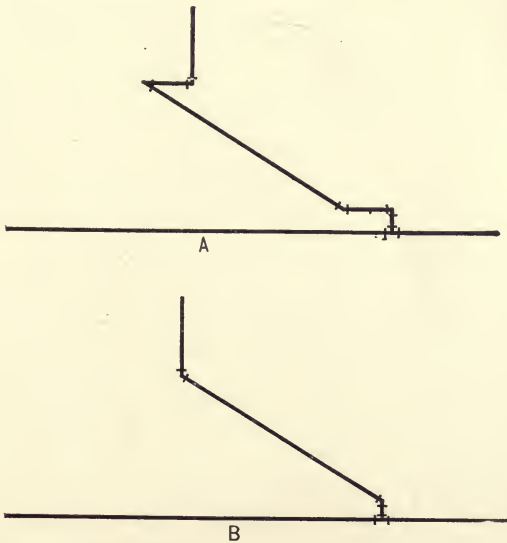


FIG. 110.—Branch Connections.

elbows are used to permit of expansion. Expansion in most heating systems can be cared for by **swinging ells**, and when

possible this should be done or expansion bends or corrugated pipes should be used. The **slip expansion joints** cause considerable trouble by leaking at the packing in the stuffing box.

The expansion to be allowed for various steam pressures is different. The table below gives the amount to be cared for in inches for each 100 ft. if the original pipe is at 0°, 30°, 60°, 70° when various steam pressures are used.

EXPANSION IN INCHES
PER 100 FEET

Original Temperature.	Gage Steam Pressure.					Water Temp.	
	150	100	40	10	5	212°	180°
0	2.84	2.63	2.23	1.87	1.77	1.65	1.40
30	2.61	2.40	2.00	1.63	1.54	1.41	1.17
60	2.38	2.16	1.76	1.40	1.31	1.18	0.93
70	2.30	2.08	1.69	1.32	1.23	1.10	0.81

The steam mains and return mains are designed in several ways. The first method is to use tables giving the amount of surface to be cared for by mains of various sizes. These tables are made for lines of a definite length, the usual length being 100 ft.

CAPACITY OF PIPES IN SQUARE FEET OF RADIATION FOR LENGTH OF 100 FEET, ACCORDING TO A. R. WOLF

Diam.	Radiation.		Diam.	Radiation.	
	2 lbs.	5 lbs.		2 lbs.	5 lbs.
1	36	60	4	1920	3200
1½	72	120	5	3720	6200
1½	120	200	6	6000	10000
2	280	480	8	12800	21600
2½	528	880	10	23200	39000
3	900	1500	12	37000	62000
			14	54000	92000

For any other length, multiply values in table by $\frac{10}{\sqrt{L}}$. For a single pipe system the pipe should be about 50 per cent larger

than the steam pipe for a double pipe system. Never use a smaller steam main than 1½ ins. or a smaller return than 1 in.

Another method is to determine the amount of steam required for a given amount of radiation, and after finding the volume of this steam, determine the area of the pipe to give a definite velocity. For large mains of 8 ins. or over 6000 ft. per minute may be used, while for smaller mains a velocity of 3000 ft. per minute. This may be written as a formula:

$$a = \frac{144Ahs}{60(H-q) \text{ vel.}} = \frac{2.4Ahs}{(H-q) \text{ vel.}} \dots \dots \dots (62)$$

- a* = area of pipe in square inches;
- A* = area of heating surface in square feet;
- h* = transmission constant for the heating surface in B.t.u. per square foot per hour;
- s* = volume of 1 lb. of steam in cubic feet;
- H* = heat content of steam in B.t.u. per pound;
- q* = heat of liquid of condensed steam in B.t.u. per pound;
- Vel. = vel. of steam in feet per minute.

In determining the diameter of the pipe, the table of actual areas on page 86 should be used rather than solving for the diameter of the circle of area *a*.

The best method of determining the area of pipe to carry a given quantity of steam is to assume the allowable drop in pressure in the given length, and then to use the method given in the Transactions of the American Society of Mechanical Engineers, Vol. XX, p. 342, by R. C. Carpenter and E. C. Sickles. The formula derived in this paper is

$$p = \frac{1}{20.663} K \left(1 + \frac{3.6}{d'} \right) \frac{W'^2 L}{D d'^5} \dots \dots \dots (63)$$

- p* = loss of pressure in pounds per square inch in length of *L* ft.;
- K* = constant = 0.0027;
- d'* = diameter of pipe in inches;
- D* = weight of 1 cubic foot of steam at given pressure;
- W'* = weight of steam in pounds per minute.

This may be written:

$$p = 0.000131 \left(1 + \frac{3.6}{d'} \right) \frac{W'^2 L}{D d'^5}, \dots \dots \dots (64)$$

or

$$d' = \sqrt{\frac{0.000131 \left(1 + \frac{3.6}{d'} \right) W'^2 L}{D p}} \dots \dots \dots (65)$$

In this the value of d' in the bracket is assumed for the first approximation, and then after substitution of this value within the radical a second approximation is found. The table given below has been prepared by the author for the weight of steam at 5 lbs. gage pressure which will be discharged for a $\frac{1}{4}$ lb. drop in the length given. If the pressure or drop is different from that given, the weight of steam at a pressure of 5 lbs. and a drop of $\frac{1}{4}$ lb. equivalent to the actual steam is given by

$$W_e = W_a \sqrt{\frac{0.049 \times \frac{1}{4}}{D \times p}} = \sqrt{\frac{0.012}{D \times p}} \dots \dots \dots (66)$$

To use lengths different from those given in the table, it is to be remembered that p varies as L and W' varies inversely as \sqrt{L} .

The paper of Carpenter and Sickles gives the loss in pressure in one globe valve to be equal to the loss in 700 diameters of the pipe, while an elbow gives a drop in pressure equal to that from 520 diameters of pipe.

To show the application of the table, suppose it is required to deliver 240 lbs. of steam per hour at 10 lbs. gage pressure through a pipe 45 ft. long, with 2 elbows and a gate valve with a drop of not over $\frac{1}{2}$ lb.

To find the equivalent steam per minute at 5 lbs. pressure and $\frac{1}{4}$ lb. drop, the formula (66) is used ($D = 0.0607$ for 10 lbs. steam).

$$W_e = \frac{240}{60} \sqrt{\frac{0.049 \times \frac{1}{4}}{0.0607 \times \frac{1}{2}}} = 2.54.$$

FLOW OF STEAM IN POUNDS PER MIN. AT 5 LBS. GAGE PRESSURE
WITH ¼-POUND DROP

Diam. in Inches	Length in Feet								
	25	50	75	100	150	200	300	500	1000
¾	0.51	0.36	0.29	0.25	0.21	0.18	0.15	0.11	0.08
1	1.00	0.72	0.58	0.51	0.41	0.36	0.29	0.23	0.16
1¼	2.46	1.61	1.31	1.14	0.93	0.81	0.66	0.51	0.36
1½	3.04	2.05	1.75	1.52	1.24	1.07	0.88	0.68	0.48
2	7.09	5.02	4.09	3.54	2.89	2.51	2.04	1.59	1.12
2½	11.70	8.33	6.79	5.89	4.80	4.16	3.39	2.64	1.86
3	21.40	15.10	12.30	11.40	8.72	7.56	6.18	4.78	3.38
4	45.60	32.20	26.30	22.80	18.60	16.10	13.10	10.10	7.20
5	84.80	60.00	48.80	42.40	34.60	30.00	24.40	19.00	13.40
6	138.00	97.60	79.60	69.00	56.30	48.80	39.80	30.90	21.80
7	202.00	144.00	118.00	102.00	83.00	72.00	58.80	45.60	32.20
8	290.00	205.00	167.00	145.00	118.00	103.00	84.00	65.00	45.90
10	525.00	371.00	302.00	262.00	214.00	224.00	151.00	117.00	83.00
12	885.00	627.00	511.00	443.00	362.00	314.00	256.00	198.00	140.00
14	1260.00	891.00	726.00	630.00	514.00	445.00	364.00	282.00	199.00
18	2370.00	1680.00	1370.00	1190.00	993.00	862.00	704.00	595.00	385.00
24	5270.00	3730.00	3410.00	2640.00	2150.00	1860.00	1520.00	1180.00	834.00

Assume as first approximation that $L = 50$ ft. Then from table $d' = 2$ ins.

Then
$$L = 45 + 2 \times \frac{520 \times 2}{12} = 225 \text{ ft.};$$

$$W_e \text{ for } 200 \text{ ft.} = 2.54 \sqrt{\frac{225}{200}} = 2.70;$$

d' from table, for second approximation = 2 ins.

This pipe will be of sufficient size.

To find the diameter by use of the table on page 138 it is necessary to change pounds of steam to square feet of heating surface. Roughly one quarter of a pound of steam is condensed per square foot of surface per hour.

$$\left(\frac{250 \text{ B.t.u. per square foot}}{1000 \text{ B.t.u. per pound}} \right).$$

240 lbs. of steam per hour will be consumed by 960 sq.ft. of

surface. From the table this requires a 3-in. pipe for the supply or a 4-in. pipe for a single pipe.

Allowing 3000 ft. per minute as the velocity, the following method is used:

$$240 \text{ lbs. of steam per hour} = \frac{240}{60 \times 0.0607} = 65.9 \text{ cu.ft. per minute;}$$

$$A = \frac{65.9 \times 144}{3000} = 3.16 \text{ sq.in.}$$

$$d' = 2 \text{ ins.}$$

In this manner the size of the supply main to various risers may be found. The pipe is reduced at various points.

To find the **size of the return**, an empirical method of using one-quarter of the area of the supply may be used until smaller sizes than 3-in. returns are found, when one-half the area may be used. In no case should a smaller size than 1 in. be used.

As another method the volume of the condensed steam may be determined, and from this after assuming a velocity the area may be found. **Chezy's formula** may be used to find v .

$$v = c\sqrt{rs}, \quad (67)$$

where

c = coefficient = 75 for iron pipe about 4 ins. in diameter;

v = velocity in feet per second;

$$r = \text{hydraulic radius in feet} = \frac{\text{area pipe}}{\text{wetted perimeter pipe}} = \frac{1}{48}d';$$

d' = diameter in inches;

s = slope of pipe in feet per foot;

v is about 2 ft. per second.

The condensed steam is accompanied by more or less air, so it is well to consider only one-half of the pipe as carrying water in the expression above. This gives the same hydraulic radius. The volume of water is much greater per pound at high temperature, and this must be considered in determining the area

to carry a given volume. The table below will be of assistance in this connection.

WEIGHT OF WATER AT DIFFERENT TEMPERATURES
PER CUBIC FOOT

Temp.	100° F.	130° F.	160° F.	190° F.	212° F.	220° F.
Pounds per cubic foot	61.9	61.5	61.0	60.4	59.8	59.4

In placing steam or return mains, care must be exercised

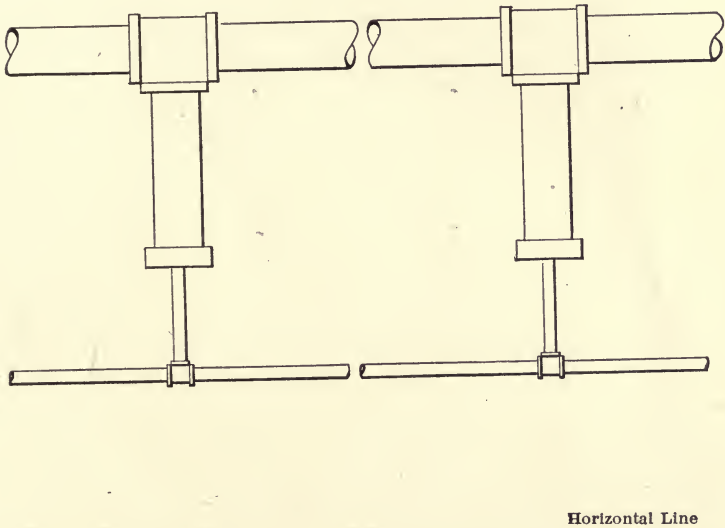


FIG. III.—Drip Pots.

to run the pipes with a decided **pitch** in one direction or the other. It is best where possible to pitch the pipe downward in the direction of flow at least 1 in. in 30 ft. Where this is not possible, and the pipe rises in the direction of flow, it is advisable to install **drip pots** at intervals, which are drained as shown in Fig. III. This enables one to cut down the amount of water which is flowing back against the steam current, and being in condition

to be taken up by the steam current whenever there is a change in the demand for steam, changing the relative velocity of the steam and water. The endeavor should be made to drain in the direction of flow, but where this would lead to complications a carefully designed line with the drainage in the opposite direction will give satisfactory results.

The branches should be arranged to care for condensation. The branches to returns or drips should enter the tops of mains. In power plant work it is customary to take the branches from the top of the main to insure drier steam. The branches from the main in heating systems should be taken from the top when the line is at the bottom of the riser, but in the Mills' system it is better to take it from the bottom of the pipe, thus dripping the main at each branch. An examination of Figs. 99, 100, 101 will show these arrangements.

Care must be exercised to have no portion of a main below the extensions from each end of it so that condensation will collect in this part and stop the flow. Such pockets may prevent the circulation of steam in a low pressure system and in any case they may produce water hammer. Pockets if necessary in any part of the system must be drained. With a vacuum system small pockets may be cared for automatically by the pump, causing sufficient vacuum to lift the water.

Valves are to be placed at high points in the line, globe valves being placed with the valve stems horizontal, while with gate valves the stems are vertical and the handle is placed above. These two positions prevent the water from collecting behind the valve.

To prevent the formation of a pocket when there is a reduction of diameter the **regular reducer** at *A*, Fig. 112, is replaced by an **offset reducer** *B*, or by an **eccentric tee** *C*, Fig. 112.

The risers may be run **exposed** in the various rooms or they may be **concealed** in **chases** behind the plaster. The former method is advisable, as repairs may be easily made. For high-class work where pipes would be unsightly the latter method is used. In this case the pipes and fittings should be care-

fully selected and the pipes should be tested before the plaster is put on.

The **radiator connections** are best carried beneath the ceiling in the room below, although in **concealed work** they are carried in the space between floor and ceiling or directly under the floor. Concealed work is undoubtedly the more attractive, but when trouble is experienced there must be considerable cutting before repairs can be made.

In many systems the **air valves** are connected to an **air line**

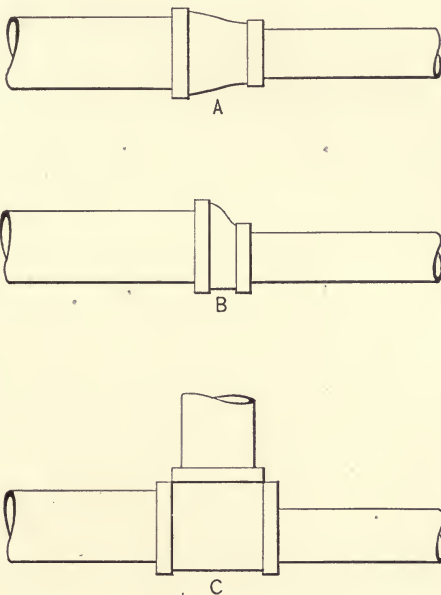


FIG. 112.—Regular and Offset or Eccentric Reducers.

which discharges some place in the basement, as shown in Fig. 113. The purpose of this was originally to deliver any drip of water into the sewer and not on the floor of the room. Lately it has been used in the **Paul system** for the attachment of a vacuum air pump to draw the air from the radiators, thus reducing back pressure in a single pipe system. The objection to an air line is the fact that an improperly set air valve may

deliver steam through this, thus interfering with the action of other valves and wasting steam unless the discharge end of the line is brought to a point where the attendant is sure to observe it.

Pipe covering is used on pipes which are put in chases or are carried through spaces which are to be kept cool. The materials used are to be good non-conductors and substances

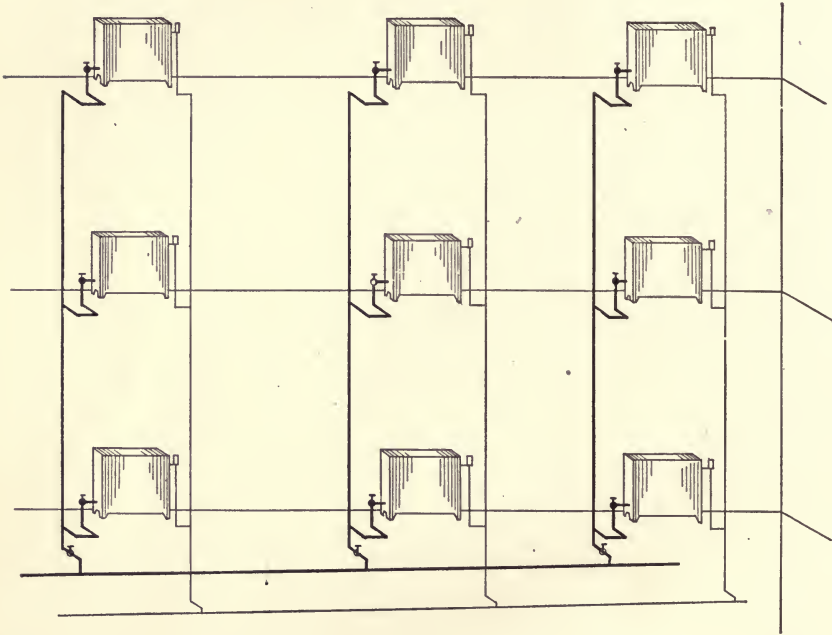


FIG. 113.—Air Line.

which will not burn or char. The substances used are **85 per cent magnesia, asbestos, hair felt, mineral wool and cork**. The heat loss is found as for any conductor by the formula $H = KA(t_s - t_a)$. Tests have been made by a number of persons to determine the values of these as heat insulators, and the results are similar. The table below gives the results of Geo. H. Barrus as reported in the Transactions of the Society of Mechanical Engineers.

LOSS OF HEAT PER SQUARE FOOT OF PIPE SURFACE PER DEGREE DIFFERENCE IN TEMPERATURE PER HOUR IN B.T.U.

10-in. pipe, 150 lbs. steam pressure.

Asbestos sponge felt, 76 laminations. . . (1 $\frac{3}{8}$ ")	0.280
“ “ “ 66 “ . . . (1 $\frac{3}{16}$ ")	0.306
Magnesia, 1 $\frac{3}{16}$ ins. thick.	0.354
Asbestos navy brand. (1 $\frac{3}{8}$ ")	0.387
Watson's Imperial, 1 in. thick (asbestos paper).	0.428
(Nonpareil cork.	0.290)
Bare pipe.	3.220

2 ins. pipe, 80 lbs. pressure.

Asbestocel 1 in. thick.	0.728
New York air cell.	0.750
Carey's moulded 1 in.	0.768
Asbesto sponge molded 1 in	0.778
Gast's air cell 1 in. thick.	0.793
(Nonpareil cork.	0.512)
Watson's Imperial.	0.548

These were reduced to same thickness of 1 in., and gave the following results:

10 in.—150 lbs.

Asbesto sponge, 66 laminations.	0.341
“ “ 76 “	0.342
Magnesia.	0.394
Watson imperial.	0.428
Asbestos navy board.	0.472

2 ins.—150 lbs.

Asbestos sponge hair felt, 3 ply.	0.497
“ “ “ “ 2 “	0.527
Asbestos sponge felt, 59 laminations.	0.527
“ “ “ 48 “	0.531
Magnesia.	0.531
Asbestos navy brand.	0.652

These coverings save about 80 to 90 per cent of the heat which would be radiated from the bare pipe. The covering will save its cost in less than half a year with steam at about 100 lbs. pressure. The kind of covering is an important item, as in many cases the more expensive covering will save much more than its original cost.

The coverings are usually prepared in **sectional form** and applied to the pipes in sections with **canvas covers**, as shown in Fig. 114, one section being an air cell covering, the other a solid covering. Each section is banded by at least two bands. At times **blocks of covering** are applied which are wire banded

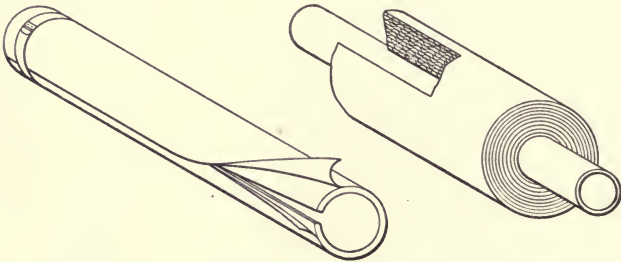


FIG. 114.—Pipe Covering.

and then a **hard plaster** is applied to the surface, making a good finish.

The covering costs about twenty to forty cents per square foot of pipe surface, depending on the diameter of the pipe, the smaller pipe costing more per square foot and will save from 90 cents to 140 cents per year of 8760 hours with coal at \$4.00 per long ton.

Using the tables of Chapter V, the various radiators for the rooms are found, and their positions determined from the plans and placed on the development of Fig. 105. After this is done the sizes of risers are determined by making the table as shown below, and then the sizes are marked on plans and development. The size of the supply is next found and marked on the plans.

RESIDENCE OF L. Q. SMITH

Riser.	A.	B.	C.	D.	E.
Radiators.....	1a. 52 7a. 48 — 100	1b. 52 7b. 48 — 100	2a. 36 8a. 48 — 84	2b. 36 — — 36	3a. $11\frac{2}{3}$ 9a. $16\frac{2}{3}$ — $28\frac{1}{3}$
Size.....	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$

Riser.	F.	G.	H.	I.	J.
Radiators.....	4a. $42\frac{2}{3}$ 10a. 20 — $62\frac{2}{3}$	10b. 20 — — . 20	5a. 32 11a. 20 — 52	5b. 32 11b. 20 — 52	6a. 75 12a. 14 — 89
Size.....	$1\frac{1}{2}$	1	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{2}$

A table is now made to insert in the specifications, giving information to the contractor in regard to the kind, size, and location of radiators.

FIRST FLOOR

Room.	Radiator.	Heating Surface.	Temp.	Ratio.
1	1a. 2 Column Peerless, 38 ins. 1b. " " 38 "	52 52 104 —	70° F.	30
2	2a. " 38 " 2b. " 38 "	36 36 72 —	70° F.	30
3	3a. " 23 "	— 11 $\frac{2}{3}$	70° F.	36
4	4a. " 26 "	— 42 $\frac{2}{3}$	70° F.	32
5	5a. " 20 " 5b. " 20 "	32 32 64 —	70° F.	26
6	6a. " 45 "	— 75	70° F.	30

SECOND FLOOR

Room.	Radiator.	Heating Surface.	Temp.	Ratio.
7	7a. 2 Column Peerless, 26 ins.	48	70° F.	30
	7b. " " "	48 96		
8	8a. " 38 "	48	70° F.	41
9	9a. " 32 "	16 ² / ₃	70° F.	45
10	10a. " 20 "	20	70° F.	42
	10b. " 20 "	20 40		
11	11a. " 20 "	20	70° F.	42
	11b. " 20 "	20 40		
12	12a. " 20 "	14	70° F.	34

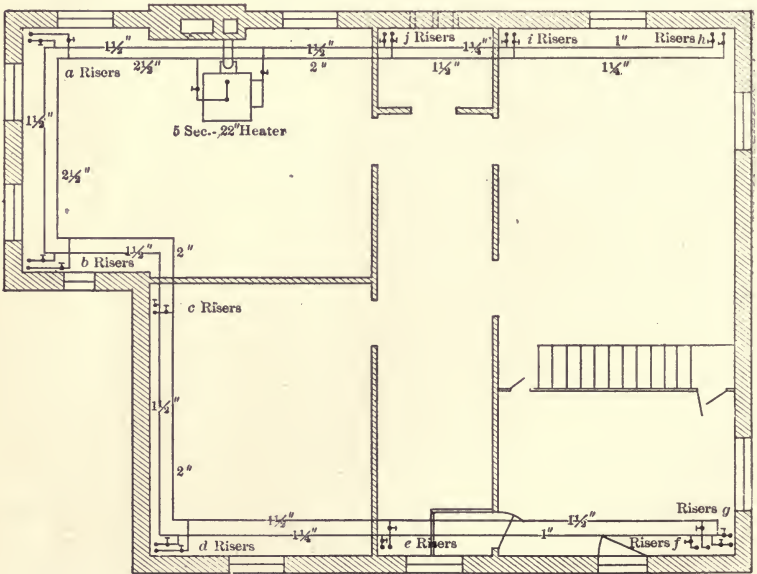


FIG. 115.—Cellar Plan of House for Direct Heating.

With the table in the specifications and the development there is no ambiguity or chance for questions to arise in the completion of the work. Each bidder knows what is wanted. At this point it is well to call attention to the fact that the heating engineer should know that what he has designed is sufficient

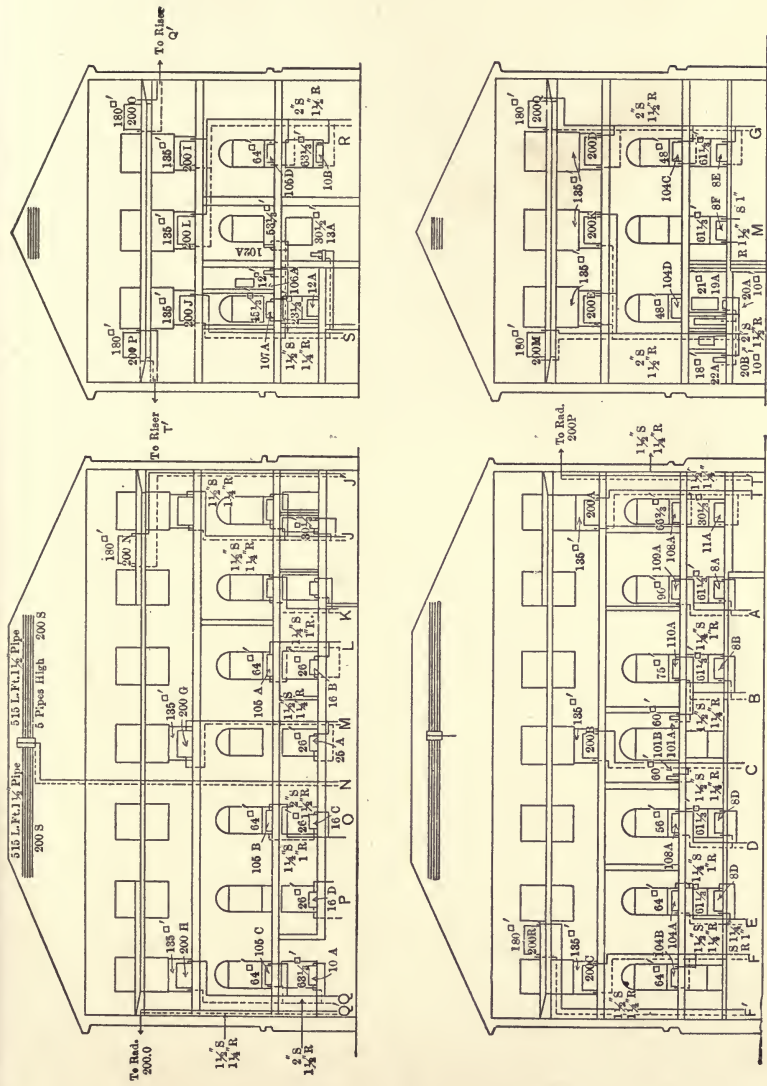


FIG. 116.—Development of the 87 Gymnasium.

for the work and not to try to shift responsibility by placing a clause in the specification that the contractor has to guarantee to heat the building to 70° in zero weather. If this is not accomplished after specifying the amount of radiation it is clearly the fault of the engineer and not of the contractor, and the engineer should assume the responsibility.

The cellar plan in Fig. 115 gives the arrangement of supply and return pipes in the cellar with the position of the boiler and flue.

Fig. 116 illustrates a development of a larger building, showing the method used to fix the sizes of risers and mains.

CHAPTER VII

HOT-WATER HEATING

IN hot-water heating there are several methods of arranging the pipes. In one system there is a **single-flow pipe or main**, Fig. 117, in the basement from which **supply and return risers** are run to the different radiators. The **flow pipe** leaves the

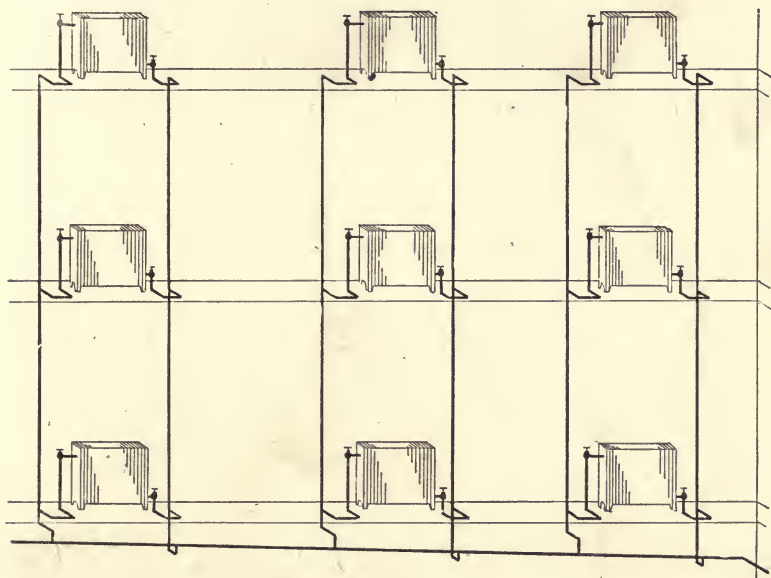


FIG. 117.—Hot Water with Single Main.

top of the water boiler and re-enters at the bottom after making its completed circuit. The radiators used in hot-water installations are usually made with the sections connected together at top and bottom as was mentioned on page 75, where the hot-water form of radiator was shown in Fig. 46. In a single-flow pipe system the water is gradually cooled as it passes through

the pipes owing to the cool return water being added. To aid the circulation and to keep the cold water from mixing too rapidly with the warm water, the branches to the supply risers are taken off from the top of the pipe, while the return is connected to the bottom of the pipe. For this purpose **special tees** are made and installed as shown in Fig. 118. The supply, being taken from the top of the flow pipe first and then the

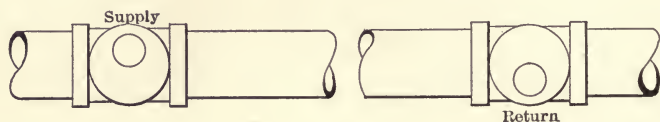


FIG. 118.—Eccentric or Offset Tees for Hot Water Main.

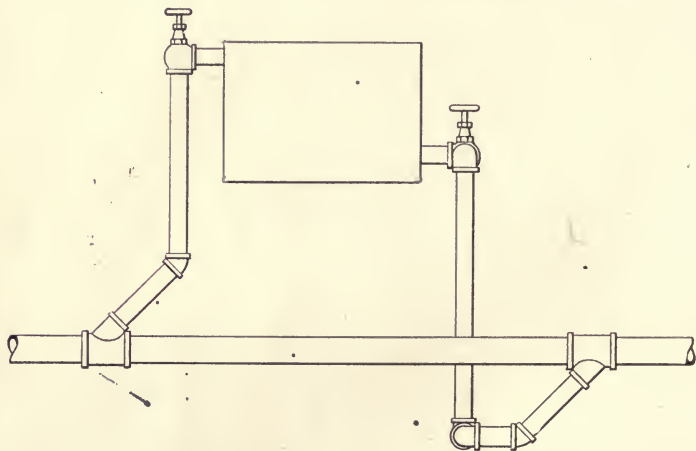


FIG. 119.—Flow Line Using Y's.

return from the bottom. Fig. 119 shows a method of using Y's to accomplish a more positive circulation.

The **separate supply flow pipe and return flow pipe** are shown in Fig. 120. In this the two pipes both rise as they leave the boiler, the supply from the high part of the boiler and the return, through a vertical leg from the bottom of the boiler. In this case there is no danger of getting the currents mixed and the branches may be taken from any part of the lines. In the figure

the risers which are connected to the part of the supply near the boiler are at the far end of the return flow line. In this

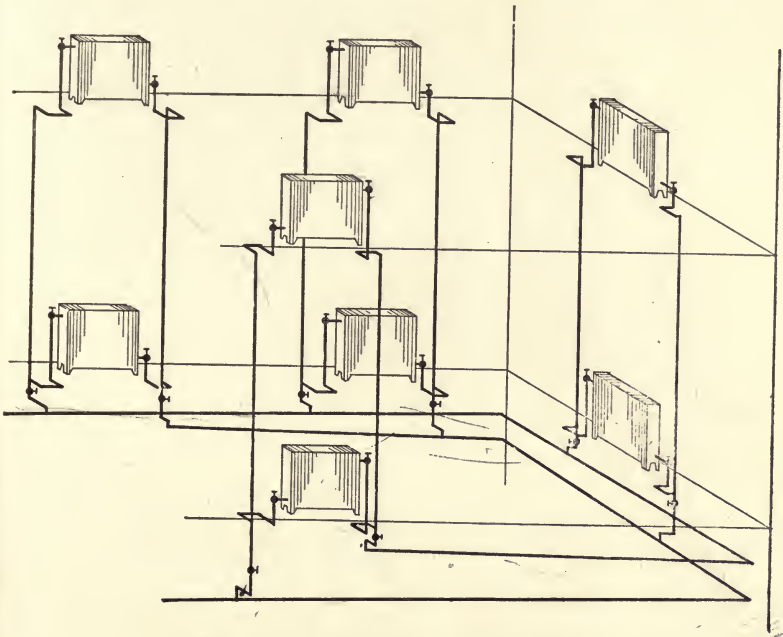


FIG. 120.—Two-pipe Hot Water System.

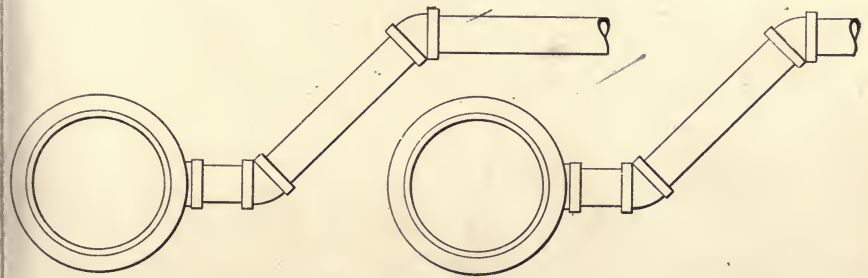


Fig. 121.—Branch Connections.

way the length of the various circuits may be equalized. Fig. 121 shows one method of taking off branches from the **flow lines** when they are placed side by side, and there is not sufficient

room beneath the joists to have one branch cross the flow pipe and to use an elbow on a tee pointing straight up with a close nipple.

One fitting may be used as shown in Fig. 122 if the tee is

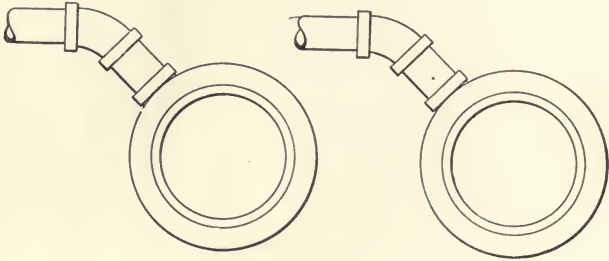


FIG. 122.—Branch Connections.

turned with the branch at 45° to the horizontal instead of in the horizontal position. In many cases there is not sufficient room to turn the tee vertically as shown in Fig. 123, and the methods of Figs. 121 and 122 are resorted to. This applies equally well to all forms of piping work for steam or water.

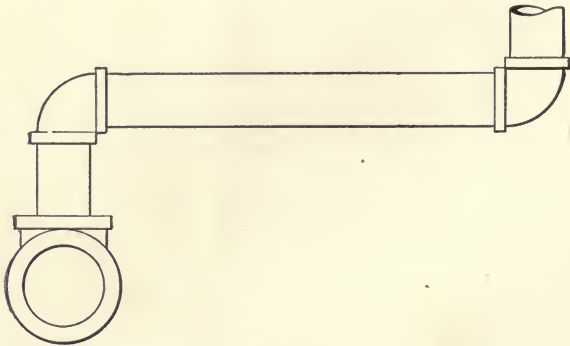


FIG. 123.—Branch Connections.

The branch is sometimes taken below the flow line as shown in Fig. 124, when it is desired to have sluggish action, although air may collect in such a branch. The Honeywell Company advise the use of connections of Fig. 121, where a reduction

occurs in the size of the main, using only a branch to a first-floor radiator. They never connect a high riser branch at such a point, as the circulation in such a riser at a reduction of section might cause excessive circulation in the section supplied by the riser. They recommend connections as shown in Fig. 124 for all branches near boilers so as to cut down circulation at these points. Drips must be provided for draining the pockets formed in this branch when the system is being drained. If this is objectionable the branch shown in Fig. 121 may be used. In any case the endeavor must be made to have the circulation good in all radiators.

The complete circuit system, Fig. 101, may be used for hot

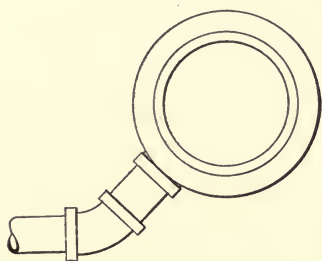


FIG. 124.—Branch Connection.

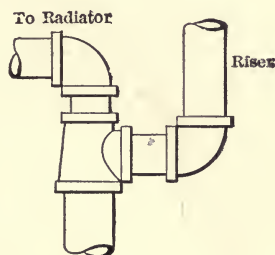


FIG. 125.—Riser.

water as well as steam. In this the supply-flow pipe at the top of the building is fed from the top of the boiler through a riser and the return-flow pipe is placed at the bottom of the building. Of course in this case there are supply risers connected to the top of the radiators on one side while the return risers are connected to the lower part of the other side of the radiator.

The risers are connected to the branches and in most hot-water installations there are no control valves on the branches to risers.

If there are a number of radiators on a riser there is some danger of the circulation being established in the riser to an upper radiator and thus prevent a proper supply from reaching the lower radiators. There are several methods of avoiding this. One method, Fig. 125, is to place the supply to the radiator at the

end of a section of the riser and continue the riser by means of a tee and an elbow. In this way there is resistance to upward flow due to the breaking of the direct path. In Fig. 125, if the radiator is assumed to the right and the continuation of the riser is assumed to the left the same result is accomplished. A third method is to reduce the riser diameter at the place where a connection is taken off to a radiator, the constriction throwing more resistance on the flow and thereby giving the lower radiator a supply of water. Fig. 125 shows the radiator connection leading at right angles from the riser without a swing ell. This can be done in the case of hot-water work if the lines are not

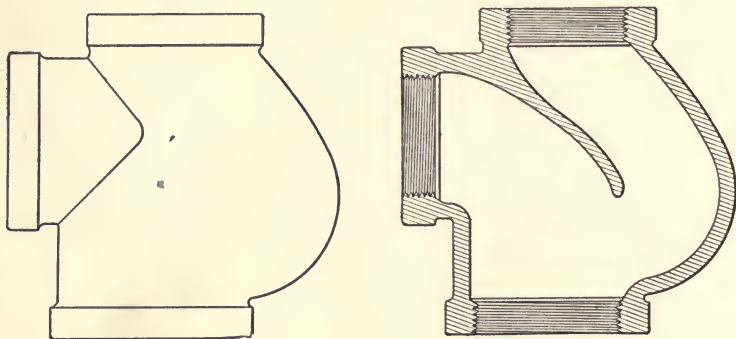


FIG. 126.—O. S. Connector.

too long. The table in Chapter VI gives the expansion of 180° heating to be less than 1 in. to 100 ft.

One of the best methods of causing water to flow into a lower radiator is to use the **O. S. distributors**, Fig. 126. These are special tees with a deflecting partition and in most cases a reduction in size of piping on the run.

When necessary to hug the wall of a building the branch may be taken off at an angle to the wall instead of parallel to it and by the use of a 45° ell, as shown in Fig. 127, the line is brought parallel to the wall.

To find the amount of radiation for a hot-water system, the same method is used as for the steam system. The amount of heat for a given room from Chapter V is divided by the amount

of heat transmitted per square foot of radiation for hot water and the result will give the amount of radiation. This number is about 170 B.t.u. per square foot per hour. Another result which is worth remembering is that 1 sq.ft. of hot water radiation requires 1 gallon of water or about $\frac{1}{8}$ of a cubic foot or 8 lbs. of water per hour, as the drop in temperature is about 20° F.

After the amount of radiation is computed it is placed on plans and developments as in Chapter VI and then the pipe sizes are found. To check the results of the amount of heating surface the following table is given:

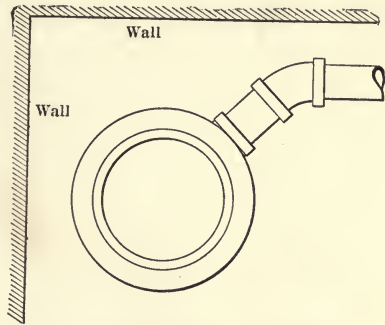


FIG. 127.—Radiator Connection.

ESTIMATED CUBIC FEET OF VOLUME HEATED BY 1 SQUARE FOOT OF HOT-WATER HEATING SURFACE

Residences:

Living rooms.....	20 to 40
Sleeping rooms.....	30 to 50
Offices.....	30 to 50
Schools.....	25 to 50
Factories.....	40 to 80
Assembly halls.....	45 to 90
Hotels.....	50 to 70
Stores.....	50 to 70
Churches.....	80 to 120
Auditoriums.....	80 to 120
Gymnasiums.....	120
Workshops.....	130

The size of the pipes in hot-water systems depends on the velocity of the water and the amount of water to be carried.

The velocity depends on the height of the various radiators

and the difference in the weight of the water on the hot side and the cold side.

Suppose the height of a radiator from the return-flow pipe line is L ft. and the temperature of the supply is t_s and that of the discharge is t_a . These are often about 180° and 160° respectively. Let the corresponding weights per cubic foot be D_s and D_a . The weight of water in the supply column is LAD_s and in the return is LAD_a and that in the flow mains is $L'A' \left(\frac{D_a + D_s}{2} \right)$. A' is the area of the pipes in square feet and L' is the length of each of the flow mains. The force in pounds causing flow is $LAD_a - LAD_s$. This is reduced to feet head by dividing by

$$\frac{A(D_a + D_s)}{2}$$

or

$$h = 2L \frac{D_a - D_s}{D_a + D_s} \dots \dots \dots (68)$$

If now this value for h be inserted in the formula

$$v = \sqrt{\frac{2gh}{f \frac{l}{d} + nm}} \dots \dots \dots (69)$$

where

- v = velocity in feet per second;
- h = head in feet causing flow;
- $g = 32.2$ = acceleration of gravity;
- f = friction factor = 0.02;
- l = total length of system;
- d = diameter of pipe (mean);
- n = number of bends;
- m = friction factor for 1 bend = $\frac{1}{2}$,

the velocity to be expected may be found. In this there will be varying velocities in the various risers so that in general an empirical table is used for the various risers after the amount of radiation is known. From the amount of radiation the amount

of water needed per hour may be found by the rules given above or to be exact the weight of water may be computed thus:

$$W = \frac{H}{q_s - q_a} \quad \dots \quad (70)$$

$$\text{vol}_s = \frac{W}{D_s} \quad \dots \quad (71)$$

$$\text{vol}_a = \frac{W}{D_a} \quad \dots \quad (72)$$

W = weight of water per hour in pounds;

H = heat to be given off per hour;

q = heat of liquid at supply or return.

Knowing the volume, the area of the pipe is given by

$$A = \frac{\text{vol}}{v} \quad \dots \quad (73)$$

For use in computing problems, the following table is given.

DENSITY OF WATER AT VARIOUS TEMPERATURES

Temperature.....	50	68	86	104	122
Rel. density.....	1.00025	1.000174	1.00425	1.0077	1.0119
Temperature.....	140	158	176	194	212
Rel. density.....	1.0169	1.0226	1.0289	1.0357	1.0431

To apply the above to a given problem suppose 7500 B.t.u.'s are to be transmitted from radiators per hour in rooms 10 20 and 30 ft. from basement floor, at which level the return water enters the boiler. The hot water is assumed at 180° and the return at 160° F. The radiators are arranged on 5 risers. The flow pipes are 40 ft. long.

1st. Heating surface approximate:

$$A = \frac{75000}{170} = 440,$$

or 88 sq.ft. per riser.

2d. Water per riser:

$$\text{wt.} = \frac{75000}{5} \frac{1}{148-128} = 750 \text{ lbs. per hour;}$$

$$\text{vol.} = \frac{750}{1.03 \times 62.5} = 11.7 \text{ cu.ft. per hour;}$$

3d. Velocities to different floors:

$$h_1 = 20 \frac{1.030 - 1.023}{1.030 + 1.023} = 0.068 \text{ ft.}$$

$$h_2 = 40 \times .0034 = 0.136 \text{ ft.};$$

$$h_3 = 60 \times .0034 = 0.204 \text{ ft.};$$

$$v_1 = \sqrt{\frac{2 \times 32.2 \times 0.068}{0.02 \times \frac{(40+10)}{\frac{1}{6}} + 8 \times \frac{1}{2}}} = 0.66 \text{ ft. per sec.};$$

$$v_2 = 0.89 \text{ ft. per sec.}$$

$$v_3 = 1.03 \text{ ft. per sec.}$$

These velocities in ft. per min. are respectively: 39.6, 53.4, 61.8.

The velocities will practically increase with the square root of the height.

4th. Areas of pipes:

Using mean velocity of 0.9 from v_2 , the following results:

$$\text{Area of riser in square inches} = \frac{11.7 \times 144}{0.9 \times 3600} = .52 \text{ sq.in.}$$

This gives a $\frac{3}{4}$ -in. pipe in which the resistance is much greater than in a 2-in. pipe used in computing V . Hence a recalculation should be made giving a large pipe.

This method is not used, as it is lengthy and instead a table such as that given below is employed generally.

SIZE OF PIPES FOR HOT-WATER MAINS AND RISERS

Sq. Ft. of Radiation.	Mean Height of Radiators.			
	10	20	30	40
50	1 inch	1 inch	1 inch	1 inch
100	1 $\frac{1}{4}$ "	1 "	1 "	1 "
150	1 $\frac{1}{2}$ "	1 $\frac{1}{4}$ "	1 $\frac{1}{4}$ "	1 "
200	2 "	1 $\frac{1}{2}$ "	1 $\frac{1}{4}$ "	1 $\frac{1}{4}$ "
250	2 "	1 $\frac{1}{2}$ "	1 $\frac{1}{2}$ "	1 $\frac{1}{4}$ "
300	2 "	2 "	1 $\frac{1}{2}$ "	1 $\frac{1}{2}$ "
400	2 $\frac{1}{2}$ "	2 "	2 "	2 "
450	2 $\frac{1}{2}$ "	2 $\frac{1}{2}$ "	2 "	2 "
500	3 "	2 $\frac{1}{2}$ "	2 "	2 "
1000	4 "	3 "	3 "	3 "

Using the table the problem above would require a 1-in. pipe for 88 sq.ft. for a mean head of 20 ft.

The flow lines are found for the total amount of radiation or water.

The Honeywell Company determine sizes of risers and flow pipes by adding together the areas of valve openings or areas of radiator connections used on any riser or supplied by the flow pipe. Their valve sizes or drilling sizes are smaller than those recommended by others, as seen by comparing tables on pages 163 and 168. Thus in the problem above they would use a $\frac{3}{4}$ -in. riser.

The connections to the radiators are fixed by the drilling table of the radiator manufacturers as given in Chapter VI. The drillings for the water radiator are repeated.

SIZE OF OUTLETS FROM RADIATORS

Sq. Ft. of Radiation.	Supply and Return.
40	1 X 1
40-72	1 $\frac{1}{4}$ X 1 $\frac{1}{4}$
72	1 $\frac{1}{2}$ X 1 $\frac{1}{2}$

Air valve vent tapping $\frac{1}{8}$ in.

Although the best circulation is obtained when two valves are used on hot-water radiators, a single valve, Fig. 128, may be used. This is known as the **Honeywell unique valve**.

The supply goes in one side of the valve and enters the radiator, being kept from short circuiting to the return by the partition which extends into the radiator. This permits the water to rise through the first section of the radiator and fall through the remaining sections. The handle moves a diaphragm which separates the two elbow openings in the valve, causing the water to pass in on one side of the partition and out of the other. A turn of one-sixth of a revolution causes the diaphragm

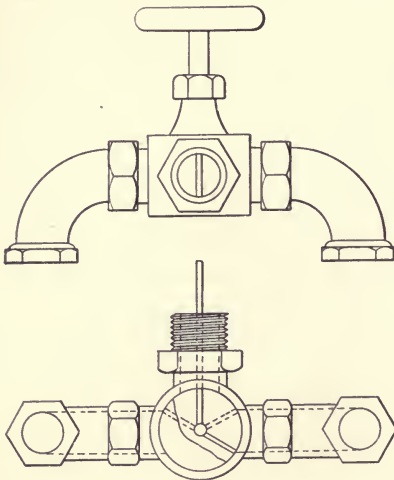


FIG. 128.—Honeywell Valve.

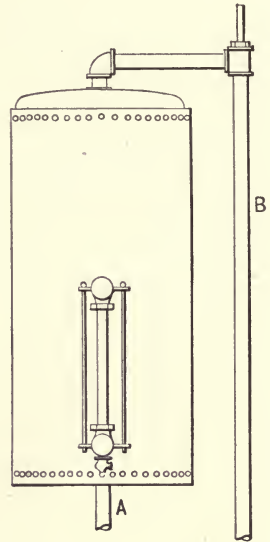


FIG. 129.—Expansion Tank.

to cut out the opening to the radiator and connect the two elbows, **by-passing** the radiator.

In this way the circulation through the supply and return is never interfered with. This valve can be used with the complete-circuit system, the supply elbow being turned up and the return down. There are several advantages in having the connections at one end; the radiator may be enlarged; the cutting is at one place and the two risers may be kept close together.

Since water expands about 3 per cent in being heated to 180° F. it is evident that there must be some provision to care

for this expansion, and hence all hot-water systems are provided with an **expansion tank**, Fig. 129, or its equivalent. The expansion tank is connected to a riser *A* at the highest part of the system and is provided with a water gage to show the level of the water and a vented overflow *B* leading to the sewer.

This tank should be of such a volume that the expansion of cold water, say at 70° F., to the higher temperature, say of 200° F., will cause the water to rise from near the bottom of the tank to the top of the tank or to the top of the water column. The size should be such that the latter is true, then at all times the level of water is shown. If the water level cannot be seen

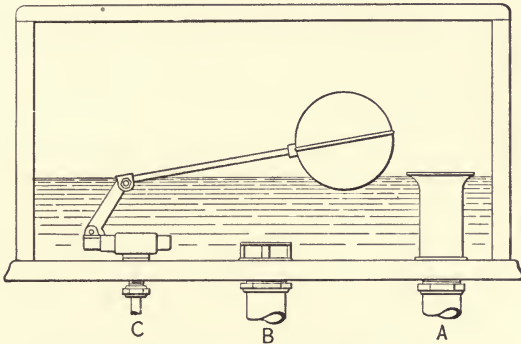


FIG. 130.—Expansion Tank with Float.

the small cock at the bottom of the gage may be opened to indicate whether water is present or not. When water is low in the tank the water is usually fed into the system from the city supply, which is connected where the return water enters the boiler.

Another method to ensure the system being kept full of water is to have a tank, Fig. 130, with a **ball float** attached to the high part of the line. Then as the water rises it is carried off by the **overflow A**, while if the water contracts more water is fed into the system from the tank through *B*, the ball float controlling the admission as the level falls.

These tanks are placed at a high part of the system, but it should always be in a warmed room, so that there is no danger

of the expansion tank freezing. The freezing of the tank would not only endanger the tank, but it will prevent expansion of the water as it is heated, thus bringing undue strains on the system and rupturing radiators or boiler.

If a **water closet** is on the top floor of a building, the water tank of the closet may be used as an expansion tank, as this will always have water over the bottom of the tank.

The size of the tank may be found by computing the water content of the system and then using 3 per cent of its volume as the volume of the tank. In general, however, the tank may be proportioned by the amount of radiating surface by the table below:

EXPANSION TANKS

Size.	Gallons Capacity.	Sq. Ft. of Radiation.	Size.	Gallons Capacity.	Sq. Ft. of Radiation.
10×20	8	250	16×36	32	1300
12×20	10	300	16×48	42	2000
12×30	15	500	18×60	66	3000
14×30	20	700	20×60	82	5000
16×30	26	950	22×60	100	6000

The maximum temperature carried on a hot-water system is fixed by the height of water carried on the boiler. Thus if the level of the expansion tank is about 40 ft. the water at the boiler could be 250° F. before it could boil, due to the pressure on it, but as soon as this heated water had reached a higher level, part of it would turn into steam and drive the water out of the expansion tank. In most cases when the water gets beyond 220° F. there is danger of driving the water out of the system.

To enable one to carry a higher temperature in very cold weather the **Honeywell Generator**, Fig. 131, is used. The point *A* is connected to the heating system, preferably near the boiler. As the water in the system is heated it expands into the chamber *B* driving the mercury *C* into the **circulating tube** *D* and the **standpipe** *E*. By the time the mercury reaches the top of the circulating pipe the lower end is open to the water and this rises through the mercury, causing an upward flow, the mercury

which is discharged into the separating chamber *F* falling back through the standpipe *E*. If the discharge is at all violent, the baffle plate *G* will deflect the mercury downward. The water then passes through *H* to the expansion tank.

When the system cools off the contraction of the water causes the mercury to be forced up in *B*, allowing water to flow back through *H* and *E* and passing up through *B* it separates from the mercury and leaves at *A*.

By this arrangement it is seen that the water in *A* and *B* may be under at least a pressure produced by a column of mercury equal to the height of the circulating tube. This usually amounts to about 10 lbs. per square inch. In this way the pressure on the water in the system may amount to 10 lbs., permitting the temperature to reach about 240° F.

The Honeywell Company claims that this device will accelerate the flow of water through the system even under low temperatures. This device does not increase the driving force unless the temperature is increased, as

the system is closed, there being the same static difference of pressure throughout the system due to temperature difference, but in case of need the possibility of getting a higher tem-

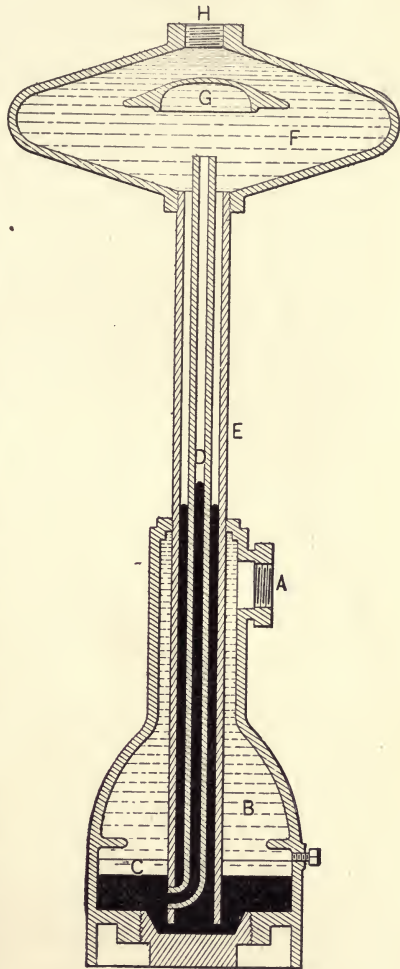


FIG. 131.—Honeywell Generator.

perature on one side means that there can be more difference between the weights of the water in the ascending riser and in the return riser, and hence there may be a more rapid circulation. The generator is a very ingenious and valuable device for increasing the temperature of the water and thus the value of each square foot of radiation, and at the same time increasing the unbalanced pressure due to the difference in density of the hotter water, so that smaller pipe may be used. It will do this with safety.

The drillings recommended by this company for radiators are as follows:

FIRST FLOOR

Up to 30 sq.ft.	$\frac{1}{2}$ inch
30 to 75 "	$\frac{3}{4}$ "
Over 75 "	1 "

SECOND FLOOR

Up to 40 sq.ft.	$\frac{1}{2}$ inch
40 to 100 "	$\frac{3}{4}$ "
Over 100 "	1 "

THIRD FLOOR

Up to 50 sq.ft.	$\frac{1}{2}$ inch
50 to 125 "	$\frac{3}{4}$ "
Over 125 "	1 "

The valves of the radiators at the ends of the mains are made one size larger than those given in the table.

These sizes are much smaller than those given in the earlier part of the chapter and the areas of risers and mains which are made equal in area to the areas of the connections which they supply are also smaller. This is possible because in times of need the temperature may be increased to such a point that the circulation is rapid enough to care for the heat needed.

Small pipes are cheaper, but beyond that there is no advantage in the small pipes except that because the amount of water in the system is less, the time taken to get the radiators heated is

not great. Against this, however, the fact must be remembered that the system of small water capacity will cool quicker. The heat put into the water, whether the mass be great or small, is to be taken out by the radiators, so that the volume of water in the system should not affect the economy of the system.

The method of attaching the generator to one of the upper radiators of a system which has been installed is shown in Fig.

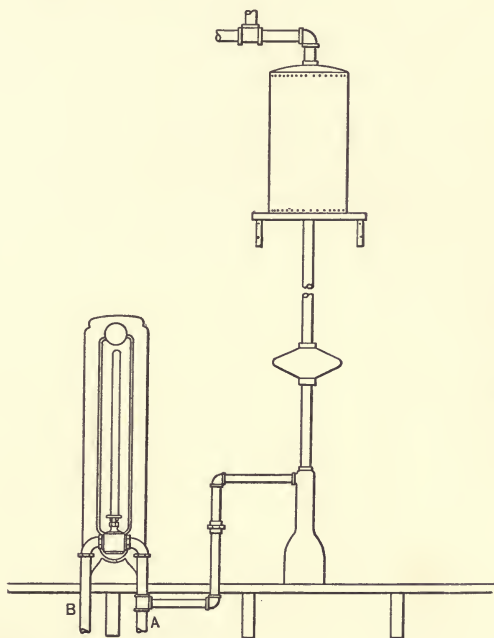


FIG. 132.—Generator Connection.

132. In this a branch has been taken from the supply of a unique valve and carried to the generator, which is then connected to the expansion tank. In no case should the generator be placed close under the expansion tank. The pressure produced in the system is independent of the position of the generator. The same water column is acting in addition to the mercury wherever it is placed.

Using the methods given above on the plans of the residence

and the tables of Chapter V, the amount of radiation for the various rooms may be found. These are given in the table below:

Room.	Sq. Ft.	Kind.	Room.	Sq. Ft.	Kind.
1	170	3 col., 38 ins.	7	157½	3 col., 26 ins.
2	120	3 col., 38 "	8	75	3 col., 38 "
3	21	3 col., 22 "	9	22½	3 col., 32 "
4	71¼	3 col., 26 "	10	66	3 col., 22 "
5	90	3 col., 20 "	11	66	3 col., 22 "
6	126	3 col., 45 "	12	27	3 col., 22 "

These are now placed on the development, and the cellar

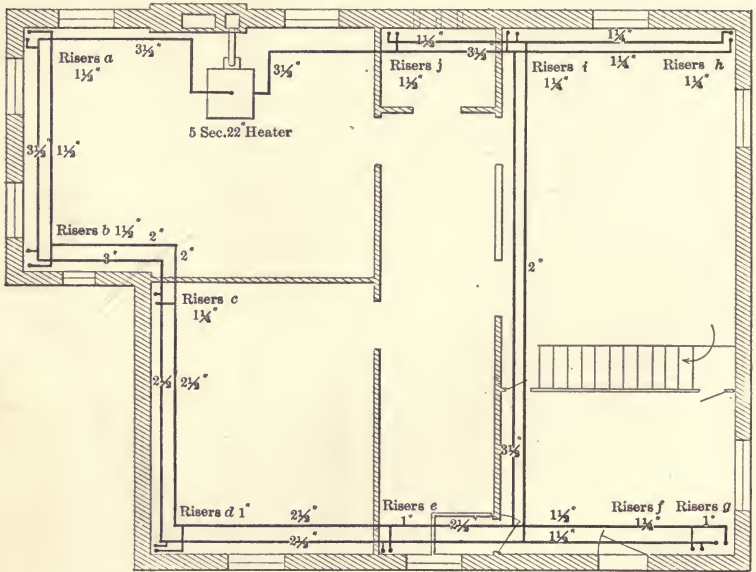


FIG. 133.—Arrangement of Pipes and Boiler for Cellar with Hot Water.

plan, Fig. 133, is made to show the arrangement of flow lines and boiler.

The table below gives the surface on each riser and the size of the same.

Risers	a.	b.	c.	d.	e.	f.	g.	h.	i.	j.
Surface	163¾	163¾	135	60	43½	104¼	33	78	78	153
Size.....	1½"	1½"	1¼"	1"	1"	1¼"	1"	1¼"	1¼"	1½"

CHAPTER VIII

INDIRECT HEATING

As mentioned in Chapter I, there are three general methods of indirect heating. First, the **natural-draft** method, in which heating coils are placed in boxes at the bases of flues leading to a room or group of rooms. This is used in small installations or residences. Second, the **plenum** or **forced mechanical method** in which air is drawn over heating coils and forced into the various parts of the building to be heated, thus producing in the rooms a pressure slightly above the atmosphere. Third, the **vacuum system** in which air is drawn from the rooms to be heated by a fan, inlet air passing from the outside over coils of pipe. In all of these methods air is used to convey the heat and in all but the first method there may be considerable force to give this air a definite path. In the first method the **heaters** are placed at the bottom of **vertical risers**, so there is little resistance to the flow of air, and this method gives good results. The **boxes** are placed at proper locations through the basement and supplied with steam from a boiler at some convenient point. There is no trouble experienced in passing steam to remote indirect radiators. The second or plenum method has the advantage of keeping the building under a pressure above the atmosphere so that leakage is outward, while in the vacuum system there is a constant leakage of cold air into the room through all loose windows or doors. There are cases, however, where the vacuum method is the only one which can be used, and hence the student should understand its peculiarities.

The method of procedure for the design of indirect installations is best illustrated by an application to a given building. The house used in Chapter V will be investigated according to the method of separate heaters.

In the indirect system the amount of air for ventilation is first determined, then the temperature of this air is found so that it will supply the heat losses to the room when the air is cooled off from the inlet temperature to the temperature desired in the room. After this, the amount of heating surface needed is computed and finally the size of the duct to properly carry the air is determined. These steps are common to all three indirect methods of heating. If the temperature of the entering air t_1 (or T_1 absolute) is too high for convenience or comfort, it will be necessary to increase the quantity of air beyond that considered necessary for ventilation.

The temperature of the air fixes the number of rows of coils or heaters in the mechanical system, although in the natural-draft system one set of indirect heaters is usually sufficient, as the velocity over the radiators is low. The curves of Chapter IV give these data.

If the quantity of air per hour is V cu.ft. and the amount of heat lost from walls and windows per hour is h in B.t.u., the following equation holds:

$$h = Vc(t_1 - t_r) \dots \dots \dots (74)$$

- c = heat necessary to raise 1 cu.ft. 1° F;
- t_1 = temperature of entrance;
- t_r = temperature of room.

Hence .

$$t_1 = t_r + \frac{h}{Vc} \dots \dots \dots (75)$$

now as has been noted earlier in the text

$$c = \frac{0.237 \times 144 \times p_b}{R(t + 459.6)} = 0.02 \text{ approximately } \dots \dots \dots (76)$$

- $R = 53.34$;
- t = mean temperature of air in deg. F. = T° absolute;
- p_b = barometric pressure in pounds per square inch.

This neglects the effect of vapor in the air, although that should be considered for great accuracy. The reason for this is the fact that if the vapor is considered in finding c for the mixture, it must be considered in Eq. (74) giving an additional term, although the value of c would be smaller. The net result would be slightly different from the above and hence this approximate method is sufficiently accurate for this type of problem.

It will be seen that the heat in the air above the room temperature is sufficient to care for the heat losses, as in most cases the hot air is delivered across the ceilings against the cold walls before it mingles with the air of the room, and hence by that time it is reduced to room temperature, and when it returns to the bottom of the wall from which it was discharged so as to pass out through the **vent flue** to the roof it is as the temperature of the room.

In computing this temperature of entrance t_1 for various rooms in a system, it is found that the value is quite different for different rooms, because the quantities V and h vary in different ways; V for one room may increase over that in another if more people occupy the room, while h might be smaller if that room were not exposed as much as the other room. For this reason it is not possible to run a main duct from the heater in the plenum or vacuum systems and take from it the flues to the various rooms. One of two methods must be used. In the first method a pair of ducts must be run in the basement, one carrying hot air and the other warm air, and from them connecting branches are run to each flue with mixing dampers, so that the proper amount of each may be had to give the desired temperature. In the second method a separate duct must be taken to each flue from the heater where both warm and hot air are supplied through mixing dampers to give the correct temperature to each duct.

The first method is known as the **double-duct system** and the second, the **single-duct system**. These were described on page 13. The requirement of air at two temperatures makes it necessary in both of these arrangements of mechanical ventila-

tion to install two heaters, one known as a **tempering coil**, to heat all of the air to a low temperature and the other to heat a portion of the air to a higher temperature after separating the air into two parts. One temperature might be fixed by the highest temperature necessary, the other by the lowest temperature necessary.

In the natural-draught method of indirect heating there is not the necessity of having the two temperatures, as the air from each box passes to its own room or group of rooms. In this system the main problem of design is to get sufficient surface to give the heat necessary for the room and to get sufficient air for ventilation. Since the air enters at about 100° F. the velocity for different heights of flow may be worked out as follows:

Weight of L ft. of air at temperature t_1 of 1 sq.ft. cross-sections:

$$W_1 = \frac{144p_b L}{R(t_1 + 459.6)} = \frac{144p_b L}{RT_1} \dots \dots (77)$$

(T is absolute temperature).

Weight of L ft. of air at the temperature of the outside air:

$$W_o = \frac{144p_b L}{R(t_o + 459.6)} = \frac{144p_b L}{RT_o} \dots \dots (78)$$

The head causing flow is the difference of these if expressed in pounds per square foot, or if divided by the weight of 1 cu.ft. of air at the flue temperature it gives the head in feet of air.

The weight of 1 cu.ft. of hot air is

$$W = \frac{144p_b}{R(t_1 + 459.6)} = \frac{144p_b}{RT_1} \dots \dots (79)$$

Hence the head in feet of the hot air which is flowing through the pipes is

$$\text{Head} = \frac{L(T_1 - T_o)}{T_1 T_o} T_1 = \frac{L(T_1 - T_o)}{T_o} \dots \dots (80)$$

This head is used in causing a velocity v in the hot air and overcoming friction. Using the general hydraulic equation for the flow of a fluid the following results:

$$\text{Head} = \frac{v^2}{2g} \left(1 + k + nm + f \frac{L}{d} \right) \dots \dots \dots (81)$$

- v = velocity in feet per second;
- k = coefficient for entrance loss = $\frac{1}{2}$;
- n = numbers of bend;
- m = coefficient for 1 bend = 0.2;
- f = friction factor = 0.02;
- L = length of pipe in feet;
- d = diameter of pipe in feet;
- g = acceleration of gravity = 32.2 ft. per sec. per sec.

If the bracket be called Z the following results:

$$v = \sqrt{\frac{2g \text{ Head}}{Z}} = \sqrt{2g \frac{L(T_1 - T_o)}{ZT_o}} \dots \dots \dots (82)$$

For $T_1 = 560$, $T_o = 460$, and $Z = 2$; this becomes

$$v = 8.02 \sqrt{0.11L} \dots \dots \dots (83)$$

These values of V for different floors are then as follows:

- $v = 6$ ft. per second for 5 ft.;
- $v = 8.4$ ft. per second for 10 ft.;
- $v = 11.9$ ft. per second for 20 ft.;
- $v = 14.6$ ft. per second for 30 ft.;
- $v = 16.8$ ft. per second for 40 ft.

Although these values have been computed with friction, the usual values taken in design are considered at about half of these: i.e., 3 ft. per second for first floor, 4.2 ft. per second for second floor 5.8 ft. per second for third floor and 7.3 ft. per second for fourth floor.

Having the velocity for any floor and the amount of air required, the size of flue may be found.

$$A = \frac{V}{3600v} \quad (V = \text{cu.ft. of air per hour}) \dots \dots (84)$$

The amount of heating surface will depend on the amount of heat required and the rate of heat transmission. Adding together the heat for ventilation and that for losses as found in the tables of Chapter V, the total amount of heat is known. The heat transmission per square foot of area of indirect surface depends on the velocity and until this is known only an approximate value can be had. Assuming this to be 300 B.t.u. the surface required will be given by

$$S = \frac{H}{300} \dots \dots \dots (85)$$

If now the area between the sections to give this surface be found then the velocity may be found:

$$v = \frac{V}{3600A_h} \dots \dots \dots (86)$$

A_h = Area of passages in heater. Experiments seem to indicate that with natural draft radiators the heat transmission coefficient equals the square root of the velocity. This means

$$h_1 = \sqrt{v} \left[t_s - \frac{(t_o + t_1)}{2} \right] \dots \dots \dots (87)$$

- h_1 = B.t.u. transmitted per hour per square foot;
- t_s = temperature of steam or mean temperature of water in degrees F.
- t_o = temperature of outside air;
- t_1 = temperature of air entering room;
- v = velocity over coils in feet per second.

In this way the heat per square foot may be found and from it, the surface. It may be necessary to make two approximations before the area found agrees with that required to give the h_1 assumed.

After this is computed the ducts leading to the outside are found. In determining these the velocity should be assumed to be that in the inlet passages. In fact this velocity might be

assumed to be the velocity over the coils or sections of the heater and the distance between sections so made that this is obtained. In figuring areas for cold air the volume of this air must be found.

The application of this method to room 1 of the house leads to the following:

Volume air per hour 6200 cu.ft.

$$t_1 = \frac{16810}{6200 \times 0.018} + 70 = 226^\circ \text{ F.}$$

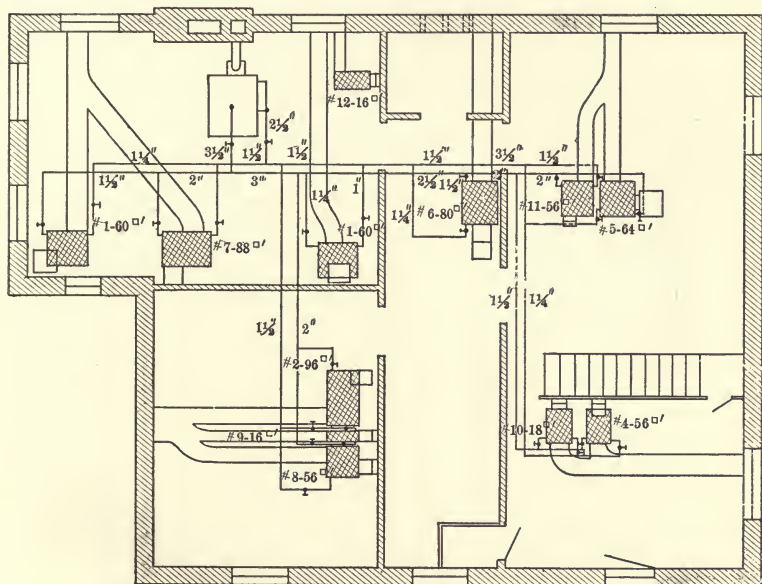


FIG. 134.—Cellar Plan for Indirect Heating.

This is too high, so the quantity of air must be increased. Assume this to be six changes per hour.

$$t_1 = \frac{16810}{18600 \times 0.018} + 70 = 120^\circ \text{ F.}$$

This is a possible value.

$$H = 16810 + 70 \times 0.018 \times 18660 = 40300.$$

Velocity 4 ft. per second.

$$h = \sqrt{4 \left(227 - \frac{0 + 120}{2} \right)} = 334;$$

$$S = \frac{40300}{334} = 120 \text{ sq.ft.};$$

$$A_h = \frac{18660 \times 144}{4 \times 3600} = 186.6 \text{ sq.in.}$$

The table below gives the data for the complete house and Fig. 134 illustrates the layout in cellar to accomplish the result.

Room	Heat Loss.	Ventilation.		t_1	Total H.	v	h	s	Actual s.*	Area Flue.	Ratio.
		Chap. V.	Used.								
1	16810	6200	18600	120	40300	4	334	120	120	187	26
2	11792	4300	14000	117	29400	4	334	90	96	140	22
3	1745	800	2000	118	4300	4	334	13	16	20	25
4	6890	2800	8000	113	16900	4	334	51	56	80	25
5	8740	3200	10000	119	21300	4	334	64	64	100	25
6	10635	3200	12000	119	25700	4	334	78	80	120	27
7	14050	2910	16000	119	34100	5.6	394	87	88	114	33
8	8750	2000	10000	119	21300	5.6	394	54	56	72	36
9	2170	720	3000	110	5700	5.6	394	15	16	22	44
10	7680	1700	9000	117	19000	5.6	394	48	48	65	26
11	8330	1710	10000	116	21000	5.6	394	53	56	72	25
12	2840	480	3500	115	6000	5.6	394	15	16	25	19

* Using Junior Indirect Radiators.

The ducts leading the air to the boxes, the boxes and the flues are made of **galvanized iron**, although in many cases the flues are made of tin as will be given more in detail in the next chapter. Each duct leading to the outside should have a **slide** so that cold air may be cut off should there be any danger of the steam being shut off. A **damper** should be put in each duct and it is well to have register faces with closing flaps.

The galvanized iron should be of proper gauge to give sufficient stiffness. For round pipes the manufacturers of heating apparatus recommend the following:

Diameter.	Gauge.	Rectangular.
up to 18"	26	up to 6×6
19 to 24"	24	7×7 to 12×12
30 to 39"	22	13×13 to 20×20
40 to 49"	20	21×21 up
50 to 70"	18	

For rectangular ducts the limiting square is given. The gauge used is the **U. S. standard sheet metal gauge**. The gauge numbers stand for the following thicknesses:

Gauge No.	28	26	24	22	20	18	16	14
Thickness	$\frac{1}{64}$ "	$\frac{3}{160}$ "	$\frac{1}{40}$ "	$\frac{1}{32}$ "	$\frac{3}{80}$ "	$\frac{1}{20}$ "	$\frac{1}{16}$ "	$\frac{5}{64}$ "
Gauge No.	10	8	6	4	2	0	000	
Thickness	$\frac{9}{64}$ "	$\frac{11}{64}$ "	$\frac{13}{64}$ "	$\frac{15}{64}$ "	$\frac{17}{64}$ "	$\frac{5}{16}$ "	$\frac{3}{8}$ "	

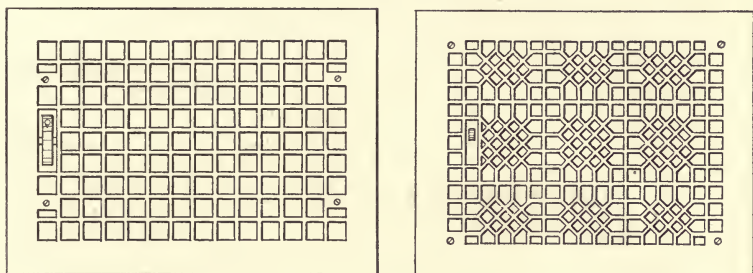


FIG. 135.—Register Faces.

The **register faces** are of various sizes and designs. Fig. 135 illustrates two typical forms of these. They are usually made with dampers. They vary from 4×6" around body ($5\frac{5}{8} \times 7\frac{5}{8}$ extreme size), to 38×42" ($40\frac{3}{8} \times 44\frac{3}{8}$). The variations are by 1 to 4" on a side so that almost any size may be obtained. For special work large sizes may be had.

The **net area** of the register amounts to about 66 per cent of the **box or body area**.

A school building has been selected to illustrate the method of calculating a plenum system, on account of the small quantity of air required for a house. After this the method will be applied to a shop building.

Figs. 136, 137 illustrate the basement and second floor of a school building. There are three floors to the school, all floors being the same.

The amount of ventilation has been computed by allowing 1800 cu.ft. per pupil, which is the amount required by law in

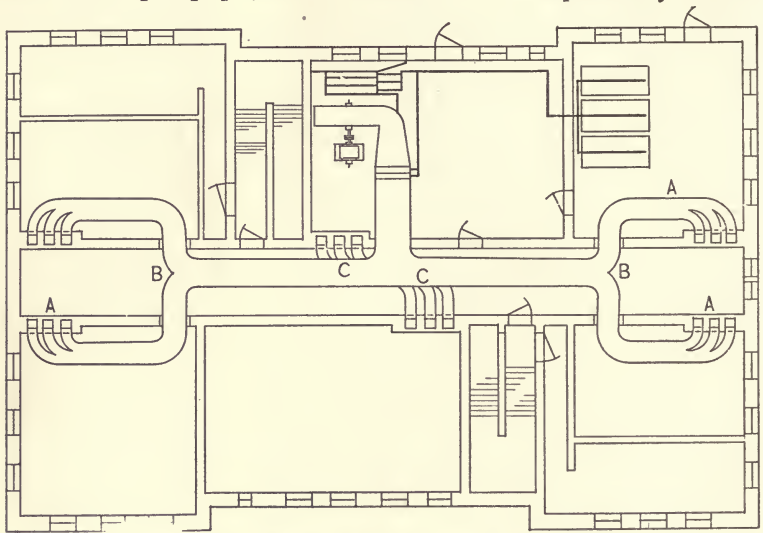


FIG. 136.—Basement of School No. 3, Rydal, N. Y.

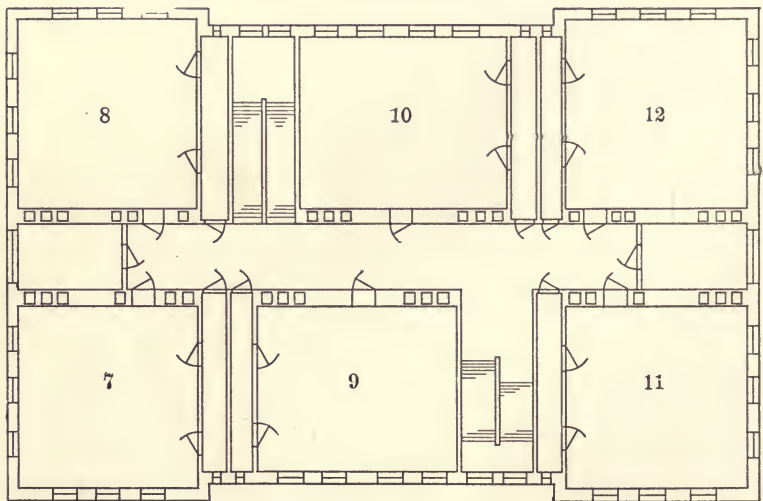


FIG. 137.—Second Floor of School No. 3, Rydal, N. Y.

some States. This is rather a large amount, especially in the lower grades and could be reduced if it were not for the law.

The amount required by various authorities has been given in Chapter II.

The heat loss from the various rooms considering the exposure and other elements has been computed in a manner similar to that described in Chapter V and the results of this computation are shown in the table on page 182.

The temperatures assumed for this school have been 70° in zero weather. The column marked "air temperature" gives the temperature of air entering the room from the flue.

The first computations to be made are those from the heater. The temperatures of the hottest air and coolest air show that the air could be heated to 71° F. and a portion to 89° F. Then by properly adjusting all dampers the necessary mixture temperatures could be had. In the present instance, though, the temperature of the tempered air will be 70° F., and the remainder will be heated to such a temperature that the total heat will be equal to that required for the building.

Total heat for building	= 3,052,000 B.t.u.
Heat for tempering coil	= 2,541,000 B.t.u.
Heat from main coil	= 511,000 B.t.u.
Temperature of air discharged from main coil if one-half is heated	= $\frac{511,000}{0.02 \times 181500 \times \frac{1}{2}} + 70^{\circ}$ = 98° F.

The problem will be worked out using Vento heaters and then with pipe coils, the steam being at 5 lbs. pressure in zero weather.

From the Vento heater curves, Figs. 89-90, it is seen that 70° is given by 2 sections at a velocity of 730 ft. per minute, or by 3 sections at 1700 ft. per minute. From Fig. 91, 1 section of 4 rows of pipes at 190 ft. per minute, 2 sections at 370 ft., 3 sections at 800 ft., 4 sections at 1600 ft. would give a temperature of 70° if pipe coils were used.

The high velocities would mean much friction and the low velocities would probably require too much space. The velocity of 1000 ft. per minute is a fair value, so that in the problem the velocity of 730 might be used for the Vento heater and 800 ft. per minute if pipe coils are used. These values will be

REQUIREMENTS FOR SCHOOL No. 3

RYDAL, N. Y.

*First Floor**Second Floor*

Room No.	Occupants.	Cubic Feet. Ventilation.	Heat Loss.	Air Temp.	Room No.	Occupants.	Cubic Feet. Ventilation.	Heat Loss.	Air Temp.
1	49	88200	29100	86	7	49	88200	29000	86
1A	—	6000	1800	85	7A	—	6000	1800	85
2	49	88200	27000	85	8	49	88200	26000	85
2A	—	6000	1500	83	8A	—	6000	1500	83
3	49	88200	21000	82	9	49	88200	20000	81
3A	—	6000	1500	85	9A	—	6000	1800	85
4	49	88200	19000	81	10	49	88200	18000	80
4A	—	6000	1500	83	10A	—	6000	1500	83
5	49	88200	33000	89	11	49	88200	32000	88
5A	—	6000	1800	85	11A	—	6000	1800	85
6	49	88200	25000	84	12	49	88200	24000	84
6A	—	6000	1800	85	12A	—	6000	1800	85
									75
Halls		60000	10000	71	Halls	30000	3000	
Total	625000	174200		Total	595000	162200	

Third Floor

Room No.	Occupants.	Cubic Feet Ventilation.	Heat Loss.	Air Temperature.
13	49	88200	30000	87
13A	—	6000	1800	85
14	49	88200	28000	86
14A	—	6000	1500	83
15	49	88200	22000	83
15A	—	6000	1800	85
16	49	88200	20000	81
16A	—	6000	1500	83
17	49	88200	34000	89
17A	—	6000	1800	85
18	49	88200	26000	85
18A	—	6000	1800	85
Halls	30000	4000	77
Total	595000	174200	
Grand total	1815000	510600	
Heat for air		2541000	
Total heat B.t.u.		3051600	

used, although in the remaining calculations it may be necessary to make a different assumption for the temperature of the tempered and heated air.

From the curves of Chapter IV the heating value of the surface may be found. Fig. 90 gives the heat per square foot per hour at 730 ft. per minute with 2 sections as 1450 B.t.u., while 1750 B.t.u. will be given off per square foot of coil surface.

The square feet of heating surface required for the tempering coil will be:

$$\text{For Vento heaters } \frac{2,541,000}{1450} = 1750 \text{ sq.ft.};$$

$$\text{For Buffalo Forge coils } \frac{2,541,000}{1750} = 1450 \text{ sq.ft.};$$

The area in square feet required through these heaters is

$$\text{For Vento heaters } \frac{1,815,000}{730 \times 60} = 41.5 \text{ sq.ft.};$$

$$\text{For coils } \frac{1,815,000}{800 \times 60} = 37.8 \text{ sq.ft.}$$

The data sheets for Vento heaters and coils of one company are given below, and from these the following is found:

1750 sq.ft. in 2 sections = 875 sq.ft. per section. Use 3 stacks of 19 sections = 912 sq.ft. For $4\frac{5}{8}$ ins. center, area = $3 \times 14.83 = 44.5$ sq.ft.

This is sufficiently close to 41.5 to give little change in velocity.

For the pipe coils 1450 sq.ft. = 4210 lin.ft.

Each section must contain 1403 lin.ft. Hence use 2-6-ft.-2E heater units in each section. The area is $21.2 \times 2 = 42.4$, which is close enough to give the proper velocity.

In each of the above cases the area was large and consequently the velocity would be low. This would reduce the heat transmission and require more surface. This is not necessary in the cases above.

The tables give the over-all dimensions of these heaters with the allowance for staggering. The Vento heater of 6 stacks will require $(3 \times 88'' + 5'' + 10'') \times (2 \times 10'' + 4'') \times 60'' = 279'' \times 24'' \times 60''$ high.

The Buffalo coils, $(2 \times 7' - 10'') \times (3 \times 8\frac{1}{2}'') \times 72''$.
 $15' - 8'' \times 24\frac{1}{2}'' \times 72''$.

To reduce the column headed lineal feet of 1-in. pipe in the table of the Buffalo Forge coils to square feet, the numbers are divided by 2.9.

In regard to the Vento heater it is to be remembered that the makers build a narrow section containing about three-quarters of the heating surface of the regular section, but having the same dimensions in height and width with the same air space.

Both forms of heaters are in common use; some designers prefer one, some the other.

To find the number of sections to use in the main heater, the number of sections must be found to give the required temperature of the hot air and from this the number used in the tempering coils must be subtracted.

Thus at 730 ft. per minute 3 sections of Vento heaters must be used to give 98° F., and at 800 ft. per minute $4\frac{1}{2}$ sections of coils must be used to get this temperature. This latter means 5 sections. The heating coils will then be made of 1 section of Vento heater or 2 sections of coils. The surface required is found as follows:

From curves, Fig. 90a, for Vento heater:

$$h = \frac{3 \times 1340 - 2 \times 1450}{1} = 1120;$$

$$\text{Area} = \frac{511,000}{1120} = 456.$$

From curves, Fig. 92a, for coils:

$$h = \frac{1580 \times 5 - 1750 \times 3}{2} = 1325;$$

$$\text{Area} = \frac{511,000}{1325} = 385;$$

$$\text{Area for air passage} = \frac{907,500}{730 \times 60} = 20.7 \text{ for Vento;}$$

$$= \frac{907,500}{800 \times 60} = 19.0 \text{ for coils.}$$

For the Vento heater 1 stack of 28 60 in. sections, $4\frac{5}{8}$ -in. centers, will be used giving 448 sq.ft. and 21.86 sq.ft. area for air passage. For the coil:

385 sq.ft. = 1120 lin.ft. per section;

2 sections of 560 lin.ft. per section;

5-ft. section 3D will be used with a $\frac{1}{2}$ section.

This gives 567 lin.ft. and 16.8 sq.ft. area.

SIZES AND DIMENSIONS OF BUFFALO STANDARD FAN SYSTEM HEATER

Length of Section.	Section No.	Extreme Height of Heater.	Width of Section.	Lineal Feet of 1-inch Pipe per Section.	Area for Air Passage, Square Feet.	Weight.
3' 4 row	1A	3' 4"	8 $\frac{1}{2}$ "	158	5.1	423
	2A	3 10	8 $\frac{1}{2}$ "	178	5.4	454
	3A	4 4	8 $\frac{1}{2}$ "	193	6.1	477
	4A	4 10	8 $\frac{1}{2}$ "	221	6.9	520
	5A	5 4	8 $\frac{1}{2}$ "	249	7.7	564
	6A	5 10	8 $\frac{1}{2}$ "	277	8.5	607
4' 4 row	1B	5' 4"	8 $\frac{1}{2}$ "	320	9.8	739
	2B	5 10	8 $\frac{1}{2}$ "	356	10.8	795
	3B	6 4	8 $\frac{1}{2}$ "	392	11.8	850
	4B	6 10	8 $\frac{1}{2}$ "	428	12.9	906
4' 6" 4 row	1C	5' 10"	8 $\frac{1}{2}$ "	396	12.0	881
	2C	6 4	8 $\frac{1}{2}$ "	436	13.0	943
	3C	6 10	8 $\frac{1}{2}$ "	477	14.0	1006
	4C	7 4	8 $\frac{1}{2}$ "	516	15.0	1066
5' 4 row	1D	6' 4"	8 $\frac{1}{2}$ "	479	14.3	1046
	2D	6 10	8 $\frac{1}{2}$ "	523	15.6	1114
	3D	7 4	8 $\frac{1}{2}$ "	567	16.8	1185
	4D	7 10	8 $\frac{1}{2}$ "	611	17.8	1251
6' 4 row	1E	7' 4"	8 $\frac{1}{2}$ "	670	19.7	1388
	2E	7 10	8 $\frac{1}{2}$ "	722	21.2	1479
	3E	8 4	8 $\frac{1}{2}$ "	774	22.7	1549
	4E	8 10	8 $\frac{1}{2}$ "	826	24.2	1630
7' 2 row	1F	8' 4"	6"	480	27.0	1039
	2F	8 10	6	512	29.0	1089
	3F	9 4	6	544	30.8	1138
	4F	9 10	6	576	32.5	1188

From Catalogue 197, Buffalo Forge System.

VENTO CAST-IRON HOT-BLAST HEATER

REGULAR SECTION, RATINGS AND FREE AREAS

Regular 40" Section, 10.75 square feet. Height 41 $\frac{1}{4}$ ". Width 9 $\frac{1}{2}$ ".

No. of Loops in Stack	Square Feet of Heating Surface.	*Equivalent in Lin. Ft. 1-inch Pipe.	5" Centers of Loops.		5 $\frac{1}{2}$ " Centers of Loops.		4 $\frac{1}{2}$ " Centers of Loops.		Actual Weight of Stack in Pounds.	Approximate Wght.
			Standard 44% of Face.		52% of Face.		37% of Face.			
			Net Air Space in Square Feet.	† Width Stack in Inches.	Net Air Space in Square Feet.	† Width Stack in Inches.	Net Air Space in Square Feet.	† Width Stack in Inches.		
7	75.25	226	4.34	35	5.12	38	3.67	32	594	7.92 lbs. per sq.ft. actual 9 lbs. per sq.ft. shipping weight
8	86.00	258	4.96	40	5.85	43	4.20	37	670	
9	96.75	290	5.58	45	6.57	48	4.72	42	728	
10	107.50	323	6.20	50	7.29	54	5.25	46	851	
11	118.25	355	6.82	55	8.02	59	5.77	51	936	
12	129.00	387	7.44	60	8.74	65	6.30	55	1022	
13	139.75	419	8.06	65	9.47	70	6.82	60	1167	
14	150.50	452	8.68	70	10.19	75	7.35	65	1193	
15	161.25	484	9.30	75	10.91	81	7.87	69	1278	
16	172.00	516	9.92	80	11.64	86	8.40	74	1364	
17	182.75	548	10.54	85	12.36	91	8.92	79	1449	
18	193.50	581	11.16	90	13.09	97	9.45	83	1535	
19	204.25	613	11.78	95	13.82	102	9.97	88	1620	
20	215.00	645	12.40	100	14.54	108	10.50	92	1706	
21	225.75	677	13.02	105	15.26	113	11.02	97	1790	
22	236.50	710	13.64	110	15.98	118	11.55	102	1876	
23	247.25	742	14.26	115	16.71	124	12.07	106	1960	
24	258.00	774	14.88	120	17.43	129	12.60	111	2045	

Regular 50" Section, 13.5 square feet. Height 50 $\frac{3}{4}$ ". Width 9 $\frac{1}{2}$ ".

			5" Centers.		5 $\frac{1}{2}$ " Centers.		4 $\frac{1}{2}$ " Centers.			
7	94.5	284	5.37	35	6.35	38	4.55	32	717	7.62 lbs. per sq.ft. actual 9 lbs. per sq.ft. shipping weight
8	108.0	324	6.14	40	7.25	43	5.20	37	810	
9	121.5	365	6.91	45	8.15	48	5.85	42	923	
10	135.0	405	7.68	50	9.05	54	6.50	46	1026	
11	148.5	446	8.45	55	9.95	59	7.15	51	1129	
12	162.0	486	9.22	60	10.85	65	7.80	55	1232	
13	175.5	527	9.99	65	11.75	70	8.45	60	1335	
14	189.0	567	10.76	70	12.65	75	9.10	65	1436	
15	202.5	608	11.53	75	13.55	81	9.75	69	1539	
16	216.0	648	12.30	80	14.45	86	10.40	74	1644	
17	229.5	689	13.07	85	15.35	91	11.05	79	1747	
18	243.0	729	13.84	90	16.25	97	11.70	83	1852	
19	256.5	770	14.59	95	17.15	102	12.35	88	1955	
20	270.0	810	15.36	100	18.05	108	13.00	92	2060	
21	283.5	851	16.13	105	18.95	113	13.65	97	2160	
22	297.0	891	16.90	110	19.85	118	14.30	102	2263	
23	310.5	932	17.67	115	20.75	124	14.95	106	2370	
24	324.0	972	18.44	120	21.65	129	15.60	111	2470	

* The actual length of 1-inch pipe per square foot of outside surface is 2.9 lineal feet but is nominally figured at 3 lineal feet, as shown in the third column of above table.

† Add to the width of stack 2 $\frac{1}{2}$ inches for staggering of stacks.

Taken from American Radiator Co. Catalogue.

Regular 60" Section, 16 square feet. Height 60 $\frac{1}{4}$ ". Width 9 $\frac{1}{2}$ ".

			5" Centers.		5 $\frac{1}{2}$ " Centers.		4 $\frac{1}{2}$ " Centers.			
7	112.0	336	6.45	35	7.62	38	5.47	32	864	7.74 lbs. per sq.ft. actual 9 lbs. per sq.ft. shipping weight
8	128.0	384	7.37	40	8.70	43	6.25	37	988	
9	144.0	432	8.29	45	9.77	48	7.03	42	1112	
10	160.0	480	9.21	50	10.85	54	7.81	46	1238	
11	176.0	528	10.13	55	11.93	59	8.59	51	1362	
12	192.0	576	11.05	60	13.00	65	9.37	55	1486	
13	208.0	624	11.97	65	14.08	70	10.15	60	1610	
14	224.0	672	12.89	70	15.15	75	10.93	65	1734	
15	240.0	720	13.81	75	16.23	81	11.71	69	1858	
16	256.0	768	14.73	80	17.31	86	12.49	74	1982	
17	272.0	816	15.65	85	18.39	91	13.27	79	2106	
18	288.0	864	16.57	90	19.46	97	14.05	83	2230	
19	304.0	912	17.50	95	20.54	102	14.83	88	2352	
20	320.0	960	18.42	100	21.62	108	15.61	92	2478	
21	336.0	1008	19.34	105	22.70	113	16.39	97	2600	
22	352.0	1056	20.26	110	23.78	118	17.17	102	2725	
23	368.0	1104	21.18	115	24.85	124	17.95	106	2850	
24	384.0	1152	22.10	120	25.93	129	18.73	111	2970	

The size of the supply and return for these heaters is found by determining the weight of steam condensed and then using one of the three methods given in the previous chapter.

$$\text{Weight of steam per hour} = \frac{\text{Heat from heater per hour}}{i - q_0};$$

i = heat content of the entering steam;

q_0 = heat of liquid at temperature of return.

The covering around the heaters is usually made of sheet iron of about No. 18 or No. 20. This is put together with stove bolts so that the apparatus may be taken apart when necessary. The casing should lead to the inlet and doors should be provided for inspection of all parts.

The air should be drawn through a wire screen to prevent leaves and sticks from entering, or air washers of the forms shown in Chapter II may be used.

The design of the flues and ducts is next considered.

Flow of Air. There is a loss of pressure due to friction as air flows through piping. This loss as in all fluids varies directly as the length of piping considered; inversely as the **hydraulic radius** of the pipe, which is the ratio of the area of the pipe to its perimeter, and directly as some power of the velocity of the air. There are two velocities to be noted. At a low velocity the air is found to travel in parallel lines and under these con-

ditions the friction varies as the first power. On increasing the velocity beyond that giving parallel flow, the limit of which is known as the **critical velocity**, the air is found to have a turbulent passage and friction varies as the square of the velocity. As most velocities used are beyond the critical velocity, this latter method is the only one to be considered.

There are several methods of stating the pressure in gases. One method is in **ounces per square inch**, or **pounds per square inch**, above the atmosphere. Another method is in **inches** or **feet of water**, which means the distance the water will rise in a **U tube** when connected to the system carrying the air on one **leg** while the other leg is connected to the atmosphere. A still further method is to express this in **feet of the air or gas** considered. This expression gives the height to which the column on one side of a U tube would rise beyond the level in the other if the substance were air or gas of constant density

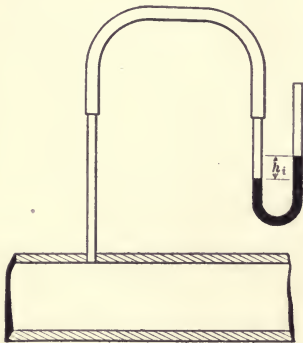


FIG. 138.—U Tube.

equal to that of the air at the point, and the U tube were connected as mentioned above and shown in Fig. 138. Another way in which to explain this is to say that the pressure or head of so many feet of a substance means the height of a column of substance one square inch in cross-section which would weigh an amount equal to the pressure per square inch. Thus if h equals the feet head of a substance and h_1 the head in inches, w the weight of a

cubic foot in pounds, W the total weight in pounds, p the pressure in pounds per square inch, and p_o the pressure in ounces per square inch, the following equations hold:

$$h = \frac{144p}{w} = \frac{144 \frac{p_o}{16}}{w} = \frac{9p_o}{w} \dots \dots \dots (88)$$

$$h_1 = 12h \dots \dots \dots (89)$$

For water

$$w = 62.5 \text{ lbs.}$$

$$h = 2.30p = 0.144p_0 \dots \dots \dots (90)$$

$$h_i = 27.60p = 1.728p_0 \dots \dots \dots (91)$$

For air or gas

$$w = \frac{144p \times 1}{RT} \dots \dots \dots (92)$$

where p = pressure in pounds per square inch.

$$R = 53.34 \text{ for air} \dots \dots \dots (93)$$

$$R = \frac{1544}{\text{mol. density for any gas}} \dots \dots \dots (94)$$

T = absolute temperature in degrees F.

The pressure p is the total pressure on the air and is equal to the barometric pressure plus that above the atmosphere.

In general for air at atmospheric pressure and 70° F.

$$w = \frac{144 \times 14.7 \times 1}{53.34 \times (459.6 + 70)} = 0.0748 = \frac{1}{13.3} \text{ lbs.} \dots \dots (95)$$

Hence

$$h_{\text{air}} = \frac{9p_0}{0.0748} = 120 p_0 = 1920 p \dots \dots (96)$$

$$h_{i \text{ air}} = 1440 p_0 \dots \dots \dots (97)$$

The relation between inches of water and inches of air at 70° and atmospheric pressure is given by

$$p_0 = \frac{h_{iw}}{1.728} = \frac{h_{i \text{ air}}}{1440} = \frac{h_{\text{air}}}{120} \dots \dots \dots (98)$$

or

$$h_{\text{air}} = \frac{120 h_{iw}}{1.728} = 69.5 h_{iw} \dots \dots \dots (99)$$

$$\begin{aligned} h_{i \text{ air}} &= 69.5 \times 12 h_{iw} \\ &= 835.0 h_{iw} \dots \dots \dots (100) \end{aligned}$$

The subscripts, "air" and "w," refer respectively to head in feet or inches of air or water.

If a set of tubes is placed in the side of a pipe line carrying air or other gas, these heads or heights represent the distances

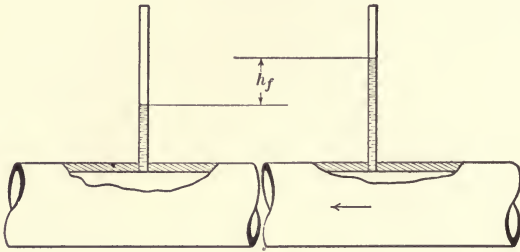


FIG. 139.—Loss in Head.

to which the air would rise if of uniform density due to the pressure of the pipe. If now there were two of these tubes, known as **piezometers**, placed at two points on a line as shown in Fig. 139, the difference in level would show the loss in pressure due to friction. The question of correctly measuring

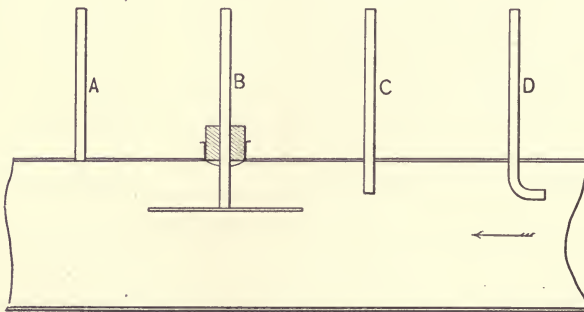


FIG. 140.—Piezometers.

the static pressure in a line carrying gas with some velocity is of importance. The best manner is to have a tube attached to the wall of the pipe absolutely perpendicular and finished flush with the surface as shown in Fig. A, 140, or a tube may be introduced as shown in the figure on which is a large, **normal disc** with sharp edges. The purpose of the disc is to cut out

eddies which form around any tube and give incorrect readings. Aspiration effect is usually produced by the tube entering as at *C*, Fig. 140, although there may be an increase of pressure due to impact of the air. The introduction of a tube as at *D*, Fig. 140, will give a pressure reading equal to the sum of the static pressure and the pressure due to velocity and for that reason it is sometimes called the dynamic pressure. The difference between the dynamic pressure and the static pressure is the velocity pressure or the **Pitot pressure**. A **Pitot tube** is a tube bent in the direction of the flow connected through a U

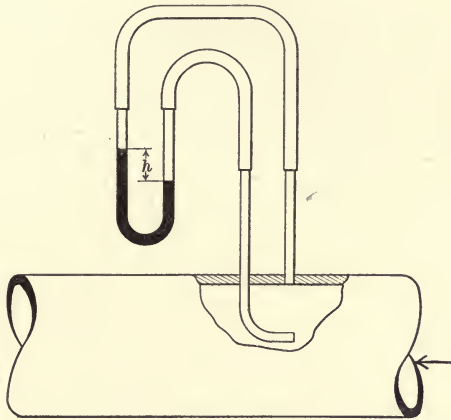


FIG. 141.—Pitot Tube.

tube to a static pressure tube, Fig. 141. In this way the pressure in inches of water due to the velocity may be found. Experiment and theory indicate that if this pressure be reduced to feet of the substance carried in the pipe the velocity is given by the equation

$$v_{\text{air}} = \sqrt{2gh_{\text{air}}} \text{ or } h_{\text{air}} = \frac{v_{\text{air}}^2}{2g} \quad \dots \quad (101)$$

h_{air} = feet of air equal to difference in pressure between the static and dynamic pressure;

v_{air} = velocity of air in feet per second;

$g = 32.174$ acceleration of gravity;

since

$$h_{air} = 69.5 h_{tw} \text{ (for } 70^\circ \text{ and atmospheric pressure)}$$

$$v_{air} = \sqrt{2g \times 69.5 h_{tw}} = 66.9 \sqrt{h_{tw}} \dots \dots \dots (102)$$

These two tubes are sometimes combined in one as shown in Fig. 142, but such a tube is apt to give incorrect readings because the static tube is subject to eddy currents. The objection to the single static tube at the wall is that the pressure may not be thought to be constant over the section of the flue, but experiment seems to indicate that the pressure is uniform.

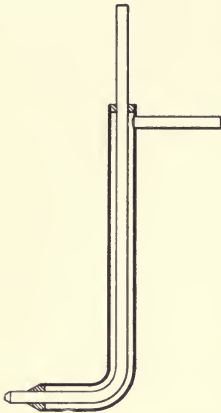


FIG. 142.—Arrangement of Pitot Tube with Static and Dynamic Tube Together.

The loss in pressure due to friction may now be discussed. The usual method is to express the loss in feet of head of the fluid being carried. Experiment then gives (beyond the critical velocity):

$$h_f \sim \frac{l}{r} v^2 \dots \dots \dots (103)$$

$$r = \frac{A}{p} = \frac{\frac{\pi d^2}{4}}{\pi d} = \frac{d}{4} \dots \dots \dots (104)$$

(A = area in square feet, p = perimeter in feet);
 l = length in feet.

Before reducing this it is well to note that since velocity pressure head is given by

$$h = \frac{v^2}{2g},$$

it is well to introduce $2g$ into the expression above for convenience and then reducing by use of the other quantities, the value of h_f becomes for round pipes,

$$h_f = f \frac{l p}{A} \frac{v^2}{2g} = f \frac{l}{d} \frac{v^2}{2g} \dots \dots \dots (105)$$

For square pipes the same expression holds, while for rectangular pipes of height d and breadth $2d$ the expression for r becomes

$$r = \frac{2d^2}{6d} = \frac{1}{3}d.$$

In this case f'' for such a pipe becomes

$$\frac{3}{4}f \text{ and } h_f = \frac{3}{4}f \frac{l}{d} \frac{v^2}{2g} \quad \dots \dots \dots (106)$$

The above discussions really apply to all fluids up to this point. In considering the problem of the flow of air, however, it must be remembered that the velocity V will vary along a pipe line, since the pressure falls and the volume increases as the air passes along the pipe line. Hence the above expression is only true for a differential length of pipe giving

$$-dh_f = f \frac{1}{d} \frac{v^2}{2g} dl = -dh \quad \dots \dots \dots (107)$$

The $-$ sign is used because h falls as l increases.

$$v = \frac{MRT}{144p} \div \frac{\pi d^2}{4}$$

M = pounds of air per second;

now

$$dh = \frac{144dp}{RT} = \frac{RT}{p} dp \quad \dots \dots \dots (108)$$

$$-dp = \frac{p}{RT} f \frac{1}{d} \left(\frac{MRT}{144p \frac{\pi d^2}{4}} \right)^2 \frac{dl}{2g} \quad \dots \dots \dots (109)$$

$$-p dp = \frac{fM^2RT}{(113)^2 d^5} \frac{dl}{2g} \quad \dots \dots \dots (110)$$

If T is assumed constant this integrates into

$$\frac{p_1^2 - p_2^2}{2} = \frac{fM^2RT}{(113)^2 d^5} \frac{L}{2g} \dots \dots (111)$$

p_1 = the pressure at entrance in pounds per square foot;

p_2 = the pressure at exit in pounds per square foot;

M = pounds of air per second;

T = absolute temperature of gas;

R = gas constant = $\frac{1544}{\text{mol. density}}$;

L = length in feet;

d = diameter in feet;

The above becomes

$$(p_1 - p_2) \frac{(p_1 + p_2)}{2} = f \frac{M^2RT}{(113)^2 d^5} \frac{L}{2g} \dots \dots (112)$$

$$\frac{p_1 + p_2}{2} = \text{mean pressure.}$$

$$p_1 - p_2 = \text{drop in pressure.}$$

Hence

Drop in pressure in pounds per square inch

$$= f \frac{M^2RT}{(113)^2 d^5 p_m} \frac{L}{2g} \dots \dots (113)$$

where p_m is mean pressure.

If $p_1 - p_2$ is reduced to feet of air at mean pressure this formula reduces to

Drop in feet of gas at density of mean pressure

$$= f \frac{L}{d} \frac{v^2}{2g} \dots \dots (114)$$

This is the same expression as that used for liquids where the specific volume does not change for a considerable change in pressure. In the case of air under pressure there may be a

considerable drop and only Eq. (111) or (113) may be used, but in cases where there is little change in pressure Eq. (114) may be used. This is the same as assuming that there is no change in velocity. Then

$$h = f \frac{l}{d} \frac{v^2}{2g} \dots \dots \dots (115)$$

The values of f vary with the velocity and the diameter of the pipe. According to Weisbach the constant is 0.0193 when h is in feet of gas, as l and d are in feet and V is in feet per second.

Weisbach proposes $f = 0.0144 + \frac{0.0172}{\sqrt{v}}$. . . (116)

Arson proposes $f = 0.03 + \frac{0.07}{v}$. . . (117)

Darcy proposes the formula $0.016 + \frac{0.005}{d}$. . (118)

and sometimes he gives this form

$$f = a + \frac{b}{d} + \frac{c}{v} + \frac{e}{vd} \dots \dots \dots (119)$$

The Green Economizer Company gives values which reduce this to

$$f = 0.0124 + \frac{0.0274}{v} + \frac{0.00145}{d} + \frac{0.0120}{dv} \dots \dots (120)$$

These formulæ give curves shown in Fig. 143, from which the average value is 0.02. For brick or concrete f is increased about 50 per cent. Formula 115 may be used for the flow of air in heating systems, since there is little change in the pressure.

The loss in pressure due to bends in pipe lines carrying fluids is usually expressed as

$$h_i = K \frac{v^2}{2g} \dots \dots \dots (121)$$

Where K is an experimental constant;

h_i = loss in feet of head;

v = velocity in feet per second;

The loss due to obstruction of various kinds, such as valves, grills or branches is given by the same kind of an expression.

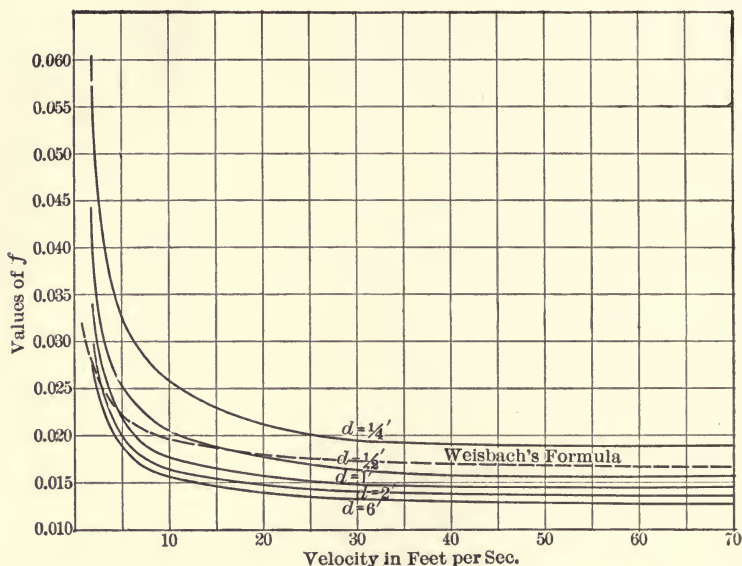


FIG. 143.—Values of f by Arson, Weisbach & Green Economizer Co.

The values of K used, as given by the Green Fuel Economizer Company are:

- 0.3 for sharp bends;
- 0.25 for bends with $R = d$;
- 0.15 for bends with $R = 2$ to $4d$;
- 0.07 for bends with $R = 5$ to $6d$;
- 0.00 for bends with $R > 6d$;
- 0.15 for branches at 135° ;
- 1.5 for grill or register, free area = $\frac{1}{2}$ total area = area of flue;
- 0.75 for grill, free area = $1\frac{1}{2}$ area flue.

At times the expression for the loss in bends is given as

$$h_b = f \frac{l}{d} \frac{v^2}{2g} \dots \dots \dots (122)$$

In which l is given as the length of an equivalent pipe. Since this expression is similar to the expression for the loss due to friction of straight pipe, one expression may be used for both if l represents the length of pipe plus the equivalent length of the bends.

N. S. Thompson gives the following equivalent lengths of bends in terms of the diameter or widths.

Radius of Quarter Bend.	Equivalent Length.
$0D$	$100D$
$\frac{1}{4}D$	$65D$
$\frac{1}{2}D$	$30D$
$1D$	$10D$
$1\frac{1}{2}D$	$6D$
$2D$	$5D$

For sudden enlargement, if there is little change in pressure the expression for loss is

$$h_e = \frac{(v_1 - v_2)^2}{2g} \dots \dots \dots (123)$$

Where v_1 is the velocity before enlargement while v_2 is the velocity after enlargement. For contraction the velocities refer to that in *vena contracta* after the contraction and that in the small pipe. These are difficult to find and this loss is, moreover, very small. The main loss is in enlargement and this with that due to contraction may be eliminated by gradual changes in section. From the above it is seen that the total head causing flow up to a given point is

$$h = f \frac{l}{d} \frac{v^2}{2g} + K_1 \frac{v^2}{2g} + K_2 \frac{v^2}{2g} + \frac{v^2}{2g} \dots \dots (124)$$

The first term is due to friction, the second to a bend, the third to a grill or branch and the last to the velocity. If there is little change in temperature and pressure the following is true approximately:

$$Q = v_1 A_1 = v_2 A_2 = v_3 A_3. \quad \quad (125)$$

Q = quantity per second in cubic feet;

v_1, v_2, v_3 = velocities at various sections in feet per second;

A_1, A_2, A_3 = areas at various sections in square feet.

Since the volume of a gas depends on absolute pressure and temperature the change from 120° F. to 110° F. would mean a change in the volume and hence in the velocity of

$$\frac{459.6 + 110}{459.6 + 120} = \frac{569.6}{579.6} = .983$$

or a difference of less than 2 per cent; while a change in pressure of 2 ins. of water to 0 in. would mean a change of

$$\frac{408 + 1}{408 + 2} = \frac{409}{410} = .995$$

or about $\frac{1}{2}$ per cent.

It is seen then that in most problems of indirect heating the pressure and temperature changes are so slight that formulae used for liquids may be employed and hence the velocity of any particular section may be expressed in terms of any other section of different size provided the quantity is the same. In most air pipes, however, the section is only changed when the quantity is changed. Hence in working out the total loss in any system the total loss may be written

$$h_l = \Sigma f_1 \frac{l_1}{d_1} \frac{v_1^2}{2g} + K_1 \frac{v_1^2}{2g} + \frac{v^2}{2g} \quad \quad (126)$$

In this the various losses are worked out for their different velocities and the $\frac{v^2}{2g}$ is the final velocity head. In this work

the sections change so gradually that there is no loss at such points.

The principal factor of the expression is that due to friction on the side walls or

$$h = f \frac{l}{d} \frac{v^2}{2g},$$

since

$$v = \frac{MRT}{\pi d^2} = \frac{K'M}{d^2} \text{ for any given condition.}$$

$$h = K'' \frac{lM^2}{d^5} \dots \dots \dots (127)$$

or

$$M = K''' \frac{h^{\frac{1}{2}} d^{\frac{5}{2}}}{l^{\frac{1}{2}}} \dots \dots \dots (128)$$

This shows that the head loss varies directly as the length and square of the mass and inversely as the fifth power of the diameter, or the mass handled varies directly as the $\frac{1}{2}$ power of the head, the $\frac{5}{2}$ power of the diameter and inversely as the $\frac{1}{2}$ power of the length. These two statements are important, as will be seen in laying out a system. For instance if the length of piping is the same, and the quantity is the same the diameter of two pipes will have to be to each other as $\left(\frac{1}{1.5}\right)^{\frac{1}{2}}$ if there is 50 per cent more drop in the first than in the second. If the drop and length are the same but the quantities are as 2 : 1 then the diameter will be as $(2)^{\frac{3}{2}} : 1$. These same statements as to diameter may be made of pipes of any form if they are geometrically similar. Thus if rectangular pipes are always made with a given ratio between the two dimensions and the smaller one is called d the expression

$$h_f = K'''' \frac{lM^2}{d^5} \text{ is true,}$$

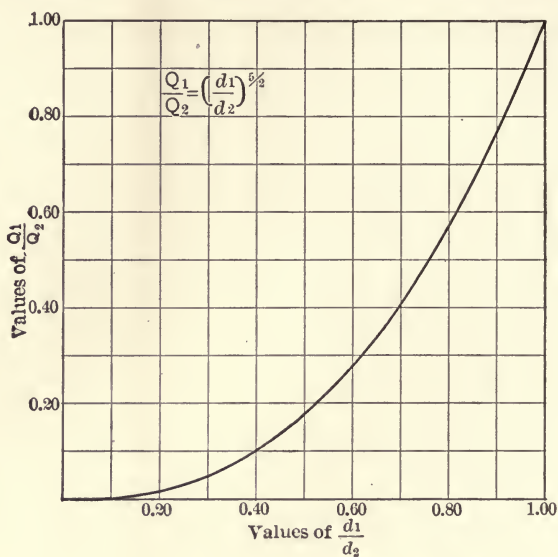


FIG. 144.—Variation of $\frac{Q_1}{Q_2}$ for Different Values of $\frac{d_1}{d_2}$ for Same Friction Loss and Same Length.

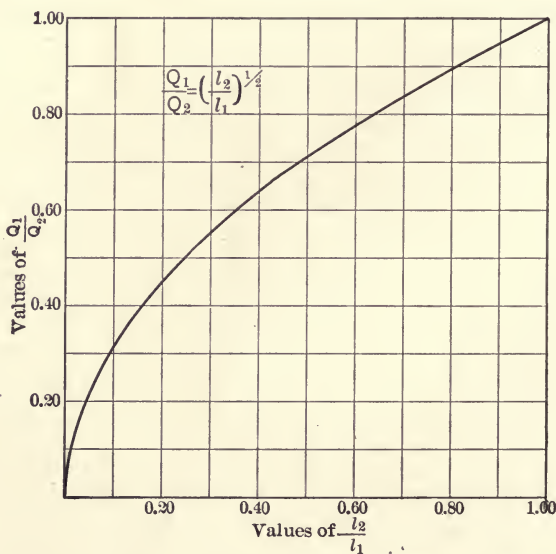


FIG. 145.—Variation of $\frac{Q_1}{Q_2}$ with Different Values of $\frac{l_2}{l_1}$ for Same Friction Loss and Same Diameter.

Fig. 144, which gives $\frac{Q_1}{Q_2}$ plotted as $\left(\frac{d_1}{d_2}\right)^{\frac{5}{8}}$, Fig. 145 with $\frac{Q_1}{Q_2}$, as

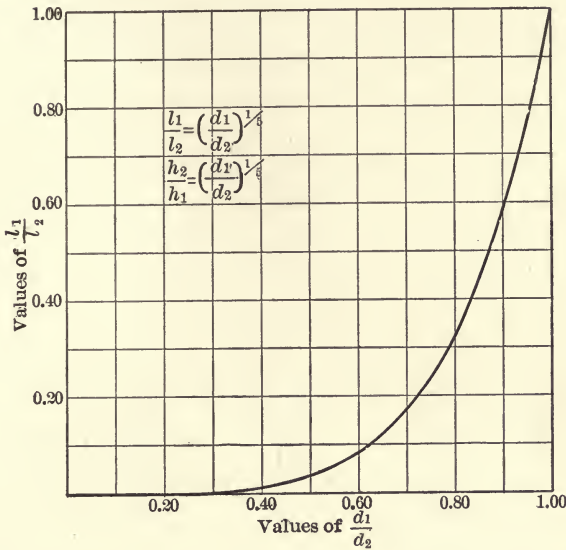


FIG. 146.—Variation of $\frac{d_1}{d_2}$ with Different Values of $\frac{l_1}{l_2}$ for Same Friction Loss and Quantity. (This curve may be used for $\frac{h_2}{h_1}$ for different values of $\frac{d_1}{d_2}$ for same length and quantity.)

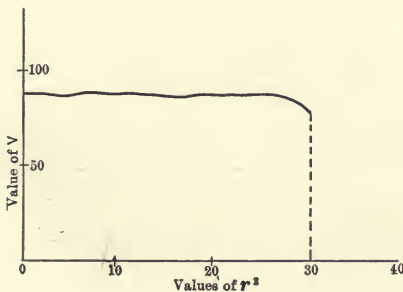


FIG. 147.—Diagram for Discharge from Pipe.

$\left(\frac{l_2}{l_1}\right)^{\frac{1}{2}}$, and Fig. 146, with $\left(\frac{d_1}{d_2}\right)$ plotted as $\left(\frac{l_1}{l_2}\right)^{\frac{1}{6}}$ are of value in working out problems of relative duct sizes.

The further use of this will be explained later.

It may be necessary to measure the quantity of air flowing and to do this there are five general methods: first the **Pitot tube**, second a **standard orifice**, third an **anemometer**, fourth a **Venturi meter** and fifth an **electric meter**.

The Pitot tube has been described. By means of this instrument the velocities at different points in the pipe are found and if these velocities are supposed to remain constant around any given radius the following expression will give the quantity Q :

$$Q = \int v_2 \pi r dr$$

$$= \pi \int_0^r v d(r^2) \dots \dots \dots (129)$$

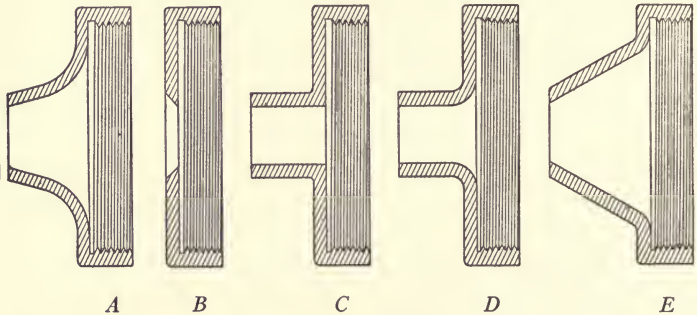


FIG. 148.—Orifices.

Hence if the values of v be plotted as ordinates against the values of r^2 of the points as abscissae the area of the curve, Fig. 147, when multiplied by π will give the value of Q . In measuring Q in this way more readings should be taken near the edge of the pipe than near the center.

In the second method, that of the **standard orifice**, Fig. 148, the velocity is determined by the thermodynamic equation:

$$\text{vel.} = \sqrt{\frac{2g}{A}(i_1 - i_2)} = \sqrt{2g \frac{k}{k-1} (p_1 v_1 - p_2 v_2)}$$

vel. = vel. in feet per second;
 i = heat content in B.t.u.;

p = pressure in pounds per square foot.
 v = vol. of 1 lb. in cubic feet;

$$k = \frac{c_p}{c_v} = 1.40 = \text{ratio of specific heats;}$$

$$A = \frac{1}{778}.$$

Now

$$p_1 v_1^k = p_2 v_2^k.$$

Hence

$$\text{vel.} = \sqrt{2g \frac{k}{k-1} p_1 v_1 \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{k-1}{k}} \right]}. \quad \dots \quad (130)$$

The quantity discharged in pounds per second is given by

$$Q = a \times \text{vel.};$$

$$M = \frac{Q}{v_2} = \frac{a \text{ vel.}}{v_2};$$

where

a = area of orifice;

$$\begin{aligned} M &= \frac{a}{v_2} \sqrt{2g \frac{k}{k-1} p_1 v_1 \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{k-1}{k}} \right]} \\ &= a \left(2g \frac{k}{k-1} \right)^{\frac{1}{2}} \left(\frac{p_1}{v_1} \right)^{\frac{1}{2}} \left[\left(\frac{p_2}{p_1} \right)^{\frac{2}{k}} - \left(\frac{p_2}{p_1} \right)^{\frac{k+1}{k}} \right]^{\frac{1}{2}} \dots \quad (131) \end{aligned}$$

When $p_2 = 0.528p$ the maximum discharge occurs.

Fliegner found by experiment that $p_2 = 0.5767p_1$ at the point of maximum discharge, and this as well as the theoretic value for maximum discharge reduces Eq. (131) to

$$M = 0.530a \frac{p_1}{\sqrt{T_1}}, \quad \dots \quad (132)$$

where

M = pounds per second;

a = area in square inches;

p_1 = pounds per square inch of high pressure

T_1 = absolute temperature.

The weight for any pressure $p_2 < 0.5p_1$ is the same as above, so that so long as the pressure p_2 is below its critical value the discharge in pounds is a fixed quantity and independent of p_2 . Above this critical value the discharge does depend on p_2 .

Fliegner gives this equation in the form,

$$M = 1.060a\sqrt{\frac{p_2(p_1 - p_2)}{T_1}} \quad (133)$$

This equation is for a rounded orifice. For a sharp-edged orifice in a thin plate a constant of 0.62 is used, by which to multiply the theoretic discharge. The constant is about 0.83 for short tubes and 0.92 when these are rounded at entrance.

The **anemometer**, Fig. 149, is only of value when the velocity is not over 1200 to 1500 ft. per minute and hence it is used often in heating work where these velocities are found. The instrument is checked by revolving it on a long arm and noting the distance moved and that recorded. In this way the instrument is calibrated. In using this apparatus the readings may be plotted as velocities on a diagram similar to that used with the Pitot tube or the area of a pipe or duct may be divided into

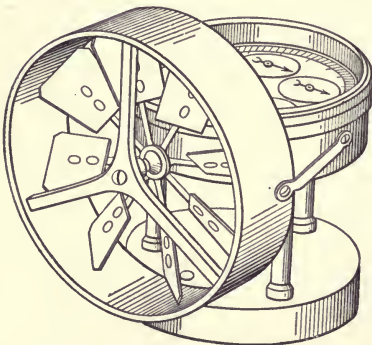


FIG. 149.—Anemometer.

square areas by imaginary lines and the velocity determined in each of them. The average of these will give the average velocity. Another method is to gradually move the anemometer over the area of the duct or pipe covering the whole pipe in this way and the recorded amount per minute will give the average velocity.

The **Venturi meter**, Fig. 150, consists of a converging and diverging section of pipe. The diverging section is more gradual on account of the loss due to enlargement being greater than that due to contraction.

By equating the sum of the energies at the large and small sections and remembering that the expansion is adiabatic the formula for the discharge may be derived. Thus:

$$\frac{p_1 v_1}{k-1} + p_1 v_1 + \frac{\text{vel.}_1^2}{2g} = \frac{p_2 v_2}{k-1} + p_2 v_2 + \frac{\text{vel.}_2^2}{2g};$$

$$p_1 v_1^k = p_2 v_2^k;$$

$$\frac{a_1 \text{ vel.}_1}{v_1} = \frac{a_2 \text{ vel.}_2}{v_2};$$

$$\frac{k}{k-1} (p_1 v_1 - p_2 v_2) = \frac{\text{vel.}_1^2}{2g} \left[\left(\frac{a_1 v_2}{a_2 v_1} \right)^2 - 1 \right];$$

$$\text{vel.}_1 = \sqrt{2g \frac{\frac{k}{k-1} p_1 v_1 \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{k-1}{k}} \right]}{\left[\frac{a_1}{a_2} \left(\frac{p_1}{p_2} \right)^{\frac{1}{k}} \right]^2 - 1}}}. \quad (134)$$

$$M = \frac{a_1 \text{ vel.}_1}{v_1} \dots \dots \dots (135)$$

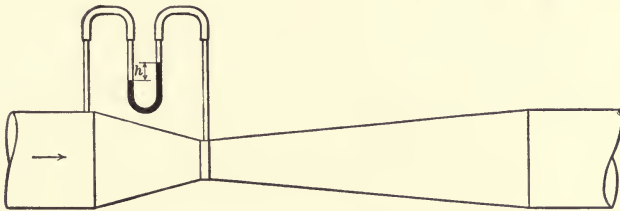


FIG. 150.—Venturi Meter.

- p_1 = pressure in pounds per square foot at section 1;
- p_2 = pressure in pounds per square foot at section 2;
- v_1 = specific volume in cubic feet;
- v_2 = specific volume in cubic feet;
- k = ratio of specific heats;
- vel._1 = velocity in feet per second;
- a = area in square feet;
- g = acceleration of gravity;
- M = mass per second.

The methods thus used are applicable under different conditions. The anemometer is applicable to low velocities up to 1500 ft. per minute, the Pitot tube to velocities of 2400 to 7000 ft. per minute, and the standard orifice to higher velocities when the pressure drop is considerable. The Venturi meter is applicable to steady flow of all velocities.

A recent method used by Prof. C. Thomas consists in heating gas by a known amount of electrical energy and then by the rise in temperature finding the mass of gas, and from it the volume and thus the velocity.

$$\frac{\text{Volts} \times \text{amp.} \times \frac{42.42}{60}}{\frac{746}{.239(t_2 - t_1)}} = M \quad (136)$$

M = mass of air per second;
 volts = average voltage;
 amp. = average current;
 t_2 = temperature of outlet;
 t_1 = temperature of inlet;

$$\text{vol.} = \frac{MRT}{144p};$$

$$\frac{\text{vol.}}{\text{area}} = \text{velocity.}$$

If the head lost is h ft. of air the work required to overcome this per second if Q cu.ft. per second are discharged and each cubic foot weighs w lbs. is

$$\text{work per second} = whQ = \frac{144p}{RT} hQ \quad . . . (137)$$

$$\text{H.P.} = \frac{whQ}{550} = \frac{144ph}{550RT} Q \quad (138)$$

The formulae may now be applied for the following tables:

TABLES TO BE USED IN DESIGN WORK

PRESSURES AND VELOCITIES OF AIR AT 70° F.

Inches Water.	Ounces Pressure.	Velocity, Feet per Second.	Inches Water.	Ounces Pressure.	Velocity, Feet per Second.	Inches Water.	Ounces Pressure.	Velocity Feet per Second.
$\frac{1}{16}$	0.04	16.7	$\frac{7}{16}$	0.25	44.2	$1\frac{3}{4}$	1.01	89
$\frac{1}{8}$	0.07	23.6	$\frac{1}{2}$	0.29	47.3	2	1.16	95
$\frac{3}{16}$	0.11	28.9	$\frac{5}{8}$	0.36	52.8	3	1.73	116
$\frac{1}{4}$	0.14	33.4	$\frac{3}{4}$	0.43	57.9	4	2.31	134
$\frac{5}{16}$	0.18	37.3	$\frac{7}{8}$	0.51	60.7	6	3.47	167
$\frac{3}{8}$	0.22	40.9	1	0.58	66.9	8	4.63	189
			$1\frac{1}{4}$	0.72	74.9	12	6.94	232
			$1\frac{1}{2}$	0.87	82.0	16	9.25	368

DIAMETER OF PIPES FOR VARIOUS CAPACITIES AND VELOCITIES

Cubic Feet of Air per Minute.	500	600	700	850	1000	1200	1400	1700	2000	2500
200	8.6	7.83	7.2	6.6	6.0	5.5	5.1	4.7	4.3	3.9
400	12.2	11.1	10.3	9.3	8.6	7.8	7.3	6.6	6.1	5.5
700	16.0	15.1	13.6	12.3	11.3	10.4	9.6	8.8	8.1	7.2
1000	19.3	17.5	16.2	14.8	13.5	12.4	11.5	10.5	9.6	8.6
1500	23.5	21.4	19.9	18.0	16.6	15.2	14.1	12.8	11.8	10.5
2000	27.1	24.7	22.9	20.8	19.2	17.9	16.2	14.8	13.6	12.2
3000	33.2	30.3	28.0	25.5	23.5	21.4	19.9	18.1	16.7	14.9
4000	39.3	35.0	32.4	29.4	27.1	24.7	23.0	20.8	19.2	17.2
5000	42.8	39.2	36.3	32.9	30.3	27.8	25.5	23.3	21.5	19.2
7500	52.5	47.9	44.3	40.2	35.8	34.0	31.5	28.7	26.3	23.5
10000	61.0	55.3	51.2	46.6	42.8	39.2	36.3	32.9	30.4	27.2
15000	74.2	67.8	62.9	57.0	52.5	47.9	43.9	40.3	37.1	33.2
20000	85.6	78.3	72.3	65.7	60.0	55.4	51.2	46.4	42.9	38.4
30000	105.0	95.8	88.6	80.5	74.2	67.7	62.7	57.0	52.6	47.0
40000	121.0	111.0	103.0	92.9	86.0	78.1	72.3	65.7	60.6	54.2
50000	135.0	124.0	115.0	105.0	95.8	87.8	81.3	73.6	68.0	60.7
62500	150.0	138.0	129.0	115.0	107.0	98.0	91.0	82.3	75.9	67.8
75000	166.0	151.0	140.0	127.0	117.0	108.0	100.0	90.2	83.0	74.2
100000	193.0	175.0	162.0	148.0	135.0	124.0	115.0	105.0	96.0	85.8

DIAMETERS OF
CIRCULAR AREAS EQUIVALENT TO RECTANGULAR DUCTS IN CARRYING CAPACITY

Length	Width.										
	3	4	6	9	12	15	18	21	24	30	36
3	3.7										
4	3.9	4.4									
6	4.6	5.4	6.7								
9	5.6	6.6	8.1	10.0							
12	6.3	7.4	9.2	11.5	13.3						
15	7.0	8.3	10.3	12.7	14.7	16.5					
18	7.6	8.9	11.2	13.9	16.2	18.1	19.7				
21	8.1	9.5	12.0	15.0	17.5	19.5	20.4	23.6			
24	8.5	10.0	12.7	15.8	18.4	20.8	22.9	24.7	26.4		
30	9.4	11.1	14.0	17.5	20.5	23.1	25.5	27.6	29.5	33.0	
36	10.1	11.9	15.2	19.1	22.2	25.2	27.8	30.0	32.3	36.4	39.9

The loss in inches of water in 100 ft. of 12-in. pipe at various velocities is given in the table below as well as the horse-power.

To use this table for any other diameter or length the values are multiplied by one one-hundredths of the length and divided by one-twelfth of the diameter for loss of head, while for horse-powers the tabular values are multiplied by the two factors. Thus for 2000 ft. per minute in 75 ft. of 8-in. pipe

$$h_l = 0.51 \times \frac{75}{100} \times \frac{12}{8} = 0.57 \text{ in.}$$

$$\text{H.P.} = 0.1269 \times \frac{75}{100} \times \frac{8}{12} = 0.0634.$$

FRICTION LOSS AND H.P. PER 100 FEET OF 12-INCH PIPE

Velocity.	Friction, Inches, Water.	H.P.	Velocity.	Friction, Inches, Water.	H.P.
200	0.005	0.0002	1200	0.184	0.0270
400	0.021	0.0011	1500	0.288	0.0536
600	0.046	0.0035	2000	0.510	0.1269
800	0.082	0.0082	3000	1.150	0.4284
1000	0.128	0.0159	4000	2.050	1.0153

This table for friction loss holds for 12-in. square pipe.

To show the effect of temperature on the various quantities the following table is given:

EFFECT OF TEMPERATURE ON AIR PROBLEMS

Temperature, Degrees F.	Relative Pressure for Same Head of Air.	Relative Velocity for Same Ounce Pressure.	Relative Weight for Same Volume.	Relative Volume for Same Weight.	Relative Power for Same Velocity.	Relative Power for Same Weight.
20	1.10	0.95	1.10	0.91	1.10	0.83
30	1.08	0.96	1.08	0.93	1.08	0.87
40	1.06	0.97	1.06	0.94	1.06	0.89
50	1.04	0.98	1.04	0.96	1.04	0.92
60	1.02	0.99	1.02	0.98	1.02	0.96
70	1.00	1.00	1.00	1.00	1.00	1.00
80	0.98	1.01	0.98	1.02	0.98	1.04
90	0.96	1.02	0.96	1.04	0.96	1.08
100	0.95	1.03	0.95	1.05	0.95	1.10
110	0.93	1.04	0.93	1.07	0.93	1.14
120	0.92	1.05	0.92	1.09	0.92	1.18
130	0.90	1.06	0.90	1.11	0.90	1.23
140	0.88	1.06	0.88	1.13	0.88	1.28
150	0.87	1.07	0.87	1.15	0.87	1.32
200	0.80	1.11	0.80	1.25	0.80	1.56
250	0.75	1.16	0.75	1.33	0.75	1.78
400	0.62	1.27	0.62	1.61	0.62	2.60
600	0.50	1.41	0.50	2.00	0.50	4.00

The data for the losses in pipes being known it now becomes necessary to outline the method of procedure for a given plant.

Register Faces. The register faces should be large enough to give a velocity of 300 to 550 ft. per minute when in the wall, while 200 to 250 is the value to be used for floor registers. These must be selected so that there is no annoyance to the occupants of the room. The net area of the register is usually 66 per cent of the area of the opening in the end of the duct while there is a border of about 2 ins. around this. A 10 by 10 opening would give a net register area of 66 sq.in. and would have an extreme register face area of about 144 sq.in.

As a guide the following table of velocities has been computed from data given in Loomis' Meteorology by Carpenter:

Just perceptible	175 ft. per minute.
Gently pleasant	330 " "
Pleasant*brisk	1100 " "
Very brisk	2200 " "

High wind	3100 ft. per minute.
Very high wind	4000 " "
Strong gale	5000 " "

Flues and Ducts. After the registers the flues may be chosen with velocities of 500 to 750 and finally the horizontal flues with 800 to 1200 ft. per minute. In any case low velocities are of value, as the loss depends on the square of the velocity, and the only limit is the cost of the ducts and the space available for them. Of course for lack of space it may be necessary to increase the velocities. The high velocities are limited by noise. The values given above will give a quiet system.

To get the size of the ducts it is well to lay out the system as shown in Figs. 136, 137, assuming the velocities to one of the rooms, usually the most remote, and from that work out the other circuits.

In Fig. 136 a **double-duct system** is assumed in which one-half of the air is hot and one-half tempered. Assuming the velocity in the various parts of the system to room 18, above room 12, the pressures are found at the various points. These calculations are now given.

$$\text{Pressure in rooms} = h_w;$$

$$\text{Velocity at register} = 300 \text{ ft. per min.} = 5 \text{ ft. per sec.}$$

$$\text{Area register} = \frac{88000}{60 \times 300} = 4.9 \text{ sq.ft.};$$

$$\text{Velocity head} = \frac{25}{64.3} = 0.4 \text{ ft. air} = 0.005 \text{ in. water};$$

$$\text{Velocity in flue} = 600;$$

$$\text{Area of flue} = \frac{88000}{600 \times 60} = 2.4 \text{ sq.ft.};$$

$$\text{Loss in brick flue with 2 bends} = \left(2 \times 0.25 + \frac{0.02 \times 38}{1.5} \right) 1.8 \frac{10^2}{64.3} =$$

$$(.5 + .5) \frac{180}{64.3} = 2.8 \text{ ft. of air} = 0.04 \text{ in. water.}$$

Velocity in ducts 1200 ft. per minute.

$$\text{Area of tempered ducts from } A \text{ to } B = \frac{151000}{1200 \times 60} = 2.1 \text{ sq.ft.};$$

(assuming 50,000 cu.ft. per room)

$$\text{From } B \text{ to } C = \frac{302000}{1200 \times 60} = 4.2 \text{ sq.ft.}$$

$$\text{From } C \text{ to heater } \frac{904000}{1200 \times 60} = 12.6 \text{ sq.ft.}$$

Area of hot ducts at 98° F.

$$\text{From } A \text{ to } B = 2.1 \times 1.05 = 2.2.$$

$$\text{From } B \text{ to } C = 4.2 \times 1.05 = 4.4.$$

$$\text{From } C \text{ to heater} = 12.6 \times 1.05 = 13.2.$$

Losses:

Tempered A to B [$12 \times 24''$].

$$\left(0.02 \times \frac{22}{0.33} + 3 \times 0.25\right) \frac{20^2}{64.3} = 12' \text{ of air.}$$

Tempered B to C [$12 \times 48''$].

$$\left(0.02 \times \frac{32}{0.4}\right) \frac{20^2}{64.3} = 10' \text{ of air.}$$

Tempered C to fan [$30 \times 63''$].

$$\left(0.02 \times \frac{28}{0.85} + 2 \times 0.25\right) \frac{20^2}{64.3} = 8.5' \text{ of air.}$$

The pressure heads for the hot air in feet of air will be the same, although when reduced to oz. pressure they will amount to 4 per cent less on account of the higher temperature of air. This decrease will be used, however, in the heating coils over which the hot air will flow so that the resistance is the same in each duct.

The flues now leading from A to second floor and from A

to first floor have to be so designed that they will give the proper discharge, for although there is the same drop in each of them the lengths are different and hence to get the proper quantity, the area of the flues leading to the lower floors is smaller. The quantities are the same for this same drop, hence

$$\frac{L}{D^5} = \frac{L_1}{D_1^5};$$

$$D_1 = D \left(\frac{L_1}{L} \right)^{\frac{1}{5}} = 1\frac{1}{2} \left(\frac{25 + 10 \times 1\frac{1}{4}}{38 + 10 \times 1\frac{1}{2}} \right)^{\frac{1}{5}};$$

$$D_1 = 1.32.$$

($1\frac{1}{4}$ is assumed as D_1 in getting the equivalent length for the quarter bends.)

$$D_2 = 1\frac{1}{2} \left(\frac{12 + 10 \times 1\frac{1}{4}}{38 + 10 \times 1\frac{1}{2}} \right)^{\frac{1}{5}};$$

$$D_2 = 1.26.$$

To get the size of the flues at *C* it must be remembered that the quantity is the same as before, but the pressure drop is greater since the pressure in the rooms is the same while the pressures at the bases of the flues are different.

$$\left(\frac{D}{D_1} \right) = \left(\frac{h_1}{h} \right)^{\frac{1}{5}} \quad \text{or} \quad D_1' = D \left(\frac{h}{h_1} \right)^{\frac{1}{5}}.$$

h for flues at *A* = 2.8 ft. air;

h for flues at *C* = 2.8 + 12 + 10 = 24.8 ft. air.

$$D_1' = 1\frac{1}{2} \left(\frac{2.8}{24.8} \right)^{\frac{1}{5}} = 0.87;$$

$$D_1' = 1.32 \left(\frac{2.8}{24.8} \right)^{\frac{1}{5}} = 0.76;$$

$$D_2' = 1.26 \left(\frac{2.8}{24.8} \right)^{\frac{1}{5}} = 0.73.$$

To aid in problems of this kind Figs. 144-146 have been added, giving the ratios of Q_s , l_s , d_s .

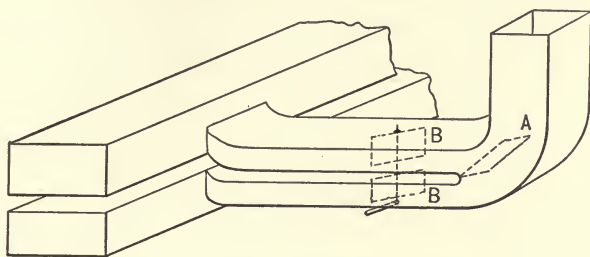


FIG. 151.—Double Dust Dampers.

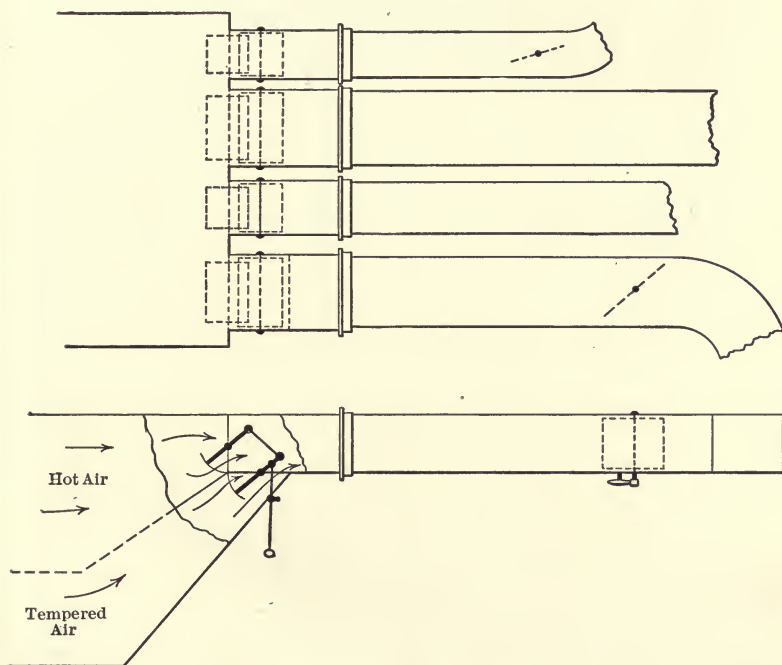


FIG. 152.—Single Duct System.

In many cases the flues are all made of the same size and in addition to the mixing damper *A*, shown in Fig. 151, there are a pair of dampers, *B*, shown in the figure, which may be locked

in any position, thus checking the flow in any room and equalizing the discharge.

If the job is equipped with single ducts, as shown in Fig. 153, then one duct is computed, for instance that to room 18, with a velocity of 840 ft. per minute, and then since the pressure

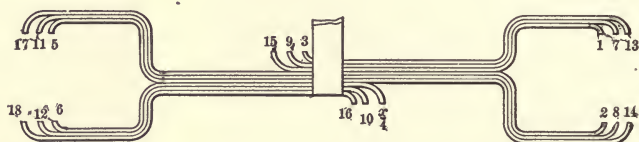


FIG. 153.—Single Ducts for School No. 3.

drop to each room is the same with practically the same quantity, the sizes of others are given by

$$D_x = D \left(\frac{L_x}{L} \right)^{\frac{1}{5}}.$$

Now the size of duct to room 18 is

$$\text{Area} = \frac{88200}{840 \times 60} = 1.7;$$

$$\text{Loss} = \left[0.02 \times \frac{(46 + 30 \times 1\frac{1}{4})}{.3} + \frac{0.03 \times (38 + 25)}{.3} \right] \frac{14^2}{64.3} = 35' \text{ of air.}$$

For room 12

$$\frac{h}{h_1} = \frac{0.02(48 + 38) + 0.03(38 + 25)}{0.02(46 + 38) + 0.03(25 + 25)}.$$

Using this the size of flue and duct are found. Fig. 152 shows the mixing dampers used in the single-duct method with the control damper for discharge.

Loss of Pressure in Heaters. The losses in the various heaters have been found by the manufacturers and some are given in the tables below:

LOSS IN PRESSURE IN BUFFALO HEATERS

Velocity through Clear Area in Feet per Minute.	Loss in Ounces per Square Inch per Section of 4 Rows.	Loss in Inches of Water per Section of 4 Rows.
700	0.027	0.047
800	0.035	0.061
900	0.045	0.078
1000	0.055	0.095
1100	0.067	0.116
1200	0.080	0.138
1300	0.093	0.162
1400	0.104	0.180
1500	0.127	0.220

The pressure-loss in heaters should not exceed one-half total loss in head.

LOSS IN PRESSURE IN GREEN POSITIVFLO HEATERS WITH PIPES ON 2 $\frac{3}{4}$ " CENTERS

Air Velocity in Feet per Minute.	Loss in Inches of Water.					
	4 Rows.	8 Rows.	12 Rows.	16 Rows.	20 Rows.	24 Rows.
600	0.04	0.06	0.09	0.12	0.14	0.15
800	0.06	0.10	0.15	0.19	0.23	0.26
1000	0.09	0.15	0.23	0.30	0.37	0.41
1200	0.12	0.21	0.31	0.43	0.56	0.58
1400	0.17	0.30	0.45	0.60	0.75	0.90
1600	0.20	0.34	0.52	0.69	0.86	1.03

$$\text{Free air space} = \frac{\text{Lineal feet of pipe}}{8.4 \times \text{rows of pipe}}$$

LOSS IN PRESSURE IN VENTO HEATERS IN INCHES OF WATER

Velocity through Clear Area in Feet per Minute.	1 Section.	2 Section.	3 Section.	n Sections.
400	0.015	0.018	0.018	0.018 + 0.008(n-2)
600	0.033	0.041	0.058	0.041 + 0.0175(n-2)
800	0.059	0.072	0.104	0.072 + 0.032(n-2)
1000	0.092	0.112	0.162	0.112 + 0.050(n-2)
1200	0.133	0.162	0.234	0.162 + 0.072(n-2)
1400	0.180	0.220	0.318	0.220 + 0.098(n-2)
1600	0.236	0.288	0.416	0.288 + 0.128(n-2)

Fans. There are several forms of fans used for mechanical ventilation. The **plate fan** used for many years consists of a **wheel** with a number of **radial paddles** or **vanes**, Fig. 154 *A*, or a number of **curved vanes**, Fig. 154 *B*, enclosed in a **metal casing**. These were used for many years and in 1897 Mr. S. C. Davidson of Belfast, Ireland, invented a **curved vane wheel** in which the

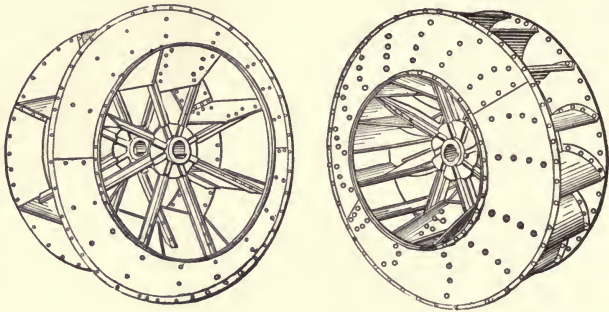


FIG. 154.—Radial Vanes on Standard Form of Fan. Curved Vanes on I-Beam Wheel.

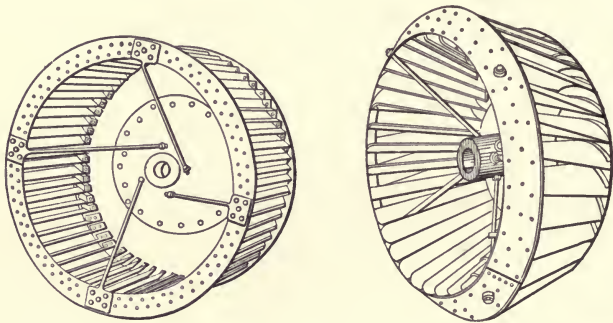


FIG. 155.—Sirocco Fan.

FIG. 156.—Conoidal Fan.

blades were not so deep, Fig. 155, which he called the **sirocco fan**. This has been followed by the **conoidal fans** of the Buffalo Forge Company, Fig. 156. These are all used for forcing air. The **cone wheel**, Fig. 157, is used mainly on the vacuum system for sucking air from a given space. The **cone center** serves to guide the air from the center. The **propellor** or **disc fan**, Fig. 157, is used to exhaust air from a room. It is usually

placed in the partition or wall of the room from which it draws vitiated air and delivers it to the atmosphere.

Fans are made with discharges in different directions and at times they may have more than one outlet. In Fig. 159 different arrangements are shown. *A* represents a full housing top horizontal discharge, *B* is a full-housed top vertical discharge, *C* is a three-quarter-housed bottom angular discharge, *D*, a three-quarter-housed top angular discharge and *F* a full-housed multiple discharge fan. *E* represents a full-housed bottom vertical discharge. Fans are made either full or three-

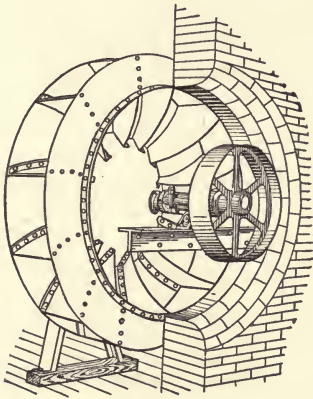


FIG. 157.—Cone Wheel.

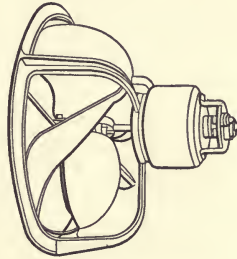


FIG. 158.—Ventilator.

quarter housed with any methods of discharge to suit conditions. They are driven by **direct-connected** motors or engines or they may be **belted** to a prime mover.

The manufacturers call a fan **right** or **left** if on facing the pulley side of the fan the discharge is to the right or left.

Fans of the type shown in Fig. 154 are usually designed so that the **peripheral speed** of the fan wheel is equal to that produced by the **dynamic pressure** at the point of maximum efficiency. At this point the **static pressure** is equal to about 75 per cent of the dynamic pressure while the **velocity head** is about 25 per cent of the pressure. As the resistance is decreased the cubic feet delivery is increased and the total pressure falls until,

when there is no resistance around the fan, free discharge occurs with total zero pressure head, 50 per cent of the total head at best efficiency. This is all velocity head. The discharge at

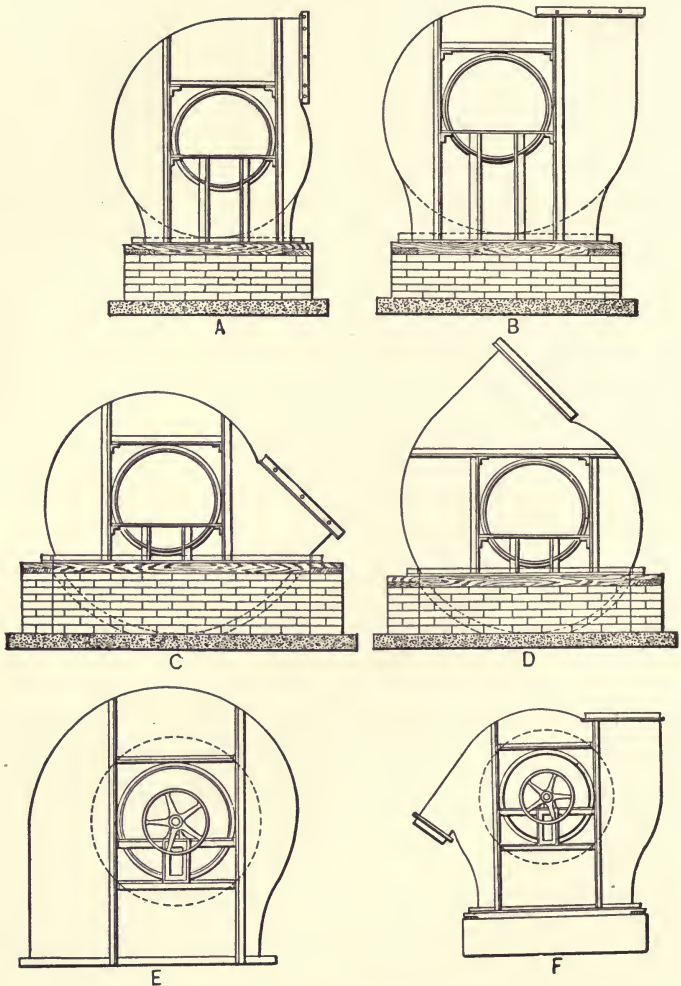


FIG. 159.—Forms of Housings.

this time is 145 per cent of the discharge at the best point. If on the other hand the discharge is closed off the static head increases until it equals the total pressure at zero discharge.

At this point the pressure is 116 per cent of that due to a velocity equal to the peripheral speed of the wheel. The curve for the Buffalo Forge Company fans showing this change in pressure as the discharge is throttled is given in Fig. 160. This curve has been prepared from one given by the company from results of tests.

If in any case the resistance in the flues, ducts, heaters and

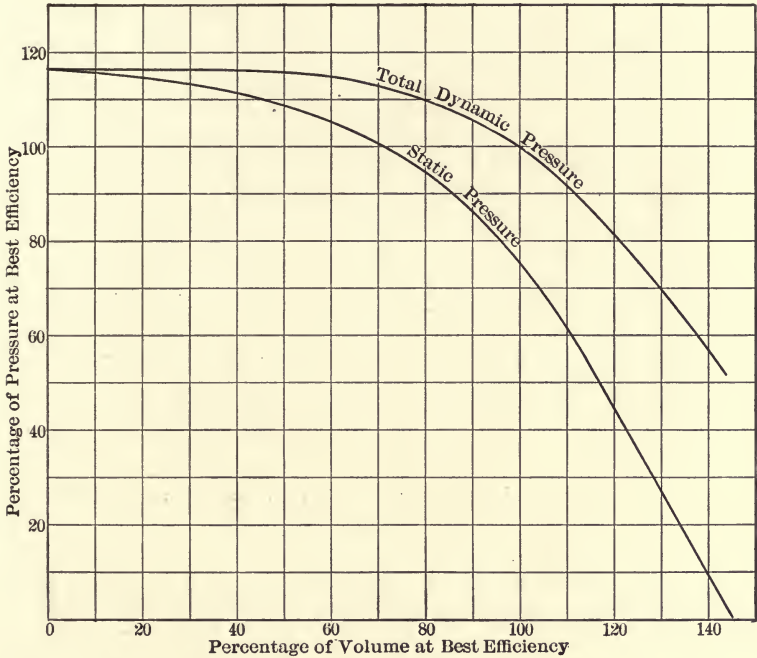


FIG. 160.—Relation between Pressure and Volume in Percentage of the Pressure and Volume from Fan at Maximum Efficiency as Given by Curves of the Buffalo Forge Co.

casing amounts to h_{tw} of water, this quantity represents 75 per cent of the total head produced by the wheel, which is equal to the velocity head of the peripheral speed.

$$\frac{h_{tw}}{1.728} = p_o \dots \dots \dots (98)$$

$$\frac{p_o}{0.75} = P \text{ (Total pressure).} \dots \dots (139)$$

$$\text{Peripheral velocity of wheel} = 66.9 \sqrt{\frac{h''_{4w}}{0.75}} \dots \dots \dots (101)$$

$$= 66.9 \sqrt{\frac{1.728}{0.75} p_0} \dots \dots \dots (140)$$

$$= \pi d \frac{N}{60}$$

where d = diameter of wheel in feet, N = R.P.M. Having the **peripheral speed** of the wheel, the diameter and number of revolutions must be found. This is determined by the **square inches of blast area** or “blast.”

This area is that through which the fan will discharge and give a velocity equal to the peripheral velocity of the wheel.

In most plate fans it is equal to $\frac{wd}{3}$ where w is the width of the blade at the tip, and d is the diameter. Now if Q is the cubic feet discharged per second, and v_p is the peripheral speed of the wheel in feet per second, the area of blast is

$$a = \frac{Q}{v_p} = \frac{wd}{3} \dots \dots \dots (143)$$

Since w is made equal to $0.4d$, while the widest part is $0.5d$, the value of a becomes

$$a = \frac{0.4d^2}{3} = 0.133d^2.$$

Hence

$$0.133d^2 = \frac{Q}{v_p},$$

and d may be found and then N . The width at body and tip are also known. With high pressures and small quantities the width is made smaller than $0.5d$, while for large quantities under small heads the width is made large.

The inlet area in square feet is given by Parsons as

$$a_{\text{in}} = \frac{0.00054Q}{\sqrt{h_{4w}}} < 0.31d^2.$$

Outlet area is given by

$A_{\text{out}} = 1.0$ to 1.25 inlet area, or this may be calculated with the flues and ducts.

The radial depth of the blade near the edge is about $0.15D$, although another way would be to have the corner *A*, Fig. 161, fit $\frac{1}{2}$ in. within the inlet opening with a clearance of about

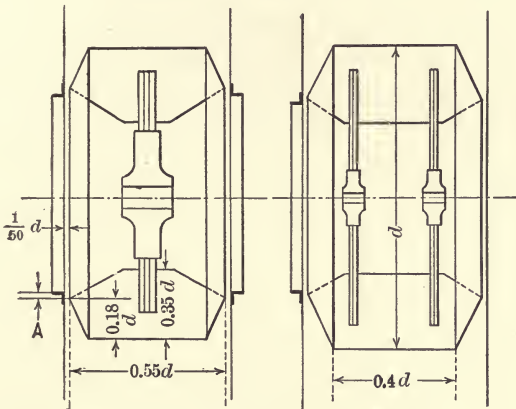


FIG. 161.—Approximate Wheel Dimensions.

$\frac{5}{8}$ in. from the casing. This with the outer diameter will fix the radial depth.

The power generated by a fan consists of three parts: (a) that required to give the air velocity, (b) that required to change the intrinsic energy, and (c) that to do the external work.

The first amount is given by

$$\text{Work per second} = M \frac{v^2}{2g} = MK^2 \frac{V^2}{2g}. \quad \dots (141)$$

M = mass per second,

v = velocity of discharge in feet per second;

V = peripheral velocity of fan;

K = percentage factor.

The second item is given by

$$\text{Work per second} = M \left[\frac{p_o v_o}{0.405} - \frac{p_i v_i}{0.405} \right]. \quad \dots \quad (142)$$

p_o, p_i , the pressure per square foot at outlet, inlet, in pounds.

v_o, v_i , the volumes of one pound of air in cubic feet.

The third amount is given by

$$\text{Work per second} = M[p_o v_o - p_i v_i]. \quad \dots \quad (143)$$

The sum of these is equal to the total work.

$$\begin{aligned} \text{Work per second} &= M \left[\frac{v^2}{2g} + \frac{p_o v_o}{k-1} - \frac{p_i v_i}{k-1} + p_o v_o - p_i v_i \right] \\ &= M \left[\frac{v^2}{2g} + \frac{k}{k-1} [p_o v_o - p_i v_i] \right], \quad \dots \quad (144) \end{aligned}$$

but

$$M = aV \frac{p_o}{RT_o} \quad \text{and} \quad v = KV.$$

Therefore

$$\begin{aligned} \text{Work} &= \frac{aV p_o}{RT_o} \left[\frac{K^2 V^2}{2g} + \frac{k}{k-1} (p_o v_o - p_i v_i) \right] \\ &= \frac{aV p_o}{RT_o} \left[\frac{K^2 V^2}{2g} + \frac{k}{k-1} R_o T_o \left\{ 1 - \left(\frac{p_i}{p_o} \right)^{\frac{k-1}{k}} \right\} \right]. \quad (145) \end{aligned}$$

Eq. (145) may be simplified by assuming that the intrinsic energy is not altered and that the specific volume does not change during the small change in pressure. This gives

$$\begin{aligned} \text{Work} &= \frac{aV p_o}{RT_o} \left[\frac{v^2}{2g} + (p_o - p_i) v_o \right] \\ &= \frac{aV p_o}{RT_o} [h_o + h_r] = \frac{aV p_o}{RT_o} H, \quad \dots \quad (146) \end{aligned}$$

H = total or dynamic head.

Now

$$H \text{ is } \frac{V^2}{2g} \dots \dots \dots (147)$$

$$\therefore \text{ Work per second} = \frac{ap_o}{RT_o} \frac{V^3}{2g} = K^1 V^3 \dots \dots \dots (148)$$

$$Q = aV \dots \dots \dots (149)$$

From Eqs. (147) (148) and (149) the important relations, that Q varies as V , H as V^2 and work per second or power, as V^3 , are seen. These relations are shown by test data and by the tables which follow. These are prepared from the catalogues of the manufacturers. Only a few values are given, but in any case the relation can be used to find the results under different conditions.

Thus from the table it is seen that as the diameter of the wheel increases, the number of revolutions per minute to produce a given pressure will decrease, the product of speed and diameter being constant. The blast area varies as the square of the diameter, and since in any column the velocity is constant, the quantity discharged will vary as the square of the diameter. The velocity being constant and the quantity varying as the square will make the power vary as the square of the wheel diameter in any column of equal pressure. To care for differences in pressure, it will be remembered that pressure varies as the square of the velocity and hence the speed of a given wheel will vary as the square root of the pressure, and the power will vary as the three halves power, while the quantity will vary as the speed or the square root of the pressure.

Thus 1 oz. pressure with a wheel 22 ins. in diameter requires 896 R.P.M. with a discharge of 2116 cu.ft. and 1.07 H.P. A 36-in. wheel will run at $\frac{22}{36} \times 896$ or 548 R.P.M. It will discharge $2116 \times \left(\frac{36}{22}\right)^2$ or 5630 cu.ft. if the same proportions for width are used, while if these are changed the result will be

different. In some tables it is seen that the discharges are not so proportioned and this means a change in the proportion of the width.

The power required will vary as the quantity, since the pressure is the same

$$\text{H.P.} = 1.07 \frac{5630}{2116} = 7.87.$$

For 2 oz. pressure for a 22-in. wheel the following should be found:

$$\begin{aligned} \text{R.P.M.} &= 2^{\frac{1}{2}} \times 896 = 1264; \\ Q &= 2^{\frac{1}{2}} \times 2116 = 2980; \\ \text{H.P.} &= 2^{\frac{3}{2}} \times 1.07 = 3.00. \end{aligned}$$

In this way new columns may be made for the table or the data for different conditions of flow may be found. If reference is made to the curve of Fig. 160, the effect of changing the pressure from that for which the table is built is seen.

Before quoting the tables of the manufacturers of fans it is well to note the results obtained from the Sirocco fan, in which shallow blades are used. This fan gives higher dynamic pressures than the peripheral speed of the wheel. This is due to the better action of properly shaped blades and to the fact that there is less friction. The space taken by the fan is less than that required for other types, also the power is less. The makers claim a saving of $\frac{1}{2}$ the space, $\frac{1}{3}$ the weight and $\frac{1}{3}$ the power. However, for this fan the same variations of power, capacity, speed and pressure as before noted is to be found in the tables.

It rarely happens that the engineer designs his fan. By an understanding of the table the proper size may be selected.

The following tables do not give all sizes made, but reference to the catalogues of builders is recommended for closer figures. The discharge is in cubic feet per minute.

SIROCCO FAN.						BUFFALO FORGE FAN.					
Fan No.	Diam. Wheel.		Total Dynamic Pressure. Oz.			Fan No.	Diam. Wheel.		Total Dynamic Pressure. Oz.		
			$\frac{1}{4}$	1	2				$\frac{1}{4}$	1	2
1	6"	Cu.ft. R.P.M. B.H.P.	155 1145 0.0185	310 2290 0.147	440 3230 0.42	30	22"	Cu.ft. R.P.M. B.H.P.	1497 634 0.37	2116 896 1.07	2989 1264 2.94
1½	9"	Cu.ft. R.P.M. B.H.P.	350 762 0.042	700 1524 0.333	1000 2152 0.95	45	32½"	Cu.ft. R.P.M. B.H.P.	3292 430 0.81	4640 607 2.30	6550 857 6.49
2	12"	Cu.ft. R.P.M. B.H.P.	625 572 0.074	1250 1145 0.588	1770 1615 1.66	70	50"	Cu.ft. R.P.M. B.H.P.	8040 280 1.99	11340 394 5.61	16010 557 16.39
3	18"	Cu.ft. R.P.M. B.H.P.	1410 381 0.167	2820 762 1.33	3980 1076 3.75	90	64"	Cu.ft. R.P.M. B.H.P.	12950 218 3.21	18300 308 9.06	25800 435 25.52
4	24"	Cu.ft. R.P.M. B.H.P.	2500 286 0.296	5000 572 2.35	7080 807 6.64	120	85"	Cu.ft. R.P.M. B.H.P.	24200 164 6.00	34300 232 17.05	48300 328 47.85
6	36"	Cu.ft. R.P.M. B.H.P.	5650 190 0.665	11300 381 5.30	15900 538 15.0	160	113"	Cu.ft. R.P.M. B.H.P.	43250 121 10.67	61100 174 30.00	86200 247 85.25
8	48"	Cu.ft. R.P.M. B.H.P.	10000 143 1.18	20000 286 9.40	28300 403 26.6	200	141"	Cu.ft. R.P.M. B.H.P.	70000 99 17.16	98900 140 48.95	139500 198 138.00
10	60"	Cu.ft. R.P.M. B.H.P.	15650 114 1.84	31300 228 14.7	44200 322 41.6	250	176"	Cu.ft. R.P.M. B.H.P.	116050 79 28.93	163960 112 80.96	231410 158 231.41
12	72"	Cu.ft. R.P.M. B.H.P.	22600 95 2.66	45200 190 21.2	63600 269 59.8	300	211"	Cu.ft. R.P.M. B.H.P.	174790 66 44.22	246950 94 129.30	348540 132 348.54
15	90"	Cu.ft. R.P.M. B.H.P.	35250 76 4.14	70500 152 33.1	99600 214 93.6	350	246"	Cu.ft. R.P.M. B.H.P.	245500 57 61.38	346770 80 173.38	489410 113 489.41

To apply the tables, use data from the school building. The friction head in the pipes is 33 ft. of air, which is $\frac{1}{2}$ in. of water. The loss in the two Vento heater stacks is 0.07 in. These with the loss in the fan may be taken as $\frac{3}{4}$ in. The quantity to be handled is 1,820,000 cu.ft. per hour.

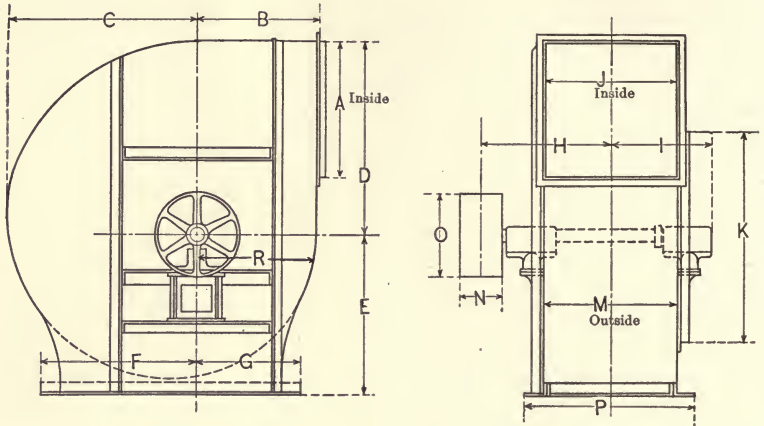
$$\frac{3}{4} \text{ in.} = 0.75 \text{ of peripheral speed;}$$

$$1.00 = \text{peripheral head;}$$

$$V = 66.9\sqrt{1.00} = 66.9 \text{ ft. per second;}$$

$$0.133d^2 = \frac{1820000}{66.9 \times 60 \times 60};$$

$$d = 7.54' = 90.5''.$$



DIMENSIONS OF SIROCCO FAN

No. of Fan.	Diam. of Wheel.	A	B	C	D	E	F	G	H	M	I	K	O	J	N
2 1/2	15	10	9 1/8	14	15	14	13	9 1/8	13	10	10	16 1/2	6 1/2	10	3 1/2
3	18	12	11 1/8	17	18	15	15	10 1/2	15	10 1/2	12	19	7 1/2	12	4 1/2
3 1/2	21	14	13 1/8	20	21	18	17	12	18	14 1/2	13	21	8 1/2	14	5 1/2
4	24	16	15 1/8	23	24	20	19	13 1/2	20	16 1/2	15 1/2	24	10	16	6
4 1/2	27	18	17 1/8	26	27	23	22	15	21	18 1/2	16 1/2	29	11	18	6 1/2
5	30	20	18 1/8	29	30	25	24	16 1/2	22	20 1/2	17	32	12	20	7 1/2
6	36	24	22 1/8	35	36	30	29	20	26	24 1/2	21	38	15	24	9
7	42	28	26 1/8	41	42	35	33	23	29	28 1/2	23	44	17	28	10 1/2
8	48	32	30 1/8	47	48	39	37	27	32	32 1/2	24	50	20	32	10 1/2
9	54	36	33 1/8	52	54	44	42	31	35	36 1/2	28	56	22 1/2	36	12
10	60	40	37 1/8	58	60	48	47	35	36	40 1/2	32	62	25	40	12
11	66	44	41 1/8	64	66	53	51	38 1/2	39	44 1/2	34	68	27 1/2	44	13 1/2
12	72	48	45 1/8	70	72	57 1/2	55	42	43 1/2	48 1/2	36	74	30	48	13 1/2

DIMENSIONS OF BUFFALO FORGE FAN

Size in Inches	Diam. Wheel.	A	B	C	D	E	F & G	H	I	J	K	P	N	O	R
30	22	11 1/4	11 1/4	14 1/4	15 1/4	14 1/4	11 1/4	11	10	11 1/4	14 1/4	15 1/4	3	7	12 1/4
35	25 1/2	13 1/4	13 1/4	17 1/4	18 1/4	16 1/4	13 1/4	13	11	13 1/4	17 1/4	17 1/4	3	7	14 1/4
40	29	15 1/4	15 1/4	19 1/4	20 1/4	19 1/4	15 1/4	14	12	15 1/4	19 1/4	19 1/4	3	8	17 1/4
45	32 1/2	16 1/4	18	22 1/4	23 1/4	21 1/4	18	15	13	16 1/4	21 1/4	20 1/4	3	8	19 1/4
50	36	18 1/4	20	24 1/4	26 1/4	23 1/4	20	17	14	18 1/4	24 1/4	24 1/4	4	9	21 1/4
55	39 1/2	19 1/4	22	26 1/4	28 1/4	26 1/4	22	18	15	19 1/4	26 1/4	26 1/4	4	9	23 1/4
60	43	22 1/4	24 1/4	29 1/4	31 1/4	28 1/4	24 1/4	19	16	22 1/4	29 1/4	29 1/4	5	10	25 1/4
70	50	26 1/4	28 1/4	34 1/4	36 1/4	34 1/4	28 1/4	22	19 1/4	26 1/4	34 1/4	30 1/4	5	11	30 1/4
80	57	29 1/4	32 1/4	39 1/4	41 1/4	37 1/4	32 1/4	24	21 1/4	29 1/4	39 1/4	35 1/4	6	12	34 1/4
90	64	33 1/4	36 1/4	44 1/4	46 1/4	44 1/4	36 1/4	26	23 1/4	33 1/4	43 1/4	38 1/4	6	14	39 1/4
100	71	37 1/4	40 1/4	48 1/4	51 1/4	47 1/4	40 1/4	28	25 1/4	37 1/4	47 1/4	43 1/4	7	16	43 1/4
110	78	41 1/4	44 1/4	53 1/4	56 1/4	51 1/4	44 1/4	31	28 1/4	41 1/4	51 1/4	47 1/4	7	18	47 1/4
120	85	44 1/4	48 1/4	58 1/4	61 1/4	56 1/4	48 1/4	34	30 1/4	44 1/4	55 1/4	51 1/4	8	20	52 1/4
130	92	48 1/4	52 1/4	63 1/4	67 1/4	61 1/4	52 1/4	36	33 1/4	48 1/4	60 1/4	54 1/4	8	22	56 1/4
140	99	52 1/4	56 1/4	68 1/4	72 1/4	65 1/4	56 1/4	39	35 1/4	52 1/4	64 1/4	59 1/4	9	24	60 1/4
150	106	56 1/4	60 1/4	73 1/4	77 1/4	70 1/4	60 1/4	42	37 1/4	56 1/4	69 1/4	64 1/4	10	26	65 1/4

This is large;

$$w = 36'';$$

$$\text{R.P.M.} = \frac{66.9}{7.54\pi} \times 60 = 169.$$

The power to drive the fan is given by

$$\text{Power} = \frac{1820000 \times 14.7 \times 144}{3600 \times 550 \times 53.35 \times 530}$$

$$\left[\frac{\frac{1}{4} \times 66.9^2}{64.32} + \frac{1.405}{0.405} (53.35 \times 530) \left(1 - \left[\frac{34}{34.1} \right]^{\frac{0.405}{1.405}} \right) \right]$$

= 2.5 H.P. with an efficiency of 35 per cent. This would require 7.1 H.P.

It is better, however, to use tables of the makers for sizes and dimensions.

From the table, since 1 in. is approximately $\frac{1}{2}$ oz., a 99-in. wheel at 141 rev. will be required to deliver 32,800 cu.ft. of air per minute. This requires 8.14 H.P. To deliver just 30,000 cu.ft. this fan will be run at

$$\frac{30000}{32800} \times 141 = 129 \text{ R.P.M.}$$

The power consumed will be

$$\left(\frac{129}{141} \right)^3 \times 8.14 = 6.2 \text{ H.P.}$$

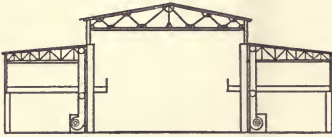
The pressure will be

$$\left(\frac{129}{141} \right)^2 \times \frac{1}{2} = 0.42 \text{ oz.}$$

If this pressure is not sufficient the fan will have to be run faster and the quantity will be changed from the point of maximum efficiency, as there will not be the same relation between

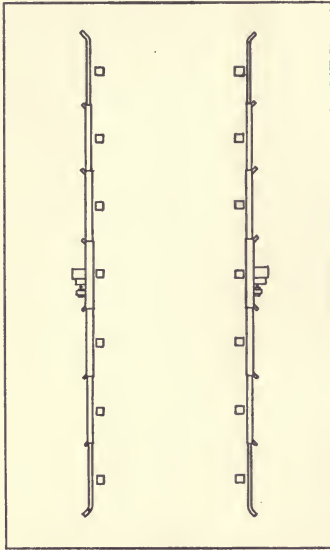
velocity head and friction head as that at a proper speed and pressure.

For a Sirocco fan a 60-in. fan at 134 R.P.M. would be required, and this would use 7.48 H.P. This would be treated in the same manner as above to get exact conditions.



The dimensions of the housings of these fans would be found from the tables.

The further application of this method of heating will be applied to a shop building shown in Fig. 162.



In this building there are 11,200 sq.ft. of wall space of 16-in. concrete, 13,600 sq.ft. of glass, 20,000 sq.ft. of concrete roof and 750,000 cu.ft. of air. There are 300 men employed in the building.

The amount of air required for ventilating would be

$$300 \times 2000 = 600000$$

cu.ft. per hour.

This is much less than the volume of the room, and with as

much window space as is here used the leakage from the windows will keep the air sufficiently fresh so that a heating system will be used in which the air is recirculated from the shop.

The heat required is found as follows:

Temperature shop, 55° F.;

K for glass, 0.96;

K for concrete, 0.26;

K for roof, 0.30.

$$H = (55 - 0)[0.26 \times 11200 + 0.3 \times 20000 + 0.96 \times 13600];$$

$$= 55 \times 21968 = 1208240 \text{ B.t.u.}$$

If the air is to be heated to 85° F., the amount of air to be circulated will be:

$$\frac{1208240}{0.02 \times 30} = 2014000 \text{ cu.ft. per hour,}$$

or about three changes of the volume of the room.

In some cases this air might be carried through a **filter** before entering the fan again, thus cleaning the atmosphere, and in case of mill heating the proper humidity might be obtained in this way.

Since 2,000,000 cu.ft. per hour is a large quantity, it might be advisable to separate the system into two parts. Moreover the **crane run-way** must be left clear, and the only way of distributing to both sides if one fan were used would be to **cross over** at the end of the building or use an **underground duct**. For simplicity and economy it is well to place heaters near the center of the building.

In this system quietness is not so important as smallness of pipes, and for that reason high velocities of 2500 ft. per minute will be used in the pipes, while 500 ft. will be used at ends of branches. There will be an opening in each bay, as shown on Fig. 162, for space beneath gallery and roof. The velocity through the heater will be 2000 ft. per minute.

The temperature of the air leaving is 85° F., which with zero air and 5 lbs. steam will require four sections of Vento heaters. The entering air is at 55°, which requires two sections. Hence the difference, or two sections, must be used to heat the air from 55° to 85°.

At 2000 ft. the heat transmission for the two sections is

$$\frac{2380 \times 4 - 2670 \times 2}{2} \text{ or } 2090 \text{ B.t.u.}$$

The heating surface required will be

$$\frac{1208240}{2000} = 578 \text{ sq.ft.}$$

The amount per heater will be 289 sq.ft. and the amount per stack will be 145 sq.ft. The area required through heater will be

$$\frac{1007000}{2000 \times 60} = 8.4 \text{ sq.ft.}$$

11 50-in. sections with 5-in. centers will give this.

To cut down the area required a higher temperature will be used. Suppose five sections be used in the heater; this means then that the condition is that of a 5-heater system where the air has passed through two sections. The temperature of the air leaving is then 120° F. and the heat transmission is

$$\frac{1950 \times 5 - 2670 \times 2}{3} = 1803 \text{ B.t.u.}$$

The heating surface is then

$$\frac{1208240}{1803} = 670 \text{ sq.ft.}$$

or 335 sq.ft. on each side of the building. This means 112 sq.ft. to each stack. The air now used will be

$$\frac{1208240}{0.02 \times (120 - 55)} = 930000 \text{ cu.ft.}$$

or a little over one change per hour. The amount for each heater is 7750 cu.ft. per minute. The area through the heater should then be

$$A = \frac{7750}{2000} = 3.9 \text{ sq.ft.}$$

A 9-sec. 50 in. high $4\frac{5}{8}$ in. centers would have the correct heating area, but the area for the air passage is 5.85 sq.ft. A lower velocity must then be taken. Suppose 700 ft. be assumed.

Then 1 sec. will give 40° F. and 4 sec. 118° F. The heat per sq.ft. for the three sections will be

$$\frac{1200 \times 4 - 1600}{3} = 1100$$

$$\text{H.S. per stack} = \frac{1208240}{2 \times 3 \times 1100} = 182.$$

This will not give the desired result. The first result is the best combination.

$$\text{Air by each fan} = \frac{2010000}{2} = 1000000,$$

or 16700 cu.ft per minute.

Air to each floor 8350 cu.ft. per min.;

Air to each branch 4175 cu.ft. per min.;

Air to each outlet 1040 cu.ft. per min.

$$\text{Area of outlet} \frac{1040}{500} = 2 \text{ sq.ft.}$$

$$\text{Area of last section} \frac{1040}{2500} \times 144 = 59 \text{ sq.in.}$$

$$D = 9.$$

Area of 2d section from end

$$= \frac{2080}{2500} \times 144 = 118 \text{ sq.in.}$$

$$D = 12\frac{1}{2};$$

Area of 3d section = 177 sq.in. $D = 15.$

Area of 4th section = 236 sq.in. $D = 17\frac{1}{2}$ in.

Area of vertical riser, 462 sq.in. $D = 24.$

Area of vertical riser from fan = 924 sq.in. = $31 \times 31.$

Drop in pressure due to friction.

$$\begin{aligned} \text{1st section, } h &= \frac{0.02 \times 25}{.75} \times \frac{42^2}{64} = 18 \text{ ft. of air} \\ &= 0.26 \text{ in. of water.} \end{aligned}$$

$$\text{2d section, } h = 0.26 \text{ in.} \times \frac{9}{12.5} = 0.19 \text{ in. of water.}$$

$$\text{3d section, } h = 0.26 \text{ in.} \times \frac{9}{15} = 0.15 \text{ in. of water.}$$

$$\text{4th section, } h = 0.26 \text{ in.} \times \frac{9}{17.5} = 0.13 \text{ in. of water.}$$

$$\text{Riser 1st to 2d, } 0.26 \times \frac{9}{17.5} \times \frac{12}{25} = 0.05.$$

$$\text{Main riser, } 0.26 \times \frac{9}{31} \times \frac{10}{25} = 0.03.$$

$$\text{Loss in bends} = 0.15 \frac{42^2}{64} = 9 \text{ ft. air or } 0.13 \text{ in. water.}$$

Total loss in pressure:

Last branch and nozzle	0.05 (assumed)
Bend	0.13
Last section	0.26
2d section	0.19
3d section	0.15
4th section	0.13
Bend	0.13
2d floor riser	0.05
1st floor riser	0.03
Bend	0.13
Fan	0.20
2 section heater	0.45
	<hr/>
Total	1.90''

This is the total resisting pressure. The dynamic pressure is $\frac{1.90}{.75} = 2.6$ ins. Use 2 oz. Then from the tables two No. 70 steel-plate fans with wheels 50 ins. in diameter will be required when run at 557 R.P.M. and each using 16.39 H.P. These each give 16,011 cu.ft. The Sirocco table gives 2 No. 6 fans with diameters 36 ins. at 538 R.P.M. and using 15.0 H.P. These each give 15,900 cu.ft. They may be speeded down to give the exact discharge, as was done with the school-house fan. If larger pipes could be used and the area through the heaters made greater to cut the velocity to one-half its value, the resistance should be $\frac{1}{4}$ of this or $\frac{1}{2}$ oz. This would require two No. 100 fans of 71 ins. diameter at 197 R.P.M. and would use 4.11 H.P. giving 16,610 cu.ft.

The Sirocco fan for this service would be a No. 9—54 ins. diam. fan at 179 R.P.M. using 4.20 H.P. and giving 17,950 cu.ft.

In all cases it is advisable to use as low a velocity as possible, but in some cases the size of pipe is the important matter.

In cases where steam engines are employed and the exhaust steam is used in coils, the excessive power loss is not important, as it is changed into heat and used in the building. This same is true in a plant using electric power when it is made on the premises and the exhaust steam is used for heating. It is simply a matter of cost of production of power, as electrical

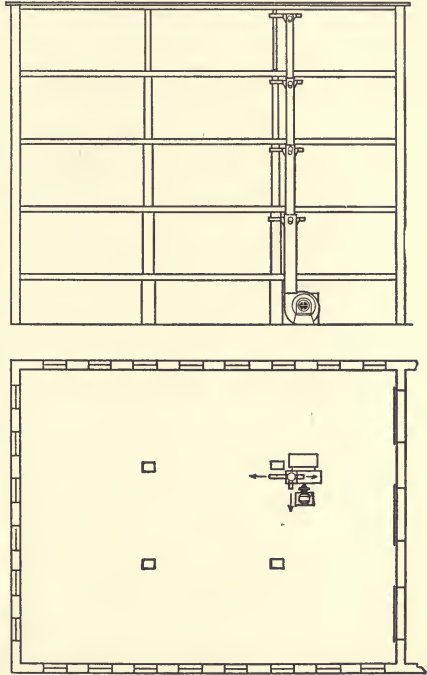


FIG. 163.—Small Factory.

power or thermal power. If 1 B.t.u. of heat can be produced as heat in the coils at $\frac{1}{50000}$ ct. while 1 B.t.u. of electrical energy costs $\frac{1}{1000}$ ct., it is evident that the power consumed

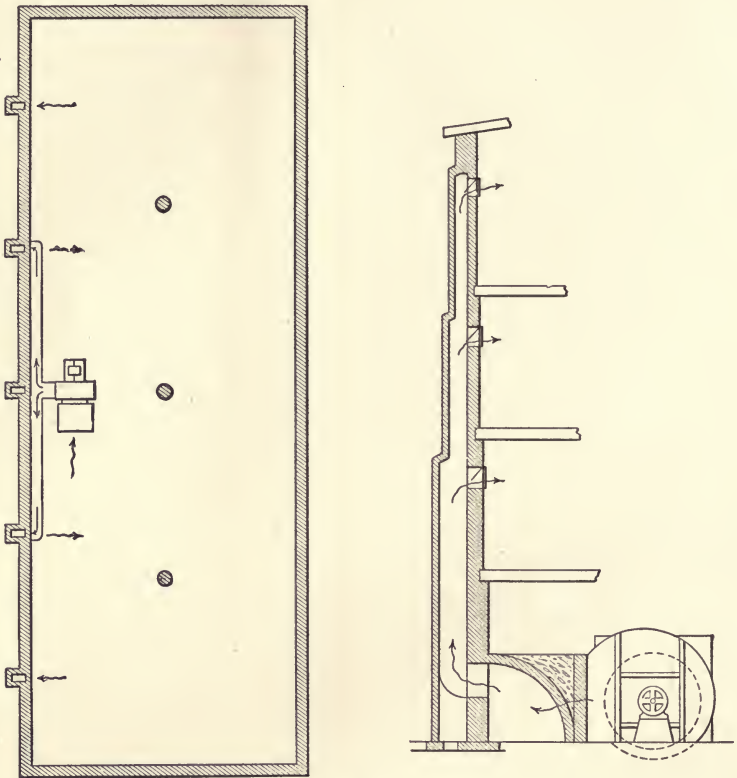


FIG. 164.—Mill Heating, Using Buttress.

by the fan should be reduced to as small a quantity as possible if an electric motor is to drive it.

The following figures are prepared from cuts showing the work of the B. F. Sturtevant Co.

Fig. 163 illustrates a method of introducing air into a small factory, while Fig. 164 shows how **underground ducts** and **buttresses** may be used in mill buildings. The use of **brick ducts**

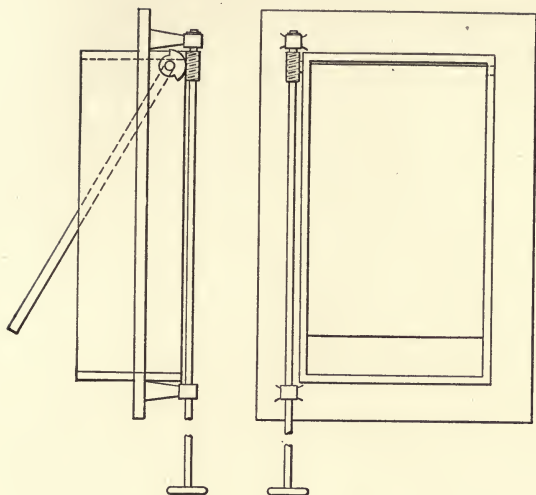


FIG. 165.—Damper for Wall Duct.

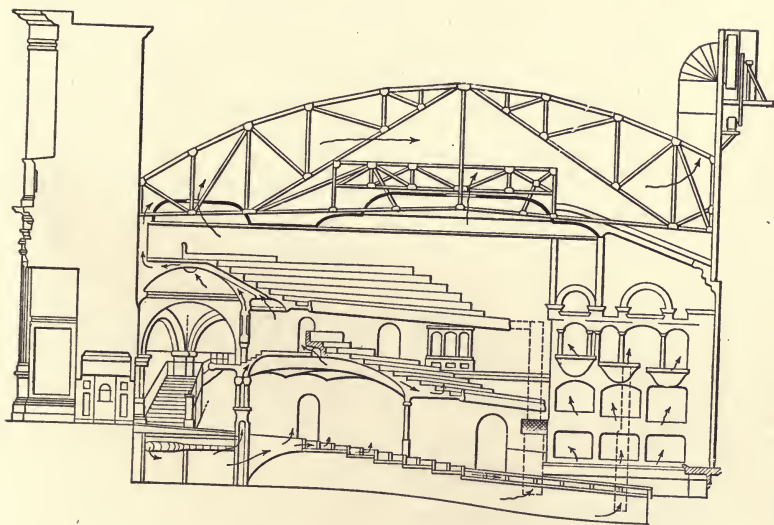


FIG. 166.—Indirect Heating of Theater.

underground is important in many cases. The problem is the same as with metal, the constant f being increased as mentioned earlier. In Fig. 165 the form of **damper** used in such a plant is given; such dampers must be operated from the floor of the room. This arrangement is not as advantageous as one in which the air is carried in the interior of the building, as there is loss of heat through the walls of the flues. Similar buttresses may be used to remove the foul air, although unless the product of manufacture vitiates the air there are so few occupants that this air may be recirculated, using screens or filters to remove dirt.

Fig. 166 illustrates the method of applying this system to a theater.

CHAPTER IX

FURNACE HEATING

As described in Chapter I, **furnace heating** is a system in which **hot air** is introduced from a **furnace** into the rooms to be heated. The air may be introduced to the heating tubes of the furnace from the **outside** or it may come from the **interior** of the building through the **recirculating duct**. This latter method may be used in residences or in buildings where there are not so many occupants, the leakage from the windows and doors giving sufficient fresh air. The former method must be employed in schools, churches or audience rooms heated by furnaces, on account of the large amount of fresh air for ventilation.

The first consideration in this method of heating, as in all methods, is the necessary heat to care for the losses, then the amount for the ventilating air, and finally one must consider the temperature at which the air must enter in order to supply the heat loss before being reduced to the room temperature.

The method of Chapter V is used to find the heat losses and then the assumed amount of air for ventilation is used to find the temperature at which this air must enter.

$$t_4 = t_r + \frac{H}{0.02 \times V}, \quad \dots \dots \dots (150)$$

where t_4 = temperature of entering air in deg. F.;

t_r = temperature of room air in deg. F.;

H = heat loss per hour in B.t.u.;

V = volume of air per hour in cubic feet.

The air assumed as the ventilating air or leakage air may give so high a value of t_4 that the amount of air must be increased. This does not necessarily mean that the air from the outside

must be increased, but that passing through the heating tube. A large part of this may come from the recirculated air in residences.

The house of Chapter V will now be examined. As mentioned in Chapter VIII an examination of the air supply to the various rooms will show that the leakage air assumed is too small, as the temperatures are all high. If 130° F. is assumed as a good temperature for this air the quantity in cubic feet at 70° will be that found in the table below which is prepared in addition to those of Chapter V for the furnace system of heating this house.

Room.	Heat Loss, B.t.u. per Hour.	Ventilation, Cubic Feet per Hour.	T	Total Heat for Various Amounts of Outside Air.		
				Full Outside Air.	One-half Outside Air.	No Outside Air.
1	16810	14000	130	36500	26600	16800
2	11792	9600	130	25000	18200	11800
3	1745	1450	130	3800	2800	1700
4	6890	5700	130	14800	10800	7000
5	8740	7300	130	19000	13900	8700
6	10635	8900	130	23200	17000	10600
7	14050	11700	130	30400	22300	14000
8	8750	7300	130	19000	13900	8700
9	2170	1800	130	4700	3300	2200
10	7680	6400	130	16600	12200	7700
11	8330	7000	130	18200	13400	8300
12	2840	2400	130	6200	4600	2800
	100432	83550	217400	159000	100300

The column marked total heat with full outside air has been computed by the formula:

$$H = 0.02V(t_i - t_o) \dots \dots \dots (151)$$

while the column for one-half outside air has been found by

$$H = \text{heat loss} + \frac{1}{2}V \cdot 0.02[t_r - t_o] \dots \dots \dots (152)$$

$$= 0.02V[T_1 - T_r + \frac{1}{2}(T_r - T_o)] \dots \dots \dots$$

$$= 0.02V\left[T_1 - \frac{T_r}{2} - T_o\right] \dots \dots \dots (153)$$

Since

$$\text{Heat loss} = 0.02V[T_4 - T_r]. \dots \dots \dots (154)$$

V in all of the cases is the volume at 70° . The computations are made with a slide-rule in all work of this nature, as the original data and the variation in conditions do not warrant a greater degree of accuracy.

The table shows well the cost of ventilation. With no fresh air 100,000 B.t.u. are required per hour. With 50 per cent efficiency of furnace and with coal of 14,000 B.t.u. this means 14 lbs. of coal per hour. With the air taken entirely from the outside, the heat required amounts to 217,000 B.t.u. and requires about 31 lbs. of coal per hour. The **leakage** mentioned in the table of Chapter V of 30,000 cu.ft. from the outside is about three-eighths of the amount necessary to keep the entering temperature at 130° F. and if this came in it would mean the supply of 144,000 B.t.u. per hour, or 21 lbs. of coal. The full amount of air would care for about 45 persons, while the leakage of Chapter V would care for 18 persons, a number which would be too large even for this house for continuous use. Hence the value, and necessity even, for **inside circulation** in residences is quite evident. In most cases the leakage, in on the windward side and out on the leeward side, is sufficient to care for the ventilation of the rooms, and this amount of leakage is likely to occur though all of the air for the furnace is taken through the inside recirculation duct.

Having found the amount of heat, the next step is to decide on the location and size of the various pipes, flues and registers.

The location of **registers** is important. If located in the wall they may interfere with the placing of furniture and are limited in size according to the size of the partition, while if placed in the floor, although better as far as size and directness are concerned, they mean a cutting of carpets or floors, to which some householders object, and they collect dust and dirt. When placed in the floor, however, they offer a more direct path and for that reason it is very advisable to place them in this manner on the first floor, where there is little head causing flow.

The head causing flow is the difference in weight between the column of hot air and cold air, and as mentioned in Chapter VIII, this head, expressed in feet of hot air, is,

$$h = L \frac{\left(\frac{p_1}{RT_2} - \frac{p_1}{RT_1} \right)}{\frac{p_1}{RT_1}} = L \frac{T_1 - T_2}{T_2} \dots (155)$$

T_1 = absolute temperature of inlet air;

T_2 = absolute temperature of air at bottom of heater;

= T_r approximately.

If this is applied as before to the various heights from registers to base of heater, say 10, 20 and 30 ft., the following heads result, for

$$t_1 = 130; T_1 = 590;$$

$$t_2 = 60^\circ \text{ F. (due to mixture of some cold air):}$$

$$H_1 = 1.35;$$

$$H_2 = 2.70;$$

$$H_3 = 4.05.$$

The pipes leading to the **vertical flues** or **stacks** are known as **leaders**. In these, the velocities will vary with the sizes of the pipe (fixing the hydraulic radii, $\frac{A}{P}$), with the length of the run, the number of bends, etc. The following problems will give an idea of the results to be expected. (See p. 198.)

1st. Find the velocity in the pipe to a first-floor register with a 15-ft. run of 12-in. pipe and a bend of radius equal to the diameter.

$$V = \sqrt{\frac{2g \times 1.35}{1 + \frac{0.02 \times 15}{1} + .25 \times 1 + 1.5}} = 8.02 \sqrt{.45} = 5.4.$$

2d. Find the velocity to be expected in a $14 \times 3\frac{1}{2}$ flue run-

ning to second floor with two sharp right-angle turns and 10 ft. of 10-in. pipe.

$$V = \sqrt{\frac{2g \times 2.70}{1 + \frac{0.02 \times 10}{\frac{10}{12}} + \frac{0.02 \times 10}{4 \frac{49}{12 \times 35}} + 2 \times .3 + 1.5}} = 8.02 \sqrt{.72} = 6.8.$$

In this way the velocity may be found for any given condition, but in general the velocity for the various floors will be assumed. The values given below are those which may be expected in practice:

- 1st floor, 4.0 to 5 ft. per sec.;
- 2d floor, 5.0 to 6.5 ft. per sec.;
- 3d floor, 6.0 to 7.5 ft. per sec.

Having the velocities to the different floors, the areas of pipes and flues are found.

$$\text{Area} = \frac{V \times c}{\text{vel.}} \dots \dots \dots (156)$$

Thus where *c* = the factor to care for temperature as found in the table of p. 209, for room 2 of the house considered in Chapter V, the area will be

$$\text{Area} = \frac{9600 \times 1.11 \times 144}{4 \times 3600} = 106 \text{ sq.in.}$$

This requires a pipe 11 ⁵/₈ ins. in diameter. A 12-in. pipe will be used.

The area of the registers will be fixed by the allowable velocity of discharge. This may be taken as 4 ft. per second on all floors, giving

$$\text{Net area reg.} = \frac{144 Vc}{14400} = \frac{cV}{100} \text{ in sq.in.} \dots \dots (157)$$

Since the net area is 66 per cent of the gross area of opening into which the register fits the area of the opening is given by

$$\text{Area opening} = \frac{144Vc}{0.66 \times 14400} = 0.015Vc. \quad \dots (158)$$

In the case above this equals

$$0.015 \times 9600 \times 1.11 = 160 \text{ sq.in.}$$

A 10'' × 16'' register would be required.

If **vent stacks** are used Hoffman recommends that their areas be made 0.8 of the **heat stack** areas.

The pipes leading to the **stacks** are in some cases made larger

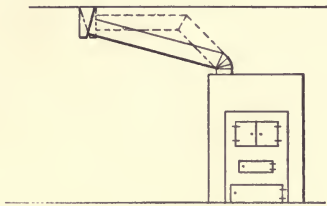


FIG. 167.—Furnace with Flat Bonnet.

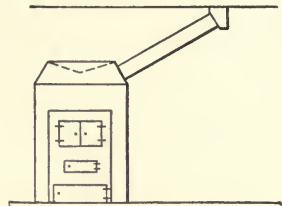


FIG. 168.—Furnace with Conical Bonnet.

in area than the stacks supplied by them to cut down resistance. This must be done in cases of long runs, although in short runs it is not necessary.

Before applying these formulæ, however, it is well to plan the location of the heater and the location of the various flues or stacks.

Stacks should be run in the inside partitions, as there would be considerable heat loss if placed in outside walls. Circulation is better cared for if applied here, as the hot air rising naturally starts the cold air across the floor from the other side and thus gets the air in motion. If placed in an outer wall the cold air from the wall may start a down current of air in the stack, thus choking off the supply. The furnace method of heating is different from steam heating, in which radiators form strong

up currents due to the higher temperature. The principal objection to furnace heating is the absence of a positive circulating force.

The **leaders** or pipes leading across the cellar from the heater are taken from the **top of the casing** around the heater. This casing may have a **flat top** as shown in Fig. 167 or it may have a **cone top** or **bonnet** as shown in Fig. 168. Although the flat top receives the hot air more directly from the furnace it means that there will be more head room needed and a full right-angle turn will have to be used. The distance above the castings to a flat top should be at least 8 or 9 ins. and this could be used to advantage on a slanting side cutting down the bend to 45 or 60° instead of a right angle. The **slant height** of the bonnet

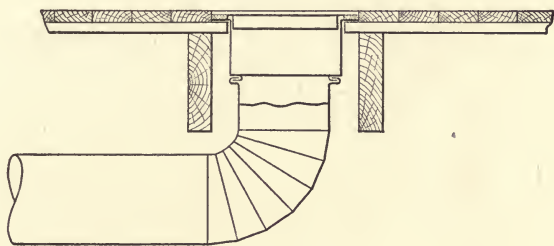


FIG. 169.—Register Floor Box.

should be about 3 ins. greater than the largest pipe to be taken from it. To aid in sending the air to the edge with a cone top, an inside conical center is used as dotted in Fig. 168. This sends the air to the outside. Sometimes a long leader is taken from the center of the top so that it may get a better supply to overcome the friction of the great length or the long pipes may be taken from the rear side of the top, but never directly over the point where the cold air enters nor near the edge of a flat top where it might start currents along the cold casing. To aid the circulation in long pipes or to pipes on the exposed sides of a building the leader is sometimes continued inside of the bonnet with an enlarged **hood**, so that the hot air may be directed into it, and in some cases positive circulation is attained by connecting certain leaders to a number of the heat-

ing tubes of the furnace. The furnace should be so placed that the runs for the leaders on the exposed sides, usually the north and west, are shorter than the others, but the heater should be placed so as to make all as short as possible.

The leaders are carried to the **register boxes**, Fig. 169, for the floor registers of the first story, or the **shoes or boots**, Fig. 170, at the bottom of the **heat flues** or **heat stacks** which run to higher floors. The boxes for the floor registers should be

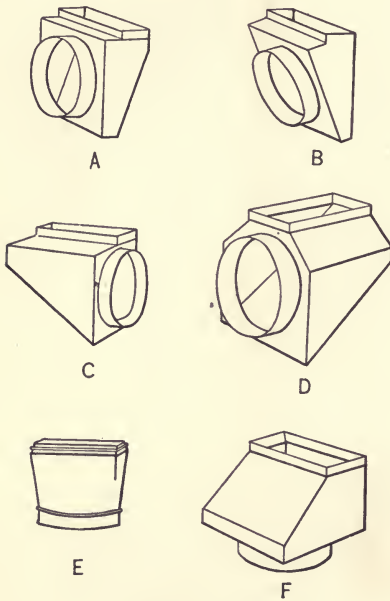


FIG. 170.—Shoes.

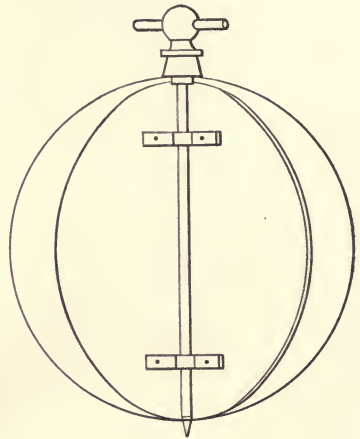


FIG. 171.—Damper.

made with no projecting fins to obstruct the flow of air and the 4-in. collar at the bottom should be connected with the leader by an elbow at least equal to the diameter of the leader. The register box is about $4\frac{1}{2}$ to 6 ins. deep to allow the air to reach the ends of the register face with little obstruction. The elbow should fit the pitch of the pipe. The pitch should be not less than 1 in. per foot. This pitch is advisable for starting the circulation, although for a pipe running full of a fluid with a definite difference in head between the ends the velocity of

the fluid is theoretically independent of the pitch. In furnace work, however, the difference of head is small and the substance has such a small density that there may be currents in two directions in the same pipe and for these reasons, although the formulæ for fluids have to be used, there is the need of considering the peculiarities of the substance. Thus pipes are pitched as much as possible and circulation is aided thereby.

The leaders are connected to collars about 5 ins. long, which are attached to the bonnet of the furnace. The tops of the collars should all be on the same level so that they will all offer the same resistance to flow. All leaders except one should be provided with tight-fitting dampers, Fig. 171, so that the heat may be cut off when necessary. Some one leader should be arranged so that it cannot be cut off in order that there may be no danger of closing all outlets and heating the air within the casing so much that fires may occur.

If there is only one outlet, as is the case at times in church heating, this connection should be without a valve or damper of any kind.

The leaders for the second floors or for registers placed in the walls are attached to boots or shoes, Fig. 170, which form the lower parts of the flues or stacks.

The shoe *A* or *B* is intended to be used when the leader is at right angles to the partition, while *C* and *D* are used when the leader runs parallel to the partition. *E* and *F* are employed when elbows are used to join the leader and shoe. Shoe *B* requires no bend at the end of the leader to accommodate the pitch and therefore should offer small resistance. The slanting back of *A* is of little value.

The shoe shown at *C* is used when it is necessary to offset for the leader. The stacks are made of single or double thicknesses of tin. The ordinary partitions are made of 3×4 or 2×4-in. studs placed 16 ins. apart. These will allow one to use 13×4- or 14×4-in. stacks as the largest single stacks and in that way the amount of heat taken for a single register face is limited. Stacks are usually made rectangular in form on account of space limitation, but the nearer the form approaches a circle

or square, the better it is for carrying a fluid. The friction of a conduit depends on the reciprocal of the hydraulic radius which is the ratio of the area of the section to the perimeter of the sections and this is larger for the circle and square ($\frac{1}{4}d$) than for any other form of area. Stacks are sometimes made with double walls, to cut down radiation and

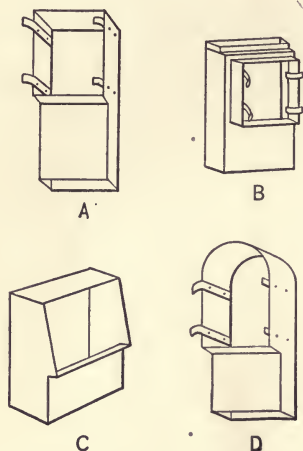


FIG. 172.—Register Boxes.

to prevent charring the wood if hot. Some advise having single-thickness stacks covered with asbestos paper. These stacks end in register boxes, Fig. 172. *A* represents the ordinary form where the box is placed at the end of the riser or stack and *B* shows one which does not require the full width of the stack. The figure also illustrates a box which takes part of the supply of a stack, the remaining part going to the room above. This method is used at times, but it is not the best way to supply heat owing to interference. For

privacy it is not well to put two wall registers opposite each other on the same stack.

The box *D* is one in which a circular top register is to be used while *C* illustrates a method for first-floor registers in order to increase the area of flue. To increase the area of the flue for first-floor wall registers, the plaster and base board are omitted at these points, and in this way an extra inch may be obtained. By using the box shown at *C*, however, in which the special register casting fits over the sides of the box as shown in Fig. 173, a much greater area necessary for large first-floor rooms is obtained. The registers are held in place by the clips shown on the boxes. The stacks are sometimes turned to enter floor boxes on an upper floor and it is then necessary to use an elbow, *A* or *B*, Fig. 174. When the partition through which the stack passes is directly over the cellar partition an offset, *C*, Fig. 174,

is used on which to attach the boot. Fig. 175 shows the application of these various fittings.

The tin used for pipes and stacks is made of soft sheet steel

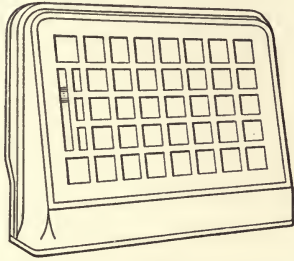


FIG. 173.—Special First Floor Wall Register.

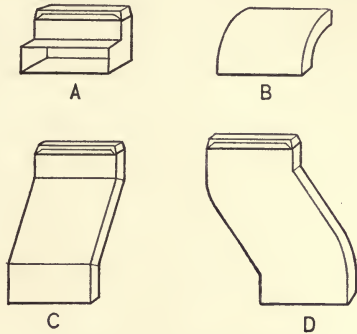


FIG. 174.—Elbows and Offsets.

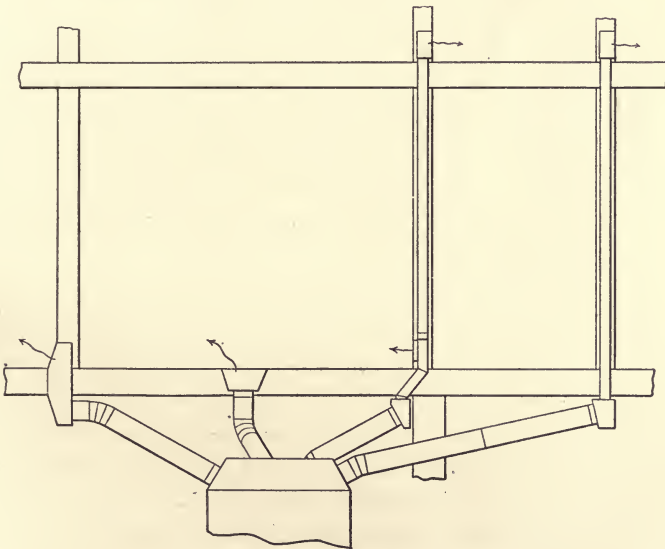


FIG. 175.—Stacks, Shoes, and Leaders.

coated with tin. Originally it was made of wrought iron. Tin is gauged by numbers or letters. The trade terms from 56 lbs. to 100 lbs. mean the weight of a box of 100 14" X 20"

sheets, while between weights of 107 lbs. and 215 lbs. to the box of 100 14"×20" sheets, the tin is known as I tin; IC being 107 lbs. to the box, IXL, 128 lbs.; IX, 135 lbs.; IXX, 155 lbs. up to IXXXXX, 215 lbs.

IC, IX or IXX tin is used for pipe work, the latter for pipes over 12 ins. in diameter. The piping and stacks are often wrapped in asbestos paper to make all the joints tight, although this does not cut down the heat loss very materially. In fact the use of non-conducting material or air space with double-thickness stacks is not necessary for stacks on interior walls, as any heat escaping is used to warm the house. To cut down loss in the cellar, some type of air-cell covering or thick pipe covering might be of value to keep the cellar cool. This heat escaping in the cellar is not wasted, however, unless the cellar is heated too much for the proper storage of vegetables and fruits. The heat from these pipes warms the cellar and thus cuts down the loss through the floor from the first-floor rooms.

In running the leaders from the heater, endeavor should be made to reduce the resistance to a minimum even though special cutting must be made on the fittings of the piping to suit the angles. Thus the dotted position for the leader of Fig. 167, although cheaper to make because of the use of standard elbows, is not the one to give the better flow of heat which should be the aim in all designs.

For school heating it is advisable to bring the air in from a wall register placed six or eight feet from the floor and to have a vent register near the same point at the floor level. In some cases this is not possible and floor registers have to be used. The ventilating stack is sometimes made effective in such installations demanding ventilation, by having a furnace or hot pipe at the base of a stack which heats the stack air and causes it to rise. This produces a draft which sucks impure air from the proper part of the rooms of the lower story while the hot air is made to pass over restricted areas at the upper floors, causing an injector action and sucking the air from the room. If the furnace gases could be carried up through a central iron flue in a brick ventilating stack, the heat from the flue

would cause the air around it in the stack to rise, drawing in air from the various rooms on the stack to replace it.

At times it is necessary even with hot-air furnaces to use fans to produce the necessary circulation, Fig. 176. This occurs in large work only when some of the rooms to be heated are at considerable distance from the heater. In school work or other large buildings, several heaters may be placed at convenient locations, each heater caring for a set of adjacent rooms. This method requires several chimneys, but these are easily built when the necessary ventilation flues are formed.

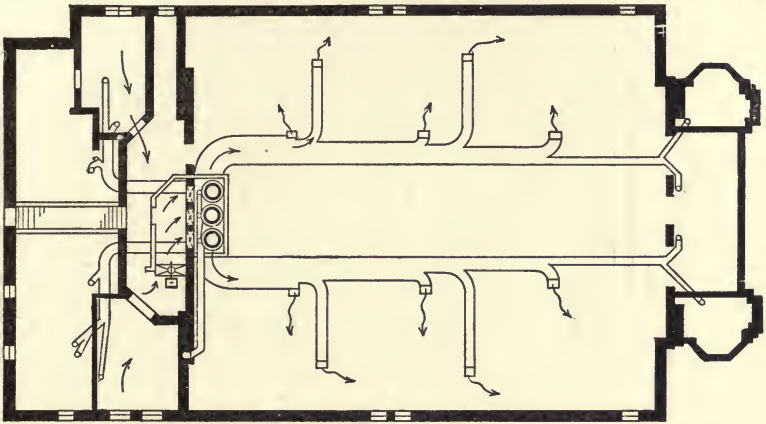


FIG. 176.—Fan Circulation with Twin Hot-air Furnaces.

The principles are now applied to the house considered earlier and the layout of heater and piping of Fig. 177 is made for the cellar, in conjunction with the plans of the rooms, Fig. 178. The heater is placed near the north and west side but the pipes are mainly on three sides of the heater. Hence the room behind the heater or the central passage of the cellar would have been far better had the use of the cellar rooms and the location of the chimney permitted it. In many cases the location of the heater is a compromise between what should be and what is needed.

The length of the pipes to the various rooms should be as short as possible, but here again the demands of the owner must

be considered. Thus for room 5, the pipe could have been made shorter had the use of room permitted the register to be

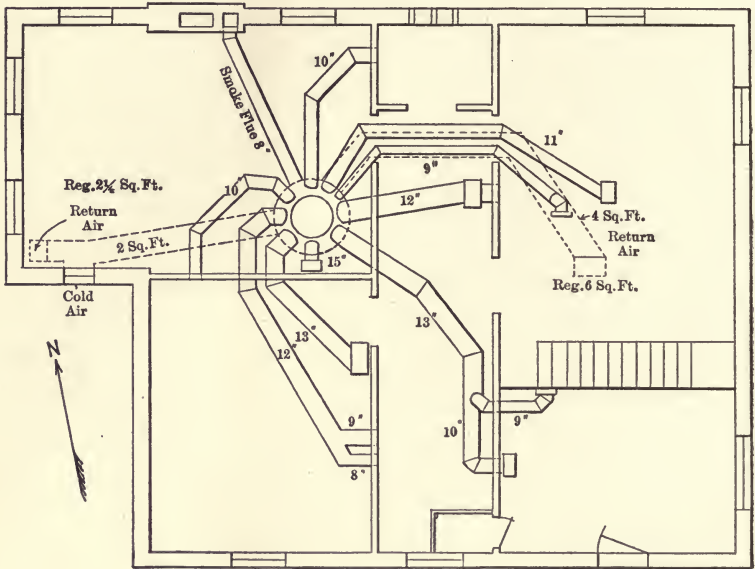


FIG. 177.—Pipe and Heater Arrangement.

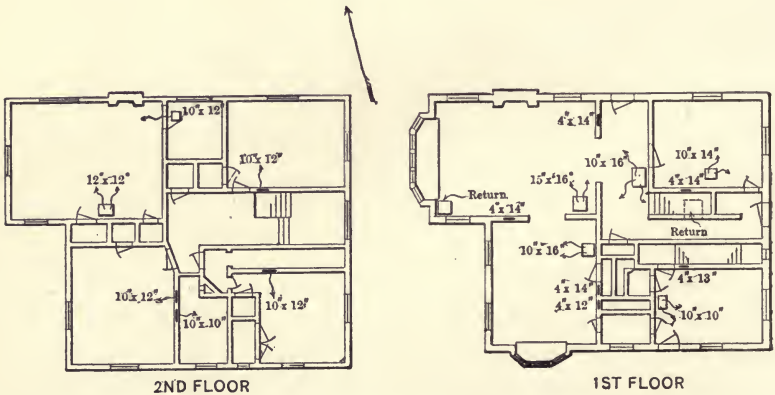


FIG. 178.—Arrangement of Outlets.

placed near the partition. Fig. 178 shows the location and size of the registers and flues.

The application of the foregoing principles to the house gives the results in the following table:

Room.	Venti-lation.	Leader.		Stack		Register.		Vent Stacks.		Vent Registers.	
		Area	Size.	Area	Size.	Area.	Size.	Area	Size.	Area	Size.
1	14000	180	15	233	15×16	125	2-4×14	188	2-10×12
2	9600	123	13	160	10×16	83	2-4× 8	128	1-10×12
3	1450	19	5	24	13	19	
4	5700	73	10	95	10×10	52	1-4×12	76	1-10×10
5	7300	93	11	122	10×14	65	1-4×14	98	1-10×10
6	8900	114	12	148	10×16	80	2-4×10	118	2-10×10
7	11700	100	(12) 10	(95)	1-4×14	(196)	12×12	(76)	2-4×12	157	2-10×10
8	7300	63	9	60	1-4×14	122	10×12	48	1-4×12	98	1-10×10
9	1800	17	(5) 8	15	1-4×12	(30)	10×10	12	24	
10	6400	57	9	52	1-4×13	107	10×12	40	4×10	86	1-10×10
11	7000	60	9	57	1-4×14	116	10×12	46	4×12	93	1-10×10
12	2400	23	(6) 10	(20)	1-4×14	(40)	10×12	(16)	32	

The first-floor heat registers are all of the floor form. The register is omitted from room 3 owing to the small size. The heat for rooms 7 and 12 is divided between two registers, that in room 12 being made larger than necessary for this room. It is placed near the door of room 7 so that the heat enters that room.

It will be seen in the table that 4×14" is the limiting size of stacks. In the actual construction no vents were used extending to the attic. The table, however, is made to include these so as to illustrate the method. In Fig. 178 the position of registers and stacks is shown. The remaining computation is that for the cold-air supply and recirculated air supply.

Of the air supply of 83,550 cu.ft. at 70° F. 30,000 enters as cold air at 0° as leakage and the full 80,000 passes back as recirculated air at 70°. The area of the circulating ducts using a low velocity of 4 ft. per second will be

$$a = \frac{80000}{4 \times 3600} = 5\frac{1}{2} \text{ sq. ft.}$$

The register faces should be

$$\frac{5\frac{1}{2}}{0.66} = 8\frac{1}{4} \text{ sq.ft.}$$

The house is planned to have one register under the stairs of 6 sq.ft. connected to 4 sq.ft. of flue and one register of $3\frac{1}{4}$ ft. in the corner of room 1, connected to a flue of 2 sq.ft. The cold air when needed enters as shown in Fig. 177. The area of the cold-air opening was $2\frac{1}{2}$ sq.ft. This, however, is rarely needed.

The use of the jib panel of a stairway as the place for the register of the recirculating flue is excellent. It is better than one in the hall floor in that it is not conspicuous and does not collect dirt, nor does it form an unpleasant place over which one must walk. The return duct must be of ample size and it must lead into the base of the heater by easy curves.

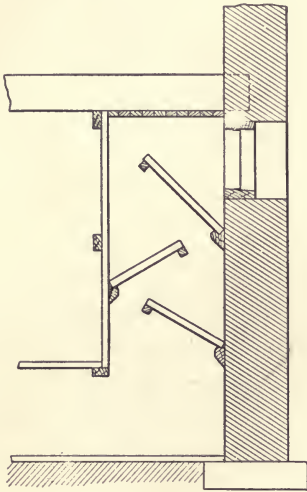


FIG. 179.—Baffle Screens.

In all cases the air supply must be ample if the heater is to perform its duty. Had the 80,000 cu.ft. been introduced from the outside, the area required would have been

$$\frac{80000 \times 0.87}{4 \times 3600} = 5 \text{ sq.ft.}$$

This air supply may be filtered by passing it through cheesecloth or by passing it over a series of baffles as shown in Fig. 179, the area of the various parts being sufficient for the passage of the air.

Some designers proportion the cold-air duct by the total area of the leader pipes. This method can be used, but it is just as well to figure it from the amount of air to be handled.

The cold-air inlet should always be placed on the side toward the prevailing winds.

The size of the fire pot is found at this point, although really belonging to the next chapter. From the heat to be developed as found in the first part of this chapter the total heat with the leakage air to be cared for is 144,000 B.t.u. per hour. If the efficiency of the furnace is taken at 60 per cent and the rate of burning coal is 4 lbs. per square foot, the area of the grate is

$$A = \frac{144000 \times 144}{0.60 \times 14000 \times 4} = 620 \text{ sq.in.}$$

$$\text{Diam.} = 28''.$$

This matter, with the size of the smoke flue, will be further considered in the next chapter. In figuring the pipes, furnaces and other parts for a room, some designers and manufacturers reduce the heat quantity to equivalent glass surface while others find equivalent volume in cubic feet.

Now Carpenter's approximate rule,

$$H = (t_r - t_o) \left[\frac{W}{4} + G + 0.02nV \right],$$

(p. 69) shows that 4 sq.ft. of wall is equivalent to 1 sq.ft. of glass and for one change of air per hour each 50 cu.ft. of volume is equal to 1 sq.ft. of glass. Hence if one-fourth of the wall area in square feet and one-fiftieth of the volume in cubic feet times the changes per hour are added to the glass area the result is called the equivalent glass area. In some methods the ventilation term is omitted and the floor and ceiling are added in giving the equivalent glass as equal to the glass plus one-quarter the wall area and one-twentieth of the area of the floor or ceiling to care for losses to basement or attic in first-floor or top-floor rooms. The equivalent is then multiplied by the heat loss per square foot of glass to find the total heat loss. This glass loss is about 70 B.t.u. per square foot per hour.

In one of the equivalent cubic feet methods the actual

cubic feet are increased by 75 cu.ft. for each square foot of glass surface and 8 cu.ft. for each square foot of wall space. This rule shows that the changes per hour are $1\frac{1}{2}$, and that the allowance for wall area is not nearly sufficient.

In both of these methods allowance is made for exposure, 10 per cent being added on the north and west sides.

Certain manufacturers list pipes and heaters to care for equivalent cubic feet or square feet, and these may be used in designing. These methods are all equivalent to the theoretical one given in the text, but they are not as valuable, as the work is too empirical.

The methods used in calculation for any form of building are similar to the above.

When distant rooms are to be heated a combination system is sometimes used in which the distant rooms are heated by hot water furnished by a coil or cluster in the fire box of the furnace, as will be shown later. The method of determining the size of flow pipes, radiators and various parts of this system have been discussed in previous chapters.

CHAPTER X

FURNACES AND BOILERS

THE ordinary hot-air furnace, Fig. 180 (Graff Co.'s Lackawanna Furnace) consists of an ash pit, *A*, a fire pot, *B*, a radiator

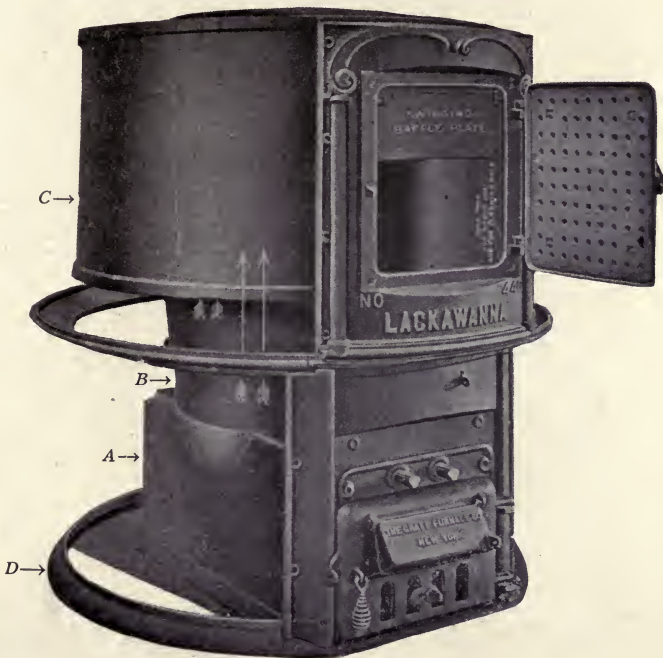


FIG. 180.—Graff Lackawanna Furnace.

C and within the radiator the combustion chamber. The radiator may be made of sheet steel between cast heads as shown, or it may be of cast iron, Fig. 181 (Graff-Comfort Furnace). The

plan of radiator is illustrated in Fig. 182. The gases enter the **dome** over the combustion chamber, which contracts in the form of a dome and is connected to the center portion of the radiator. The hot gases thus pass around the radiator to the **smoke outlet**. In most cases there is a **by-pass valve** or **cut-off plate** which on being moved gives a free passage from the fire pot



FIG. 181.—Graff Comfort Heater.

to the smoke outlet to be used on starting fires. The **base ring** *D* of the furnace is placed on the brick foundation which in many cases forms the **cold-air pit** to which the air supply is brought from the atmosphere or house. A central pier is usually built to carry the weight of the ash pit, fire box and radiator. The grates are mostly made of triangular-shaped toothed bars, Fig. 183, supported on their ends and so arranged that by turning one of the bars of the grate, one-half of them turn,

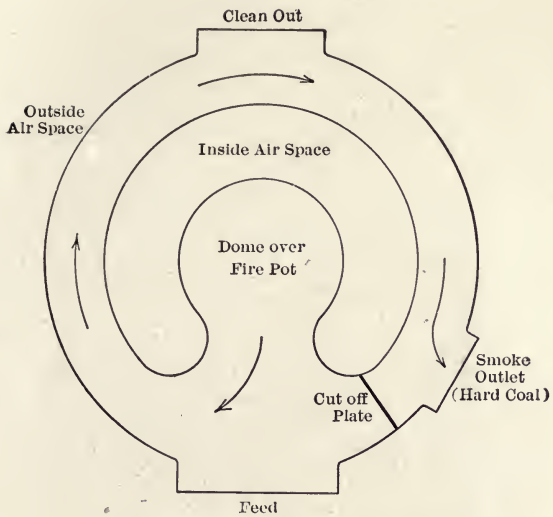


FIG. 182.—Plan of Furnace Radiator.



FIG. 183.—Grate and Ash Pit of Fuller, Warren & Co.

cutting clinkers and allowing clinkers and ashes to fall. A **galvanized-iron casing** is now placed around the furnace and is carried up to the bonnet. This casing should be of bright iron to cut down the radiation loss and in some cases it is made double with an **air space** for this purpose.

The **air duct** leading to the pit beneath the furnace is made of brick or concrete and is covered by wood or reinforced concrete, Fig. 184. In case the concrete form is used **manholes** should be left for cleaning or examination.

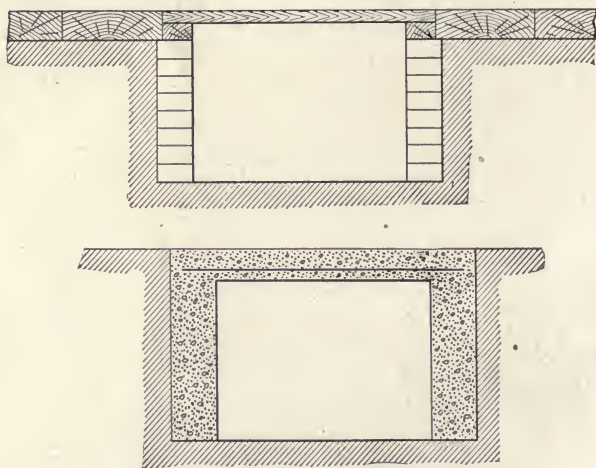


FIG. 184.—Cold-air Ducts.

The main objection to the type of furnace shown in Figs. 180, 181, which is probably the most common form, is the fact that the heating surface is not sufficiently large for the grate area. To increase the heating surface in their heater, the Graff Co. in their "Faultless Furnace," use a number of **air-heating flues**, Fig. 185, which are surrounded by the hot gases and are exposed to the radiant heat and through which the air passes. The hot gases pass through openings at the top of these air-heating flues to the exterior surface and, by means of the baffle plates, have a long path to the smoke outlet, giving up their heat to the walls of the boxes. The outside of this hot-gas

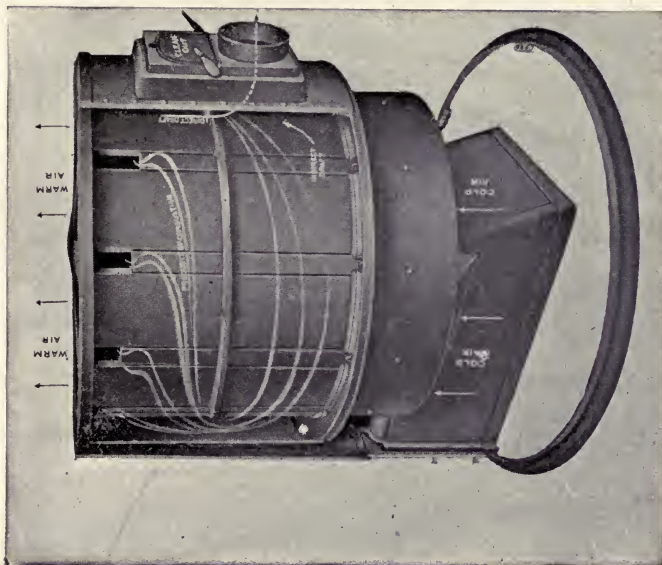
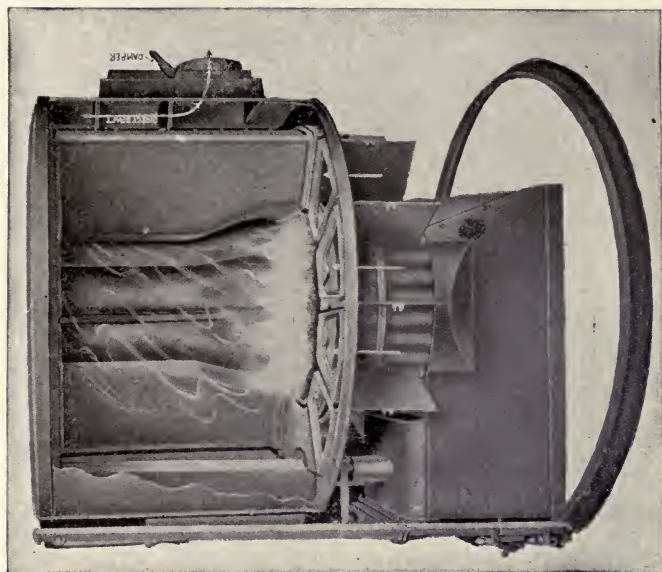


FIG. 185.—Graff Faultless Heater.

passage is made of sheet iron and this serves as the outer surface of the radiator. The casing is placed outside of this so that

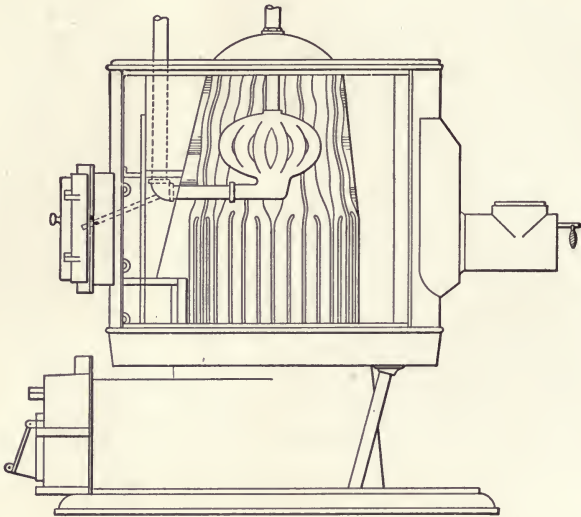


FIG. 186.—Kelsey Warm-air Generator.

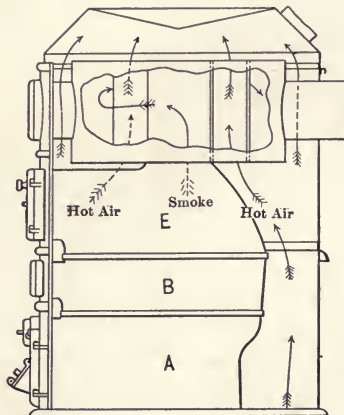


FIG. 187.—Tubes in Radiator.

this furnace has much the same appearance as the Lackawanna furnace of this company.

The fins guiding the air into the air-heating flues also add heating surface.

The Kelsey warm-air generator or furnace, Fig. 186, shown with a **hot-water generator** at the top of the combustion chamber, is quite similar to the Graff furnace. In this the **air-heating flues** or **elements** are made corrugated to increase the heating surface. The action of this furnace is similar to that just examined.

The same result may be accomplished partially by putting **tubes** through the hot gas space of the radiator, Fig. 187, and allowing the air to pass through the tubes while the gases pass around them. This method should improve the efficiency of the heater. This figure illustrates the method of putting the casing together and the **bonnet**. The three castings forming the ash pit, fire pot, and dome of fire box are clearly seen. These sections are fitted together by the edge of one casting fitting in a groove in the other, which joint is filled with a **stove cement**, making it tight. The same substance is used in putting the sections of the other furnaces together. It is important that this work be carefully done, as a leak of coal gas might poison the occupants of the house. The top of the furnace is often covered with **sand** to cut down the radiation from the top and a **sand ring** is placed at the edge of flat tops to hold this.

Fig. 188 is the special Novelty Furnace of the Abram Cox Stove Co. in which the increased heating surface is obtained by complex castings of considerable length. The method of making gas-tight joints is illustrated as well as the use of a double casing. The **direct draft** for cutting down the resistance is shown near smoke outlet and the **water pan** near the bottom at *B*. The water pan should be installed on all hot-air furnaces to humidify the air, giving it the necessary amount of moisture. The pan should be located where it will not be in contact with the hot air, as that air could take up so much moisture that this would be deposited on furniture and windows when cooled to the room temperature. It is usually placed near the bottom of the furnace where the air temperature and consequently the moisture content is not high.

Sectional heaters or boilers, Fig. 189, are used for either hot water or steam. They are made of **sections** which may be

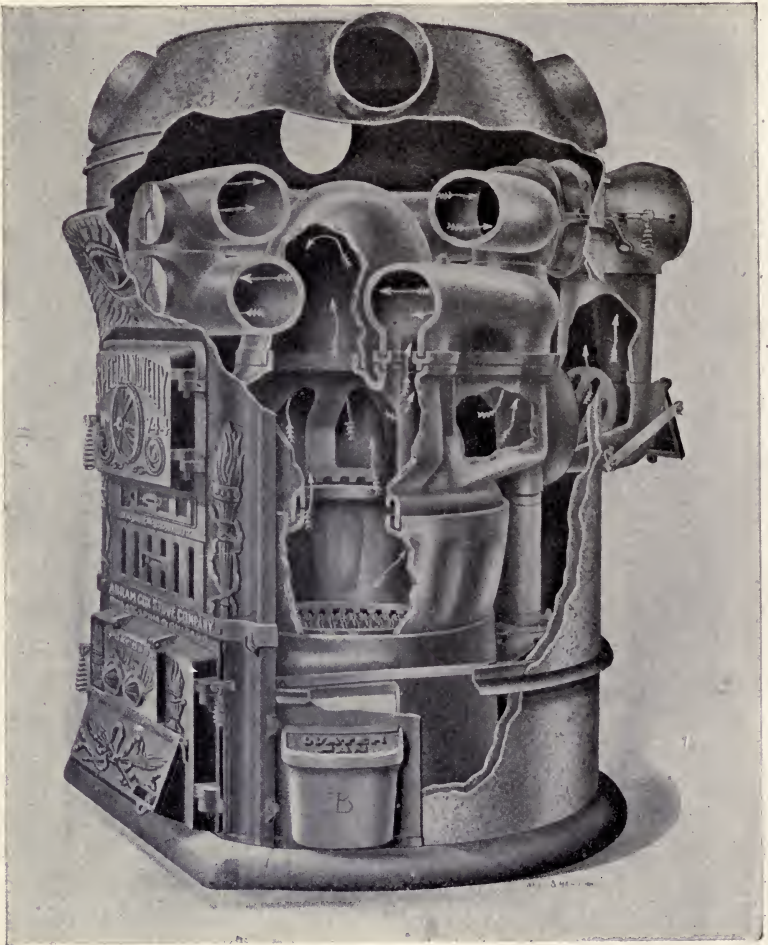


FIG. 188.—Special Novelty Heater.

increased in number to form heaters of different capacities. The middle sections are the same in form except for side or top outlets, while special shapes of sections form the rear and front. The sections are so made that there is a large amount of surface

exposed to the fire. This is done by having considerable space between the lower parts of the sections, although at the top the

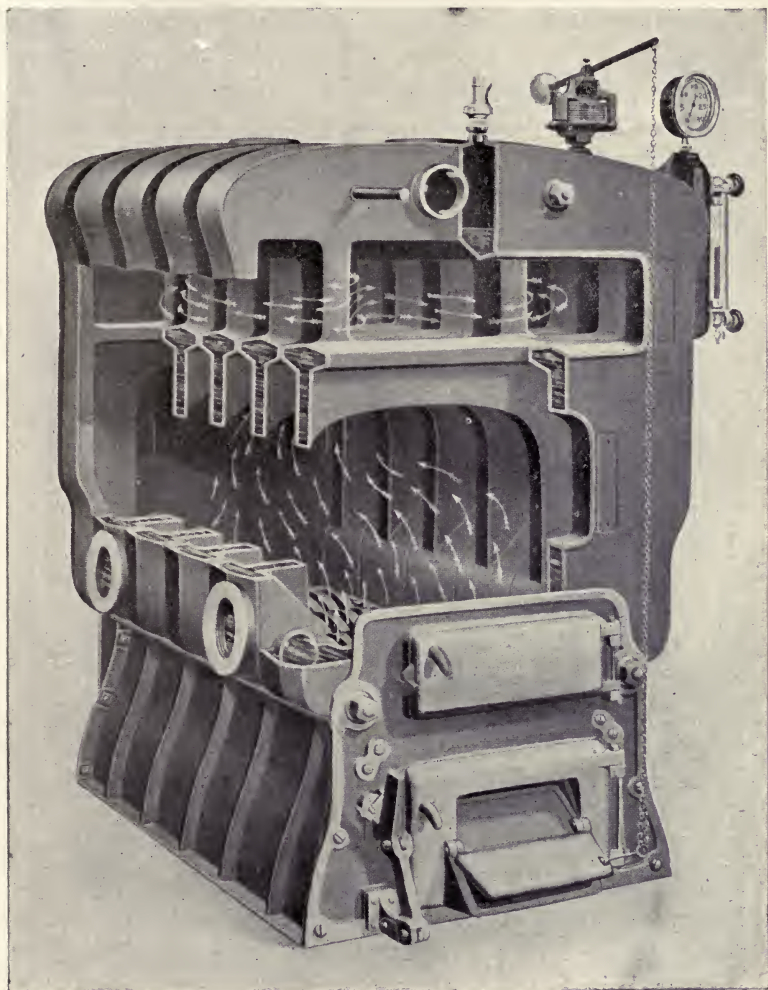


FIG. 189.—No. 5-15-6 Ideal Sectional Boiler.

sections come so closely together that three passages are formed through which the gases must pass to the chimney. The sections of the boiler shown, the Ideal Sectional Boiler of the

American Radiator Co., are held together by bolts, the sections being connected by three **conical-faced push nipples**. The sections *A*, Fig. 190, are supported by the casting forming the grate and ash pit. Some other boilers, as the American of the Pierce Co., *B*, Fig. 190, are made of sections which are connected by three **manifolds**, attached to flanged nipples screwed in at *a*, *b* and *c*. The Ideal Boiler has an outlet at the top and the bottom as shown in the figure for steam or water and feed. The steam boilers are usually provided with damper regulators which are attached as shown to the damper and

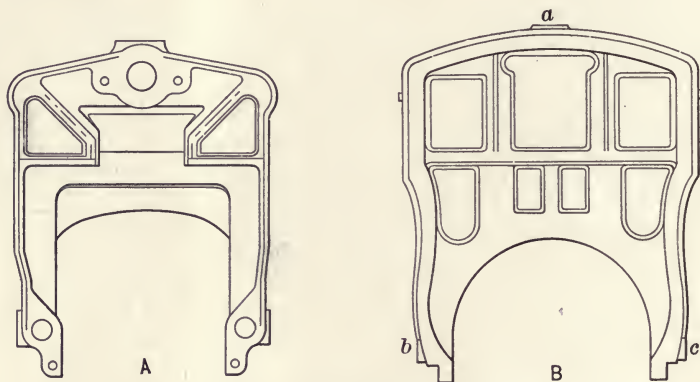


FIG. 190.—Sections of Boiler.

ash-pit door. They are operated by the pressure in the steam boiler and shut draft and ash-pit doors when the pressure rises, while at low pressures both are opened. Fig. 189 shows the equipment on one of these used as a steam boiler. Fig. 195 illustrates the dimensional sheet of one type of this boiler, the table of which will be of service in laying out plans before letting contracts. For small installations small circular boilers are used. Several of these are illustrated in Fig. 191. *A* represents the Pierce Boiler, in which the sections are united by screwed nipples. The **water leg** on the side of the fire box forms a good heating surface. The outlet for steam (or water) is at the top, while the return enters at the bottom.

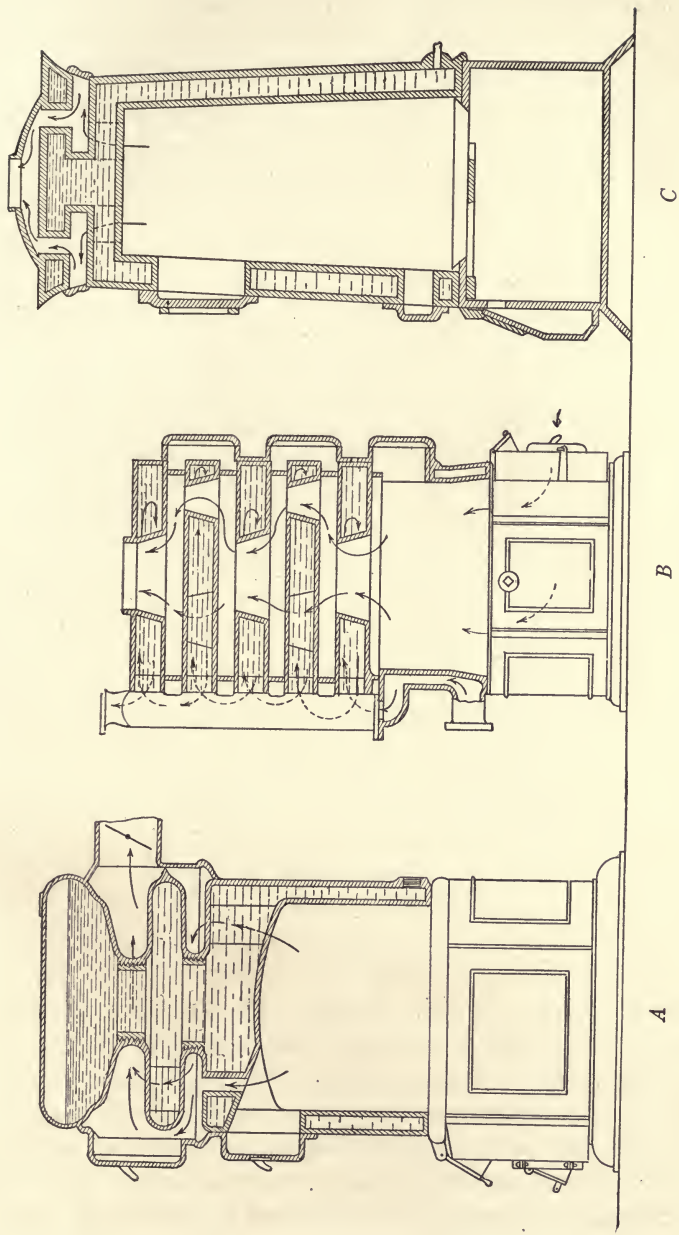


FIG. 191.—Small Boilers.

The Spence water boiler, as shown at *B*, Fig. 191, consists of five cast circular sections attached to the base section by a cast manifold on one side. The manifold is so made that there is continuous circulation from the bottom to the top, in a definite path as shown by the arrows. *C* illustrates the Ideal Junior Water Heater of the American Radiator Co. used for heating

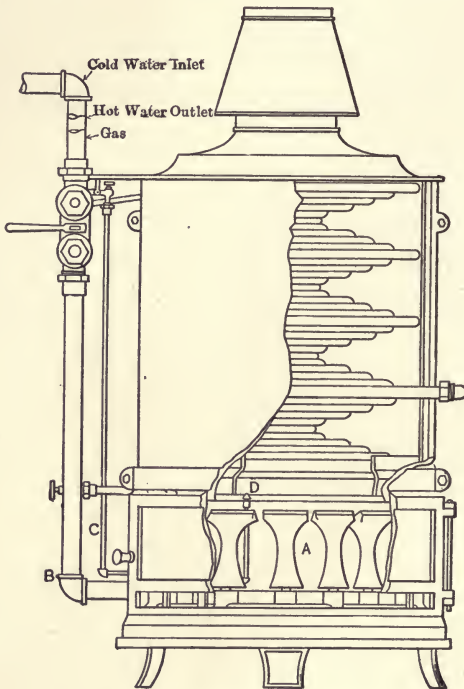


FIG. 192.—Humphrey Heater.

water for laundry purposes or domestic service. This is not large and is intended to be used when a quantity of hot water is needed in the laundry or home. However, the gas heater has come into extensive use where hot water is needed for domestic service. There are a number of these heaters on the market. The Humphrey Gas Water Heater is shown in Fig. 192. The gas burners *A* are supplied through pipe *B*. The small pipe *C* supplies gas to the small pilot light *D*. The pilot light can be

shut off by the valve at the top of the line C and the main gas line may be closed by a cock.

The two water lines are behind the gas line. The cold water from the city supply or cool water from the storage tank enters at the bottom of the coils. The water circulates from the tank as soon as water is not taken from the faucets by an automatic

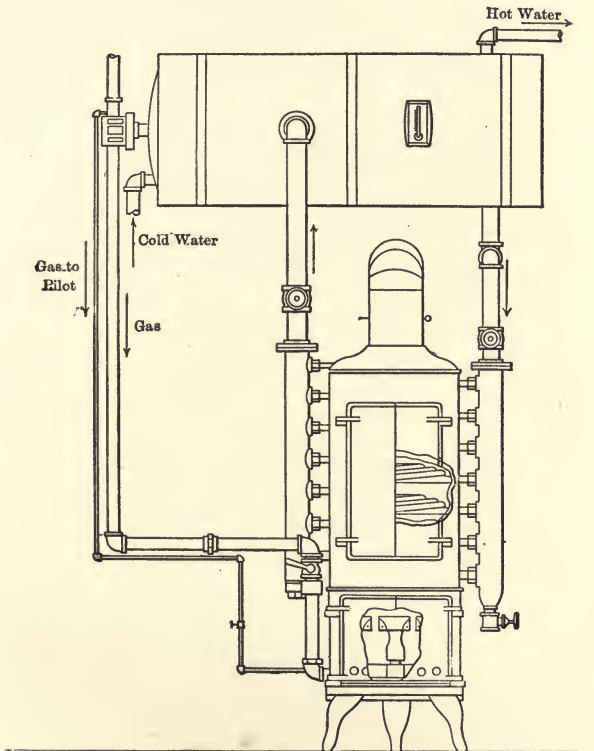


FIG. 193.—Rudd Heater and Tank.

valve which allows cold city water to enter when a faucet is open. When the water from the tank reaches 140° F., a thermostat cuts off the gas supply except for the pilot light. As soon as cold water is used this thermostat turns on the gas supply. The Rudd Heater which is somewhat similar to this is also a good one to give an instantaneous supply of hot

water. The demand for gas is so great in these heaters that special services are often run for them. Fig. 193 illustrates the attachment of one of these heaters to a storage tank with the thermostat attached to the gas supply at one end of the tank.

For small plants where steel boilers are needed on account of a desire for high-pressure steam a locomotive type of small boiler, Fig. 194, may be employed. This is the Acme Boiler of the American Radiator Co., and they are built from about 10 to 100 H.P. For large plants the water-tube or fire-tube boilers are used and these are considered in books on steam boilers and do not form a part of this work. For guidance of the student

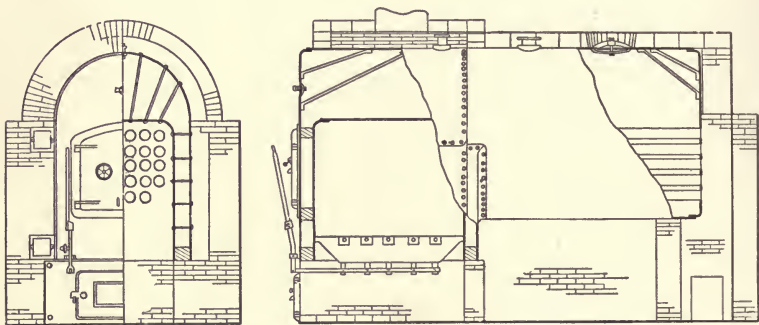


FIG. 194.—Locomotive Type of Boiler.

the following tables of a few standard furnaces and boilers are appended. Every engineer should have a supply of catalogues giving dimensions, sizes and capacities of various kinds of apparatus. The hot-air furnaces are rated by the cubic feet of volume, equivalent cubic feet or equivalent square feet of glass which the heater will care for. The hot-water and steam boilers are rated in square feet of radiation which the boiler will supply with heat. This includes the square feet of surface in radiators and piping combined. The numbers are the result of tests and in the catalogue of the American Radiator Company, they are found by taking four times the steam produced by the boiler per hour on a test in which the fire box has to be charged once

in eight hours. The allowance for piping should be made even though the pipes be covered. The piping in the ordinary direct system for steam amounts to about 25 per cent of the radiator surface, while in hot-water work the amount is from 30 to 50 per cent.

DIMENSIONS AND CAPACITY OF SPECIAL NOVELTY HEATERS

No.	Capacity in "Equivalent Cubic Feet" Heater will Heat, on the Basis of Maintaining a Temperature of 70° Above Zero in the Building when the Outside Temperature is Zero.*	Height of Heater Cased Complete. Inches.	Diameter of Firepot, Inches.	Diameter of Casings, Inches.	Height to Top of Radiator, Inches.	Diameter of Smoke Pipe, Inches.	Weight, Less Casings, Pounds.	Size of Cold-air Duct Required, Inches.
732	32000	62	19	32	50	7	820	10×20
736	41000	64	21	36	52	8	1060	12×20
740	50000	65	23	40	53	8	1280	12×26
744	61000	66	26	44	56	9	1675	12×32
748	73000	67	28	48	57	9	2125	14×32
752	76000	68	28	52	58	9	2270	14×33
756	105000	70	32	56	60	9	2850	14×48

* For definition of "Equivalent Cubic Feet" see p. 254.

DIMENSIONS AND HEATING CAPACITY OF FULLER, WARREN & CO.'S B SERIES FURNACES

(Similar to Fig. 180 but with steel radiator).

Size.	Diameter Firepot, Inches.	Diameter Casing, Inches.	Height Furnace, Inches.	Recommended Size of Air Box.	Diameter Smoke Pipes, Inches.	Number of Average Size Hot-air Pipes.	Diameter of Single Hot-air Pipe, Inches.
B 18-32	18	32	46 $\frac{5}{8}$	10×15	7	3 to 4	20
B 20-36	20	36	50 $\frac{1}{4}$	10×22	7	4 to 5	22
B 22-42	22	42	51 $\frac{7}{8}$	12×25	8	5 to 7	26
B 24-48	24	48	54 $\frac{1}{4}$	13×28	8	6 to 8	28
B 26-53	26	53	56 $\frac{1}{4}$	14×32	8	7 to 9	30
B 28-58	28	58	58 $\frac{3}{4}$	14×35	10	9 to 11	32
B 31-60	31	60	59 $\frac{3}{8}$	14×45	10	11 to 12	36
B 36-65	35	65	61	16×42	10	12 to 15	40

DIMENSIONS AND HEATING CAPACITIES, GRAFF FAULTLESS
HEATER

Size.	Diam. Firepot	Diam. Casing.	Height, Casing.	Height with Cone Top Casing.	Grate Area, Sq.ins.	No. of Flues.	Heat- ing Sur- face, Sq.ft.	Usual Cold Air Duct.	Cubic Feet Capacity.	
									Houses	Halls
19-48	19	48	48	66	283	9	131	12×36	15000	25000
22-54	22	54	54	69	380	10	152	14×40	25000	45000
25-60	25	60	60	70	491	11	176	14×48	40000	60000
30-70	30	70	70	70	707	12	219	16×60	60000	100000

DIMENSIONS AND HEATING CAPACITIES—KELSEY WARM-AIR
GENERATORS

Size Genera- tor.	Diameter of Base, Inches.	Height of Castings, Inches.	Regular Height Generator. Cased Com- plete, Inches.	Diameter Grate, Inches.	Area of Grate, Square Feet.	Heating Surface, Square Feet.	Square Feet Heating Surface to Each Square Foot of Grate Area.
24	56	59	69	24	3	161	51
27	60	60	72	27	4	176	44
30	64	64	76	30	5	211	43

Size Genera- tor.	Free Area Square Feet.	Cubic Feet Air Heated per Minute Mechan- ical.	Cubic Feet Air Heated per Minute Gravity.	Thickness of Brick Walls, Heater Case. Inches.	Inside Dimen- sions, Brick Housing One Heater. Inches.	Size of Smoke Pipe, Inches.	Weight with Cast Casing.
24	4	2800	1900	8	54×68	9	2520
27	5	3500	2300	8	57×70	9	2975
30	6	4100	2900	8	60×72	9	3425

The ratings in the following table for boilers are for hard coal. With soft coal use one size larger than tabular value. Ratings include all pipes, covered or uncovered. These amount to about 30 per cent of radiator surface. The width of the grate is equal to the number of the boiler; 15, 22, 28, 36 and 48 represent the grate width. The length is the length of the fire box.

LEADING DIMENSIONS, IDEAL HEATERS FOR STEAM AND HOT WATER

Number and Sections.	Length, Total.	Height, Total.		Width, Total.		Water Line.	Firepot	Outlet. Ins.	Smoke Pipe.	Square Feet Capacity.	
		Steam	Water	Steam	Water					Steam.	Water.
15-4	40 $\frac{7}{8}$	61 $\frac{1}{2}$	42 $\frac{1}{2}$	38 $\frac{3}{4}$	27 $\frac{1}{2}$	38 $\frac{1}{4}$	19×18	2-3	8	300	500
15-5	47 $\frac{1}{8}$	61 $\frac{1}{2}$	42 $\frac{1}{2}$	38 $\frac{3}{4}$	27 $\frac{1}{2}$	38 $\frac{1}{4}$	19×25	2-3	8	425	700
15-6	53 $\frac{3}{8}$	61 $\frac{1}{2}$	42 $\frac{1}{2}$	38 $\frac{3}{4}$	27 $\frac{1}{2}$	38 $\frac{1}{4}$	19×31	2-3	8	550	900
22-5	53 $\frac{1}{4}$	67 $\frac{1}{4}$	52 $\frac{1}{4}$	45 $\frac{1}{4}$	36	45	25×28	2-4	10	800	1300
22-6	60 $\frac{1}{4}$	67 $\frac{1}{4}$	52 $\frac{1}{4}$	45 $\frac{1}{4}$	36	45	25×35	2-4	10	1000	1650
22-7	67 $\frac{1}{4}$	67 $\frac{1}{4}$	52 $\frac{1}{4}$	45 $\frac{1}{4}$	36	45	25×42	3-4	10	1200	2000
22-8	74 $\frac{1}{4}$	67 $\frac{1}{4}$	52 $\frac{1}{4}$	45 $\frac{1}{4}$	36	45	25×49	3-4	10	1400	2350
28-5	60	75 $\frac{3}{8}$	60 $\frac{5}{8}$	53 $\frac{1}{2}$	44	52	33×32	2-4	12	1300	2150
28-6	68	75 $\frac{3}{8}$	60 $\frac{5}{8}$	53 $\frac{1}{2}$	44	52	33×40	2-4	12	1625	2675
28-7	76	75 $\frac{3}{8}$	60 $\frac{5}{8}$	53 $\frac{1}{2}$	44	52	33×48	3-4	12	1950	3200
28-8	84	75 $\frac{3}{8}$	60 $\frac{5}{8}$	53 $\frac{1}{2}$	44	52	33×56	3-4	12	2275	3725
36-5	69 $\frac{3}{8}$	83	70	64	53 $\frac{1}{4}$	60	41×36	2-5	15	2100	3450
36-6	78 $\frac{7}{8}$	83	70	64	53 $\frac{1}{4}$	60	41×45	2-5	15	2625	4325
36-7	88	83	70	64	53 $\frac{1}{4}$	60	41×54	3-5	15	3150	5200
36-8	97 $\frac{1}{8}$	83	70	64	53 $\frac{1}{4}$	60	41×63	3-5	15	3675	6050
36-9	106 $\frac{1}{4}$	83	70	64	53 $\frac{1}{4}$	60	41×73	4-5	15	4200	6925
48-6	92	97	81 $\frac{3}{4}$	80	68	70	50×53	2-6	21	4750	7825
48-7	102 $\frac{3}{4}$	97	81 $\frac{3}{4}$	80	68	70	50×64	2-6	21	5700	9400
48-8	114	97	81 $\frac{3}{4}$	80	68	70	50×75	3-6	21	6650	10975
48-9	124 $\frac{1}{4}$	97	81 $\frac{3}{4}$	80	68	70	50×86	3-6	21	7600	12550
48-10	135	97	81 $\frac{3}{4}$	80	68	70	50×90	3-6	21	8550	14125

The dimensions in this and the following table will be found to be approximately the same for other makes of sectional boilers and for that reason no other tables will be given.

In finding the size of a hot-air furnace one method is to find the amount of heat for the heat loss and for the heating of the ventilating air up to 70° and to divide this by the product of the efficiency of the furnace and the heating value of the coal and the result is the amount of coal required per hour. This is then divided by the rate of combustion, pounds of coal per square foot per hour, to find the area of the grate.

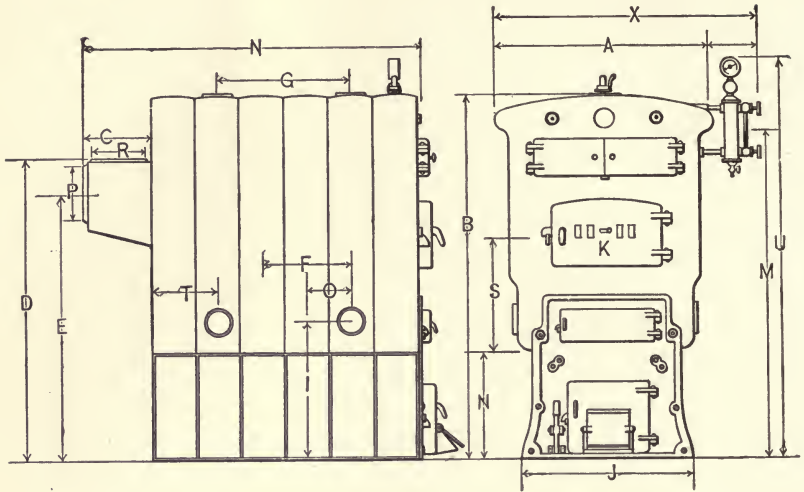


FIG. 195.—Dimensional Views of Sectional Boiler.

DIMENSIONS OF IDEAL BOILERS OF THE AMERICAN RADIATOR CO.

Size.	A	B	C	D	E	F	G	H
15" boilers...	28½	46 ³ / ₁₆	13 ⁵ / ₁₆	41 ¹ / ₈	34 ³ / ₄	12½	18 ³ / ₄	25
22" " ...	36¼	52¼	15¼	47 ³ / ₄	40½	14 ³ / ₈	21¼	28¼
28" " ...	44½	60 ⁵ / ₈	18 ³ / ₈	55 ⁷ / ₈	46¼	16	24	32
36" " ...	54¼	69 ³ / ₈	21 ¹ / ₁₆	63 ³ / ₈	52 ⁹ / ₁₆	18¼	27 ³ / ₈	36½
48" " ...	69	81 ³ / ₄	27¼	73 ¹ / ₈	59½	21½	32¼	43

Size.	I	J	K	N	O	P	S	T
15" boilers...	16 ³ / ₁₆	23 ³ / ₄	8×14	11 ³ / ₄	6¼	8	13¼	7½
22" " ...	16 ³ / ₈	29 ³ / ₈	8×14	9½	7 ³ / ₈	10	14½	8½
28" " ...	17 ⁷ / ₈	37 ¹ / ₈	9×18	10	8	12	16	9½
36" " ...	18 ⁷ / ₁₆	45 ⁷ / ₁₆	10×20	10 ¹ / ₁₆	9 ¹ / ₈	15	18 ¹ / ₁₆	10 ⁷ / ₈
48" " ...	22 ³ / ₈	58 ³ / ₈	11×19	14 ¹ / ₁₆	10 ³ / ₄	21	17 ³ / ₄	12 ³ / ₄

For X, U, and M see previous table.

If H = heat loss per hour;

V_l = leakage air in cubic feet at 70° per hour;

t_r = room temperature;

- t_o = outside area;
- h = heat per pound of coal in B.t.u.;
- eff. = efficiency of furnace = 65 per cent;
- O = rate of combustion = 4 or 5 lbs.;
- A = area of grate.

$$A = \frac{H + 0.02 V_i(t_r - t_o)}{\text{eff.} \times h \times O} \dots \dots \dots (159)$$

This area should be compared with the area of the manufacturer's heater for the same volume of building. The results should be approximately the same.

For some large buildings the area will be found to be greater than the amount to be obtained from the largest heater. In that case two or more heaters must be used. These may be placed at convenient locations in the cellar, thus shortening runs and making the heating more positive if it is possible to have proper chimneys, although in some cases twin furnaces or batteries have to be used. In this method several furnaces are placed side by side and a common bonnet is used connecting all casings (Fig. 176). This of course has the advantage over the separate arrangement of allowing one to run a single furnace at full capacity when a small amount of heat is needed in the early or late heating season.

The application of this method to the residence of Chap. V gives the following area:

1st. From heat required:

Total heat = 144000;

$$\text{Grate area} = \frac{144000}{0.60 \times 14000 \times 4} = 620 \text{ sq.in.}$$

Diam. grate = 28 ins.

2d. By method of volume:

Total volume = 27600 cu.ft.; use 25-60 Graff.

Diam. grate 25 ins. (Graff table).

3d. By method of equivalent volumes:

Room.	75 × Glass.	8 × Wall.	Volume.	Total Equivalent Volume ₂
1	9,100	2,250	3,110	14,460
2	8,000	2,700	2,160	12,860
3	2,100	750	400	3,250
4	2,600	1,540	1,390	5,530
5	3,600	1,550	1,620	6,770
6	5,850	620	2,110	8,580
7	6,400	2,400	2,910	11,710
8	3,600	1,680	2,000	7,280
9	750	390	720	1,860
10	3,150	1,520	1,700	6,370
11	3,150	1,620	1,710	6,480
12	1,650	300	480	2,430
Total	49,950	17,320	20,310	87,580

This requires a No. 752 Special Novelty Heater with 28-in. fire pot.

In selecting a furnace the endeavor should be made to get as much heating surface per square foot of grate surface as possible. There are certain furnace books which state that unless the galvanizing is burned off of the smoke pipe the furnace has not been operated to its full capacity. This burning means a high temperature of the exhaust gases and hence a great loss in them. There should be enough surface to remove the heat before these gases leave the furnace. The flue will be hot in the coldest weather, but this is no proof that the furnace is working properly. The furnace giving the lowest temperature of exhaust gases, other things being equal, is always the best furnace. The Kelsey and Graff furnaces give ratios of heating surface to grate surface of 45 : 1, and the student should have this in mind as a possible ratio. In steam-boiler work 35 to 40 is the value often used.

The flue leading from the furnace to the chimney is usually fixed by the size of the fire pot or the capacity of the furnace. An area of one-twelfth of the grate for furnaces and one-eighth of the grate area for small boilers may be used, although the method employed by many manufacturers is to use a table which gives the B.t.u. cared for by various sizes of flue.

The chimney should be at least 30 to 40 ft. high and the chim-

ney flue if round should be 2 ins. larger in diameter than the smoke pipe, while if square the size of the square is $1\frac{1}{2}$ ins. larger than the diameter of the smoke pipe or flue. If the chimney flue is rectangular the dimensions b and d should be such that

$$\frac{bd}{2(b+d)} = \text{or } > \frac{1}{4} \text{ side of square of desired size.}$$

The inside of the flue should be as smooth as possible and tight. It is well to line it with tile for fire protection, the space between tile and brick being filled with mortar. This chimney can then be built of 4-in. brick work, while an unlined flue must be made of 8-in. brick work. The top of the chimney must pass above the highest part of the building. The smoke flue must not extend beyond the inner surface of the chimney flue. It is well to have a pocket at the bottom of the chimney with a clean-out door to remove soot when necessary.

In figuring the size of the boiler to be used for a steam or hot-water installation the amount of heat required is reduced to pounds of coal by a method used above, taking the efficiency at 66 per cent. The rate of combustion may be taken as 5 to 8 lbs. of coal per square foot per hour. This gives the area of the grate, and from a table the size of boiler may be found.

Another way as mentioned above is to compute the surface of the radiation and pipes and then select size from a catalogue. These methods are now applied to the residence for a steam boiler and after that for a hot-water boiler.

For the steam system of the residence the total amount of radiation is 624 sq.ft., and the piping amounts to about 150 sq.ft., giving a total of 774 sq.ft. This requires a 22-in. 5-section Ideal Sectional Boiler. Computing this from the size of the grate surface the following results:

$$\text{Total heat supply} = 188500;$$

$$\text{Grate area} = \frac{188500}{14000 \times .66 \times 5} = 4 \text{ sq.ft}$$

Using 22 ins. width,

$$\text{Length} = \frac{4 \times 144}{22} = 26\frac{1}{4} \text{ ins.}$$

The grate surface of a 22-5 boiler is 22'' \times 28''. This method gives the same result. For the hot-water equipment the amount of heating surface is 1009 sq.ft. with 150 sq.ft. of pipe surface. The total surface is 1159 sq.ft., requiring the same size of boiler as before.

The table on page 272 gives the leading dimensions of this boiler.

The flue of the chimney for the boiler is now determined from the builder's table or else as one-eighth of the grate area. The same remarks apply here as to chimneys and flues of hot-air furnaces.

The flue in this case is to have an area of

$$A = \frac{1}{8} \times 4 \times 144 = 72 \text{ sq.in.} \quad \text{or} \quad d = 10 \text{ ins.}$$

This is the value given in table.

The boiler should be set on a foundation which forms the bottom of the ash pit. This is made of concrete. After the boiler is connected and tested it should be covered with at least two inches of asbestos or magnesia plaster.

The furnaces and boilers should be provided with the necessary gauges, shovels, pokers, cleaners and brushes or scrapers.

The question of the relative merits of the various systems is one which is difficult to decide, and one about which the various manufacturers are usually prejudiced. As far as efficiency is concerned, if the proper amount of heating surface is used, the systems are all equally good. If the loss in the chimney gases amount to 35 per cent, due to the unburned gases, dilution, hot gas or any other cause, 65 per cent of the heat of the coal must be used in the house somewhere. From 60 to 65 per cent is obtained with all furnaces or boilers if properly designed, and even better results if the fire is operated steadily. The great trouble with most hot-air furnace work and the one which is the cause of hot water and steam showing better results, is the fact that the furnace does not contain sufficient heating surface. It is not the fact that a given furnace will heat a house which should count, but that it will do it without a great loss of heat

to the chimney. The same result would happen, although it is not so common, when the boiler is much too small for an installation.

As far as convenience in installing is concerned without the use of valuable space the hot-air methods using furnace or indirect radiators are better than the direct systems, which take valuable space. The indirect system is better than the furnace system for large buildings, as the heating coils may be placed under the stack carrying the air, or air under pressure is used, while the furnace gives such a small driving pressure that there is difficulty in getting the proper flow on long lines.

The hot-air methods will supply air for ventilation, and this air can be brought in at the proper humidity. The direct-steam or hot-water system does not permit of ventilation, and when needed with this system, tempered air must be brought in by use of a fan blower and coil.

The hot-air furnace systems are out of the question for large buildings, as the circulation is not possible. The direct system is suitable in that the piping is easy to run and does not take much space.

The indirect system of heating requires the operation of a steam engine or electric motor and large air ducts and unless ventilation is necessary the direct systems are easier and more cheaply installed. In residences and office buildings where there are not many occupants in the various rooms, this method is quite extensively used. The method is positive even to the remotest parts.

Steam systems are usually cheaper than hot-water systems, because less surface is required owing to the greater unit value of the steam surface. The steam may be raised more quickly than the hot water, but unfortunately it drops more quickly also.

A hot-water system gives a more uniform temperature over a long period. It is not subject to sudden changes, and the radiators are at a lower temperature. It consumes time in bringing the house to a proper temperature after the house has been chilled.

The radiators in these two systems take valuable space, while if concealed they are difficult to repair.

As far as cost of installation is concerned the hot-air furnace system is the cheapest; then follow direct steam, direct hot water, indirect steam and indirect hot water. The furnace system for a dwelling house costs about one-half to two-thirds the cost of a steam system.

CHAPTER XI

DISTRICT HEATING

District heating or heating from a central station has been used for a long time in institutions where a number of buildings are within the radius of several hundred feet, and this same method has been extended to heat towns or portions of towns when the remote buildings have been several miles from the power house.

There are two general methods: hot water and steam. In the hot-water system a complete circuit is usually made with a pump to force the water through the **feed-water heater** into the **supply main**, from which the water passes into the buildings through a **service** or **branch pipe** and after passing through the radiators it leaves through a service and enters the **return pipe**, passing back to a **discharge tank** and thence to the pump. The heater may be an ordinary boiler using hot gases to heat the water, or a feed-water heater using steam or hot gases from a boiler to heat the water. Any form of heater may be used.

In the steam system, live steam from boilers or exhaust from engines passes through the supply main and service pipes to the buildings and in some cases the returns from the buildings are carried back to the power house, while in other cases this water of condensation is sent to the sewer.

In the hot-water system there is no reason why the pipe should be laid on a definite grade. It may follow the contour of the surface of the ground through which it passes. In this system when closed there is no power consumed in raising water to the tops of tall buildings, as the **down legs** will balance the weight of the **up legs**. In fact there is really some motive power due to the greater weight of cold water in the down legs. The water may be measured by meter and if the temperatures of inlet and

outlet are known the heat used by a building is determined. This latter factor is the difficult one to find at the consumer's building in a hot-water district system during the whole season.

The hot water may be stored in times of peak load if the system uses exhaust steam to heat the water and this water may be used in time of small steam load on the engines.

In the steam system, the steam and return mains must be put on a definite grade if the mains are to be dripped and all drips and condensation are to be returned to the power house. This grade may mean considerable cutting to accommodate the contour of the ground surface, or if this is not done all low spots must be drained and a pump used to return the drips and condensation. These two reasons have resulted, in many cases, of employing a single pipe in steam systems allowing all drips and condensations to pass into the sewer. To cut down the loss of heat in such cases, the drips from the low points of the line are taken into a customer's property and passed through a tempering coil so as to heat some of the ventilating air of the building. The same method is used with the condensation of the building and thus the water is discharged into the sewer at a very low temperature. In this system much of the exhaust steam at peak load must be wasted unless live steam is largely used for the heating plant at small loads and thus the advantages of the use of the heating plant as a by-product plant are not attained. In many cases where boilers are used for heating, the exhaust of engines at certain times is used to do part of it, as in an office building, but in a true district-heating system there is generally a waste of steam at peak engine load when exhaust steam is used.

The steam system in most cases dispenses with the use of the distributing pump, as 5 or 6 lbs. back pressure will carry the steam a considerable distance.

The pipes in a district system are installed so as to properly drain, so that expansion is cared for and so that the heat losses may be reduced to a minimum.

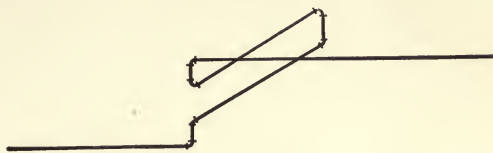
To properly drain the pipes a uniform grade must be estab-

lished, or high and low points are fixed and the pipe grade is suited to these. The low points are usually placed at manholes so that traps may be placed there which are accessible for adjustment or repair for

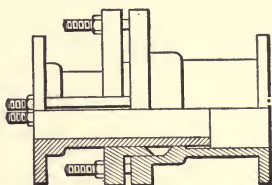
steam systems or so that drains may lead to the sewer in water systems. Drainage is important, as corrosion and rusting are the main troubles in district systems. It is claimed by advocates of the steam system that the pipes will not corrode as rapidly with steam as they will when hot water is used. The rusting out of pipes is one important objection to this system.

The expansion is cared for by **swinging ells**, by **slip expansion joints**, by **pipe bends**, by **corrugated pipes** or by a special contrivance known as a **variator**. These are shown in Fig. 196. They are all of value. The slip expansion joint is objectionable in that the leakage from the packing is hard to care for and at times the sleeve becomes incrustated so that it does not slip easily. The swinging ells, although efficient in caring for expansion, have a large amount of resistance, and this is objectionable.

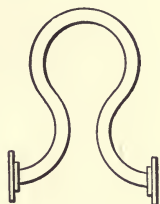
The corrugated pipe section is one which offers little resistance and allows the expansion to take



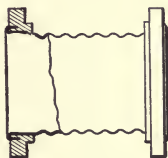
A



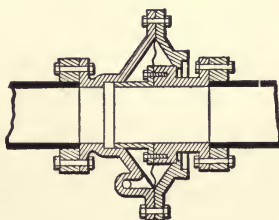
B



C



D



E

FIG. 196.—Arrangements for Expansion.

place easily. The same may be said of the pipe bend, on account of the large radius of the bend.

The variator has a **flexible diaphragm** to make the moveable joint steam tight. When used it is provided with outlets at top and bottom so that service pipes and drip pipes can be run from this point as the main casting is anchored and does not move. The variator shown is a single one in which the expansion occurs in one side. Double variators permit the pipes on each side to move relative to the casing.

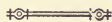
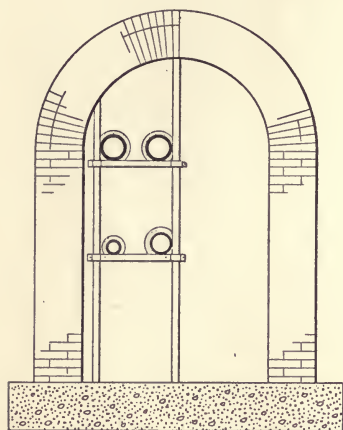


FIG. 197.—Tunnel.

To care for the radiation loss several methods are used. In some cases pipes are covered with pipe covering, while in other cases the pipe is buried in a wooden box and surrounded by shavings, or the pipe may be placed in a wooden pipe made up of thick staves joined together.

The pipe covering is used when the pipe is carried in **tunnels**, Fig. 197, or in certain forms of **conduits**, Fig. 198. Tunnels are so expensive to construct that they are rarely used except between buildings of a manufacturing plant where steam mains

are carried from a central plant to a group of buildings, or in a district system where a number of branch mains are carried from the power house to a point from which the lines radiate. The tunnel should be made sufficiently large for men to walk through and to care for the pipe lines. The clear passageway outside of the standards should be at least 24 ins. wide and $5\frac{1}{2}$ or 6 ft. high. The crowded or small tunnel may be right for the original installation when the pipes and tunnel are cold,

although even then work is difficult, but when a broken or leaky main needs repairing, work is almost impossible on account of the heat. Brick arches are used for the roofs of tunnels, beams may be used with flat brick arches or reinforced concrete may be employed. The walls should be 12 ins. thick. The floor of all tunnels should drain to one side and the gutter should drain to proper **sumps** or **sewers** so that no water will remain on the floor. Water is not only hard on workmen, but the dampness is likely to cause the covering to mildew and rot.

The pipe supports are made by fastening $1\frac{1}{2}$ - or 2-in. pipes into the floor and roof and then using a pair of strap irons clamped into position by bolts. In this way any alignment may be had with ease.

The arrangement of underground conduits as shown in Fig. 198 with pipe covering around the pipe gives a very satisfactory construction. **Split tile** or **gutter tile** are made of regular terra cotta and are manufactured with a cut from the inside surface almost to the outside, so that on striking them they break into two parts.

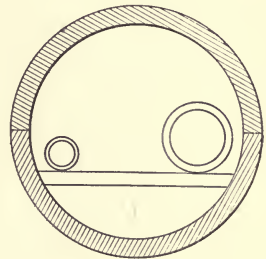


FIG. 198.—Split-tile Conduit.

One-half is then placed in the trench and well rammed into position for alignment and level, with the joints between the lengths of tile made up with mortar of one part cement and three parts sharp sand. The pipe is supported at 10- or 12-ft. intervals in one of several ways. A simple method of support is to place a piece of pipe across the tile in grooves after cutting them in the side of the tile so that the steam pipes are at the proper level in the conduit. After the cover is put on at this section concrete of one part cement, three parts sharp sand and six parts broken stone of 2 ins. size is placed around this section, filling a hole about 15 ins. long in the direction of the pipe and extending out about 10 or 12 ins. and reaching the full height of the tile. This is shown in Fig. 199. Another method is to use concrete sections at the proper distance apart in which rollers or rods as shown in Fig. 199, are used

to carry the pipe. For conduit work, the pipe should have screw connections, as flange connections leak. After the pipe is installed it should be tested with water pressure to a pressure a little above that to be carried, and if found tight, the covering is placed around the pipe, the top half of the split tile put on and joints cemented. Then the ditch is back filled, care being taken to **ram** or **tamp** the earth. At times the earth is **puddled** with water during filling to make the earth pack properly.

This tile is usually water-tight if all joints at sides and ends are properly cemented together, but at times it is well to drain the conduit by placing crushed stone beneath it and placing an unglazed tile at the center of the stone to give a waterway for

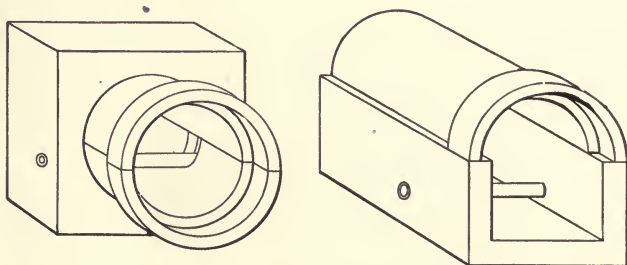


FIG. 199.—Supports for Pipes in Conduits.

the seepage. This tile of course leads to a sump or sewer. This is necessary in some cases to keep the water from working into the conduit. Of course as shown in the figure any water reaching the conduit would flow along the bottom to the manhole. Figs. 197, 198 illustrate installations where the return is brought back to the power-house, while Figs. 200–203 show systems when there is no return.

Manholes should be placed at intervals, especially at branches, and by making the floor at a lower level, the seepage water collects here and may be removed. Fig. 200 shows the sections of manholes. The tops of these should be made as tight as possible with a double cover to cut down radiation losses.

At times the insulating material, consisting of loose asbestos or magnesia, is filled into the conduit around the pipe, and in that

case no water should be allowed to enter. Figs. 201, 202 and 203 show various methods of carrying single pipes. In Fig. 201 a wooden box is made up with air spaces. These spaces are filled with some substance to break up the air currents. The pipe is surrounded by some filler to stop air circulation and the whole box is covered with pitch. In Fig. 202 the box is made of concrete and

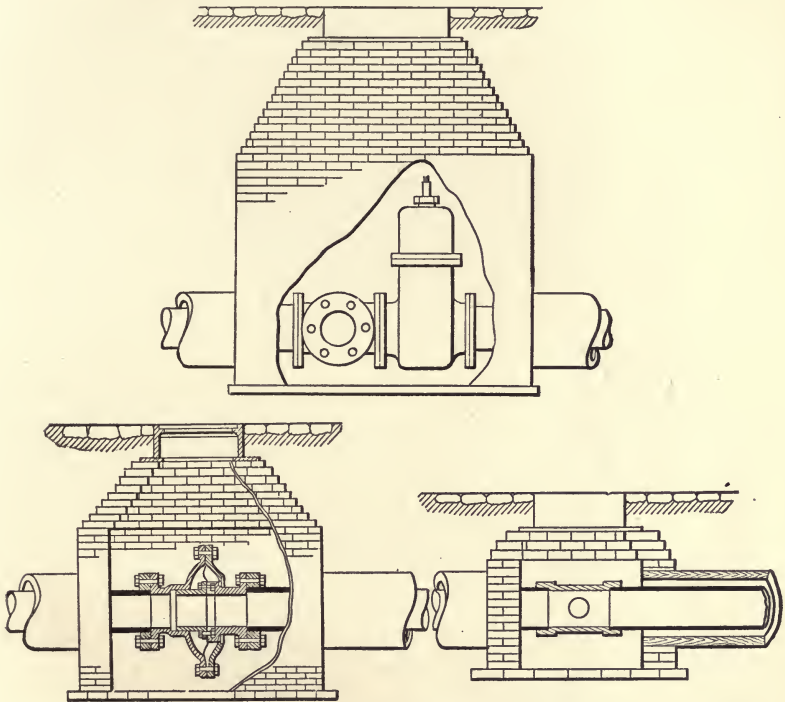


FIG. 200.—American District System.

in Fig. 203 the pipe is surrounded by a wooden covering and the whole is surrounded by concrete. In the last three methods the pipe is carried by some form of roller. Fig. 200 shows the method used by the American District Steam Co. The pipe or casing, Fig. 204, is made of wood staves 4 ins. thick locked together by tongues, wrapped with wire and lined with tin. The pipe has an air space of 1 in. between the tin and the sheet asbestos which is placed around the pipe. The staves are treated with creosote

after shaping and after banding with a spiral of $\frac{3}{16}$ in. galvanized wire embedded into the wood. A $3\frac{1}{2}$ -in. bell and spigot is formed at the end by turning after the pipe is made. The exterior is then treated with asphaltum, pitch and sawdust.

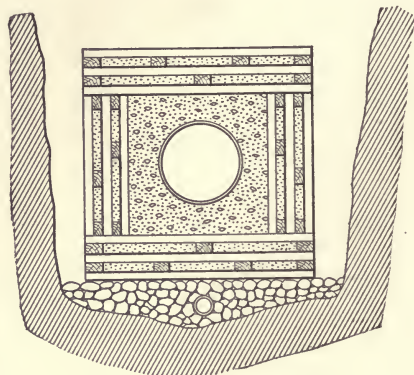


FIG. 201.—Wooden Conduit.

The American District Steam Co. install their apparatus as shown in Fig. 200, using manholes at all special fittings such as variators (see Fig. 196) or at anchor specials. These anchor

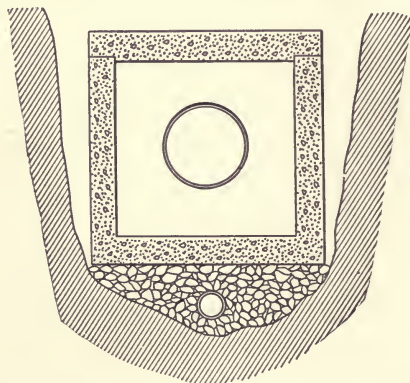


FIG. 202.—Concrete Conduit.

specials are usually tees or crosses which supply the service connections. The variators also are points for connecting service mains or drips, as the main body is always so anchored that it

is a fixed point. The variators care for 50 ft. of pipe so that double variators are 100 ft. apart with an anchor special between them. In this way outlets may be made at 50-ft. intervals. The third manhole is used to control certain sections of the pipe.

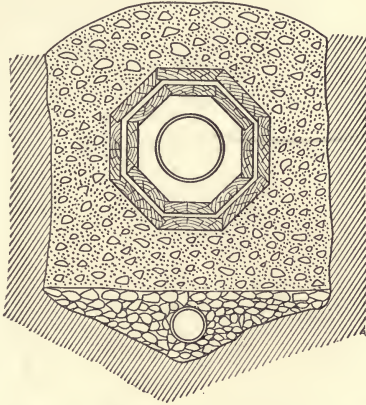


FIG. 203.—Wooden Conduit in Concrete.

In laying out a district system the map of the buildings to be supplied by a plant should be laid out and then the profile of the line made, giving the cuts at the various points. In many cases these lines may be placed in alleys or small streets parallel to the main streets, in which the pavement is not so expensive

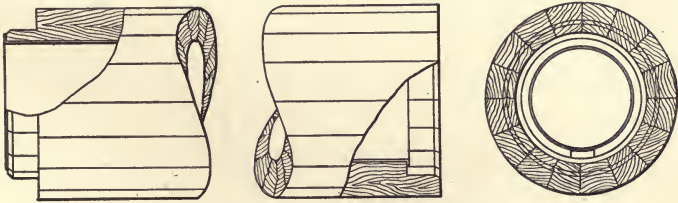


FIG. 204.—Wooden Covering.

and which will not interfere with traffic during construction or repairs.

The steam or water used by the various buildings is then found and allowance is made for future growth and extension, and then the pressure to be expected at the various points

is assumed. By starting at the dead ends of the mains and working toward the source, the amount of steam or water and the pressure drop on any line are known. From this the size of the pipe may be found for the assumed pressures.

There is a heat loss from the pipe line due to radiation, but in pipes lying buried in conduits surrounded by earth this quantity is much smaller than in pipes carried in a room. Experiment seems to indicate a formula for the heat per hour in the form

$$H = R \times \text{sq.ft. of external pipe surface.} \quad \dots \quad (160)$$

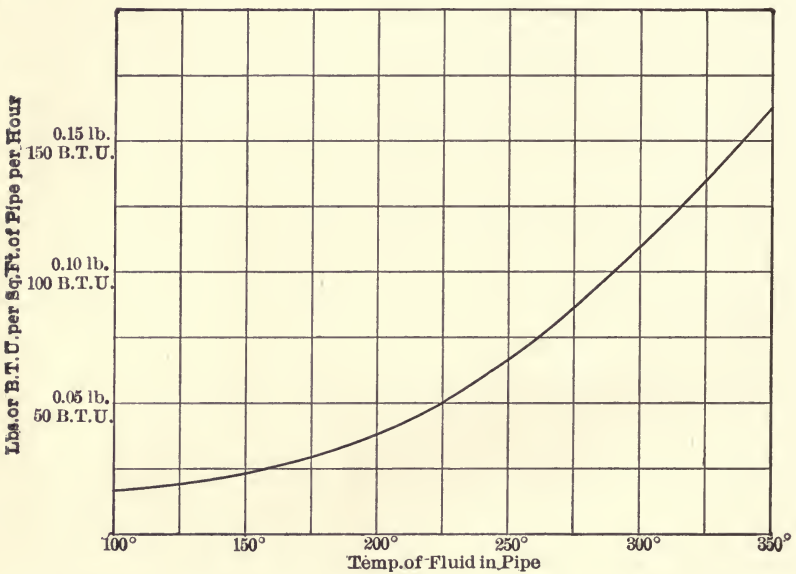


FIG. 205.—Curve for Values of *R*.

R depends on the steam temperature and is given by the curve shown in Fig. 205. This curve has been constructed from various experimental results. It is given in pounds of steam condensed per square foot per hour or in B.t.u. per hour. The value of 0.03 lbs. per hr. is used for water and 0.05 for 5 lbs. steam. The constant *R* may vary considerably, increasing as the covering becomes old or water-soaked.

Some authors consider a loss due to friction, but this is not a loss, as the heat produced by this friction remains in the steam and although the pressure may drop there is no diminution of heat outside of radiation.

This method of assuming pressures will not give the best size necessarily, unless the assumed pressure drop from the plant to the end of the line is the maximum possible amount. In that case the pipe is the best size, but when the pressure can be changed a smaller pipe with more drop might be more economical. This problem of economical size of pipe divides itself into two problems, one for water and one for steam.

In the case of the hot-water system it is well to determine the cost of installing a certain size of main and then the cost of pumping the water through the main and the heat loss from it. After this is done a larger pipe is taken and in this the cost of pumping will be less, due to the lower velocity, but the installation will cost more and there will be a greater radiation loss. If the sum of the cost of pumping, of radiation and of interest, depreciation and taxes be less for the large pipe than for the smaller one, the large pipe should be used. If, however, the interest, taxes and insurance, radiation and pumping cost more than the sum of these quantities for the small pipe, the latter should be used in preference to the larger pipe, and moreover it would pay to investigate a still smaller pipe. In this way one size is found which gives the most economical result. As an example suppose 100,000 lbs. of water is to be carried per hour to a building 3000 ft. from the power house at 170° and returned at 140° F. The size of the main is required.

1st. Suppose the pressure fall is fixed at 10 lbs. per square inch and this can and should be used. This permits of one answer only, and assuming a 4-lb. drop in the building, 6 lbs. may be used; 6 lbs. per square inch is equivalent to $\frac{6 \times 144}{61.08} = 14.1$ ft. of water at 155° F. (at the mean temperature).

$$Q = \frac{100000}{60 \times 60 \times 61.08} = 0.45 \text{ cu.ft. per sec.}$$

The formula to be used to determine the quantity depends on that used to determine the friction head. The formulae mentioned in Chapter VII are not considered, as the loss in district systems is so great that more accurate methods must be employed. The approximate formula of Chapter VII was sufficient, as the runs were short in that work. The formula from the work of Williams and Haven as given by I. N. Evans is

$$h = 0.00035l \frac{v^{1.85}}{d^{1.166}}, \dots \dots \dots (161)$$

while the formula

$$h = f \frac{l}{d} \frac{v^2}{2g}, \dots \dots \dots (162)$$

which has been used for so many years, has a constant term *f* which depends on the *v* and *d*. The tabular values of *f* which have been used indicate approximately

$$f = \frac{0.025}{v^{\frac{1}{2}} d^{\frac{1}{4}}}, \dots \dots \dots (163)$$

Using this the formula becomes

$$h = 0.0004l \frac{v^{1.875}}{d^{1.25}}, \dots \dots \dots (164)$$

$$h = 0.0004l \frac{v^{1.875}}{d^{1.25}} = 0.0004 \frac{Q^{1.875}}{\left(\frac{\pi}{4}\right)^{1.875} d^5} = 0.0006l \frac{Q^{1.875}}{d^{5.00}},$$

$$d = \left(\frac{0.0006lQ^{1.87}}{h} \right)^{\frac{1}{5}}$$

For the problem

$$d = \left(\frac{0.0006 \times 6000 \times 0.45^{1.875}}{14.1} \right)^{\frac{1}{5}} = 0.57 = 7 \text{ in.}$$

2d. Suppose the pressure is not limited and it is desired to know whether a 6-in. or 8-in. pipe would be better than a 7 in., supposing 1 H.P. hr. is worth $1\frac{1}{2}$ cts. to this company.

$$\text{Velocity in 6-in. line} = \frac{0.45}{0.201} = 2.2 \text{ ft. per second;}$$

$$\text{Velocity in 7-in. line} = \frac{0.45}{0.269} = 1.7 \text{ ft. per second;}$$

$$\text{Velocity in 8-in. line} = \frac{0.45}{0.3474} = 1.3 \text{ ft. per second;}$$

$$\text{Loss in 6-in. line} = \frac{0.0004 \times 6000}{0.5^{1.25}} \times 2.2^{1.875} = 25.2 \text{ ft.};$$

$$\text{Loss in 7-in. line} = \frac{0.0004 \times 6000}{0.583^{1.25}} \times 1.7^{1.875} = 12.7 \text{ ft.};$$

$$\text{Loss in 8-in. line} = \frac{0.0004 \times 6000}{0.667^{1.25}} \times 1.3^{1.875} = 6.5 \text{ ft.}$$

Friction horse-power:

$$\text{6-in. line, } \frac{25.2 \times 61.08 \times .45}{550} = 1.26;$$

$$\text{7-in. line, } \frac{12.7 \times 61.08 \times .45}{550} = 0.61;$$

$$\text{8-in. line, } \frac{6.5 \times 61.08 \times 0.45}{550} = 0.32.$$

Total horse-power to drive pump, assuming a 50 per cent overall efficiency, is as follows:

6-in. line, 2.52;

7-in. line, 1.22;

8-in. line, 0.64.

Cost of power, using 200 heating days at an average of 15 hrs. at full capacity:

$$6\text{-in. line} = 2.52 \times 200 \times 15 \times 1\frac{1}{2} = \$113.50;$$

$$7\text{-in. line} = 1.22 \times 45.00 = 54.90;$$

$$8\text{-in. line} = .64 \times 45.00 = 33.80.$$

Radiation loss:

$$6 \text{ ins., } 0.03 \times 1.734 \times 6000 = 312 \text{ lbs. of steam;}$$

$$7 \text{ ins., } 0.03 \times 1.996 \times 6000 = 359 \text{ lbs. of steam;}$$

$$8 \text{ ins., } 0.03 \times 2.255 \times 6000 = 406 \text{ lbs. of steam.}$$

With heat worth 35 cts. per 1,000 lbs. of steam the cost of the radiation is

$$6 \text{ ins., } \frac{312 \times 200 \times 15}{1000} \times 0.35 = \$327.60;$$

$$7 \text{ ins., } \frac{359 \times 200 \times 15}{1000} \times 0.35 = 376.50;$$

$$8 \text{ ins., } \frac{406 \times 200 \times 15}{1000} \times 0.35 = 426.00.$$

The cost of excavation and back filling for the pipe will cost 20 to 30 cts. a cubic yard, but this item will not vary much, if at all, for the different sizes of pipes.

The list prices per foot of the pipes and covering are as follows:

	For Pipe	For Wood Casing
6 ins.,	\$1.88	\$1.94
7 ins.,	2.35	2.16
8 ins.,	2.82	2.44

Assume discount as 70-10-10 on pipe, 50-20 on casing.

Cost of casing:

$$6 \text{ ins.} = \$1.94 \times 6000 \times .40 = \$4660.00;$$

$$7 \text{ ins.} = 2.16 \times 6000 \times .40 = 5180.00;$$

$$8 \text{ ins.} = 2.44 \times 6000 \times .40 = 5860.00;$$

Cost of pipe:

$$6 \text{ ins.} = 1.88 \times 6000 \times 0.243 = \$2740.00;$$

$$7 \text{ ins.} = 2.35 \times 6000 \times 0.243 = 3420.00;$$

$$8 \text{ ins.} = 2.82 \times 6000 \times 0.243 = 4115.00.$$

If interest amounts to 5 per cent, taxes to $1\frac{1}{2}$ per cent, and depreciation to 4.6 per cent (if the life is taken at fifteen years), the yearly cost of the investment will be:

$$6 \text{ ins.} = 7400.00 \times 11.1 = \$821.40;$$

$$7 \text{ ins.} = 8600.00 \times 11.1 = 954.60;$$

$$8 \text{ ins.} = 9975.00 \times 11.1 = 1107.22.$$

The cost of installation will be taken to be the same for each line, as will practically be the case, or it might be considered to be included as a per cent of the cost of the pipe and casing and figured in as part of investment just computed. Yearly cost then becomes:

	For 6 ins.	For 7 ins.	For 8 ins.
Interest	\$821.40	\$954.60	\$1107.22
Radiation	327.50	376.50	426.00
Power	113.50	54.90	33.80
Total cost	<u>\$1272.40</u>	<u>\$1386.00</u>	<u>\$1567.02</u>

This shows that the smallest pipe is best. The yearly cost should now be worked out for a 5-in. pipe and the result would show a slight decrease in the cost (\$1242.61). The use of a 4-in. pipe would increase the cost of power so much that the total cost would be increased. The difference between the yearly cost of the 5 and 6 in. is so slight that it would be well to use the 6 in., since if more water were needed this would care for it with less cost of power and the other items would be the same. For that reason it would be cheaper to operate than the 5 in. except under the conditions of the problem.

For steam pipe there is not much power consumed, and if the pressure can be taken to a high point it may pay to use high-pressure steam, since this steam is more dense. The main

consideration in steam-pipe work is whether or not the small pipe with a higher temperature difference will radiate so much heat that its cost added to the yearly cost on the investment will be more or less than on a larger pipe with low-pressure steam, in which the cost of installation is greater, but the heat loss is less. If the computation is made it will be found that the yearly cost is less on the small pipe carrying high-pressure steam. If possible live steam should be used in the transmission main. When exhaust steam is to be used, large pipes have to be employed to accommodate the low-density steam. The method of calculation is similar to that used before. The items considered are the cost per year for investment and loss of heat at different pressures, using the various sizes of pipes resulting from the use of the formula for size in terms of the drop, steam pressure and length (see p. 139). These systems are applied as shown in Figs. 206, 207, the first being for a town while the second is for an institution. Each of these is used in the same manner by assuming pressure drops and quantities from which the diameters are found, and then an economic study is made with a change in certain of the conditions, the idea being to make the operating cost and fixed charges as small as possible.

The costs are usually figured per 1000 lbs. of steam, per 1,000,000 B.t.u. or per square foot of radiation. The charges are about 60 cts. per 1000 lbs. of steam and 35 cts. per square foot of steam radiation per season and 20 cts. per square foot of hot-water radiation.

Gifford states that the annual charge for heat should be divided monthly in the following proportion:

October	3	per cent
November	12	“
December	18	“
January	21	“
February	19	“
March	13	“
April	8	“
May	3	“

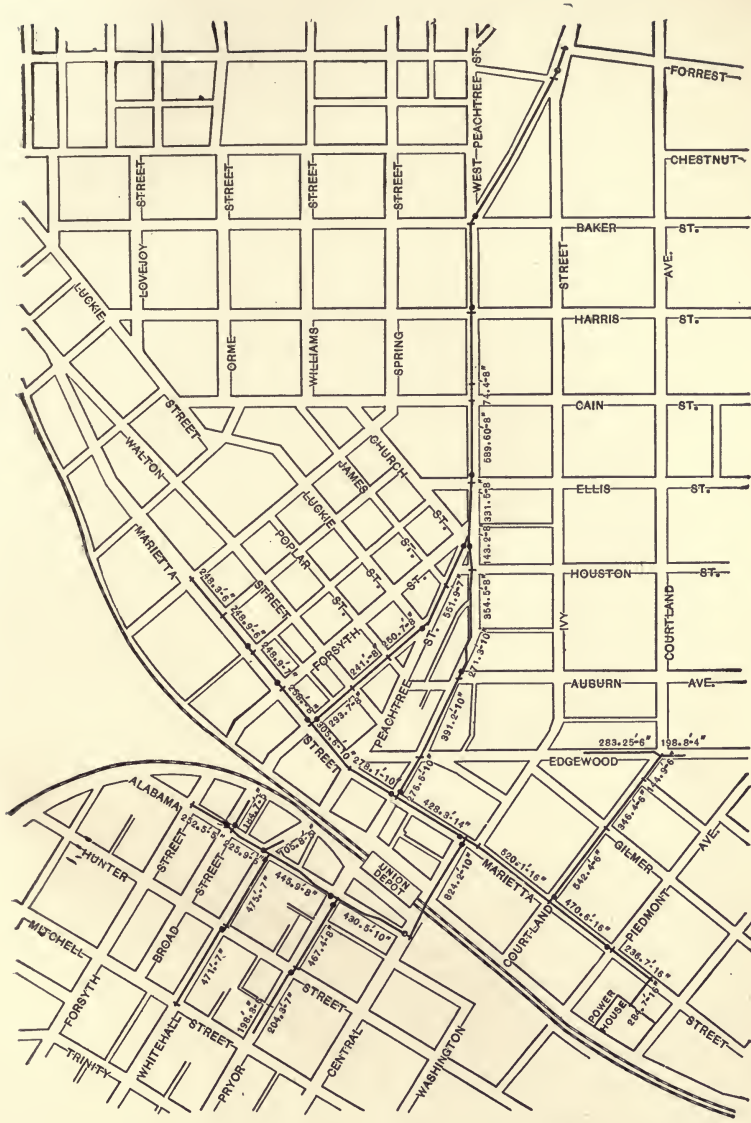


FIG. 206.—District Heating of a City. (From Catalogue of American District Heating Co.)

The following might also be used in northern climates, where March is a hard month:

October.....	5 per cent
November.....	10 “
December.....	15 “
January.....	20 “
February.....	20 “
March.....	15 “
April.....	10 “
May.....	5 “
	100 per cent

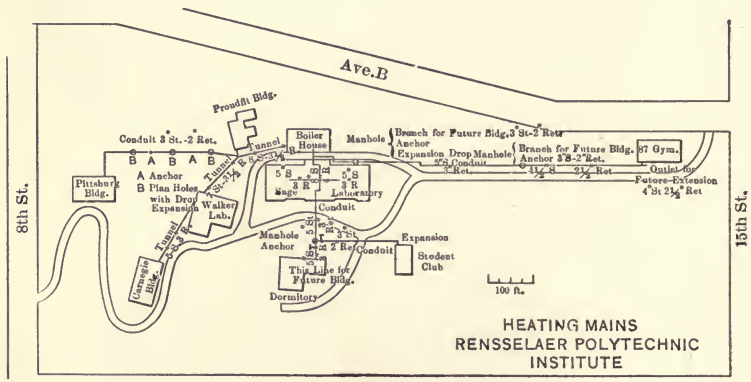


FIG. 207.—District Heating of an Institution.

The value of district systems is due to the fact that a uniform heat is always available and there is no dirt or expense for maintaining or caring for the furnace, ashes and coal. The expense may be higher than the cost of coal, but should not exceed the cost plus the cost of operating the furnace. The absence of dust from coal and ashes has undoubtedly a money value which, although it is not possible to fix, nevertheless should be considered in estimating the value of this method of heating.

This system also gives a cleaner town in that the numerous smoking chimneys are taken away from the location of the high-grade property.

In maintaining mains care must be taken to stop all leaks at any place, for very small openings are great sources of loss. A $\frac{1}{4}$ -in. hole, for instance, at 5 lbs. steam pressure, would discharge about 36,000 lbs. of steam per month, using over two tons of coal.

The pipe should be full-weight pipe (specially selected to see that all is uniform and of proper weight), and the fittings should be extra heavy. All breaks are guarded against, as it is very expensive to repair pipes installed in the usual manner. In designing tunnels it is well to place a number of inclined openings leading to the ground level at intervals for the introduction of new pipes when necessary.

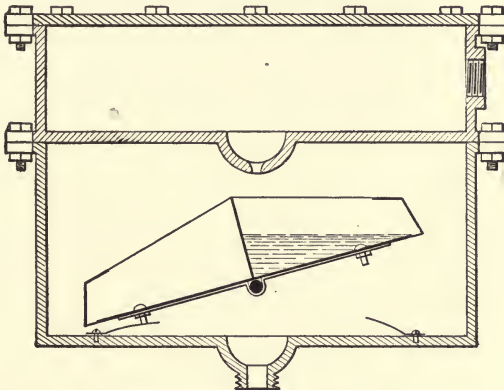


FIG. 208.—Oscillating Water Meter.

When pipes and conduits are placed in streets the back fill must be well rammed and puddled to avoid subsequent settling, and on top of this fill a good concrete base should be put over the hole to support the pavement.

The meters used for steam condensation should be of the oscillating bucket type, Fig. 208, so that no water will leak at times of small discharge. The trouble with many meters is the fact that they allow a small rate of flow to remain unrecorded and for that reason the type above is of value. To regulate the flow of water into buildings in the hot-water system it has been suggested to insert standard orifices in the service con-

nections. These are made smaller near the station, where the difference of pressure between main and return is greater than at the end. These smaller orifices prevent a large quantity from passing into the services near the station, which would mean a higher temperature on the return than should exist at this building. It would indicate that the water was not doing its proper work, as its return temperature was too high. By observing the return temperature from a building the correctness of the orifice may be known. If low, the orifice is too small; if high, the orifice is too large.

There should be constant inspection of all manholes for the evidences of leakage from expansion joints, valves, conductors, and to see if traps are clean. These should be blown out and cleaned at regular intervals.

One form of district heating is that used for **car heating** on railroad systems. The heat is supplied from the locomotive. A 1-in. extra-heavy pipe is taken from the dome through a stop valve and a pressure regulator, the outlet of which is $1\frac{1}{4}$ and after passing the T for the gauge, the line enlarges to $1\frac{1}{2}$ ins. It is connected to $1\frac{1}{2}$ -in. covered pipes under the tender and cars by means of special **steam hose couplers**. Each coupler is provided with a **trap** which opens to drain the line as soon as steam is cut from it.

The pressure regulator is a spring-controlled reducing pressure valve of such a construction that it will operate properly on the moving locomotive.

The $1\frac{1}{2}$ -in. train pipe has a $1\frac{1}{4}$ -in. branch under the center of each car and this is split into two 1-in. branches, each one running to a 1-in. angle valve under a seat near the center of each side. The line on each side enlarges to 2 ins. and a loop is taken along the side of the car, care being exercised to use a right-angle section at one end to care for expansion, as was done in the corner coil of Fig. 63. The condensation in this line is then carried through a trap and discharged from the system beneath the car. This trap is made on the same principle as was used in the thermograde motor valve, Fig. 79. As soon as steam strikes it the valve closes. There is usually a trap

which opens as soon as the pressure is cut off, to ensure that all water is drawn off, thus preventing freezing.

Each car is equipped with a **train pipe valve** at the center of the car or at the end. These valves are used to close off the train line at the last car. They are provided with traps to relieve the line of water.

The direct steam-heating system is valuable in that it is rapid, effective, and when cut out there is no danger of freezing. The objection is that there is no storage and as soon as steam is cut off the car begins to cool. For storage and for a less intense heat hot water is used. This is heated in a coil placed in a fire pot of a heater, or steam from a train line is introduced into a small pipe within the coil and heats the water. This avoids the cracking noises which occur with steam lines at times. By using brine the liquid is prevented from freezing. At times for storage terra cotta bricks are surrounded by a large pipe and these are heated by the steam. The tile retains the heat for some time.

CHAPTER XII

TEMPERATURE CONTROL AND DRYING BY AIR

THE **control of temperature** is more or less important for the comfort of those occupying the rooms and for the economical operation of a plant. The great trouble with various systems for this purpose is the disarrangement of parts due to the delicacy of the apparatus and to the maltreatment which they receive from those who do not understand the adjustment of them. When they cease to operate properly the radiator valves

cannot be opened or closed and hence the occupants of a room may be driven out by the cold or heat without power to correct the evil.

There are several simple as well as some complicated systems applied to all methods of heating.

The Johnson system of temperature control consists of a **thermostat** which controls the admission of air from a compressed-air supply into a case above a diaphragm at the top of the valve, Fig. 209. This admission of air closes the valve. The air is

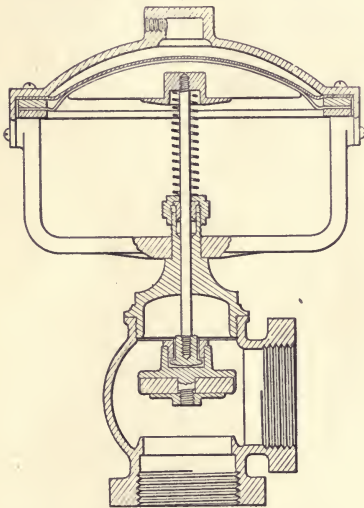


FIG. 209.—Section of Diaphragm Valve.

compressed by electric motors or small water motors.

At a given pressure of 14 or 15 lbs. in the storage tank the motors are shut off. The air is then carried to the thermostats

in small pipes and from these it is conducted through other small pipes to the valves. The pipes are small in diameter and are usually concealed in the plaster. The air used is screened when entering the apparatus to prevent foreign particles entering the system.

The thermostat is really a valve controlling the admission of air under pressure into the pipe leading to the radiator valve, or permitting the air to escape the valve line. The thermostat, Fig. 210, is carried on the plate *A*, which is attached to the wall and the brass block *B*, which forms the end of the supply pipe *C* and the valve pipe or motor pipe *D*. The upper of the two pin valves *EE* is used to shut off the air if the thermostat is taken off, while the lower one regulates the flow to the valve motor. From the upper valve *E* the air is led by a crooked passage to the air valve *F*. This valve is closed in the figure and if there is any excess air in *D* it can escape around the spindle *G*. When the spindle is moved out the disc *F* opens the pipe *C* but closes the annular space around *G* so that the air cannot escape, and hence it passes into *D*.

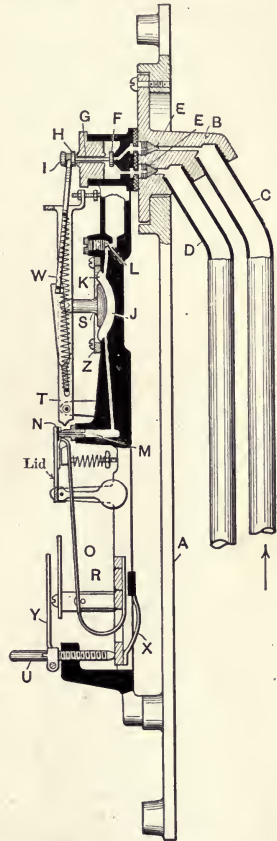


FIG. 210.—Mechanism of Johnson Thermostat.

The rubber diaphragm *K* rests against a disc *S*, forming an air motor at *J*. By means of a side tube air is brought to the valve *L* from *C*. The valve *L* is adjustable, so that the amount of air admitted into the space *J* may be fixed. *M* is the outlet to this space, closed by the valve *N*. *N* is fastened to the lid, which is controlled by a bent piece of metal *O*. This piece is fastened rigidly to the block *R* at one end, but the other end

is free to move and operate the seat *N*. The strip is composed of strips of steel and brass and bends as its temperature changes, due to a difference in the coefficients of expansion of the two metals. The strip will move to the right with a fall in temperature and this will close the valve *N*, causing the pressure in *J* to increase until it is sufficiently great to move the piston *S* and the lever *T*, thus moving *F* and *I* to right and closing the

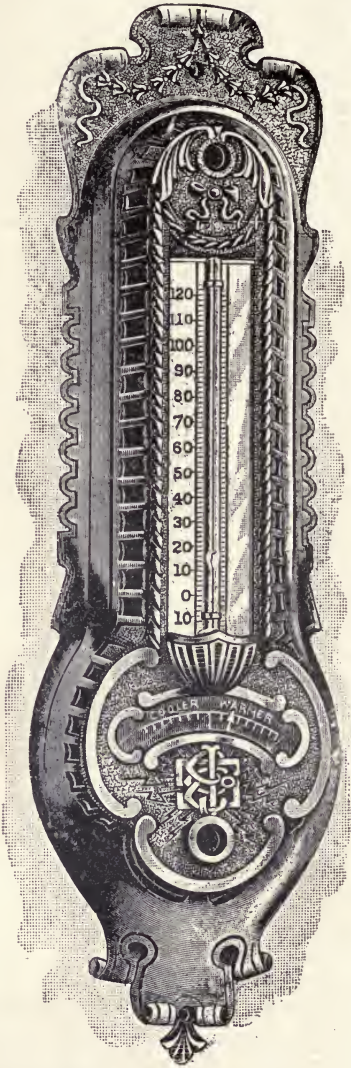
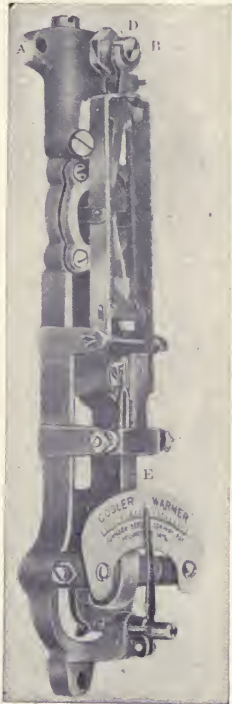


FIG. 211.—Johnson Regulator.

opening from *C*, the springs causing the motion of *F* to be rapid when it occurs. The steam valve now opens as air leaks from

D and after the room becomes hot *O* moves to the left; air leaks from *M* and *L*, *S* moves to the right, moving *H*, *G*, and *F* to the left. This allows air to enter *D* and shuts off the valves on the radiator. The turning of the screw at *U* operates on the spring

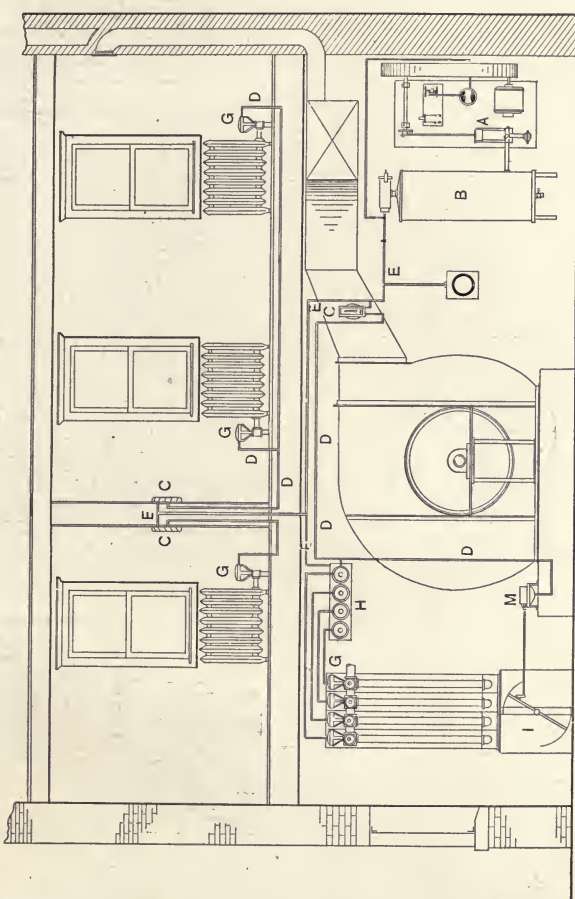


FIG. 212.—Powers System of Temperature Control.

X and so fixes the temperature at which *O* will open the valve *N*. This system as here described is applied to any number of thermostats. A compressed-air main is connected to all of the thermostats and these connect the air to a group of radiators in a room. The same system may be applied to regulate dampers or coils in indirect heating, as will be described in the next system.

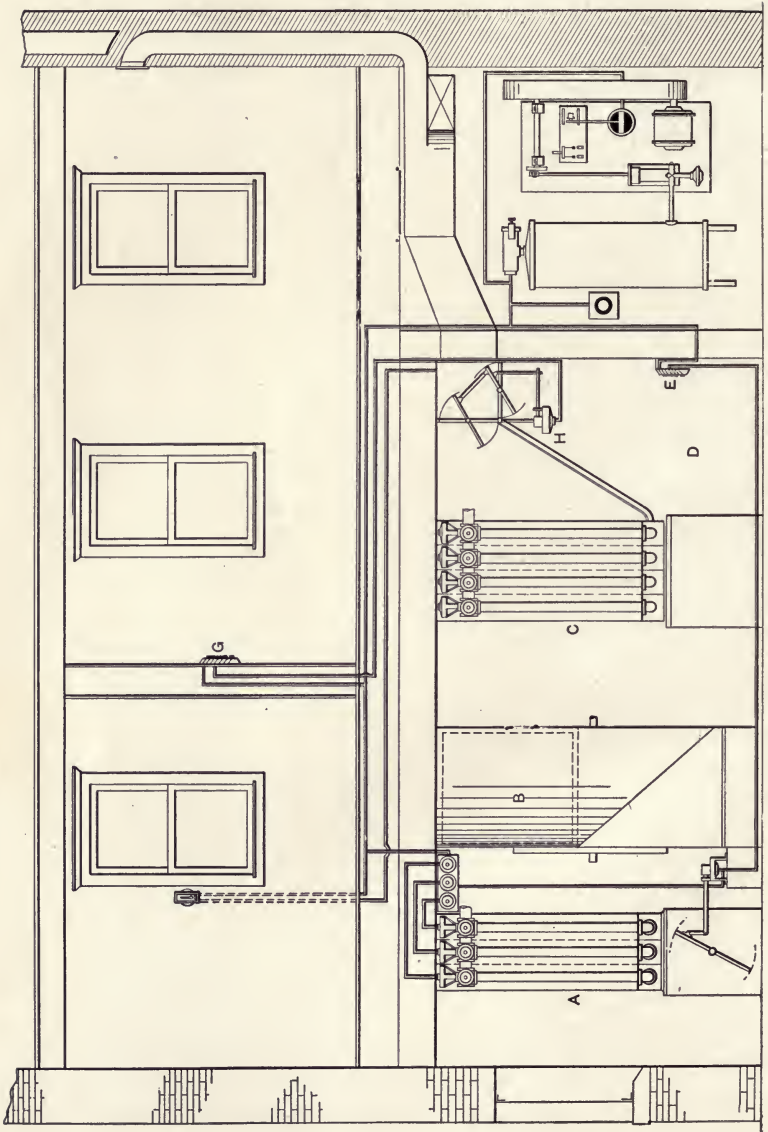


FIG. 213.—Powers System Applied to Indirect Heating.

The external appearance of the thermostat is shown in Fig 211. The hole at the bottom is for the key adjustment for the temperature at which action takes place.

The Powers system, as applied to a plant using radiators for the heat loss and tempered air for ventilation so that the air for all rooms is at the same temperature, is given in detail in Fig. 212. It consists of motor-driven air compressor *A* and storage tank *B* connected by pipes to a series of thermostats *C*, and the thermostats are then connected by pipes to the motor valves *G*, *G* and *G* and the damper operator *M*. The ther-

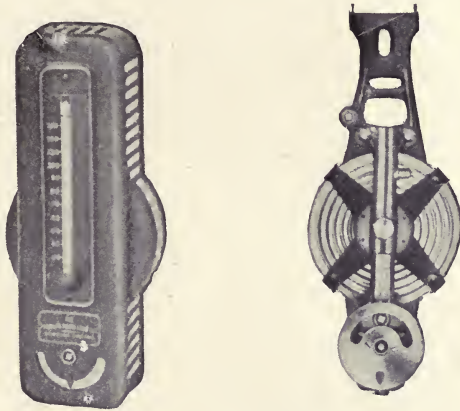


FIG. 214.—Powers Thermostat.

mostat allows air to enter the casing at the top of the valves *G* when the temperature is high and this air presses against a diaphragm attached to the valve stem, closing the valve, as described before. The thermostat on the side of the tempered air duct is heated by the metal of the case and operates the relays *H*, by which the valves on the various coils are thrown into action in succession, the action of the air when thrown on by the hot thermostat immediately opening the damper *I* by-passing the coil and then shutting down the sections.

Fig. 213 illustrates a single-duct system of indirect heating. The fan *B* draws air through the tempering coil *A*, sending part of it through the main coil *C* and part beneath the coil into the tempered air chamber *D*. The thermostat *E* keeps the tem-

perature at the right point by acting on the by-pass damper through the diaphragm motor and by acting on the coils *A* through the diaphragm valves.

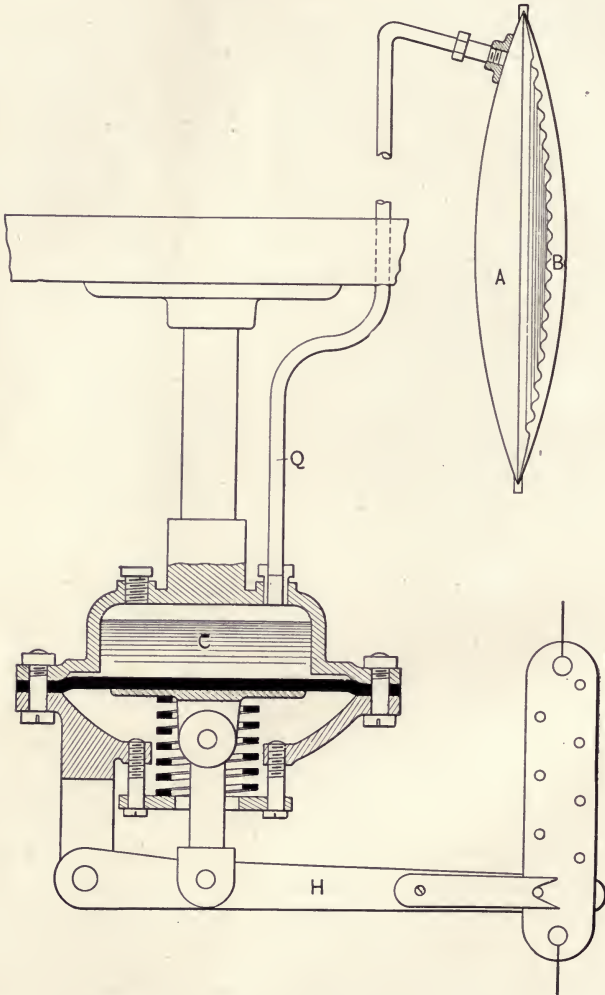


FIG. 215.—Powers Damper Regulator.

The thermostats *G* in the rooms act on the diaphragm motors *H* and operate the mixing dampers, cutting off the hot air when

more tempered air is needed. Each room will have a diaphragm motor acting on the mixing chamber dampers for its line.

The motor part of the thermostat consists of a hollow corrugated vessel filled with a volatile fluid. The frame, Fig. 214, of the apparatus is covered with a casing to protect the motor parts. The pointer is attached to a screw which so moves the frame that a higher temperature is required for the volatile fluid to produce sufficient pressure to operate the two valves.

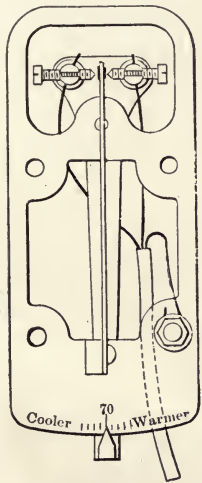


FIG. 216.

Honeywell Thermostat.

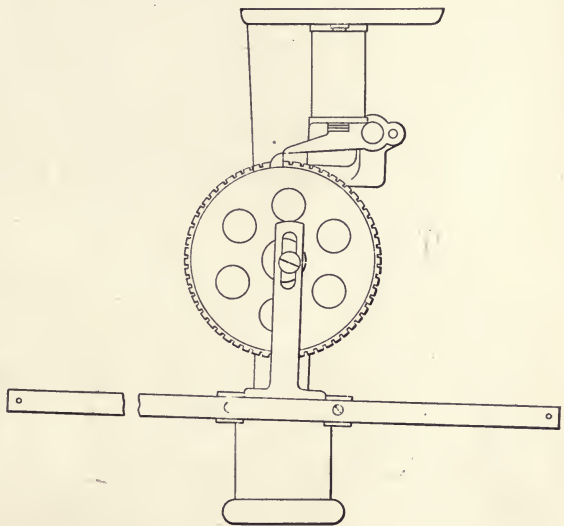


FIG. 217.

Honeywell Solenoid Regulator.

Fig. 215 illustrates a method of using this corrugated vessel to produce the pressure to operate a diaphragm regulator. The metal casing, 12 ins. diameter, has a corrugated partition within it, the space on one side containing a volatile liquid. When this heats to a definite temperature some of the liquid volatilizes and compresses the air on the other side of the diaphragm and this acts on the water seal on top of the rubber diaphragm of the motor below and forces the lever down. In this way boiler draft or furnace draft may be regulated. The temperature

around the vessel fixes the amount of volatilization and thus the position of the lever. For controlling heaters or furnaces the Honeywell and the Minneapolis systems may also be used.

The Honeywell temperature regulator consists of a thermostat, Fig. 216, in which the relative expansion of two dissimilar

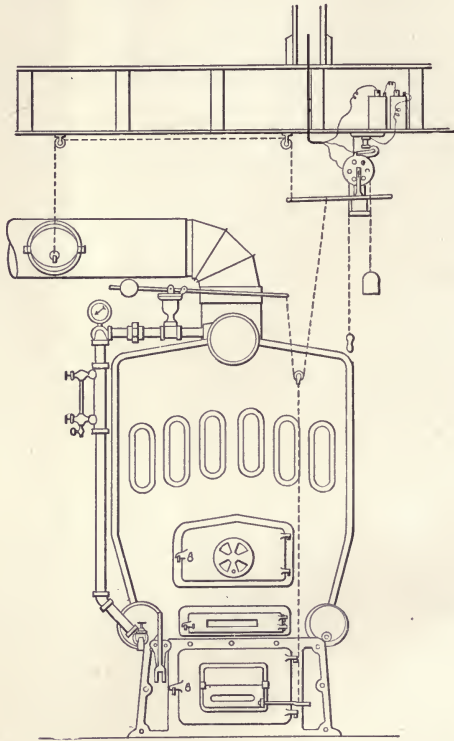


FIG. 218.—Honeywell Regulator.

metals causes a strip to touch one of the contact points, the temperature at which the change from one to the other occurs depending on the position of the pointer at the bottom. This thermostat operates a solenoid, Figs. 217, 218, on the damper motor. If the temperature is high one point touches and the current causes the solenoid to lift the pawl, and a weight on a chain, Fig. 218, will cause the wheel to rotate, closing the draft

and opening the check. The driving weight in this system has to be pulled up at intervals. This system is governed by one thermostat which is placed in the living room or one of the rooms which will have a fair average temperature. The system is simple and not expensive to maintain or control.

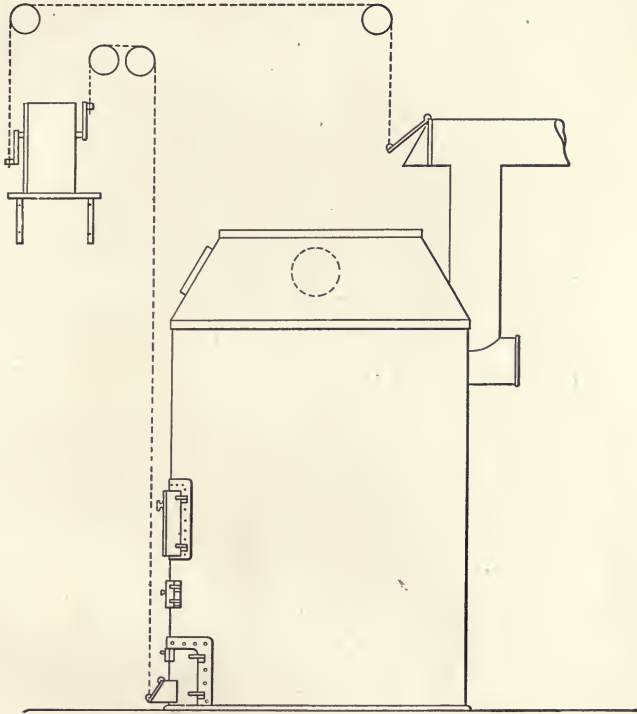


FIG. 219.—Minneapolis Regulator.

In the Minneapolis heat regulator the thermostat, Fig. 221, releases a pawl on a clock which then revolves a driving wheel one-half revolution. The clock mechanism, Fig. 220, is such that the pawl is controlled by one point of the two contact points during one-half of the revolution while the other one controls the other. Thus, in Fig. 219, the position of the clock motor corresponds to the high temperature contact as the check damper is open and the draft is closed on a hot-air furnace.

If the room gets chilled the thermostat, Fig. 221, touches on the other point and the cranks are moved through one-half of a revolution. These last two systems are applied with one thermostat for the heater at some central point.

In residences the doors are mostly open during the day, and hence for such this single thermostat is satisfactory. Such apparatus is of value for night heating where perishable objects

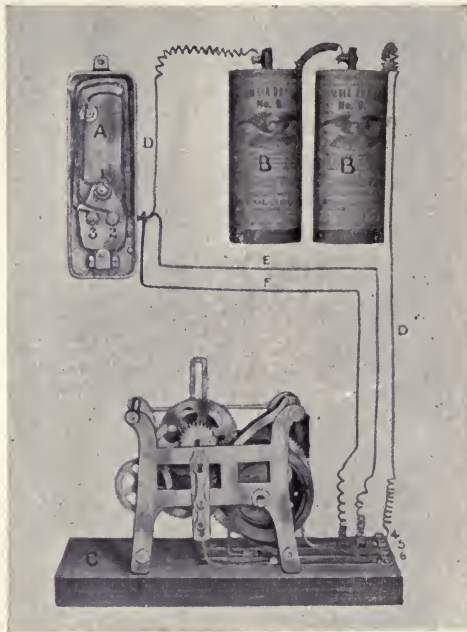


FIG. 220.—Minneapolis Thermostat and Motor Works.

are in rooms. Fig. 222 illustrates the application of the siphon bellows *A* to a thermostatic control of temperature in a tank. The bellows *A* is connected by a tube *B* to the closed tube *C*, the whole system containing a volatile hydrocarbon. When the temperature is low in the tank the weight *D* opens the valve and permits the steam to enter the heating coils. Then as the temperature rises the vapor pressure from the liquid in *C* increases and pushes down on the valve, shutting off the

steam supply. By moving the weight various temperatures may be obtained in C.

Drying. The use of air for drying such things as bricks, lumber, woven goods or any other article, is one which warrants considerable attention.

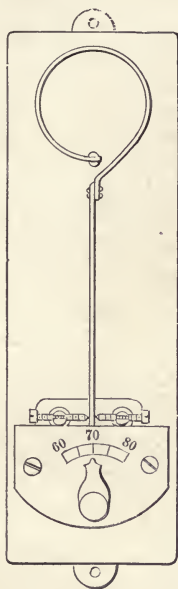


FIG. 221.
Minneapolis Thermostat.

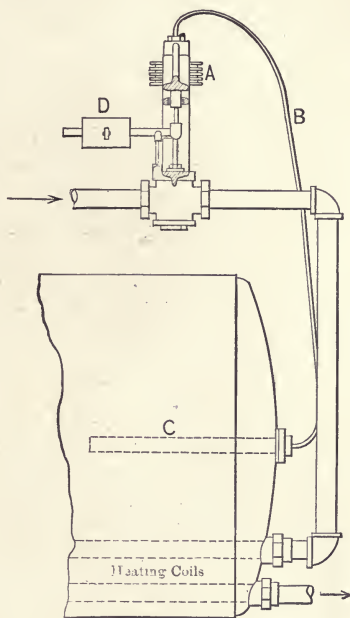


FIG. 222.
Syphon Thermostat Regulator.

If air is heated its capacity for moisture increases because the weight of a cubic foot of moisture is greater at a higher temperature. Each cubic foot can be occupied by air and moisture, each constituent sustaining part of the atmospheric pressure. If, for instance, air saturated with moisture at 60° F. is heated to 120° F., this air can hold more moisture because moisture at 120° F. occupies less space than that at 60° F., as it is under a greater pressure of saturation. Moisture at 60° is under 0.20 lb. pressure and weighs 5.8 grains per cubic foot,

while at 120° F. the pressure is 1.69 lbs. and the weight is 34.5 grains per cubic foot. This means then that in time the air could absorb more than five times the original amount of moisture and carry it away from the article from which it absorbed the moisture.

It must be remembered that this evaporation of water requires heat, and although the hot air can take it up this act will cool the air unless there is a certain amount of heat supplied from some source. This cooling action reduces the temperature of the air and with it its capacity for moisture. It is to be remembered that moisture will always enter any space until there is enough present in that space to saturate it at the given temperature. It is absolutely independent of the presence of the other substances in the space. Moisture when present will exert the pressure corresponding to its temperature. The cooling action is the feature of air conditioning used in summer weather. To illustrate these principles—suppose that bricks at a temperature of 80° weigh when wet 600 lbs. and that water is 20 per cent of this weight. The problem is to find how much air is required to dry this if the air is heated to 190° by exhaust steam in a coil when taken from the atmosphere in which the dry- and the wet-bulb readings are 70° and 65° respectively.

From Fig. 19, for air at 70° with a 5° drop for the wet bulb the relative humidity is 77 per cent and the moisture content is 6.4 grains per cubic foot.

If now the air and its moisture are cooled to t ° F. when the necessary heat is given up to warm the clay and its water and to evaporate the water, the heat thus removed per cubic foot entering is

$$Mc_p(190-t) + mc_{ps}(190-t), \quad (165)$$

where M = mass of 1 cubic foot of entering air;

c_p = specific heat of air;

m = mass of moisture in 1 cu.ft. of entering air;

c_{ps} = specific heat of steam;

t = temperature of mixture.

Now

$$M = \frac{144(p_b - p_m)}{650R} \dots \dots \dots (166)$$

p_b = barometric pressure;

p_m = pressure from moisture or vapor tension
 = steam pressure at $70^\circ \times$ rel. humidity (approximately).

$$m = 6.4 \text{ grains, } 0.0009 \text{ lb. at } 70^\circ, \text{ or } 0.0009 \times \frac{530}{650} \\ = 0.0007 \text{ at } 190^\circ.$$

The heat removed from the brick per cubic foot of air is

$$M'c(t - 60) + nM'(t - 60) + nM'r. \dots \dots (167)$$

M' = weight of bricks per cubic foot of hot air;

c = specific heat of material;

n = per cent of weight which is water;

r = heat of vaporization.

The moisture nM' should be sufficient to saturate the air at the temperature t . This air will not occupy the original volume, as the temperature is increased and the pressure is decreased. Both of these actions cause the volume of the air to increase. If the quantity of air is too great the discharged air will not be saturated, while if the quantity is driven through at a low rate it will become saturated and this will continue until all the moisture is removed. The following equations must therefore hold when the leaving air is saturated.

$$M c_p(190 - t) + m c_{ps}(190 - t) = M'c(t - 60) \\ + nM'(t - 60) + nM'r. \dots (168)$$

$$nM' = \frac{(p_b - p_m)(460 + t)}{(p_b - p_{mt})(460 + 70)} m_t - m. \dots (169)$$

p_{mt} = vapor tension for saturation at temperature t ;

m_t = weight of moisture per cubic foot.

In these two equations M' and t are unknown, and by trial the resulting values can be found. The best method is to assume

t and solve for M' in each equation. When t is so assumed that the two values of M' are the same the value will be the correct ones.

$$M = \frac{144(14.7 - 0.77 \times 0.363)}{650 \times 53.37} = 0.06 \text{ lb.};$$

$$c_p = 0.239;$$

$$c_{ps} = 0.5;$$

$$c = 0.2;$$

assume

$$t = 150;$$

$$0.06 \times 0.239 \times 40 + 0.0007 \times 0.5 \times 40$$

$$= M' \times 0.2 \times 90 + \frac{1}{3}M' \times 90 + \frac{1}{3}M' \times 1007.2;$$

$$M' = \frac{.586}{237} = .00247 \text{ lb.};$$

$$\frac{1}{3}M' = \frac{(14.7 - 0.77 \times 0.363)(610)}{(14.7 - 3.715)(530)} = 0.01032 - 0.0007;$$

$$M' = 0.07.$$

If now the temperature be made smaller, M' from the first equations will be larger and that from the second will be smaller. Suppose 100° is used; this gives as the two values of M' :

$$M' = 0.0058;$$

$$M' = 0.0123.$$

A smaller value will give the correct amount. Suppose 85° is tried; the values are:

$$M' = 0.00705;$$

$$M' = 0.0061.$$

For 90° F.:

$$M' = 0.00668;$$

$$M' = 0.00790.$$

By sketching the curves assumed to be straight lines between the last two values as shown in Fig. 223, the values of M and t at the points of intersection are found to be $t = 87.0^\circ$ and $M' = 0.0069$ respectively. This shows that the air at 190° F.

is cooled to 87° F. by the drying action and that each cubic foot of air will dry 0.0069 lb. of brick material and will leave at 87° F. The total quantity for air required will then be

$$\text{cubic feet air} = \frac{600}{0.0069} = 87,000 \text{ cu.ft.}$$

Of course if this remains here after the moisture is driven out the brick will warm up and probably leave at 190° F. This will change the problem somewhat, in that the amount of air will

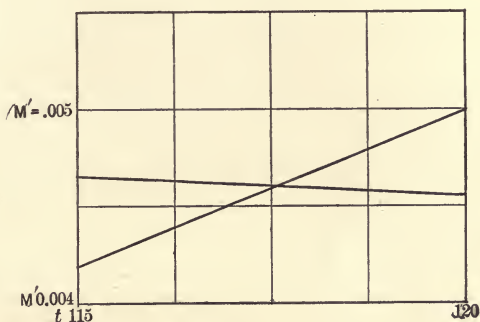


FIG. 223.—Method of Finding Values of t and M' .

have to be increased. The equation first used might have been put in the form:

$$Mc_p(190-t) + mc_{pb}(190-t) = M'c(190-60) + nM'(t-60) + nM'r \quad (170)$$

Problems similar to this may be handled for drying any kind of substance. A similar method may be used to investigate the exhaust gases from a boiler or from kilns to find whether or not this gas will be sufficient to dry the material from the clay presses. The heat and moisture content of the gases in the hot and cold condition are considered in connection with the heat to be added and the moisture to be removed from the wet material.

INDEX

A

- Absolute humidity, 20
- Air, 20
 - allowance, 23
 - , changes per hour, 23
 - cooling, 43
 - duct, 257
 - , flow of, 187
 - , free, 20
 - line, 145
 - per hour per person, 23
 - , properties of, 38
 - space, 58
 - , tempered, 12, 13
 - valve, 101, 145
 - , vitiated, 21
- Allen, 111
 - automatic air valve, 101
- American District System, 285
 - Radiator Co., 74, 111
- Anchors, 136
- Anemometer, 204
- Angle valve, 92
- Arch, 51
- Area flues, 176, 207, 208, 231
- Area pipes, 86, 138, 141, 163, 207, 208
- Argon, 20
- Asbestos, 146
- Auto valves, 96, 98

B

- Barrus, 146
- Base ring, 256
- Bends, 281
- Billings, 26
- Blast area, 220
- Boiler, 2
 - , locomotive, 268
 - , sectional, 262
 - sizes, 271-2
 - size determinations, 274-5

- Boltzman, 106
- Bonnet, 242, 243, 259
- Book tile, 62
- Boots, 1, 244
- Bottomley, 105
- Bottom support, 75
- Boxes, 178
 - for registers, 243-6
- Box radiators, 76
- Branch, 3, 155
- Bricks, 58
 - , Haverstraw, 58
 - , hollow, 58
- British thermal unit, 45
- Buffalo Forge Co., 117
 - coils, 86, 115-117
 - , size, 185
 - , loss in, 215
- By-pass plate, 256

C

- C, values of, 49
- Calculations, boiler size, 289
 - coefficient K , 53
 - , direct heating, 148
 - , district system, 289
 - , fans, 223
 - , flue size, 210
 - , furnace heating, 249, 274
 - , furnace size, 273
 - , heat, 118
 - , hot-water system, 169
 - , indirect heating, 171
 - , mill heating, 227
 - , natural draft, 178
 - , pipe size, 139
 - , plenum system, 183-228
- Calorie, 45
- Carbon dioxide, 20
 - apparatus, 24
 - bottle, 25

- Carbon dioxide, exhaled per person, 22
 , permissible amount 21
- Car heating, 298
- Carpenter, R. C., 69, 111, 139
- Carpenter's rule, 69, 253
- Carrier, W. H., 32
- Carrier's air washer, 39
- Casing, 1, 257
- Ceiling, flush, 60
 plates, 135
 values of K , 65
- Cellar plan for direct heating, 150
 for hot-air furnace, 250
 for hot water, 170
 for indirect heating, 177
- Changes of air per hour, 23
- Chases, 144
- Chezy's formula, 142
- Chimneys, 274, 276
- Circulation, sluggish, 129
- City, district heating of, 295
- Clapboards, 58
- Coefficient of transmission, 48
 , C , 49
 , d , 51
 , e , 51
 , K , 64-65
- Coils, 86, 88
 , corner, 85
 , main, 174
 , miter, 92
 , resistance, 9
 , size, 185
 , tempering, 174
- Cold-air duct, 252
 pit, 256
 supply, 238, 253
- Collars, 245
- Combustion chamber, 255
- Complete circuit system, 129, 157
- Concealed work, 145
- Concrete, value of K , 64 ↓
- Conduction, 46
 constant, 47
- Conduits, 282-4
 , loss from, 288
- Connections, 2, 132
 , branches, 155, 137
 , radiators, 132, 145
- Constant of conduction, 47
 of transformation, 45
- Convection, 46
- Cooling air, 43
- Cork, 146
- Corner coils, 85
 valves, 96
- Corrugated pipes, 281
- Cost of installing, 293
 of operation, 293
- Covering, 287
 , block, 148
 , pipe, 146
 , plaster, 148
 , saving from, 148
 , sectional, 148
 , value of, 147
- Cox, Abram, Stove Co., 261
- Cubic feet per sq.ft. radiation, 131, 159
- Curves, cooling air, 43
 , equivalent pipes, 200
 , friction factors, 196
 , heating value of coils, 112-116
 , method of solving equations,
 315
 , moisture in air, 29-32, 37
 , pressure in fans, 219
 , relative humidity, 29-32, 37
 , temperature from coils, 12-116
 , values of heat loss from con-
 duits, 289
- Cut-off plate, 256
- D
- Dalby, 106
- Damper, 13, 178, 234
 , when omitted, 245
- Design, district mains, 288
 , direct heating, 148
 , hot-water heating, 169
 , indirect heating, 171, 227, 183-
 228
- Determination of K , 110
 , fan size, 223
 , pipe size, 139
- Development of building, 134-151
- Dew point, 26
 apparatus, 26
- Direct heating, 127
- Direct-indirect heating, 8
 radiator, 76, 7
- Direct radiation, 1, 4, 127
- Distributing main, 2, 129
- Distributors, O. S., 158
- District heating, 279
 , value of, 296
- Dome, 1
- Double ducts, 13
- Down takes, 75

Drip pots, 143
 Drop, pressure, 193
 Drying, 309
 Ducts, 12, 178
 , single system, 14, 173, 214
 , double system, 13, 174, 210, 213
 Dulong, 105
 Dynamic pressure, 191

E

Eccentric fittings, 154
 Effect of exposure, 68
 of intermittent heat, 68
 Elbows, 247
 , resistance, 140
 Electric heating, 8
 meter, 206
 radiators, 9
 Eliminator, 40
 , glass surface, 253
 Equivalent, lengths, 197
 , pipes, 197
 , temperatures, 71
 , volume, 253
 Evans, 290
 Evaporation requires heat, 310
 Excelsior indirect radiator, 90
 Expansion apparatus, 281
 joints, 138, 281
 of pipes, 138
 tank, 165
 Exposure factor, 68

F

Factory heating, 233
 Fan blowers, 6, 8
 Fans, 6, 8, 216
 , dimensions, 221
 , power for, 221
 , power capacity and speed, 223
 , pressure from, 217
 , tables for, 225, 226
 with furnace, 249
 Ferrel, 28
 Filters, 41
 Fire pot, 1, 255
 Fire proof floor, 69
 Flat arch, 61
 Floors, 59
 Floor box, 243
 plate, 135
 , slow burning, 60
 , reinforced concrete, 60

Floor tile arch, 61
 , values of K , 65
 and ceiling vent, 10
 Flow of air, 187
 of steam, 141
 of water, 290
 , main, 2, 153-4
 Flue area, 175, 207, 208, 210, 231
 radiator, 76
 smoke, 274-6
 velocities, 210

Flues, 240
 Forced circulation, 171
 Free air, 20
 French windows, 123
 Friction coefficient, 195
 factor, 195-6
 loss, 192
 loss, table, 208
 Fuller, Warren & Co.,
 Furnace, hot-air, 1, 255
 heating, 1, 237
 position, 250
 , size, determination, 273
 , size of, 253, 269, 270
 , with fan, 249
 Furred, 58
 Furring strips, 58

G

Gauge metal, 179
 Generator, Honeywell, 166
 Gib panel, 80, 252
 Gifford, 294
 Glass area, equivalent, 253
 Glower, luminous radiator, 9
 Government tile, 62
 Graff Co., 255-7
 Grashof, 49, 51

H

Hair felt, 146
 Hammering, 128
 Hangers, 136
 Haven, 290
 Head, loss of, 190
 , total, 197
 Heater, 2
 , loss of pressure in, 214
 , main, 174
 , positivflo, 88
 , sectional, 262
 , tempering, 174
 , Vento, 89

Heat for ventilation air, 69
 from lights, 70
 from motors, 70
 from persons, 70

Heating, direct steam, 127
 , district, 279
 , furnace, 237
 , hot water, 153
 , indirect, 171
 , surface per cubic foot, 131, 159
 , surface required for natural
 draft stacks, 176
 systems, merits of, 276

Heat loss, 45
 with covered pipes, 147
 through walls, 47, 69
 stacks, 1
 transmission through radiators,
 105

Helium, 20

Hoffman, 242

Honeywell Co., 163
 generator, 166
 temperature regulator,
 306
 valves, 164

Horse-power for blowing, 208

Hot-air furnace, 1, 255-262
 heating, 237

Hot-water heating, 153
 radiators, 75

House plans, 122

Housing, 217, 218

Humidifiers, 39

Humidity, 26
 , absolute, 20
 , relative, 20

Humphrey gas heater, 267

Hydraulic radius, 142

Hydrodeik, 30

Hygrometer, 26

I

Ideal Junior heater, 265

Indirect heaters, transmission, 111
 heating, 171
 , calculations, 171, 183,
 228, 178
 radiators, 90-92
 systems, 5

Inside circulation, 239

Intermittent heat factor, 68

Isometric drawing, 132

J

Jenkins Bros. valves, 94
 Johnson's system, 300

K

Kelsey Co., 259
 Kinealy, 67
 Krypton, 20

L

Lambrecht polymeter, 31
 Lath, 58
 Law, Stefan Boltzman, 106
 , Stewart Kirchhoff, 106
 Leader, 1, 240, 243
 Leakage, 11
 air, 46
 Leg, detachable, 81
 loops, 81
 section, 75
 , vertical, 130
 Libra automatic air valve, 102
 List price, 86
 Lock and shield radiator valve, 96
 Loops, 74
 Loss from conduits, 288
 ducts, 232
 water pipes, 290
 of head, 190, 192
 of heat, 45
 of pressure, 214
 Luminous radiator, 9
 Lunge, 26

M

Magnesia, 146
 Main, 127, 128
 Manholes, 284
 Masonry, values of *K*, 64
 Massachusetts fan coils, 86
 fans, 216
 Measurements, radiators, 83
 , Vento heaters, 90
 Metal gauge, 179
 Metargon, 20
 Meter, electric, 206
 , oscillating, 297
 , Venturi, 204
 Mill heating, 233
 or slow-burning construction, 60
 Mills system, 129
 Mineral wool, 146
 Minneapolis temperature regulator, 307

Miter coil, 92
 Modulation valve, 96, 98
 Moisture, 26
 Monash valve, 100
 Monthly percentage of heat, 294
 Motor discharge valve, 99

N

Natural draft circulation, 171
 Neon, 20
 Newton, 105
 Nicholson, 106
 Nipples, hexagon, 89
 , push, 264
 Nitrogen, 20
 Norwall automatic air valve, 103
 packless valve, 92
 Numbers, designating, 119

O

Offset fittings, 154
 Offsets, 247
 Orifice, standard, 202
 O. S. connector or distributors, 158
 Outlets, 163, 168
 , size, 132
 Overhead system, 129
 Oxygen, 20
 Ozonator, 14
 Ozone, 14-19

P

Packless valves, 92
 Partial pressure, 28
 Partitions, 63
 , values of K , 65
 Paschen, 105
 Paul system, 145
 Perfection pin radiator, 92
 Peripheral speed, 217-220
 Personal amount of air, 23
 exhalation of CO_2 , 22
 Petavel, 105
 Petit, 105
 Pettenkofer, 21
 Pettersson's CO_2 apparatus, 24
 Phenolphthalein, 26
 Pierce Co., 264
 Piezometers, 190
 Pipe coils, 84
 coil radiators, 82
 , concealed, 144
 covering, 146

Pipe, double system, 128
 , exposed, 144
 , full weight, 297
 , list price, 86
 , single system, 127
 , size of hot-water, 159
 , sizes, 86
 , wrought iron, 86
 Pitch of hot-air pipes, 244
 of steam pipes, 143
 Pitot pressure, 191
 tubes, 191
 Plate warmers, 81
 Plenum, 8, 171
 Pockets, 128, 144
 Polymeter, 31
 Positivflo heater, 88
 Power for fans, 221
 Powers system, 302
 Pressed steel radiator, 82
 Pressure in fans, 217
 , dynamic, 191, 217
 , loss of, 191-3
 , measures of, 188
 , partial, 28
 , Pitot, 191
 , saturation, 29
 , static, 191, 217
 , velocity, 191, 217
 Properties of dry and saturated air, 38
 Psychrometer, 27

R

Radiation, 46
 from conduits, 288
 Radiator, 1, 73, 255
 , circular, 80
 connections, 132, 145
 , corner, 80
 designation, 135
 dimensions, 83
 , dining room, 81
 , electric, 9
 foot ups, 82
 , furnace, 255
 , heat transmission, 105, 159
 heights, 80, 83
 , hot-water, 153
 , low, 80
 outlets, 132, 163, 168
 , pantry, 81
 pedestal, 82
 , pipe coils, 82
 , pressed steel, 82

Radiator selection, 80, 132
 , size required, 131
 table, 149
 tapping, 132, 163-8
 , test of, 110
 , values of K , 110
 , wall, 81
 , window, 80

Radifier, 100

Rate of transmission, 47

Recirculating duct, 237

Reducers, 144

Register, 1

Register box, 243, 246
 faces, 179, 209
 location, 329
 , special, 247

Reheater, 12

Reinforced cinder concrete floor, 60

Relative humidity, 20, 26

Resistance, valves and elbows, 140

Return, 3
 for different buildings, 130
 , size of, 142

Reitschel, H., 47, 49

Right and left nipples, 73

Risers, 1, 2, 127, 128
 , arrangement of, 134
 control, 135
 designation, 135
 table, 149

Rococo wall radiator, 81

Rosetti, 105

Rudd gas heater, 268

S

Sand ring, 259

Saturation, 20
 pressure, 29

School building, 180
 heating, 248

Screens, 252

Sealed return, 130

Sections of boiler, 264

Sheathing boards, 58

Shingles, 58

Shoes, 1, 244

Sickels, 139

Single-duct system, 14
 -flow system, 153
 -pipe system, 127

Size of flues, 179
 furnace, 253
 mains, 138, 141, 163

Size of pipes, hot-water, 159
 supply and return, 187
 Vento heater, 186

Skew backs, 61

Sleepers, 60

Sleeves, 136

Sling psychrometer, 27

Slow-burning construction, 60

Smoke flue, 274-276
 outlet, 256

Sodium carbonate, 26

Spangler, 119

Spence boiler, 265

Stacks, 1, 5, 8, 240

Stateroom type electric radiator, 9

Static pressure, 191

Steam flow, 141
 pipes, pitch of, 143

Stefan, 105

Stefan-Boltzman Law, 106

Sterling Radiator, 92

Stove cement, 259

Studs, 58

Sturtevant, B. F. & Co., 86
 coils, 86

Supports, 282-4

Swinging ells, 137, 281

Sylphon, 94
 air and vacuum valves, 104
 air valve, 103
 packless radiator valve, 94
 vent valves, 104

T

Tables, air per person,
 , properties, 38
 , velocity, 209
 , boilers, 271-2
 , Buffalo Forge Co. heater, 185
 , capacity pipes, 138
 , coefficients, 64, 65
 , conversion factors, 45-6
 , covering loss, 147
 , cubic feet per square foot heat-
 ing surface, 131, 159
 , equivalent pipes, 208
 temperature, 72
 , expansion of pipes, 138
 tanks, 165
 , exposure allowance, 68
 , fans, 225-6
 , friction factor, 196-7
 , loss, 208

- Tables, furnace heating, 238, 251
 , size, 269-70
 , gauge thickness, 179
 , heat changes, 294-6
 , heaters, 186, 187
 heat given out, 70
 transmission coefficient, 64,
 65
 , house heating, 120-1
 , indirect heating, 178
 , loss pressure in heaters, 215
 , pipe sizes, air, 207
 , steam, 86, 138, 141
 , water, 163
 , pressure, 207
 , properties of air, 38
 , radiator list, 149, 170
 , radiators, 81, 83
 , riser list, 149, 170
 , room temperature, 66-7
 , school building, 181
 , size, iron, 179
 , pipes, 86, 138, 141, 163, 207
 , risers, 135
 , tin, 247
 , tappings, 132, 163, 168
 , temperature effect, 209
 rooms, 66-7
 , tin sizes, 247
 , Vento heater, 186
 , velocity, 20, 209
 , water weight, 143, 161
- Tank, expansion, 165
- Tappings, radiator, 163-8
- Temperature allowance for height, 68
 atmosphere, 66
 control, 300
 , effect of, 198
 , entrance, 172
 , equivalent, 71
 , inlet, 237
 , maximum, on hot water,
 166
 outlet from heater, 112
 rooms, 66
- Tempered air, 12, 13
- Test of radiator, 110
- Theatre heating, 235
- Thermal units, 45
- Thermomgrade system, 96
- Thermostat, 300, 303, 308, 312
- Thermostatic motor, 13
- Thickness of flues, 179
- Thomas, 206
- Tiles, hollow, 58
- Tin, size of, 247
- Tongued and grooved boards, 58
- Top guide, 75
- Total head, 197
- Transformation factors, 45
- Transmission, rate of, 47
 coefficient, 48
 of heat through radiators,
 105
- Traps, use of for buildings on same re-
 turn, 130
- Tunnel, 282
- Two ducts, 13
- Two-pipe system, 128
- V
- Vacuum system of heating, 129
 of ventilation, 8, 171
 valves, 103-4
- Valves, 92
 , air, 101, 102
 , corner, 96
 , lock and shield, 96
 , Monash, 108
 , motor discharge, 99
 , positions of, 144
 , Q. O. water, 95
 , resistances, 140
 , sylphon packless, 94
- Vapor tension, 36
- Variators, 281
- Velocities, 209, 210
- Velocity, effect on transmission, 107
 for natural draft, 175
 in ducts, 183
 in hot water-pipes, 159
 pressure, 191
- Vent flue, 173
 stack, 1, 242
- Vent valves, 104
- Ventilation for schools, 181
 , heat for, 69
 , method of, 11
- Vents, 92
- Vento heater, 89, 90
 , measurements, 90
 , size, 186
- Venturi meter, 204
- Vitiated air, 21
- Volume equivalent, 253
- Voussoirs, 61

W

Wall, 58

section, 52, 57

, type electric heater, 9

 , values of K , 64

, wooden, 59

Warren, Webster & Co., 40, 43

washer, 40

valves, 98-9

Washers, 39

Water line, 127

pan, 261

vapor, 20

Water, weight of, 143, 161

Weber, 105

Wet- and dry-bulb hygrometer, 27

Williams, 290

Window constants, 64-5

, French, 123

Wood floor, 54

walls, 59

 , value of K , 65

X

Xenon, 20



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