STEAM-BOILER CONSTRUCTION.

A PRACTICAL HANDBOOK

FOR ENGINEERS, BOILER-MAKERS, AND

STEAM-USERS.

RANGE OF LANCASHIRE BOILERS BY HICK, HARGREAVES & CO., BOLTON.

[For Description see page 253.

STEAM-BOILER CONSTRUCTION.

A PRACTICAL HANDBOOK

FOR

ENGINEERS, BOILER-MAKERS, & STEAM-USERS,

CONTAINING

A LARGE COLLECTION OF RULES AND DATA RELATING TO RECENT PRACTICE IN THE DESIGN, CONSTRUCTION, AND WORKING OF ALL KINDS

OF

STATIONARY, LOCOMOTIVE, & MARINE STEAM-BOILERS.

BY

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PREFACE.

THIS work is issued in continuation of the series of HANDBOOKS written by the Author, viz., "THE WORKS' MANAGER'S HANDBOOK," and "THE PRACTICAL ENGINEER'S HANDBOOK," which are much appreciated by engineers for the practical nature of their information. It is, therefore, written in the same style as the works which it is intended to supplement.

In the present volume, the Author has endeavoured to present in a concise, condensed, and readily accessible form, the latest and most useful information on steam-boilers, comprising the results of numerous investigations and researches in this particular branch of engineering.

Much useful information is given on heat; combustion; the composition, calorific power, and evaporative power, of fuels; chimneys for steam-boilers; evaporation; the properties of steam; and the transmission of heat.

Numerous rules and data are given for riveted-joints; furnace-tubes; heating-surfaces; fire-grates; and for the water-capacity, steam-capacity, efficiency, and power of steam-boilers.

Many different kinds of steam-boilers are described and illustrated; and rules and data are given by which nearly every kind of steam-boiler may be readily proportioned.

Steam-pipes; the flow of steam; fittings of steam-boilers; incrustation; corrosion; feed-water-heating;

feed-water-heaters; evaporators; inspecting and testing steam-boilers; are fully treated.

The usual method of conducting the evaporative test of a steam-boiler is described; and the results are given of a number of evaporative tests of steam-boilers of different types.

The principal causes of the explosions of steam-boilers are explained; and particulars are given of a number of boiler-explosions.

The Author believes that the concentration, in a convenient form for easy reference, of such a large amount of thoroughly practical information on steam-boilers, will be of considerable service to those for whom it is intended, and he trusts the book may be deemed worthy of as favourable a reception as has been accorded to its predecessors.

W. S. HUTTON.

103, SUNDERLAND ROAD,
FOREST HILL, LONDON, S.E.,
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SECTION I.

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heat, must be reckoned from a point at which heat does not exist. This point is 461° below zero of Fahrenheit scale. It is the absolute zero of temperature, and temperatures reckoned from it are called absolute temperatures. When heat is applied to a substance it is absorbed and stored partly as sensible heat, which can be measured by a thermometer, and partly as latent heat, of which the thermometer gives no indication of its existence.

Quantity of Heat is the measure of the energy of molecular motion. Quantities of heat are measured by a standard unit. The British thermal unit is the quantity of heat required to raise one pound of water one degree Fahrenheit, that is, from 32° to 33° Fahr.

Dr. Joule found that the quantity of heat necessary to increase the temperature of water by one degree, requires for its production the expenditure of a force measured by the fall of 772 pounds from a height of one foot. Heat-energy and mechanical energy are mutually convertible, therefore the heat applied and the work done are measured by the same unit. A British unit of heat is equal to 772 pounds raised one foot high, and 772 units of work are equivalent to one unit of heat.

Heat, as a quantity, is independent of temperature, because the same quantity of heat produces different temperatures in different substances. These variations are denoted by the term specific heat.

The Specific Heat of a body is its capacity for heat, or the quantity of heat-units required to raise the temperature of one pound of the body one degree Fahr., compared with that required to raise an equal weight of water one degree. For instance, the quantity of heat necessary to raise the temperature of a pound of charcoal one degree, is only about one-fourth of that required to raise one pound of water one degree.

As water has a higher specific heat than any known solid, and than nearly all liquids, it is taken as the standard of comparison of specific heat. The specific heat of all metals is very low.

The specific heats of solid and liquid bodies are nearly constant for temperatures up to 212° Fahr., but above this point the specific heat increases as the temperature rises. The specific heat of a solid increases very rapidly as its temperature approaches the melting point.

It is sufficiently accurate for most practical purposes to consider specific heat to be constant at constant pressure, and the quantity of heat required to heat various substances may be found from Table 1.

As an example of the use of Table 1, to heat a plate of mild steel weighing 100 pounds to a temperature of 367° Fahr., requires 100lbs. × 367° × ·1158, specific heat = 4,250 units of heat. To heat the same weight of water requires 100lbs. × 367° = 36,700, or nearly nine times as much heat as that required for the steel.

Heat is converted into mechanical work through the agency of some body which is expanded by heat, such as air or water. Heat cannot enter and leave a body without a certain loss, because more heat is required to give elasticity to matter than is converted into work by elasticity. The conversion of heat into work is always accompanied by a fall of temperature, and the proportion of work obtainable from heat depends upon the ratio which the fall of temperature in the working substance bears to the original absolute temperature. Hence, the greater the working-range of temperature the greater the efficiency.

TABLE 1.—SPECIFIC HEAT OF SOLIDS AND GASES, BEING THE QUANTITY OF HEAT IN BRITISH THERMAL UNITS REQUIRED TO RAISE THE TEMPERATURE OF THE BODY ONE DEGREE FAHRENHEIT.

METALS.		Coal, when heated to a dull cherry-red, or a temperature of 1470° Fahr.	·4187
Water at 32° Fahr. =	1·0000	**GASES.**	
Cast-iron, white	·1298	Water at 32° Fahr. =	1·0000
Cast-iron, grey	·1216	Sulphurous acid	·1553
Cast-steel, hard	·1185	Carbonic acid	·2164
Cast-steel, soft	·1166	Oxygen	·2182
Mild-steel	·1158	Atmospheric air	·2380
Wrought-iron	·1146	Sulphuretted hydrogen	·2424
Copper	·0965	Nitrogen	·2440
Gun-metal	·0952	Carbonic oxide	·2460
Brass	·0940	Products of combustion of coal at a temperature of 570° Fahr.	·2650
FUELS.		Olefiant gas	·3694
Pine-wood	·6510	Vapour of benzine	·3754
Oak, English	·5709	Steam	·4810
Carbon	·2500	Ammoniacal vapour	·5080
Charcoal	·2415	Carburetted hydrogen, or marsh-gas	·5929
Coal, averages	·2412	Hydrogen	3·4046
Anthracite	·2017		
Coke	·2000		
Ashes of coal	·1920		
Clinker of coal	·1674		

The maximum quantity of heat theoretically capable of being transmuted to motion is expressed by the following formula:—

$$\text{Available heat} = \frac{T - t}{T}$$

In which T represents the highest absolute temperature, and t the lowest absolute temperature.

The mechanical equivalent of the available heat is expressed by the following formula:—

$$\text{Mechanical energy of heat} = 772\,\frac{T - t}{T}$$

The greater the fall of temperature during the working of a heat-engine the greater the proportion of the work obtained.

The proportion of the work in the case of a steam-engine depends upon the initial and final temperatures of the steam in the cylinder.

The function of a steam-engine is to convert the heat of the steam into work, and the function of a steam-boiler is to convert the heat-energy contained in the fuel into the heat-energy resident in the steam.

A steam-boiler is, therefore, a means of adapting the calorific effect of fuel into useful work, and it may be considered to be a heat-engine. If the specific heat of the products of combustion be assumed to be constant, the efficiency of a steam-boiler may be calculated approximately by the foregoing rule. In which T is to be taken as representing the absolute temperature of the furnace, and t that of the products of combustion on escaping from the boiler.

For instance, if the absolute temperature of the furnace of a steam-boiler be = 3120° Fahr., and that of the products of combustion on leaving the boiler = 1020° Fahr., then the proportion of heat available for work is approximately =

$$\frac{3120^\circ - 1020^\circ}{3120^\circ} = \cdot 67.$$

That is, the heat utilized is 67 per cent. of the heat supplied, the maximum efficiency theoretically possible with this range of temperature.

The mechanical equivalent of the heat supplied is 772 units of work, and the mechanical equivalent of the available heat is =

$$772 \times \frac{3120^\circ - 1020^\circ}{3120^\circ} = 518 \text{ units of work, or foot-pounds.}$$

The efficiency of a boiler obtained by the above rule is only approximately correct. The efficiency of a steam-boiler is represented more correctly by the following expression :—

$$\text{Efficiency of boiler} = \frac{\text{Units of heat absorbed by the water in the boiler.}}{\text{Units of heat developed by combustion of the fuel.}}$$

The previous rule is, however, sufficiently accurate for most practical purposes.

Heat is transferred from one body to another in three ways:—By right lines, called radiation; by direct contact, called conduction; by carrying, called convection. The quantity of heat transferred by these processes depends upon the difference of temperature between the hot and the cold body, and the extent and nature of the bodies.

Radiation of Heat is the direct transfer of heat across an intervening space, from one body to another. The rays of heat travel in straight lines through a uniform medium, and traverse the space between the bodies with very great rapidity. As radiant heat only travels in straight lines like light, it cannot turn round a corner, or pass round an obstacle.

Radiant heat traverses air without raising its temperature, and will pass through gases without warming them, in the same manner as the rays of the sun pass through atmospheric air without heating it. The heating effect of radiant heat is inversely as the square of the distance from the hot body. For instance, if at any given distance a certain effect is produced, at twice that distance the effect will only be one-fourth.

Lampblack and finely powdered charcoal are good radiators and good absorbers and receivers of heat. Hence, surfaces intended to receive, give up, or radiate heat, should be coated with dull black.

The black-coloured, or carbonaceous, portion of the gases produced by combustion, radiate heat much more efficiently than the light, or yellow, coloured portion. Dry air neither absorbs nor radiates heat. Dry transparent gases radiate no heat. Water and saturated steam are good radiators and good absorbers of heat.

The radiating powers of substances vary considerably, as will be seen from Table 2.

TABLE 2.—RELATIVE RADIATING AND ABSORBING POWERS OF VARIOUS MATERIALS.

Lampblack	100	Tarnished zinc	30
Soot	100	Mercury	21
Water	100	Bright galvanized sheet-iron	20
Powdered charcoal	95	Polished cast-iron	19
Dull black paint	90	Polished lead	18
Ordinary sheet glass	89	Polished zinc	17
Dull red paint	80	Platinum	16
Plumbago	75	Polished wrought-iron	15
Tarred surface	70	Polished steel	14
Dark lead-colour paint	65	Bright tin-plate	12
Dark brown paint	60	Polished copper	11
Dark green paint	55	Polished brass, fine	10
Dark chocolate-colour paint	50	Polished nickel	9
Tarnished lead	40	Polished gold	8
Tarnished pewter	35	Polished silver	6

Absorption is the power of receiving or taking in heat. Coated surfaces absorb heat more readily than uncoated surfaces. Good radiators are good absorbers, and bad radiators are bad absorbers of heat. Hence, the same numbers which express the relative radiant powers of any series of substances will also express their relative absorbent powers. The absorbing power of metals depends upon the degree of their polish. The higher the surfaces are polished the less the absorbing power and the greater the reflecting power.

Brickwork, of either common bricks or fire-bricks, is an excellent absorber and radiator of heat. A brick-arch is frequently used in locomotive fire-boxes to absorb radiant heat from the fuel and give it up to assist the combustion of the gases from the fuel, and also to protect the tube-plate from destructive heat. The brickwork-casing of water-tube boilers absorbs heat, part of which is transmitted by radiation to the water-tubes, and the remainder is lost by dissipation through the brickwork to the external air.

Reflection of Heat.—When radiant heat falls upon a polished surface, it absorbs a portion of it and the remainder is reflected in the same way as rays of light. The quantity of heat absorbed is the measure of its absorbing power, and the quantity reflected is the measure of its reflecting power. Hence, the sum of the absorbing and reflecting powers of all bodies is the same. Good reflectors of heat are bad absorbers and bad radiators of heat. Therefore, vessels intended to retain heat should be covered with highly polished metal. Highly polished silver reflects nearly the whole of the radiant heat that falls upon it, the amount absorbed being extremely small. The reflecting powers of metals are given in the following Table:—

TABLE 3.—RELATIVE REFLECTING POWERS OF METALS.

Highly polished fine silver	100	Polished zinc	83
Polished silver	94	Polished lead	82
Polished gold	92	Polished cast-iron	81
Polished nickel	91	Bright galvanized sheet-iron	80
Polished brass, fine	90	Mercury	89
Polished copper	89	Tarnished zinc	70
Bright tin-plate	88	Tarnished pewter	65
Polished steel	86	Tarnished lead	60
Polished wrought-iron	85	Ordinary sheet glass	11
Platinum	84	Lampblack	0

Heat Radiated in the Furnaces of Steam-Boilers.—In the furnace of a steam-boiler, as very little of the fuel is in contact with the plates, practically no heat is transmitted by conduction, and the heat radiated from the fuel forms a large proportion of the total heat evolved. The furnace-plates receive direct rays of heat radiated from the fire on the fire-grate.

It is estimated that one-half of the total heat evolved by the combustion of coal on the fire-grate of a steam-boiler, is given out by radiation to the heating surfaces over the fire, and the remainder is absorbed by the air in passing over and through the fire. The heat thus absorbed is partly recovered when the products of combustion pass through flues surrounded by absorbing or water-heating surfaces.

The proportion of the total heat evolved by combustion radiated by different fuels is, on an average, as follows:—

Charcoal	·57	Bituminous Coal	·50
Coke	·55	Wood	·25
Anthracite	·53	Oil	·20

The heat radiated from the fire is transmitted through the furnace-plates of a boiler by conduction.

TABLE 4.—RELATIVE HEAT-CONDUCTING POWER OF MATERIALS.

Silver	100	Chalk	5·900
Gold	84	Asphalt	4·600
Copper	80	Oak	3·310
Gun-Metal	73	Fir	3·201
Brass	70	Elm	3·200
Mercury	67	Ash	3·102
Aluminium	65	Sawdust	3·008
Zinc	61	Chestnut	3·000
Cadmium	56	Walnut	3·000
Wrought-iron	42	Mahogany	2·837
Tin	41	Apple	2·829
Steel	40	Fine Ashes of Coal	2·800
Platinum	38	Ebony	2·231
Cast-iron	35	Plaster of Paris	2·174
Lime	25	Cement	2·012
Lead	20	Plaster	1·907
Antimony	19	Sand	1·836
Coal, Anthracite	19	Straw, chopped	1·120
Coke	19	Bran	1·102
Limestone	19	Flannel	·415
Granite	18	Horsehair	·317
Building Stone, average	17	Cow's-hair	·256
Sandstone, hard	16	Powdered Charcoal	·200
Sandstone, medium	14	Lampblack	·112
Marble	12	Slag-wool wrapt in felt	·108
German Silver	11	Silk, dressed	·104
Slate	10	Hair-felt, wood-lagged	·102
Glass	10	Hare's Fur	·095
Stock-Bricks	9	Lint	·085
Asbestos	8	Wood-ashes	·084
Porcelain	8	Cotton	·084
Terra-cotta	7	Sheep's Wool	·080
Fire-clay	7	Raw Silk	·070
Fire-brick	6	Beaver's Fur	·068
Bismuth	6	Eiderdown	·067

Conduction is the process by which heat passes from a hotter part to a colder part of the same body, or from one to another of bodies in contact.

The flow of heat from a hot to a cold body in contact, depends upon the perfection of their contact, and upon the conducting power of the substances. Heat flows more quickly through some bodies than others; the rate of flow varies according to the nature of the material.

The rate of conduction of a body, or its conductivity, is its power of transmitting heat. It is measured by the number of units of heat that will flow in a unit of time across a cube of the material one inch in volume, the opposite surfaces of which differ in temperature by one degree.

For instance, if an inch be taken as the unit of length, and a second as the unit of time, and the temperature of the inner surface of a boiler-plate be one degree less than that of the outer surface, then the number of units of heat that pass, under a steady flow of heat, through each cubic inch of metal in a second, is termed the specific thermal conductivity of the metal.

Good Conductors of Heat transmit heat readily. Metals are good conductors of heat. Silver is one of the best conductors. The conducting power, or conductivity, of different substances compared with this metal is given in Table 4.

The conducting power of pure metals is nearly the same for heat and electricity.

Bad Conductors of Heat offer a greater or less resistance to the transmission of heat. Animal and vegetable substances with fine fibres, such as hair, felt, cotton, wool, fur, and eiderdown, are the slowest or worst conductors of heat, owing chiefly to their interstices being filled with stationary air, which is a still worse conductor. Wood, sawdust, chalk, sand, stone, and brick are slow or bad conductors of heat. Brickwork of either common bricks or fire-bricks is a bad conductor of heat.

Non-conducting Coverings for preventing radiation from steam-pipes, steam-cylinders, and steam-boilers, are composed of materials which conduct heat slowly. Fine hair-felt is probably the best non-conducting material for this purpose.

TABLE 5.—RELATIVE HEAT-SAVING VALUES OF NON-CONDUCTING COVERINGS, EACH 1½ INCHES THICK, FOR STEAM-PIPES AND STEAM-BOILERS.

A perfect non-conducting covering	1·00
Loose wool, wood-lagged	·97
Loose feathers, wood-lagged	·97
Hair-felt, wood-lagged	·96
Loose cotton-wool, wood-lagged	·96
Loose lamp-black, wood-lagged	·96
Fine slag-wool wrapped in felt	·95
Loose powdered-charcoal, wood-lagged	·94
Paper, hair-felt	·93
Air-space, hair-felt	·93
Hair, silicate, magnesia	·93
Chopped straw, silicated	·92
Bran, silicated, thin felt	·91

TABLE 5 *continued*.—RELATIVE HEAT-SAVING VALUES OF NON-CONDUCTING COVERINGS, EACH $1\frac{1}{2}$ IN. THICK, FOR STEAM-PIPES AND STEAM-BOILERS.

Loose calcined magnesia, wood-lagged	·91
Air-space, bran, hair	·90
Loose slag-wool, wood-lagged	·90
Brown-paper, wood-lagged	·90
Fossil-meal, hair-plaster	·89
Air-space, fine wool	·89
Straw-rope, wound spirally, covered with canvas	·88
Loose chaff, wood-lagged	·88
Loose fossil-meal, wood-lagged	·88
Air-space, fine cotton	·87
Air-space, goat's hair	·86
Cork-chips, plaster	·85
Air-space, paper-pulp, hair	·84
Clay, hair, flour, flax-fibre	·84
Larch twinings, hair, flour	·82
Clay, sawdust, paper-pulp, flour	·80
Flax-fibre, clay, paper shavings, flour	·79
Moss, hair, sawdust, flour	·79
Thin hair-felt and straw-rope	·78
Chalk, hair, flour	·78
Charcoal, sawdust, hair, flour	·76
Plaster of Paris, fine ashes, flax	·75
Peat, sawdust, hair, flour	·74
Pumice-stone, sawdust, clay, flour	·74
Ashes, hair, cement	·72
Sawdust, plaster, paper	·71
Asbestos, cement, paper	·71
Cocoa-nut fibre with paper covering	·70
Sawdust, sand, cement	·70
Brick-dust, sand, flax, cement	·70
Air-space, tin-plate case, paper	·69
Clay, flax-refuse, and cement	·69
Asbestos-paper, brown paper	·68
Slag-wool, or silicate cotton	·67
Charcoal, cement, paper	·66
Charcoal	·63
Wood-chips, small, and loam	·55
Loam, dry and loose	·53
Marble reduced to very small pieces	·52
Powdered-chalk	·50
Gas-works carbon	·45
Asbestos	·36
Fine coal-ashes	·35
Plaster of Paris	·35
Brick reduced to very small pieces	·33
Flax-fibre and clay	·31
Coke-breeze, small	·30
Air alone	·20
Sand	·15

Many tests have been made to ascertain the quantity of heat saved by non-conducting coverings on steam-pipes relatively to the bare pipes; the heat-saving value of coverings composed of different slow-conducting material is given in Table 5.

Non-conducting Coverings for the Prevention of Radiation of Heat.—Non-conducting coverings for preventing the radiation of heat, when composed of animal and vegetable substances, are liable to spontaneous combustion if they become greasy and damp.

Hair-felt is liable to injury from high temperature. When it becomes charred it rapidly crumbles to dust of a very inflammable nature.

Mineral-wool, slag-wool, or silicate cotton, is a mineral fibre formed by blowing jets of steam into fluid slag, by which it is drawn into very fine glass-like threads. It frequently contains impurities which in the presence of moisture corrode metals. One cubic foot of slag will produce about 12 cubic feet of slag-wool, containing about 90 per cent. of its volume of air in confinement. One cubic foot of slag-wool weighs, on an average, 12 lbs. One square foot of slag-wool one inch thick weighs one pound. Slag-wool is a good non-conductor of heat, and an excellent fire-proof and vermin-proof material. It is also a very effective deadener of sound.

Asbestos is a mineral substance, nearly inconsumable by fire. It is composed on an average of lime and magnesia, 37 parts; oxide of iron, 3; potash, 1; soda, 1·4; alumina, 3, and 11·4 parts of organic matter and water. It is very durable, but does not possess much power of preventing radiation. When it is used as a covering for steam-boilers it is necessary to mix it with some other slow-conducting material, otherwise it does not make an efficient covering.

Before applying a covering the plates of the boiler should be covered with several coats of plaster of Paris, which is a non-corrosive and non-conducting material easy of application. When a covering is applied in the form of plaster, wire-netting may be placed on the surface to be covered, in order to facilitate removal of the covering when required. Or laths of wood laid on strips of asbestos-board may be placed a few inches apart on the surface to be covered and the cement applied between them. Coverings are most effective when an air-space is formed between the plate and the covering as shown in Fig. 1.

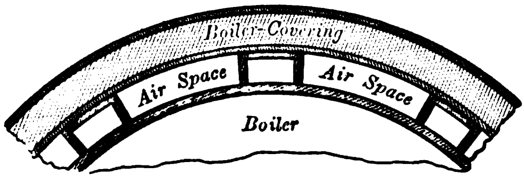


Fig. 1.—Section of a covering for a steam-boiler.

The outside of non-conducting coverings should be painted white, because this colour radiates the least heat.

Non-conducting coverings should not be less than one inch thick, but rather $1\frac{1}{2}$ inches thick.

Quantity of Heat Transmitted by Non-Conductings Coverings.—Careful experiments were made by Professor Ordway to determine the quantity of heat transmitted per hour through various non-conductors of heat, each used in a mass one inch thick, placed on a flat surface of iron heated by steam to 310° Fahr., the results of which are given in the following Table. The first column of the Table gives the loss by the measure of lbs. of water heated ten degrees. The second column gives the amount of solid matter in the mass one inch thick. The third column gives the amount or bulk of included or entrapped air.

TABLE 6.—HEAT TRANSMITTED PER HOUR THROUGH VARIOUS NON-CONDUCTING COVERINGS, RECKONED IN POUNDS OF WATER HEATED 10° FAHR., THE UNIT OF AREA BEING ONE SQUARE FOOT OF COVERING.

Description of Non-Conducting Material one inch thick.	Lbs. of water heated 10° Fahr., per Hour through one square foot.	Solid matter in one square foot, one inch thick, parts in 1000.	Air included parts in 1000.
Loose wool	8·1	56	944
Goose feathers	9·6	50	950
Carded cotton-wool	10·4	20	980
Hair-felt	10·3	185	815
Loose lamp-black	9·8	56	944
Compressed lamp-black	10·6	244	756
Cork-charcoal	11·9	53	947
White-pine charcoal	13·9	119	881
Anthracite coal-powder	35·7	506	494
Loose calcined magnesia	12·4	23	977
Compressed calcined magnesia	42·6	285	715
Light carbonate of magnesia	13·7	60	940
Compressed carbonate of magnesia	15·4	150	850
Loose fossil-meal	14·5	60	940
Crowded fossil-meal	15·7	112	888
Ground chalk or Paris-white	20·6	253	747
Dry plaster of Paris	30·9	368	632
Fine asbestos	49·0	81	919
Air alone	48·0	—	1000
Sand	62·1	527	471

Non-conducting coverings should be maintained perfectly dry, for not only is water a good carrier of heat, but it was found in these experiments that still water conducts heat about eight times as rapidly as still air.

A Composition for covering Boilers consists of sawdust, 2 parts; cement, 1 part; mixed dry and moistened with water. It is applied in layers to a thickness of from 3 to 6 inches, one layer being allowed to dry before another is applied. Another composition for the same purpose consists of sawdust mixed with starch; a mixture of two-thirds wheat-starch and one-third rye-starch is the best. A string should be wound spirally round steam-pipes to secure adhesion of the composition. Copper-pipes should be primed with a solution of potter's-clay before applying the composition. Another covering for steam-boilers and pipes consists of waste products from the manufacture of paper, mixed in a dry state with potter's-clay in the proportion of 4 to 1, and made into a plastic compound with water. It is applied in thin layers.

Hair-Felt Coverings for Steam-Boilers and Steam-Pipes should be of best quality, $1\frac{1}{2}$ inch thick, weighing about one pound per square foot, stitched on strong canvas and secured to the boiler by bands of hoop-iron. For very high temperatures, the metal should be covered with asbestos-board before applying the felt, to prevent it becoming charred.

Radiation from Steam-Boilers.—The loss of heat by radiation from unclothed steam-boilers and steam-pipes is very great. It may be estimated approximately by the following formula:—

Let T = the temperature of the steam in the boiler in degrees Fahr.
t = the temperature of the air outside the boiler in degrees Fahr.

The units of heat lost by radiation per square foot of unclothed surface of a steam-boiler per minute $=(T-t)\times \cdot 058$.

The units of heat lost by radiation per square foot of unclothed surface of a steam-boiler per hour $=(T-t)\times \cdot 058\times 60$.

Example.—Required the heat lost by radiation from the unclothed surface of a steam-boiler having 250 square feet of surface exposed to the atmosphere for 10 hours. The temperature of the steam inside the boiler is 358° Fahr., and that of the atmosphere outside the boiler is 58° Fahr.

Then $(358°-58°)\times \cdot 058\times 60=1044$ units of heat emitted by radiation per hour per square foot of exposed surface; and 1044 units $\times$ 250 square feet $\times$ 10 hours $=2,610,000$ units, the total quantity of heat lost during the working of the boiler by radiation from the surface of this shell exposed to the atmosphere.

The loss of heat by radiation from long steam-pipes is considerable. For instance, the steam at the engine-end of an uncovered steam-pipe of 7 inches diameter and 60 feet long, was carefully tested and found to contain 31 per cent. of moisture. After protecting the pipe with an efficient non-conducting covering the steam was again tested and found to contain 3·6 per cent. of moisture.

Radiation from Steam-boilers with non-conducting Coverings.—The quantity of heat lost by radiation from a steam-boiler protected by a non-conducting covering of hair-felt, applied in the best manner, may be estimated approximately by the following formula, deduced from experiments:—

Let T = the temperature of the steam in the boiler in degrees Fahr.
t = the temperature of the external atmospheric air in degrees Fahr.

The units of heat lost by radiation per square foot of external clothed surface of the boiler per minute are $= (T - t) \times {\cdot}02$.

The units of heat lost by radiation per square foot of external clothed surface of the boiler per hour are $= (T - t) \times {\cdot}02 \times 60$.

Example: Required the quantity of heat lost by a well-clothed marine return-tube boiler, presenting an external surface of 760 square feet to the atmosphere. The temperature of the steam inside the boiler is 367° Fahr., and that of the atmosphere outside the boiler is 67° Fahr.

Then $(367° - 67°) \times {\cdot}02 \times 60 = 360$ units, the heat lost by radiation per square foot of external surface per hour. The total quantity of heat lost by radiation from the external surface of the clothed boiler is $= 360$ units $\times$ 760 square feet $= 273{,}600$ units per hour. Assuming the area of the fire-grate to be 49 square feet, and the consumption of coal 20 pounds per square foot of fire-grate surface per hour. Then the quantity of coal burnt per hour is $= 49$ square feet $\times$ 20 pounds $= 980$ pounds, and the heat lost by radiation is $= 273{,}600$ units $\div$ 980 pounds $= 280$ units per pound of coal. If the heat actually expended in the production of steam is, say, 11,000 units per pound of coal consumed, then the heat lost by radiation is $= 280 \div 11{,}000 = {\cdot}025$, or $2\frac{1}{2}$ per cent. of the total heat utilised.

The heat lost by radiation from steam-boilers protected by coverings of ordinary non-conducting compositions averages 6 per cent. in internally-fired boilers, and 12 per cent. in externally-fired boilers.

Gaseous Fluids conduct heat very slowly. When gases are restrained in motion, their thermal conductivity is extremely small. Dry air is incapable of absorbing or radiating heat. An air-space $1\frac{1}{2}$ inches deep formed round a steam-cylinder or steam-pipe by means of sheet-iron covered with hair-felt, is an excellent retainer of heat and preventer of radiation.

Liquids, with the exception of mercury, conduct heat slowly. The conducting powers of liquids compared with water are as follows:—

Table 7.—Relative Heat-conducting Powers of Liquids.

Water	1·00	Sulphuric acid	1·74
Proof spirit	·85	Sulphuric ether	2·15
Alcohol, pure	·93	Mercury	2·82
Nitric acid	1·51	Turpentine	3·14

The Heat-conducting value of Metals depends upon their purity and homogeneousness of texture. The relative heat-conducting value of metals, or their evaporative efficiency, as determined by careful experiments, when of good quality, free from impurities and laminations, compared with copper, is given in the following Table:—

Table 8.—Relative Internal Heat-conducting Power of Metals.

Copper-sheet	100	Cast-steel hammered	56
Silicium-bronze	85	Siemens mild-steel	54
Aluminium-bronze	80	Bessemer-steel	52
Phosphor-bronze	77	Wrought-iron, fine	50
Gun-metal	75	Wrought-iron, good ordinary	46
Brass, fine	72	Cast-iron, fine	42

These values are for clean surfaces. When the surface of metal is coated with a hard substance the conducting power is diminished.

Expansion of Substances by Heat.—All substances expand more or less when heated. The co-efficients of expansion of various substances, or their lineal expansion for every degree Fahr., are as follows:—

						Coefficient of linear expansion.
Copper	at temperatures between	32°	and	212°	Fahr.	·00000958
Copper	„ „	32°	„	572°	„	·00001095
Brass, cast	„ „	32°	„	212°	„	·00001047
Brass, sheet	„ „	32°	„	212°	„	·00001078
Gun-metal	„ „	32°	„	212°	„	·00001062
Phosphor-bronze	„ „	32°	„	212°	„	·00001067
Cast-iron	„ „	32°	„	212°	„	·00000630
Wrought-iron, average	„ „	32°	„	212°	„	·00000658
Wrought-iron, best	„ „	32°	„	212°	„	·00000685
Wrought-iron	„ „	32°	„	572°	„	·00000892
Cast-steel	„ „	32°	„	212°	„	·00000615
Mild-steel boiler-plates	„ „	32°	„	212°	„	·00000672
Mild-steel boiler-plates, Siemens-Martin, best	„ „	32°	„	450°	„	·00000700
Stock-bricks	„ „	32°	„	212°	„	·00000306
Fire-bricks	„ „	32°	„	212°	„	·00000235

The expansion of a long bar of metal is considerable. For instance, if a bar of wrought-iron 15 feet long be heated to a temperature of 170° Fahr. Then, the lineal expansion of the bar is = 15 feet × 12 = 180 inches × ·00000658 × 170 degrees = ·2013, or nearly $\frac{7}{32}$ of an inch.

Force exerted by Substances in Expanding and Contracting.—The force exerted by a bar in expanding by heat, or in contracting by cold, may be found by the following formula:—

Let A = the sectional area of the bar in square inches.
C = the coefficient of expansion from the above Table.
H = the number of degrees the bar is heated above 32° Fahr.
E = the modulus of elasticity, or resistance to stretching, of the bar,

= 18000000 for cast-iron; 28020000 for wrought-iron; and 30000000 for mild-steel.

F = the force in pounds exerted by the bar in expanding or contracting.

Then F = A × C × H × E.

Example: A wrought-iron girder having a cross-sectional area of 10 square inches is walled firmly into a building used as a drying-room. What force will be exerted on the walls of the building if the girder be raised to a temperature of 152° Fahr.?

Then the girder is raised 152° − 32° = 120 degrees in temperature, and 10 square inches × ·00000658 × 120° × 28020000 = 221246 pounds, the force exerted by the expansion of the girder.

Combustibles or Fuels.—Carbon and hydrogen are the two principal combustibles, and form the chief constituents of all solid, liquid, and gaseous fuels. The combustibles or fuels in general use are coal, coke, wood, peat, straw and other vegetable-refuse, charcoal, crude petroleum, creosote, mineral oil, and combustible gases obtained by distillation of solid kinds of fuel.

Coal, the staple fuel, is fossilized vegetable matter. It is composed chiefly of those elements which enter into the composition of organisms, or, carbon, hydrogen, and oxygen.

TABLE 9.—COMPOSITION OF FOSSIL VEGETABLE MATTER AT DIFFERENT STAGES OF TRANSFORMATION TO COAL.

State of Fossilization.	Carbon.	Hydrogen.	Oxygen and Nitrogen.
Vegetable matter, decayed	35	5·1	59·9
Cellulose	46	5·3	48·7
Peat	57	5·4	37·6
Lignite, or wood-coal	60	5·5	34·5
Lignite, or earthy brown coal	70	5·6	24·4
Coal, secondary	75	5·7	19·3
Coal, Bituminous	82	5·2	12·8
Anthracite	93	3·8	3·2

The transformation of the vegetable matter to coal is, briefly, as follows:—When wood and vegetable matter are buried in the earth, exposed to moisture, and partially or entirely excluded from the air, they decompose slowly and evolve carbonic acid gas, thus parting with a portion of their original oxygen. By this means they become gradually converted into lignite or brown coal, which contains a larger proportion of hydrogen than wood does. A continuance of decomposition changes the lignite into bituminous or common coal, chiefly by the discharge of carburetted hydrogen. The gaseous contents of the coal—carbonic acid, carburetted hydrogen, nitrogen, and olefiant gas—are continually escaping, and the disengagement of the gases gradually transforms bituminous or ordinary coal into anthracite or hard coal.

The composition of the fossil vegetable matter at different stages of transformation to coal is given approximately in Table 9, in which earthy matter is not included.

It will be seen from the foregoing Table that the quantity of oxygen decreases rapidly during the process of fossilization of the vegetable matter, and that the constituents of coal are chiefly carbon and hydrogen, which are united and solid.

All coals contain more or less impurities, as silica, alumina, lime, magnesia, oxide of iron, and earthy matter, the latter of which greatly detracts from the heating power of coal.

Classes of Coal.—There are two classes of coal,—anthracite and bituminous. The effect of bitumen is to facilitate ignition, and cause coal to cake and smoke. Hence the more bituminous the coal, the greater the liability to cake and to emit smoke. The following are the principal varieties of coal:—

Anthracite, or stone-coal, is of hard stone-like structure, contains no bitumen, but consists almost entirely of free carbon. The finest quality of anthracite is found in the Pembrokeshire, Gwendraeth Valley, and Vale of Neath coalfields. It has a beautiful and lustrous appearance, is compact and hard, gives a ringing sound when struck, and is of such a cleanly nature that it does not soil the hand.

This coal is difficult to ignite, burns slowly or sluggishly, and breaks into small pieces when rapidly heated, but yields intense heat without smoke when dry. It requires a very large fire-grate, or about double the grate-area required for bituminous coal, and heavy charges of coal at each firing. It also requires considerable attention, a thin fire of regular thickness, a well-distributed supply of air through the fire, and a strong draught, and the fire must remain undisturbed for a long interval; therefore it is not adapted for burning economically in the furnace of a steam-boiler with natural draught.

The evaporative power of anthracite in ordinary boiler-furnaces is less than that of bituminous coals. It is difficult in practice to evaporate more than 8 pounds of water per pound of anthracite in a steam-boiler with a strong draught under ordinary working conditions, while one pound of good bituminous coal will easily evaporate 10 pounds or more of water under the same conditions.

Semi-Anthracite, anthracitic, or slightly bituminous coal, such as the best Welsh steam-coal, has less density than anthracite. It requires little attention, burns freely with a short flame, and yields great heat with very little clinker and ash; but it decrepitates much in burning, and wastes considerably in falling through the bars of the fire-grate. It swells considerably, but does not cake, and emits only a light vapoury smoke. It is very tender and will not bear rough usage, and crumbles rapidly after long exposure to the atmosphere, but is excellent for steam-producing purposes.

Semi-Bituminous Coal is of dull black colour, and contains a considerable amount of hydro-carbons. When of good quality it ignites easily, burns freely with a short flame with ordinary draught; but it emits a considerable quantity of smoke, unless means be taken to prevent its formation. It requires little attention, makes very little clinker and ash, and is an excellent steam-coal.

Bituminous Clear-burning Coals are generally very tender, ignite with difficulty, and burn slowly with a short, clear, bluish flame.

Bituminous Flaming Coals are black and glossy, ignite with difficulty, but burn somewhat rapidly with a long white flame.

Bituminous Fuliginous Coals contain a large proportion of volatile matter, ignite easily, and burn rapidly with a long yellow fuliginous flame.

Bituminous Gaseous Coals are hard, compact, and strong, of a brownish-black colour and dull lustre. They contain a large amount of bitumen, and yield about two-thirds of volatile matter, and one-third of clinker and ash. They average in composition from 60 to 78 per cent. of carbon, and from 5·75 to 9 per cent. of hydrogen. When distilled they yield a large amount of hydro-carbons, and are principally used for making coal-gas.

The quantity of carburetted hydrogen evolved varies from 5,000 to 12,000 cubic feet per ton of coal. This kind of coal is not suitable for the furnaces of steam-boilers.

Storage of Coal.—Coal should be kept dry. When exposed to the atmosphere it rapidly deteriorates in quality owing to the gradual escape of part of its volatile constituents.

Distillation of Coal.—One ton of Lancashire coal, distilled in gas retorts, is stated to be capable of yielding the following products:—

1. Coal-gas, 10,000 cubic feet.
2. Ammonia-liquor, 5 per cent., 20–23 gallons = 30 pounds.
3. Coal-tar, 12 gallons, or 139·2 pounds.
4. Coke, 13 cwt.

Twelve gallons of coal-tar yield:—

1. Benzene, 1·1 pound; aniline, 1·1 pound.
2. Toluene, ·9 pound; toluidine ·77 pound.
3. Phenol, 1·5 pound; aurine, 1·2 pound.
4. Solvent naphtha, 2·4 pounds.
5. Naphthalene, 6·3 pounds; naphthol, 4·75 pounds; naphtha, yellow, 9·5 pounds.
6. Creosote, 17 pounds.
7. Heavy oils, 14 pounds.

8. Anthracene, ·46 pound; alizarin, 20 per cent., 2·25 pounds.
9. Pitch, 69·6 pounds.

These are the average products, but the yield varies with the quality of the coal.

Lignite, or Brown Coal, is of more recent formation than bituminous coal, and is an imperfectly formed coal. It is frequently of a woody texture, and generally contains considerable moisture and a large proportion of oxygen. When dry, it ignites easily and burns very rapidly. It weighs less and develops less heat than coal, and is not a valuable description of fuel.

Slack is the screenings of coal. Large-sized screenings are termed rough slack, and small-sized screenings small slack. The dust of screenings, or fine siftings from slack, is termed duff or smudge.

The Composition of Coal varies considerably. The average results of a number of careful analyses of British and foreign coals are given in the following table. The heat evolved by the combustion of any of the coals, and their equivalent evaporative power, may be calculated by the rules on page 42. The evaporative power of coals is given at page 45.

TABLE 10.—COMPOSITION OF BRITISH AND FOREIGN COALS.

Description of Coal.	Carbon	Hydrogen.	Oxygen.	Nitrogen.	Sulphur.	Ash.	Specific Gravity.
Welsh Anthracite . . .	91·7	3·78	1·30	1·00	·72	1·5	1·37
Welsh Coal, Penrikyber . .	89·0	4·25	1·45	1·00	·80	3·5	1·32
Ditto, Powell-Duffryn's .	88·1	4·20	2·17	1·63	·90	3·0	1·32
Ditto, Aberdare . .	88·4	3·65	3·35	·65	·77	3·18	1·32
Ditto, Good Ordinary .	86·0	4·64	2·96	1·40	1·70	3·2	1·31
Newcastle Coal	82·9	5·30	5·24	1·26	1·30	4·0	1·26
Ditto, Good Ordinary.	78·0	5·15	8·63	1·30	1·32	5·6	1·27
Durham Coal	80·0	5·10	7·23	1·27	1·40	5·0	1·27
Lancashire Coal. . . .	79·5	5·42	8·51	1·18	1·50	4·7	1·27
Derbyshire Coal . . .	78·9	4·88	10·34	1·38	1·10	3·4	1·27
Staffordshire Coal . . .	78·2	5·28	10·32	1·50	1·20	3·5	1·28
Yorkshire Coal . . .	78·1	4·84	10·53	1·43	1·10	4·0	1·29
Nottinghamshire Coal . .	75·2	5·60	12·34	1·33	1·23	4·3	1·28
English Coal, Best, Average .	80·4	5·20	7·77	1·13	1·40	4·1	1·27
Ditto, Second Quality, Average	72·9	4·97	13·47	1·06	1·60	6·0	1·26
Scotch Coal	70·0	4·85	13·50	1·35	1·70	6·0	1·27
Ditto, Good Ordinary .	77·4	5·56	9·68	1·26	1·50	4·6	1·27
Gaseous Coal and Brown-Black	63·8	8·85	4·77	·95	·33	21·3	1·22
FOREIGN COALS.							
Sydney, N.S.W., Bituminous .	82·4	5·30	8·30	1·20	·70	2·1	1·30
Australian Brown Coal .	73·2	4·71	12·35	1·11	·63	8·0	1·27

TABLE 10 *continued*.—COMPOSITION OF BRITISH AND FOREIGN COALS.

Description of Coal.	Carbon	Hydrogen.	Oxygen.	Nitrogen.	Sulphur.	Ash.	Specific Gravity.
Foreign Coals *continued*.							
Australian Coal, Good, Average	71·0	4·65	18·15	1·25	1·00	4·0	1·28
American, Rhode Island . .	85·0	3·71	2·39	1·00	·90	7·0	1·42
Ditto, Maryland . .	80·0	5·00	2·70	1·10	1·20	8·3	1·33
Ditto, Penna . . .	75·5	4·93	12·35	1·12	1·10	5·0	1·32
Ditto, Tennessee . .	70·0	5·21	8·06	1·53	1·40	13·8	1·39
Ditto, Indiana . . .	69·7	5·10	19·17	1·23	1·30	3·5	1·25
Ditto, Illinois . . .	61·4	4·87	35·42	1·41	1·20	5·7	1·31
Ditto, Virginia . . .	57·0	4·96	26·44	1·70	1·50	8·4	1·32
Ditto, Alabama . .	53·2	4·81	32·37	1·62	1·30	6·7	1·30
Ditto, Kentucky . . .	49·1	4·95	41·13	1·70	1·40	7·2	1·28
Californian Brown Coal . .	49·7	3·78	30·19	1·00	1·53	13·8	1·32
Austrian Lignite	58·10	4·64	30·43	·10	·73	6·0	1·20
French Anthracite . . .	90·9	1·47	1·53	1·00	·80	4·3	1·35
Ditto, Bituminous . . .	83·8	4·82	4·86	1·22	1·30	4·0	1·31
Russian, Grouchefski, Anthracite	90·7	3·50	1·40	1·00	·80	2·6	1·40
Ditto, Miouche, Bituminous	83·9	4·10	4·50	1·00	1·50	5·0	1·32
Saxony, ,, .	79·4	4·65	9·48	1·37	1·10	4·0	1·29
Borneo, ,, .	70·3	5·41	19·16	·66	1·17	3·3	1·37
Cape Breton, ,, .	67·2	4·26	20·16	1·07	1·21	6·1	1·33
Vancouver's Island ,, .	66·9	5·32	8·76	1·02	2·20	15·8	1·28
Formosa Island ,, .	78·3	5·70	9·91	1·60	·49	4·0	1·29
Chilian, ,, .	70·6	5·71	13·26	·95	1·98	7·5	1·28
Patagonian, ,, .	62·3	5·00	17·84	·63	1·13	13·1	1·29
Nova Scotia, ,, .	60·8	5·70	19·45	·65	·90	12·5	1·32
Brazil, ,, .	58·0	5·42	29·04	1·14	1·41	5·0	1·29
Silesia, ,, .	57·5	5·34	30·72	1·21	1·23	4·0	1·26
Hindostan, ,, .	50·0	4·26	27·50	1·34	1·40	15·5	1·36
Queen Charlotte's Island, Anthracite	55·1	1·72	3·20	·60	·38	39·0	1·32
Indian Coal, from Northern Districts	57·2	4·20	31·10	1·30	1·20	5·0	1·34
Indian Coal, from Southern Districts	64·6	4·75	18·40	1·25	1·00	10·0	1·36
Lignite, or Brown Coal . .	69·0	5·30	19·00	·20	·60	5·9	1·28

Mixed Coal.—It is frequently advantageous to mix a little soft coal with hard coal. Coal from South Wales mixed with a small quantity of bituminous coal makes excellent fuel for steam-boilers.

Flame of Coal.—Flame is considered to be a hollow cone having the heat concentrated at its apex. The heating-power of flame is much greater than that of heated gas. Coal, which evolves the largest quantity of heavy carburetted hydrogen gas, and liberates the largest quantity of carbon by combustion, gives the most luminous flame. As flame increases in temperature, it diminishes in length and becomes whiter in colour. The greatest heat is radiated from flame when it has space for free development.

The Quantity of Ash yielded by the combustion of coal varies, according to Table 10, from 1½ to 39 per cent. Good Welsh steam coal only yields 3·2 per cent. of ash.

Clinker is produced by fusion of ash. Coals which produce much clinker rapidly destroy the fire-grate bars.

The quantity of clinker and ashes produced from coal consumed in the furnace of a steam-boiler is, under ordinary working conditions, seldom less than 10 per cent. It varies according to the description and quality of the coal used, and is, on an average, approximately as follows:—

Welsh coal . . .	7 per cent.	Bituminous slack . .	12 per cent.
Bituminous coal . .	10 per cent.	Anthracite, small	6 to 18 per cent.

Small slack of inferior quantity seldom produces less than 20 per cent. of clinker and ash in the furnace of a steam-boiler.

The expense of getting rid of clinker and ash is so considerable that it is frequently more economical to use best steam-coal yielding little ash, than a cheaper coal making a quantity of clinker and ash.

At rates of combustion of 40 pounds of coal per square foot of fire-grate surface per hour and upwards, clinker accumulates very rapidly in the furnace, and is an obstacle to economical combustion.

The Composition of Ash varies considerably. The nature of the incombustible ingredients of ash may be seen from the average results of several analyses of ash of coal from different districts, given in the following Table:—

TABLE 11.—COMPOSITION OF COAL-ASH.

Description of Coal.	Silica.	Alumina and Oxide of Iron.	Lime.	Magnesia.	Sulphuric acid.	Phosphoric acid.	Total per centage.
Anthracite . .	44·7	48·87	2·96	·58	2·00	·62	99·73
Welsh Coal . .	52·2	36·30	3·86	2·27	4·55	·78	99·94
Newcastle Coal . .	60·4	26·64	3·25	1·67	7·00	·84	99·80
Lancashire Coal .	48·7	35·56	6·34	1·54	6·83	·69	99·66
Derbyshire Coal .	42·5	40·00	7·87	1·37	7·46	·76	99·96
Staffordshire Coal .	56·4	26·61	6·37	1·76	7·91	·87	99·92
Yorkshire Coal .	62·0	20·40	5·73	2·18	8·40	·89	99·60
Scotch Coal . .	54·5	29·38	8·94	1·87	4·26	·85	99·80
Lignite . . .	20·1	18·42	22·10	2·48	33·00	3·84	99·94

The Heat Lost in the Clinker and Ash, withdrawn from the furnace of a steam-boiler carefully-fired with good coal, averages from ¼ to 2 per cent. of the heat developed by the combustion of the coal. It may be roughly estimated from the rise of temperature observed in quenching a known weight of clinker and ash in a known weight of water. For instance, if 20 lbs. of clinker and ash, when quenched in 120 lbs. of water at 55° Fahr., raised its temperature to 118° Fahr., then the rise of temperature is $= 118° - 55° = 63°$, and $\frac{63° \times 120 \text{ lbs. of water}}{20 \text{ lbs. of clinker and ash}} = 378$ units of heat were given up to the water, and if the weight of clinker and ash produced by each pound of coal was, say, ·12 lb., then 378 units × ·12 = 45·36 units of heat were lost in the clinker and ash per pound of coal consumed.

The Weight and Bulk of Coal varies considerably. The average weight and bulk of a number of samples of coal from various districts is given in the following Table, which also gives their burning qualities.

TABLE 12.—WEIGHT, BULK, AND BURNING QUALITIES OF COAL.

Description.	Weight of one cubic foot of loose coal heaped.	Bulk of one ton of loose coal heaped.	Draught required.	How it burns.	Quantity of smoke.
	lbs.	Cubic feet.			
Anthracite . . .	58·2	38·4	Quick	Difficultly	None
Welsh Coal . .	52·1	43·2	Quick	Free and clear	Little
Newcastle Coal . .	50·3	45·5	Ordinary	Quickly	Large
Lancashire Coal .	48·6	46·0	Ordinary	Quickly	Large
Derbyshire Coal .	46·8	47·4	Moderate	Quickly	Large
Staffordshire Coal .	47·0	47·5	Ordinary	Quickly	Large
Yorkshire Coal . .	47·6	47·0	Brisk	Quickly	Large
Scotch Coal . .	49·2	43·0	Ordinary	Quickly	Verylarge

Coke is the solid carbon and earthy matter remaining after coal has been deprived of its volatile constituents by partial combustion, or by slow distillation in retorts. The composition of coke averages as follows:—

Carbon	from 85 to 95 per cent.
Sulphur	„ 1 „ 2 „
Ash	„ 3 „ 15 „

The best quality of coke yields the least ash. The quantity of coke yielded by coal averages two-thirds of the weight of coal. Coke is screened into three sizes—large coke, small coke, and breeze.

The Heating Power of Coke is calculated from its constituent carbon, the sulphur being usually neglected. For instance, coke containing 90 per cent. of carbon will develop 14500 × ·90 = 13050 units of heat per pound. Its evaporative power may be calculated by this *Rule*:—

$$\frac{\text{Units of heat evolved per lb. of fuel}}{\text{Heat absorbed by water supplied at } 212^\circ \text{ Fahr.}}$$

$$= \frac{\text{Heat evolved per lb.}}{966}.$$

This coke will evaporate $\frac{13050 \text{ units of heat per lb.}}{966} = 13{\cdot}51$ lbs. of water from and at 212° Fahr.

Coke has the advantage of absence from dust and smoke, and it yields most of the heat developed in the furnace to the heating-surfaces above the fire-grate. The heating-power of a good gas-coke is from 10,143 to 11,109 units per pound, and its evaporative power is from 10·5 to 11·5 pounds of water per pound of fuel from and at 212° Fahr. It yields from 8 to 12 per cent. of ash and clinker.

To Burn Coke Economically, the greatest available amount of heating surface should be exposed to the furnace, the fire should not be forced, and the fire-grate should be proportioned to burn not more than eight pounds of coke per square foot of the area of the fire-grate surface per hour. There should be a strong draught, increasing in strength as the size of the coke diminishes. The fire-bars should be of wrought-iron, and the fire-grate protected from burning by water-pans placed in the ash-pit.

Breeze from coke of good quality will develop from 6,279 to 7,245 units of heat per pound, and its evaporative power is from 6½ to 7½ pounds of water per pound of breeze, from and at 212° Fahr. It yields from 15 to 25 per cent. of ash and clinker.

Peat is the decayed vegetable matter, or organic matter, of bogs and marshes. It is light, spongy, and fibrous, and contains in its natural state about 85 per cent. of water. When air-dried it retains permanently about 20 per cent. of moisture, and when kiln-dried about 7 per cent. of moisture. It is very bulky, and occupies about four times the space, for equal weight, of coal, and as its maximum evaporative power is little more than one half that of coal, about eight cubic feet of peat are required to evaporate an equal quantity of water as one cubic foot of coal. The composition of peat averages as follows:—

Carbon	45 to 60 per cent.
Hydrogen	4 „ 6 „
Oxygen	25 „ 45 „
Nitrogen	1 „ 1¾ „
Ash	4 „ 7 „

Peat is consumed so rapidly as to necessitate almost constant firing.

The Heating Power of Peat in a moderately dry state, containing ·47 carbon, ·05 hydrogen, and ·32 oxygen, is as follows:—The constituent oxygen will combine with $\cdot 32 \div 8 = \cdot 04$ hydrogen, leaving $\cdot 05 - \cdot 04 = \cdot 01$ hydrogen in excess, and carbon $\cdot 47 \times 14500 = 6815$ units $+ (\cdot 01 \times 62535)$

= 625 units of hydrogen = 7440 units, its maximum calorific power, and its evaporative power is $7440 \div 966 = 7 \cdot 7$ pounds of water from and at 212° Fahr. The evaporative power of peat of average quality is very much less than this. In a test of a boiler fired with peat the evaporation was only 4·2 pounds of water per pound of peat.

Peat is frequently compressed into blocks weighing about 70 lbs. per cubic foot, and occupying a space of about 48 cubic feet per ton.

Peat-Charcoal is light and porous, and burns feebly. Its average composition is ·82 carbon and ·12 ash; its heating power is $\cdot 82 \times 14500 = 11890$; its evaporative power is $11890 \div 966 = 12 \cdot 31$ pounds of water from and at 212° Fahr. Peat yields about one-third of its weight of charcoal.

Wood-Charcoal is prepared by exposing wood to a red heat until deprived of its gases and volatile matter, the product being nearly pure carbon.

It is necessary to heat the wood for a considerable time to a temperature of at least 400° Fahr. to drive off the gases; at 550° Fahr. half formed or brown charcoal is produced; at 700° Fahr. the wood is converted into a strong, brittle, black charcoal of wood-like structure, which makes a bright, clear fire that radiates heat strongly, and burns without flame or smoke. By using forced draught with a small supply of air to the fire, intense heat may be developed by the combustion of charcoal. Hard wood makes better charcoal for fuel than soft wood.

The composition of charcoal varies with the temperature at which it is produced. Charcoal of hard wood is composed approximately as given in the following Table:—

TABLE 13.—AVERAGE COMPOSITION OF CHARCOAL OF HARD WOOD AT VARIOUS STAGES OF CARBONIZATION.

Temperature of carbonization.	Carbon per cent.	Hydrogen per cent.	Approximate yield of charcoal in percentage of weight of wood.
220° Fahr.	50	6·0	48
400° ,,	60	5·1	40
550° ,,	70	4·6	35
700° ,,	77	4·0	28
850° ,,	80	3·1	21
2100° ,,	85	1·5	19
2300° ,,	90	1·0	18
2500° ,,	93	·3	17
2800° ,,	95	——	16

The Heating Power of Wood-Charcoal containing ·95 carbon is $\cdot 95 \times 14500 = 13775$ units per pound, and its evaporative power is $= 13775 \div 966 = 14 \cdot 26$ pounds of water from and at 212° Fahr.

The quantity of ash yielded by wood-charcoal is from 2 to 10 per cent.

Compressed Charcoal is composed of two-thirds of powdered charcoal and one-third gas-tar. It is moulded under pressure into short round pieces, and baked at a high temperature. It burns slowly, without flame or smoke.

Patent Fuel, or artificial fuel, is composed of small coal, or other refuse fuel, mixed with adhesive and combustible substances, such as pitch and tar, and moulded by machinery into blocks.

Coal-dust Briquettes, or block-coal, of good quality, generally contain from 86 to 90 per cent. of coal-dust, mixed with from 8 to 10 per cent. of pitch, and from 2 to 4 per cent. of tar. It is moulded under pressure into blocks, and baked at a high temperature. Its average composition is as follows:—

Carbon	from	70	to	87	per cent.
Hydrogen	„	4·25	„	5·6	„
Oxygen	„	1	„	2	„
Nitrogen	„	1	„	1·8	„
Sulphur	„	1	„	1·6	„
Ash	„	3	„	5·0	„

It burns freely, and develops on an average 9,660 units of heat per pound. This is equal to an evaporation of $9660 \div 966 = 10$ pounds of water per pound of fuel, from and at 212° Fahr.

Coke-dust Briquettes, or block-coke, are generally composed of washed coke-dust 100 pounds, pitch 7 pounds, tar $2\frac{1}{2}$ pounds. It is made from gas-coke refuse, and has on an average a heating power of 8,970 units of heat per pound. This is equal to an evaporation of $8970 + 966 = 9·28$ pounds of water per pound of fuel, from and at 212° Fahr.

Sawdust Briquettes, or block sawdust, are a mixture of sawdust, tar, and clay moulded into blocks. It burns freely, and develops about 6,300 units of heat per pound, and its evaporative power is $6,300 \div 966 = 6·52$ pounds of water from and at 212° Fahr.

Vegetable-refuse Fuel, such as sugar-cane refuse, cotton-stalks, reeds, coarse grass, fibrous plants, and straw, may be used in steam-boilers having sufficient capacity of furnace and fire-box.

They develop on an average 3570 units of heat per pound, and evaporate $3570 \div 966 = 3·7$ pounds of water from and at 212° Fahr. Spoilt grain has sometimes been used as fuel; its evaporative value is less than one-fourth that of good coal.

Steam-Boilers Fired with Vegetable Refuse-Fuel.—To effect economical combustion of refuse-fuels, they require on an average, a boiler with double the area of fire-grate surface, double the area of fire-box surface, and one-half greater area of total heating-surface per indicated horse-power of the engine, than is required for a boiler burning coal.

Various Fuels.—The heating powers of numerous other fuels are given in Table 20, pages 44–46.

Straw-Fuel for Steam-Boilers.—Straw may be economically burnt as fuel for steam-boilers, but it requires a furnace specially arranged for the purpose. The fire-box of the boiler of a portable engine, arranged for burning straw, is shown in Fig. 2. The straw is fed with a pitchfork through a wide aperture placed below the ordinary fire-door.

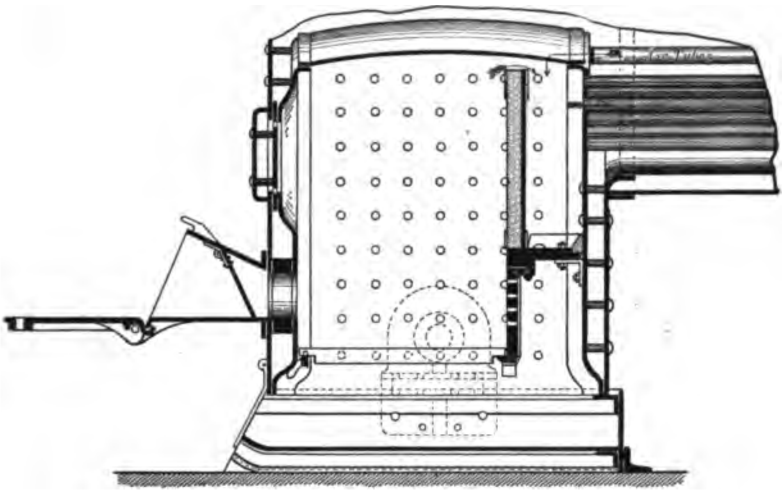


Fig. 2.—Garrett's straw-burning fire-box.

The straw in burning fills the fire-box full of flames, which pass over the top of a high fire-brick bridge before entering the tubes. The flames are met at the top of the bridge by streams of heated air coming from the smoke-box through tubes provided for the purpose, as shown in the engraving.

In a test of a compound portable engine having a boiler with fire-box arranged in this manner, the straw burnt per square foot of fire-grate per hour was 55·1 pounds; the straw consumed per indicated horse-power per hour was 7·78 pounds, and each pound of straw evaporated 2·6 pounds of water. In the test of the boiler of a simple portable engine with a similar fire-box, the straw burnt per square foot of fire-grate per hour was 57·6 pounds; the straw consumed per indicated horse-power per hour was 11·9 pounds, and each pound of straw evaporated 2½ pounds of water.

For ordinary portable engine-boilers three pounds of straw may be assumed to be equivalent in heating power to one pound of coal when burnt in a furnace of this kind.

Wood-Fuel for Steam-Boilers.—Green wood contains about 50 per cent. of moisture, and loses about 50 per cent. in weight when dried at a temperature of 280° Fahr. Air-dried wood contains from 20 to 25 per cent.

of moisture. As the heating-power of wood decreases with the quantity of moisture it contains, it should be dried in a kiln at a temperature of not less than 380° Fahr. Soft woods require to be more dried than hard woods.

The composition of wood does not vary to the same extent as that of coal. The average composition of wood is given in the following Table:—

TABLE 14.—AVERAGE COMPOSITION OF WOOD.

Elements.	Composition of Wood.		
	In a dry state.	In its ordinary state of dryness, or containing 20 per cent. of water.	Imperfectly dry, or containing 25 per cent. of water.
	Per cent.	Per cent.	Per cent.
Hygrometric Water	——	20	25·00
Carbon	51	40·8	38·25
Hydrogen	6	4·8	4·50
Oxygen	40	32	30·00
Nitrogen	1	0·8	0·75
Ash	2	1·6	1·50
	100	100·0	100·00

Hard woods yield more intense, more prolonged, and steadier, heat than soft woods. Soft woods kindle more readily, and burn more rapidly, than hard woods.

Wood, on an average, contains approximately 1 pound of hydrogen for each 8 pounds of oxygen, and, as the hydrogen is thus neutralized by the oxygen, it develops no heat.

Wood, in its ordinary state of dryness, develops $\cdot 408$ carbon $\times$ 14500 units of heat per pound of carbon $= 5916$ units per pound, and evaporates $5916 \div 966 = 6\cdot 124$ lbs. of water from and at 212° Fahr.

English oak, kiln-dried, averages $\cdot 50$ carbon, and develops $\cdot 50 \times 14500 = 7250$ units of heat per pound, and will evaporate $7250 \div 966 = 7\frac{1}{2}$ lbs. of water from and at 212° Fahr. Wet wood on an average develops 2898 units of heat per pound, and evaporates $2898 \div 966 = 3$ lbs. of water from and at 212° Fahr.

Boiler-Furnace for Burning Wood.—A cord of wood can be burnt per hour on 60 square feet of fire-grate surface. Wood fuel requires one-third more fire-grate surface and two-thirds more space in the furnace than is required for coal, for equal generation of steam.

A Cord of Wood contains $4 \times 4 \times 8 = 128$ cubic feet, of which 73 cubic feet are solid wood, and the remainder, 55 cubic feet, is space. The weight of coal that one cord of different kinds of dry wood is equivalent to, in evaporative power in a steam-boiler, is given in the following Table :—

TABLE 15.—APPROXIMATE WEIGHT OF ONE CORD OF DIFFERENT KINDS OF KILN-DRIED WOODS, AND THEIR EVAPORATIVE POWER COMPARED WITH COAL OF AVERAGE QUALITY.

Kind of Wood.	Approximate weight of one Cord of the Wood.	Weight of Coal that one Cord of Wood is approximately equivalent to in evaporative power.
	lbs.	lbs.
English oak	3850	1560
Ash, beech and thorn . . . . . . each	3520	1420
Red oak, hard maple and walnut . . . each	3310	1340
Apple-tree, pear-tree, cherry-tree and plum-tree, each	3140	1260
Birch, elm, plane-tree and hazel . . . each	2880	1190
Chestnut, brushwood and yellow pine . . each	2520	1130
Pitch-pine, alder, aspen and poplar . . each	2130	1050
Willow, white pine or deal . . . . . each	1920	970
Hemlock	1220	580

The quantity of ash yielded by wood averages from 1 to 4 per cent. The quantity of ash yielded by English oak is generally 1·7 per cent.

The Heating Power of Soft Wood, such as pine, or of mixed soft woods, not kiln-dried, but in a moderately dry state, varies considerably. It may, in a general way, be assumed that one pound of wood of this description is equivalent in heating power to one quarter of a pound of coal, or that four pounds of wood are equivalent in evaporative power to one pound of good coal.

Gaseous-Fuel.—There are five classes of gaseous-fuels, which may be briefly described as follows.

Illuminating Gas, or coal-gas, is the product of the distillation of coal in closed retorts. The quantity of gas produced is generally 10,000 cubic feet per ton of coal distilled. The gas-companies undertake to supply gas of 16 candle-power, or gas giving a light equal to that of 16 sperm-candles burning 120 grains of sperm per hour, the consumption of gas being 5 cubic feet per hour through an approved burner.

Generator-Gas is the product of the furnaces in which coke is gasified for heating retorts in coal-gas works, wherein the carbon of the solid fuel is transferred into carbonic-oxide by combination with atmospheric oxygen.

Water-Gas is formed by the combustion of carbon in aqueous vapour or steam.

Siemens-Gas is a mixture of illuminating and generator gases, produced in the ordinary generators by the gasification first of coal, and afterwards of the coke resulting therefrom.

Generator-Water-Gas is produced when steam and air together are admitted to an ordinary coke-generator, as is often done in the usual way of working.

Production of Gas.—Illuminating gas, like water-gas, is produced by the extraneous heating of the generating apparatus, either through the heating of retorts from the outside, or from the bottom of the generator respectively. The evolution of the gases themselves does not develop heat, and the gases are produced at a low temperature. The three other gases are formed in conjunction with a development of heat. They do not require external heating, and are consequently made by a simple selfacting process.

Composition of Gaseous-Fuels.—The chemical constitution of these five descriptions of gases, as determined by Dr. Bunte, is in round numbers as follows, their percentage being in volume :—

TABLE 16.—COMPOSITION AND HEATING-POWER OF GASEOUS-FUELS.

Constituents.	Illuminating-Gas.	Generator-Gas.	Water-Gas.	Siemens Gas.	Generator Water-Gas.
Carbonic oxide	9·0	34·3	50·0	20·0	38·0
Hydrogen	47·0	—	50·0	6·0	12·0
Marsh-gas	34·0	—	—	2·0	—
Heavy hydrocarbons	5·0	—	—		—
Carbonic acid, nitrogen, etc.	5·0	65·7	—	72·0	50·0
	100·0	100·0	100·0	100·0	100·0
CALORIFIC VALUE (CALORIES) PER UNIT OF BULK.					
Carbonic oxide	275	1048	1527	611	1161
Hydrogen	1209	—	1286	154	309
Marsh-gas	2916	—	—	280	—
Heavy hydrocarbons	1111	—	—		—
Heating power per cubic metre of gas	5511	1048	2813	1045	1470
Comparative value as fuel.	5·3	1	2·7	1	1·

Heating Power of Gaseous-Fuels.—It will be seen from the above Table that the heating power of water-gas is only about one-half that of coal-gas, and that of generator-water-gas is only about one-half that of water-gas. The superiority of illuminating gas and water-gas is due to their high percentages of hydrogen, which is more than 30 times superior in inflammability and burning power to carbon monoxide. A considerable portion of the heating effect of coal-gas is due to the heavy hydrocarbons contained in it.

The heating power of ordinary coal-gas is from 630 to 700 heat-units per cubic foot of gas. As 30 cubic feet of coal-gas at 62° Fahr. weigh 1 lb. the heating power of ordinary coal-gas of average quality is $= 630 \times 30 =$ 18900 heat-units per pound of gas.

The heating power of water-gas being one-half that of coal-gas is $=$ $630 \div 2 = 315$ heat-units per cubic foot of gas.

The temperature produced by the flame from a properly proportioned mixture of coal-gas and air is about 3670° Fahr., and that of water-gas is about 4900° Fahr.

Natural Gas.—Natural gas, or that issuing from wells of mineral oil, is composed, on an average, of hydrogen 23 per cent., marsh-gas 67 per cent., and 10 per cent. of other bodies. One thousand cubic feet of this gas are equal in heating-power to one-half a cwt. of good bituminous-coal.

Steam-Boilers Fired with Gaseous-Fuel.—When gas is employed for heating a steam-boiler, it is necessary to employ a furnace specially arranged for the purpose. The quantity of gas required, in a multitubular-boiler with special furnace, to generate the steam necessary for the development of one indicated horse-power per hour in a good engine is as follows:—

Coal-Gas . . .	20 cubic ft.	Natural-Gas . .	50 cubic ft.
Water-Gas . . .	40 " "	Generator Water-Gas .	80 " "

These are the minimum quantities required on an average with well-arranged burners.

Liquid-Fuel.—Liquid-fuel gives special facilities for the development and maintenance of intense steady heat, for the quick control of the applied heat, and for the rapid generation of steam.

Liquid-fuels, such as petroleum, petroleum-refuse, tar, and creosote-oil or tar-refuse, have a much higher calorific power than coal, because they contain a much larger quantity of hydrogen.

Petroleum is a natural hydrocarbon oil, having, in its crude state, a calorific power one and a half times as great as that of coal. Petroleum oil is obtained by distillation from petroleum. Its calorific power is from two and a half to three times as great as that of coal.

The best petroleum fuel-oil has a specific gravity of ·818, and weighs 10 pounds × ·818 = 8·18 pounds per gallon. Its composition averages as follows:—

Carbon	85·34
Hydrogen	13·51
Oxygen and impurities	1·15
	100·00

It contains about three times as much hydrogen as is contained in good coal.

The theoretical heating power of this fuel-oil is 20822 thermal units, and it has a theoretical evaporative power = 20822 ÷ 966 = 21·56 pounds of water, from and at 212° Fahr., per pound of oil. Its actual evaporative power in practice is from $15\frac{1}{2}$ to 17 pounds of water, from and at 212° Fahr., per pound of oil. Its flashing-point is about 217° Fahr.

In a general way, 104 gallons, or 104 × 8·18 = 851 pounds, of this oil are equal in evaporative power to one ton of good coal.

Method of Burning Fuel-Oil.—Petroleum fuel-oil is burnt, in a boiler furnace, in the form of spray, after being pulverised or atomised by steam or compressed air. It cannot be burned in an ordinary furnace as arranged for coal-burning, because it makes an enormous quantity of smoke, which coats the water-heating surfaces with a sticky, sooty, non-conducting deposit.

To prevent the production of smoke it is necessary to burn the oil with a large supply of air in a furnace so arranged as to accumulate heat, or in a muffle-like, brick-lined chamber. The high temperature of the chamber prevents cooling of the gases and partial extinction of flame.

When the oil is sprayed into a furnace of this kind, practically complete combustion may generally be obtained without the production of soot.

Air Required for the Combustion of Fuel-Oil.—The quantity of air required for the complete combustion of fuel-oil is at least one-third greater than that required for good coal. The minimum quantity of air that should be provided in practice is 22 pounds of air per pound of oil, but it is generally necessary to provide a larger quantity than this, in order to prevent the production of smoke.

TABLE 17.*—CHEMICAL COMPOSITION AND THEORETICAL EVAPORATION OF PETROLEUM-FUEL COMPARED WITH COAL.

Description of Fuel.	Specific Gravity at 32° Fahr.	Chemical Composition.		
		Carbon.	Hydrogen.	Oxygen.
	water = 1.	per cent.	per cent.	per cent.
Pennsylvanian heavy crude oil	.886	84·9	13·7	1·4
Caucasian light crude oil	·884	86·3	13·6	0·1
Caucasian heavy crude oil	·938	86·6	12·3	1·1
Petroleum-refuse	·938	87·1	11·7	1·2
Good English coal, mean of 98 samples	1·380	80·0	5·0	8·0

Description of Fuel.	Specific Gravity at 32° Fahr.	Heating power in Thermal units.	Theoretical Evaporation in pounds of water per pound of fuel.	
			From and at 212° Fahr.	At 8½ Atmospheres effective pressure.
	water = 1.	units.	lbs.	lbs.
Pennsylvanian heavy crude oil	·886	20736	21·48	17·80
Caucasian light crude oil	·884	22027	22·79	18·90
Caucasian heavy crude oil	·938	20138	20·85	17·30
Petroleum-refuse	·928	19832	20·53	17·10
Good English coal, mean of 98 samples	1·380	14112	14·61	12·16

* The Author is indebted for this Table and for engravings of the apparatus for burning petroleum-fuel, to a paper read before the Institution of Mechanical Engineers, by Mr. Thomas Urquhart, Locomotive Superintendent, Grazi Tsaritsin Railway, South-East Russia.

Steam-Boilers Fired with Liquid-Fuel.—As a steam-boiler well arranged for the combustion of petroleum fuel-oil should produce no soot, the tubes may be of smaller diameter than is necessary for coal, and the heating surface of the boiler may be increased to the extent of from 33 to 50 per cent.

In a general way, a boiler fired with this oil may be from one-third to one-fourth smaller than one fired with coal, and still possess the same evaporative power, with the same strength of draught.

Petroleum-Refuse or Astaki is the dead oil or refuse left in the stills after the crude oil has been refined. It makes an excellent fuel for steam-boilers. A system of burning this and other liquid fuels has been devised and brought to great perfection by Mr. Thomas Urquhart. The composition, heating-power, and evaporative power of different kinds of liquid fuel used on his locomotives are given in Table 17.

The flashing point of petroleum-refuse is about 212° Fahr. The highest evaporative duty of the petroleum-refuse used in Mr. Urquhart's locomotives is 14 pounds of water per pound of fuel. The theoretical evaporative value of this fuel is 17·1 lbs. of water per lb. of fuel; the actual efficiency of petroleum-refuse is therefore = 14 lbs. ÷ 17·1 lbs. = 82 per cent. of the theoretical efficiency. This shows that the combustion is very complete, and that the apparatus for effecting the combustion of the fuel is practically perfect.

Urquhart's Spray-Injector for Liquid-Fuel.—Urquhart's spray-injector is shown in Fig. 3. It is placed outside the fire-box, and its nozzle is connected to the fire-box by a tube in the water-space. The orifice through which the pretroleum flows is adjustable. It can be readily swept out by the steam in case of the outlet becoming choked.

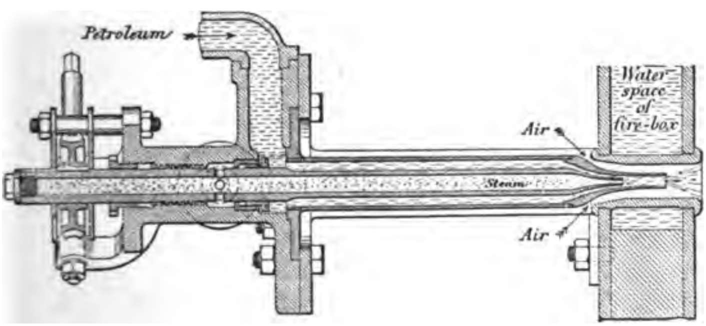
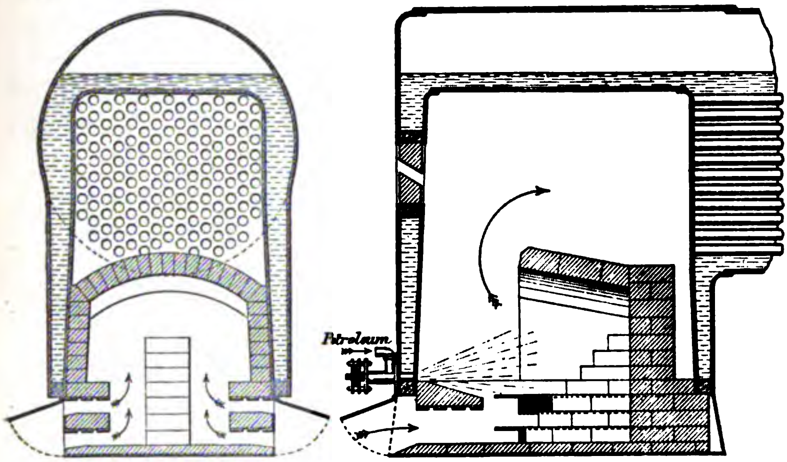


Fig. 3.—Urquhart's liquid-fuel-injector.

The regenerative or accumulative combustion-chamber used by Mr. Urquhart for six-wheeled goods and passenger locomotive engines is shown in Figs. 4 and 5. The bottom portion of the fire-box is lined with brick walls covered in by an arch, as clearly shown in the engravings.

Creosote-Oil is the heavy oil from the distillation of coal-tar. It varies in composition, and has a less evaporative value than petroleum. The composition of creosote is from 78 to 82·5 of carbon, 6 to 10 of hydrogen, and 7·5 to 16 of oxygen.



Figs. 4 & 5.—Urquhart's combustion-chamber for burning liquid-fuel.

Holden's System of Burning Liquid-Fuel.—In this system a liquid fuel is employed composed of 1 part creosote oil and 2 parts ordinary gas tar. The fire-grate is covered with a layer of chalk on which a thin fire of coal is placed to assist the combustion of the liquid-fuel. The liquid-fuel is injected into the fire-box by the injector shown in Fig. 6. The ordinary form of locomotive fire-box with a brick-arch is employed for burning the liquid-fuel, as shown in Fig. 7. One injector is placed on each side of the fire-box, below the level of the fire-door.

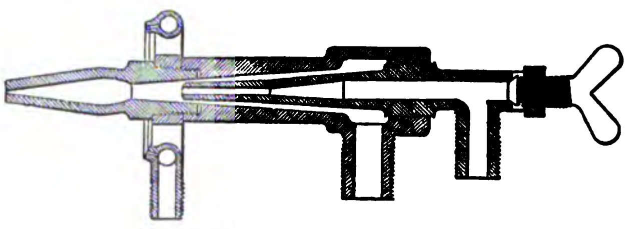


Fig. 6.—Holden's injector for liquid-fuel.

The evaporative efficiency of boilers employing this system of burning liquid-fuel is about one-fourth greater than that obtained when burning coal.

Steam used in Pulverising Fuel-Oil.—The system of effecting combustion by pulverising the oil with steam involves the expenditure of a considerable quantity of steam. From 8 to 13 per cent. of the total quantity of steam produced by the boiler is expended in atomising the oil, or converting it into spray.

Combustion.—The elements of combustion are carbon, hydrogen, nitrogen, and oxygen.

Carbon, the basis of most fuels, is a finely divided pulverulent mineral substance in its natural state. It is obtained from coal in the form of coke; from wood as charcoal; and from oil-lamps as lamp-black. Carbon is considered as the next most abundant body in nature to oxygen. The carbon of fuel produces the glowing heat of combustion, and the purer the carbon the more intense the heat.

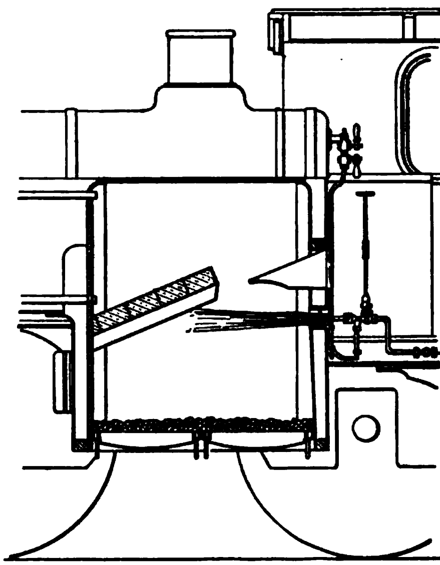


Fig. 7.—Holden's system of burning liquid-fuel.

Hydrogen, the source of all common flame, is a permanent but combustible gas. It is the lightest known body in nature, being sixteen times lighter than oxygen. When combined with sulphur it becomes explosive.

Nitrogen does not support combustion and does not burn, but passes through fire without chemical alteration. It is lighter than air, and has no taste or smell. It dilutes the products of combustion and lowers their temperature.

Oxygen is a permanent, colourless, transparent gas, without smell. It is 1·106 times heavier than air, and is the supporter of combustion, for which purpose it is supplied from the atmosphere.

Atmospheric Air is a mechanical mixture of oxygen and nitrogen in the proportion of 1 pound of oxygen to $3\frac{1}{2}$ pounds of nitrogen, or by volume 1 cubic foot of oxygen to 4 cubic feet of nitrogen. Nitrogen being a neutral gas is present simply as a diluent.

A cubic foot of air weighs ·08072 lb. at 32° Fahr., and ·076098 lb. at 62° Fahr. The volume of air is 12·386 cubic feet at 32° Fahr., and 13·14 cubic feet at 62° Fahr. under the pressure of one atmosphere, or 14·7 lbs. per square inch.

A Mechanical Mixture is one in which the substances have been brought together, but each retains its original qualities, such as heat and water in steam, or sand and water, all of which can be readily separated and restored again to their original state.

Chemical Combinations.—When two bodies unite to form a third body distinct from either of the combining bodies, this is called chemical union. All substances combine chemically in certain fixed proportions, both in volume and weight, which are termed their chemical equivalents, combining volumes, or atomic weights; some of which are given in the following table:—

TABLE 18.—ATOMIC WEIGHT OF THE PRINCIPAL ELEMENTARY CONSTITUENTS OF FUEL.

Element.	Symbol.	Combining Equivalents.		
		By Volume.	By Weight.	By Weight.
Oxygen . . .	O	1	8	100
Hydrogen . . .	H	1	1	12·5
Carbon . . .	C	1	6	75
Nitrogen . . .	N	...	14	175
Sulphur . . .	S	...	16	200

Chemical combinations are accompanied by elevation of temperature; the intensity of the heat evolved depending upon the rapidity of the combinations.

Products of Chemical Combinations.—The combination of one atom of oxygen with one atom of hydrogen produces water; and the combination of two atoms of oxygen with one atom of carbon produces carbonic acid. Taking the combining equivalents from the above table, one atom, or 12·5 lbs., of hydrogen combining with one atom, or 100 lbs., of oxygen, forms $12·5 + 100 = 112·5$ lbs. of water. One atom, or 75 lbs., of carbon, combining with two atoms, or $100 \times 2 = 200$ lbs., of oxygen, forms $75 + 200 = 275$ lbs. of carbonic acid.

The heat evolved by these combinations is 12,906 units for each pound of carbon, and 62,535 units for each pound of hydrogen, according to Dulong. One atom, or 75 lbs., of carbon combining with one atom, or 100 lbs., of oxygen form $75 + 100 = 175$ lbs. of carbonic oxide: the heat evolved per pound of carbon being 2,495 units, or only about one-fifth that of carbonic acid, according to Dulong. Two atoms of nitrogen, or $175 \times 2 = 350$ lbs., combining with one atom, or 100 lbs., of oxygen, form $350 + 100 = 450$ lbs. of atmospheric air.

Temperature of Ignition of Fuels.—Some fuels ignite at a temperature of 637° Fahr., and others at from 700° to 950° Fahr. At a temperature of 750° Fahr. some fuels become luminous and emit a dull-red heat. The igniting temperatures of good coal from different districts generally average as follows:—

	° Fahr.		° Fahr.
Gaseous coal ignites at . .	700	Durham coal ignites at . .	790
Scotch coal „ . . .	760	Derbyshire coal „ . . .	800
Newcastle coal „ . .	770	Yorkshire coal „ . .	810
Lancashire coal „ . . .	780	Welsh coal „ . . .	875
Staffordshire coal „ . .	785	Anthracite „ . .	925

The igniting temperature varies with the quality of the coal.

Combustion, or burning, is chemical combination, or the rapid union of any oxidisible substance with oxygen. Combustion cannot be efficiently maintained at a lower temperature than 800° Fahr. Combustion requires heat, air, space, and time.

Gases are combustible only in proportion to the degree of mixture which is effected between them and the oxygen of atmospheric air. There is considerable disproportion between the volumes and weights of gases. For instance, an atom of hydrogen is double the volume of an atom of carbon-vapour, but the latter is six times the weight of the former. An atom of hydrogen is double the volume of an atom of oxygen, but the latter is eight times the weight of the former. An atom of nitrogen is double the volume of an atom of oxygen, but in weight it is as 14 to 8.

Fuel-Gases.—The constituents of the gas disengaged from coal by heat are hydrogen and carbon; they unite and form two gases, viz., carburetted hydrogen and bicarburetted hydrogen, commonly called olefiant gas.

Carburetted hydrogen gas consists of two volumes of hydrogen and one of carbon-vapour, the volume of these three being condensed into that of a single atom of hydrogen, or into two-fifths of their previous volume. Bicarburetted hydrogen consists of two volumes of hydrogen and two of carbon-vapour, the volume of these four gases being equal to that of a single atom of hydrogen. The other ingredient of combustion, atmospheric air, is composed approximately of two atoms of nitrogen and one atom of oxygen, each of the former being double the volume of an atom of the latter, while their relative weights are as 14 to 8.

The constituents of coal-gas are condensed into two-fifths of their gross volume, but this is not the case with air, an atom of which is the same, both in volume and weight, as the sum of its constituents. The oxygen bears a proportion in volume to that of the nitrogen as 1 to 5, there being approximately only 20 per cent. of oxygen in atmospheric air and 80 per cent. of nitrogen.

In the Combustion of Coal the bituminous portion is convertible to the purposes of heat in the gaseous state only, and the carbonaceous portion is combustible only in the solid state; neither can be consumed while they remain united. The processes incident to the combustion of these two portions will now be considered separately in order to simplify the explanation.

Combustion of the Gaseous portion of Coal.—When heat is applied to coal, resulting in lighting the gas when duly mixed with air, the hydrogen separates from the carbon and unites with oxygen, the produce of which is water.

The saturating equivalent of an atom of hydrogen is one-half its volume of oxygen, the product being aqueous vapour; the relative weights of the combining volumes being one of hydrogen to eight of oxygen; the volume when combined being two-thirds of that of both taken together.

The carbon on meeting its equivalent of oxygen unites with it, forming carbonic acid gas, composed of one atom of carbon, by weight 6, and two atoms of oxygen, by weight 16, the latter being double that of the former in volume.

The quantity of oxygen required for the saturation of the two constituents of coal-gas to effect perfect combustion, is determined by their chemical constituents. The quantity of atmospheric air required to supply the oxygen is found from the proportion which oxygen bears in volume to that of the air, five volumes of the latter being required to produce one volume of the former, and as two volumes of oxygen are required for each volume of coal-gas, ten volumes of air are required to produce these two volumes.

Combustion of the Carbonaceous portion of Coal, or that portion remaining in a solid form on the fire-grate after the gaseous matter has been evolved. Carbon unites with oxygen in two proportions, by which two distinct bodies are formed: first, carbonic acid; second, carbonic oxide. Carbonic acid is a compound of one atom of carbon with two atoms of oxygen; carbonic oxide is composed of the same quantity of carbon with only half the above quantity of oxygen, but it has the same volume as carbonic acid.

The proportions of these compounds are usually expressed as follows:—

	Symbol.	Parts by weight. Carbon.	Oxygen.
Carbonic oxide,	CO	12	16
Carbonic acid,	CO_2	12	32

The direct effect of the union of carbon and oxygen is the formation of carbonic acid. If, however, one of its portions of oxygen be abstracted, the remaining portions would then be those of carbonic oxide. If a second portion of carbon be added to carbonic acid the result would be the same, or carbon and oxygen combined in equal proportions as in carbonic oxide. By the addition then of a second portion of carbon two volumes of carbonic oxide will be formed, which, if they cannot find the oxygen required to complete their saturating equivalents, will pass away only half consumed, a circumstance which is constantly taking place in all furnaces where the air has to pass through a body of incandescent carbonaceous fuel.

If the carbonaceous constituent of coal while at a high temperature encounters carbonic acid, this latter, taking up an additional portion of carbon, is converted into carbonic oxide and again becomes a gaseous and invisible combustible.

The most prevailing operation of the furnace, however, and by which the largest quantity of carbon is lost in the shape of carbonic oxide is the following:—The air, on entering from the ash-pit, gives out its oxygen to the glowing carbon on the fire-grate, and generates much heat in the formation of carbonic acid. This acid, necessarily at a very high temperature, passing upwards through the body of incandescent solid matter, takes up an additional portion of the carbon and becomes carbonic oxide. Thus, by the conversion of one volume of acid into two volumes of oxide, heat is absorbed and the portion of carbon taken up during conversion is lost.

Carbonic oxide gas, by reason of its already possessing one half its equivalent of oxygen, inflames at a lower temperature than ordinary coal-gas; the consequence of which is, that the latter on passing into the flues, is often cooled down below the temperature of ignition; while the former is sufficiently heated even after having reached the top of the chimney and is there ignited on meeting the air. This is the cause of the flame frequently seen at the tops of chimneys and funnels.

If the carbon either of the gas or of the solid mass of coal on the fire-grate passes away in any other form than carbonic acid, the result is a commensurate loss of heating effect.

Air to support Combustion.—It will be seen from the foregoing explanation, that the air required to support combustion must be delivered in two distinct ways, viz., above the fire to effect the combustion of the bituminous or gaseous portion of the coal, and below the fire-grate to effect the combustion of the solid carbonaceous portion resting on the fire-grate.

It was found in one case that by altering the furnace to give a better supply of air above the fire, the evaporative power of the fuel was increased 20 per cent.

With a supply of unvitiated atmospheric air containing 20 per cent. of oxygen, 10 cubic feet of air are required to supply 2 cubic feet of oxygen to effect the combustion of 1 cubic foot of coal-gas. It is necessary to provide the air in such a manner as to ensure a pure supply without deficiency of oxygen.

Many experiments have been made to determine the saving to be gained by heating the air before it enters the furnace, the results of which show that a saving in fuel of from 4 to 8 per cent. may be obtained by heating the air to a temperature of from 150° to 300° Fahr.

The Weight of Atmospheric Air required to support combustion in the furnace of a steam-boiler varies with the amount of the constituent hydrogen and carbon of the fuel. It may be calculated from Table 18 as follows.—Hydrogen in burning produces water composed of $12{\cdot}5 \div (12{\cdot}5 + 100) = {\cdot}112$ lb. of hydrogen, and $100 \div (12{\cdot}5 + 100) = {\cdot}89$ lb. of oxygen per pound of water: and one pound of hydrogen requires ${\cdot}89 \div {\cdot}112 = 8$ lbs. of oxygen.

The source of oxygen, as the supporter of combustion, is atmospheric air, one pound of which only contains $100 \div (175 \times 2 \text{ atoms} + 100) = {\cdot}223$ lb. of oxygen per pound of air. Therefore, to obtain 8 lbs. of oxygen from the air to support the combustion of the hydrogen, 8 lbs. $\div$ ·223 lb. $= 35{\cdot}78$ lbs., or say 36 lbs. of atmospheric air are required.

Carbon in burning produces carbonic acid, composed of $75 \div (75 + 200) = {\cdot}273$ lb. of carbon, and $200 \div (75 + 200) = {\cdot}728$ lb. of oxygen per pound of carbonic acid, and one pound of carbon requires ·728 ÷ ·273 = 2·667 lbs. of oxygen. Therefore, the quantity of air required to support the combustion of the carbon is, 2·667 ÷ ·223 lb. of oxygen per pound of air, = 11·96 lbs., or say 12 lbs. of atmospheric air.

The weight of a cubic foot of air at the standard temperature, 62° Fahr. is ·076098 lb.; hence one pound of hydrogen requires 35·87 ÷ ·076098 = 471·44, or say 472, cubic feet of air; and one pound of carbon requires 11·96 ÷ 076098 = 157·16, or say 158, cubic feet of air for combustion.

This is the minimum theoretical quantity chemically consumed by each combustible, and the weight of air in lbs., W, theoretically required for the combustion of any kind of fuel, may be found according to the above data by the following formula:—

$$W = (12 \times \%\ \text{carbon}) + \left(36 \times \left[\%\ \text{hydrogen} - \frac{\text{oxygen}}{8}\right]\right).$$

Example: What quantity of air is theoretically required for the combustion of coal containing ·8 carbon, ·05 hydrogen, and ·08 oxygen?

Then the deduction to be made from the constituent hydrogen is = ·08 ÷ 8 = ·01, and (12 × ·8 carbon) + 36 (·05 — ·01) = 9·6 + 1·44 = 11·04 lbs. the weight of air theoretically required for the combustion of one pound of this coal.

The volume of air in cubic feet, V, theoretically required for the combustion of any kind of fuel may be found by the following formula:—

$$V = (158 \times \%\ \text{carbon}) + \left(472 \times \left[\%\ \text{hydrogen} - \frac{\text{oxygen}}{8}\right]\right).$$

Applying this formula to the coal described in the previous example,— Then (158 × ·8 carbon) + 472 (·05 — ·01) = 126·4 + 18·88 = 145·28 cubic feet, the volume of air theoretically required for the combustion of one pound of this coal.

The weight of air required in practice to support combustion is much larger than that theoretically required, in order to effect complete combustion, and prevent the formation of carbonic oxide instead of carbonic acid. Complete combustion can be obtained with a supply of air not greater than fifty per cent. in excess of the quantity necessary for theoretical combustion with natural draught, but it is usual to provide double the quantity of air theoretically required.

On this basis, calculating, for example, the volume of air for coal composed of ·83 carbon, and ·04 available hydrogen, then (·83 × 158 cubic feet × 2) + (·04 × 472 cubic feet × 2) = 300 cubic feet of air at 62° Fahr. are required for each pound of coal consumed with natural draught.

The weight of air per pound of fuel necessary in practice for different kinds of fuel, to secure sufficient dilution of the gases to effect their combustion, both with natural or ordinary chimney draught and with forced draught, is given in the following Table.

TABLE 19.—QUANTITY OF ATMOSPHERIC AIR REQUIRED IN PRACTICE FOR THE COMBUSTION OF ONE POUND OF DIFFERENT FUELS.

Description of Fuel.	Air required for combustion with natural draught.		Air required for combustion with forced draught.	
	Weight in lbs.	Cubic feet at 62° Fahr.	Weight in lbs.	Cubic feet at 62° Fahr.
Petroleum	36	474	30	394
Creosote, or Tar-Refuse . . .	32	434	24	316
Coal-Gas	29	382	22	290
Coal, average	24	316	18	237
Coal, best, very carefully stoked .	17	224	16	210
Charcoal	23	303	17·25	227
Patent Fuels, average . . .	22	290	16·5	217
Coke	21	275	15·75	207
Coal-Gas	16	211	—	—
Peat, well-dried	16	211	12	158
Peat, moderately dry	13	170	9·75	128
Sawdust, dry	13	170	9·75	128
Wood, well-dried.	12	158	9	119
Straw, dry	11	145	8·25	109
Wood, moderately dry. . . .	10	132	7·5	99

When the supply of air is greater than is required to ensure complete combustion, it lowers the temperature of the furnace and carries heat to waste among the products of combustion.

Heat of combustion, or the calorific power of fuel, is expressed by the number of thermal units developed by one pound of the combustible in burning. The quantity of heat evolved by each of the elements of combustion has been accurately determined by experiment, and forms a correct basis for calculations of the calorific value of fuels.

The calorific power of numerous combustibles is given in Table 20, which also contains the equivalent quantities of water which would be evaporated from and at 212° Fahr. per pound of the combustible. These quantities of water are obtained by dividing the units of heat evolved by 966, which is the number of units of heat absorbed by water supplied at 212° Fahr., and evaporated at the same temperature. It will be seen that the heat of hydrogen, according to Favre and Silbermann, is $62032 \div 14544 = 4·265$ times as great as that of carbon. Therefore hydrogen $\times$ 4·265 will represent an equivalent amount of carbon.

Oxygen exists in fuel in combination with the hydrogen in the form of water, and abstracts its combining equivalent of hydrogen from the constituents of the fuel; therefore a deduction equal to one-eighth of the constituent oxygen must be made from the constituent hydrogen in calculating the calorific power of fuel.

Taking the carbon, hydrogen, and oxygen at their percentage of the fuel, the heat of combustion may be found by the following formula.

Heat of combustion in thermal units, U :—

$$U = 14544 \times \left\{ \% \text{ carbon} + 4{\cdot}265 \times \left(\% \text{ hydrogen} - \% \frac{\text{oxygen}}{8} \right) \right\}.$$

Water evaporated in pounds from and at 212° per pound of fuel, W :—

$$W = \frac{14544}{966} \times \left\{ \% \text{ carbon} + 4{\cdot}265 \times \left(\% \text{ hydrogen} - \% \frac{\text{oxygen}}{8} \right) \right\}.$$

Example. Required the heat of combustion, and the equivalent quantity of water evaporated from and at 212° Fahr., of coal composed of ·82 carbon; ·054 hydrogen; ·072 oxygen?

Then ·072 ÷ 8 = ·009, the deduction to be made from the constituent hydrogen, or that portion which forms steam with the constituent oxygen. And ·054 — ·009 = ·045 × 4·265 = ·191 + ·82 = 1·011 × 14544 = 14704 units, the heat of combustion.

Again, ·054 — ·009 = ·045 × 4·265 = 191 + ·82 = 1·011 × (14544 ÷ 966) = 15·22 lbs. of water evaporated per pound of fuel from and at 212° Fahr., the maximum theoretical evaporative power of this fuel.

When the water is supplied at 62° Fahr. divide by 1116 instead of 966.

If Dulong's value for the hydrogen and Peclet's value for the carbon be taken, then, the heat of the hydrogen is 62535 ÷ 14500 = 4·31 times as great as that of carbon. Then 14500 ÷ 966 = 15 lbs. of water, the equivalent evaporation, and on this basis the heat of combustion is obtained by the following formula :—

Calorific power of fuel in thermal units, U :—

$$U = 14500 \times \left\{ \% \text{ carbon} + 4{\cdot}31 \times \left(\% \text{ hydrogen} - \% \frac{\text{oxygen}}{8} \right) \right\}.$$

Evaporative power of fuel in pounds of water evaporated per pound of fuel from and at 212° Fahr. W :—

$$W = 15 \times \left\{ \% \text{ carbon} + 4{\cdot}31 \times \left(\% \text{ hydrogen} - \% \frac{\text{oxygen}}{8} \right) \right\}.$$

Applying these rules to the previous example, the result is = ·045 × 4·31 = ·194 hydrogen + ·82 carbon = 1·014 × 14500 = 14703 units, the heat of combustion, or maximum calorific power of the fuel; and 1·014 × 15 = 15·21 lbs. of water, or about the same result as that obtained by the previous formula.

Although the above formulæ are generally used for calculating the calorific power of fuel, they are not strictly correct because no deduction has been made from the hydrogen for the heat absorbed by the internal work done in vaporisation. The total heat developed by hydrogen in combination with oxygen, according to Favre and Silbermann, 62032; this includes the heat absorbed or that becomes latent in vaporising the water formed. One lb. of hydrogen burns to 9 lbs. of water in a gaseous state, which, if condensed at 212° Fahr., would yield 996 × 9 lbs. = 8694 units of heat, so that the available or effective heat, in the gaseous state, is = 62032 − 8694 = 53338 units.

In ordinary calculations the sulphur of the fuel is usually neglected, but in exact calculations it is necessary to include the heat developed by the sulphur, which may be found by multiplying the percentage of sulphur by 4032, its calorific power from Table 20. Take, for example, coal containing ·78 carbon, ·05 hydrogen, ·09 oxygen, and ·02 sulphur. Then the oxygen in the coal will combine with ·09 oxygen ÷ 8 = ·0112 hydrogen, leaving ·05 − ·0112 = ·0388 hydrogen in excess, to develop heat.

The calorific power of 1 lb. of this coal is:—

Carbon	·78 × 14544 =	11345 units.
Hydrogen, uncombined with oxygen .	·0388 × 53338 =	2070 „
Sulphur	·02 × 4032 =	81 „
		13496 „

of heat, and it will evaporate 13496 ÷ 966 = 13·97 lbs. of water from and at 212° Fahr.

Taking Favre and Silbermann's value for the carbon, the heat of hydrogen is 53338 ÷ 14544 ≐ 3·668 times as great as that of carbon; the heat of sulphur is 4032 ÷ 14544 = ·28 that of carbon, and the formulæ become as follows:—

$$\text{Heat of combustion or calorific power of fuel} = 14544 \times \left[\%\ \text{carbon} + 3{\cdot}668 \times \left(\%\ \text{hydrogen} - \%\frac{\text{oxygen}}{8}\right) + \%\ \text{sulphur} \times {\cdot}28\right].$$

$$\text{Evaporative power of fuel from and at } 212^\circ \text{ Fahr.} = 15{\cdot}06 \times \left[\%\ \text{carbon} + 3{\cdot}668 \times \left(\%\ \text{hydrogen} - \%\frac{\text{oxygen}}{8}\right) + \%\ \text{sulphur} \times {\cdot}28\right].$$

Applying these rules to the previous examples, the result is = ·0388 hydrogen × 3·668 = ·1424 + ·78 carbon + (·02 × ·28) = ·0056 sulphur = ·928 × 14544 = 13496 units of heat in 1 lb. of the coal, and its evaporative power is ·928 × 15·06 = 13·97 lbs. of water from and at 212° Fahr.

Calorimeter for determining the Calorific Power of Coal.— The simplest apparatus for determining the heating-power of coal is the small brass calorimeter shown in Fig. 8. The tube A is filled with a mixture of coal, potassium chlorate, and potassium nitrate, which is lighted by means of a fuse. The cylinder B is placed over the tube A, and is held in position by spring-clips; near the bottom of this cylinder are two rows of holes of one thirty-second of an inch in diameter, and at the top is a small tube and air-cock. The outer cylinder contains a known weight of water, and is fitted with sockets to hold a thermometer.

In testing a sample of coal, 2 grammes of pulverised coal are thoroughly mixed with 7·5 grammes of potassium chlorate, and 2·5 grammes of potassium nitrate. This mixture is rammed in the tube A, and a fuse, consisting of a small piece of filter-paper coated with a mixture of the chlorate and nitrate of potassium, is inserted. The tube A is then placed in its socket, the fuse lighted, the cylinder B is slipped into its place, the air-cock being closed, and the whole is placed under water in the outer cylinder. As combustion takes place in the tube A, the gases find their way through the small holes in the cylinder B, and thence up through the water, giving up most of their heat to it. When the combustion is completed, which is indicated by the cessation of bubbles of gas, the air-cock is opened, and water flows into the cylinder B through the small holes therein, and cools the firing-tube A. Then, by moving the cylinder B up and down, the water is thoroughly agitated and its temperature rendered uniform.

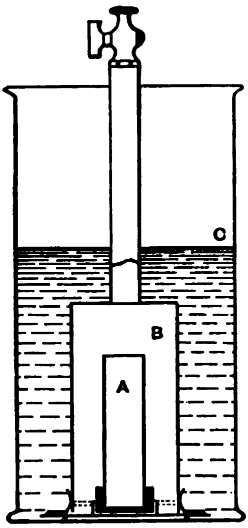


Fig. 8.—Calorimeter for testing the heating power of coal.

The heating-power of the coal is determined from the weight of coal and water, and the initial and final temperature of the water; allowance being made for the heat taken up and radiated by the metal of the calorimeter, and for the heat of combustion of the fuse.

The Calorific Power and Theoretical Evaporative Power of Fuels are given in the following Table :—

TABLE 20.—CALORIFIC POWER, AND THEORETICAL EVAPORATIVE POWER OF COMBUSTIBLES.

Description of Combustible.	Calorific Power, or Units of Heat developed per Pound of Fuel.	Evaporative Power in lbs. of Water evaporated to Steam, from and at 212° Fahr.
Hydrogen burning to water . . . . (Dulong)	62535	64·74
Ditto, ditto (Favre and Silbermann)	62032	64·21
Marsh-Gas . . . . ,, ,,	23513	24·34
Petroleum, light	21560	22·32
Ditto, heavy	19400	20·08
Petroleum . . . . . . Medium Quality	18000	18·63
Paraffin	21460	22·21
Olefiant Gas . . . . . (Favre and Silbermann)	21343	22·09

TABLE 20 *continued.*—CALORIFIC POWER AND THEORETICAL EVAPORATIVE POWER OF COMBUSTIBLES.

Description of Combustible.	Calorific Power, or Units of Heat developed per Pound of Fuel.	Evaporative Power in lbs. of Water evaporated to Steam, from and at 212° Fahr.
Coal-Gas . . . . Average	21000	21·74
Olive-Oil . . . . (Lavosier)	20153	20·86
Naphtha-Refuse	19200	19·87
Olive-Oil . . . . (Dulong)	17752	18·37
Wax	19950	20·65
Sperm	19680	20·37
Turpentine . . . . (Favre and Silbermann)	19534	20·22
Ditto . . . . (Dulong)	19505	20·19
Solar-Oil	18990	19·65
Colza-Oil	18840	19·50
Linseed-Oil	18738	19·39
Neatsfoot-Oil	18546	19·19
Whale-Oil	18315	18·95
Tallow	18120	18·75
Stearine	17950	18·60
Shale-oil	17871	18·50
Creosote, or Tar-Refuse	17388	18·00
Creosote-Oil, from Tar-Refuse, averages . . .	14490	15·00
Sulphuric Ether	16974	17·56
Asphalte	16596	17·18
Welsh Coal, Best . . Average of 24 Samples	15865	16·42
Ditto, Aberdare . . „ 12 „	15213	15·74
Ditto, Penrikyber	15000	15·53
Ditto, Powell-Duffryn's	14945	15·47
Ditto, Good Ordinary . Average of 24 Samples	14826	15·30
Welsh Steam-Coal of Medium Quality, Averages .	13428	13·90
Newcastle Steam-Coal . Average of 18 Samples	14168	14·66
Ditto, of Good Quality . . Averages	14000	14·50
Lancashire Steam-Coal . Average of 24 Samples	13963	14·45
Derbyshire Steam-Coal . . „ 18 „	13876	14·36
Yorkshire Steam-Coal . „ 12 „	13762	14·24
Steam-Coal, English, Average of a large number of Samples	13524	14·00
Slack, Good Clean Rough, English . . Averages	13300	13·76
Scotch Steam-Coal . . Average of 12 Samples	13493	13·96
Ditto, Average of a large number of Samples	12800	13·25
Half Coke and half Newcastle Small Coal . . .	14302	13·77
Half Welsh Coal and half Newcastle Coal . . .	13041	13·50
Newcastle Steam-Coal, small, good quality . Average	12800	13·25
Coke, good quality . . . . . Averages	12558	13·00
Gas-Coke, good . Average of a number of Samples	10626	11·00
Breeze from Gas-Coke, good . . . . Averages	6762	7·00
Carbon burning to Carbonic Acid (Favre & Silbermann)	14544	15·06
Ditto, ditto . . . (Peclet)	14500	15·00
Ditto, ditto . . (Despretz)	14040	14·53

TABLE 20 *continued*.—CALORIFIC POWER, AND THEORETICAL EVAPORATIVE POWER OF COMBUSTIBLES.

Description of Combustible.	Calorific Power, or Units of Heat developed per Pound of Fuel.	Evaporative Power in lbs. of Water evaporated to Steam, from and at 212° Fahr.
Carbon burning to Carbonic Acid . . . (Dulong)	12906	13·36
Phosphorus . . . . (Laplace)	13500	13·97
Naphtha	13208	13·66
Alcohol	12339	12·77
Charcoal from Wood	11592	12·00
One Pound of Carbon in the form of Carbonic Oxide burning to Carbonic Acid	10411	10·77
Animal-Fat	8694	9·00
Slack, good, small . . . . Averages	7844	8·12
Lignite	9660	10·00
Peat, well-dried . . . . Kiln-dried	9500	9·83
„ moderately dry . . . . Air-dried	7245	7·50
English Oak . . . . Kiln-dried	7516	7·78
Ash, Beech, and Thorn . . . . . . „	7245	7·50
Red Oak, Hard Maple, and Walnut . „	6888	7·13
Apple Tree, Pear Tree, Cherry Tree, Plum Tree „	6610	6·84
Birch, Elm, Plane Tree, and Hazel . . . „	6536	6·76
Chestnut and Yellow Pine . . . . „	6482	6·71
Pitch-Pine, Alder, Aspen, and Poplar . „	6436	6·66
Willow, White-Pine, or Deal . . . . „	6400	6·62
Wood, air-dried, containing 20 °/₀ of water, averages .	4830	5·00
Cork . . . . Kiln-dried	6347	6·57
Tan-Refuse, or Oak-Bark . . . . . Dry	6279	6·50
„ „ , moderately dry, averages .	4830	5·00
„ „ , in a damp state	3024	3·13
Sawdust from Oak or other hard-wood . . Dry	5912	6·12
Ditto, Pine-Wood, or other soft woods „	5217	5·40
Ditto, ditto, and ditto, , in a moderately dry state . . . . . Averages	3961	4·10
Brush-Wood . . . . Dry	5000	5·17
Cotton-Stalks . . . . Dry	4916	5·08
Carbonic Oxide burning to Carbonic Acid (Dulong)	4478	4·63
Ditto, ditto (Favre and Silbermann)	4325	4·47
Carbon burning to Carbonic Oxide „ „	4453	4·61
Ditto, ditto . . (Dulong)	2495	2·58
Sulphur . . . . „	4682	4·84
Ditto . . . . . (Favre and Silbermann)	4032	4·17
Flax-Refuse . . . . Averages	4106	4·25
Ramie-Refuse	3980	4·12
Straw . . . . Dry	3864	4·00
Ditto . in a moderately dry state, Averages	2898	3·00
Wood-Chips and Sawdust mixed . Moderately dry, Averages	3671	3·80
Wood-Chips and Green Twigs, in a damp state, or containing 50 °/₀ of moisture . . Average	1932	2·00

Efficiency of Fuel in the Furnace of a Steam-boiler.—The full value of the heating-power of the fuels given in the foregoing Table is not realised in practice, in the evaporation of water to steam, owing to losses from various causes, but principally by conduction, radiation, and imperfect combustion, and a portion of the heat is necessarily expended in creating a draught in the chimney.

The efficiency of the firing, combustion in the furnace, and evaporation by the boiler, may be found by the *Rule*:—

$$\text{Efficiency of firing, furnace, and boiler} = \frac{\text{Actual quantity of water evaporated per pound of fuel}}{\text{Theoretical evaporative power of the fuel.}}$$

Example: Required the efficiency of the firing, furnance, and boiler, in a test where 12·18 lbs. of water were evaporated per lb. of coal from and at 212° Fahr., with coal having a theoretical evaporative power of 14 lbs. of water from and at 212° Fahr. per lb. of coal?

Then $\frac{12{\cdot}18 \text{ lbs.}}{14 \text{ lbs.}} = {\cdot}87$ per cent., showing that the actual evaporation is 13 per cent. less than the theoretical evaporation.

Firing Steam-boilers.—The fire should be maintained at as great a heat as possible. A high and uniform temperature of the furnace is essential to economical combustion. It effects rapid diffusion and combination of the gases, secures their combustion, and conduces to the prevention of the formation of carbonic oxide and of the discharge of smoke. When the temperature of the furnace is low, a considerable portion of the gases escape to the chimney unconsumed. The rate of combustion should be moderate, so as to afford time for the fuel to be effectively burned.

Thickness of the Fire.—The thickness of fire required for economical combustion varies with the size of the pieces, quality, and description of fuel used; and also with the strength of the draught, the stronger the draught the thicker may the fire be. A fire of anthracite may be from 4 to 6 inches thick, and of bituminous coal from 7 to 15 inches thick, according to the nature of the coal, and the available draught. A thick fire is necessary for the production of a high temperature; the more freely the coals burn the thicker may the fire be. Coals which develop little flame, and small coals of all kinds, burn best in a thin fire.

A thin fire facilitates combustion by offering the least resistance to the passage of air through the bed of fuel, but it is difficult to maintain a regular thickness of fire and the fire-grate evenly covered with fuel with a thin fire, because the bed of fuel burns into holes. This causes waste of fuel, because the rush of air through the uncovered portions of the grate reduces the temperature of the furnace and flues, and results in a large volume of air passing through the furnace without having its oxygen consumed.

Hence, a thick fire is generally more economical than a thin fire, but the fire should be replenished with thin layers or moderate charges of coal.

Improper Firing.—The introduction of heavy charges of coal at long intervals is objectionable, as it has a severe damping effect upon the fire and lowers the temperature of the surface excessively, and is not conducive to economical combustion. It is a common practice to throw a large quantity of coal on the fire at one firing. That which is on the top is rapidly coked by the heat underneath, and the gases evolved escape unconsumed. An equal quantity of coal placed in the furnace in three or four firings, at intervals of from three to six minutes, would, in many cases, evaporate twice as much water as when it is all thrown on the fire at one firing, and the quantity of smoke produced would be considerably less with this method of firing. Irregular and reckless firing results in waste of fuel.

Spread Firing.—The firing should be regular, and consist of small or light charges of coal, distributed evenly over the surface of the fire, and delivered at short intervals of, say, from four to ten minutes, according to the character of the coal used, and the demand upon the boiler for steam. The quantity of coal delivered at each firing should be in proportion to the work being performed by the boiler. When a furnace is fired in this way, the gases become rapidly disengaged, and the greatest heating effect is obtained from the fuel with the least reduction of temperature of the furnace, due to the damping effect on the fire of fresh charges of fuel.

The only objection to this method of firing is that, the number of times the furnace-door is opened permits the admission of an excessive quantity of air. This may be obviated by employing a damper of light construction connected to the furnace-door in such a manner, that on opening the door the damper closes sufficiently to prevent the admission of much air, and the consequent reduction of the temperature of the furnace and flues of the boiler.

This method of firing permits the use of a small fire-grate, and enables the greatest quantity of steam to be produced with the combustion of the smallest quantity of fuel. The fire should be saucer-shaped, that is, thicker at the sides than at the middle.

Side-Firing is frequently adopted for moderate charges of coal. The coal, instead of being spread over the fire, is thrown on each side of the fire alternately, leaving one side always bright to effect the combustion of the gases from the coal freshly charged on the opposite side of the fire, by which means the temperature of the furnace is lowered as little as possible, and economical combustion may be effected with the production of little smoke. When there are two or more furnaces they should be fired alternately.

Coking-Firing is most effective for coal of a very smoky nature. It is effected by providing a broad dead-plate at the entrance of the furnace, on which each charge of coal is placed and allowed to remain during the intervals of firing, in order that the volatile ingredients may be expelled by the heat of the furnace and the coal become partly converted into coke before it is put on the fire, and thus prevent as much as possible the emission of smoke from the chimney.

Smokeless-Firing.—The fire should not be roused with a rake. If the coal cakes together, a slicer should be run in on the top of the bars and the burning mass gently broken up. The fire-bars should be maintained covered all over and not allowed to be bare at the back. With a good draught and careful hand-firing by any of the previously described methods, and the admission of air by opening the grid of the fire-door for about a minute after firing, no smoke need be made. The admission of air in small streams above the fire is of great advantage in affording efficient combustion of the fuel-gases, and lessening the amount of smoke produced. In burning coal of a very smoky nature, a little air should be admitted at the fire-bridge as well as at the front of the furnace.

In Stoking and Cleaning the Fires, the fire-door should be open as short a time as possible, in order to prevent the inrush of cold air, causing loss of heat, and variation in the supply of steam. In cleaning the fire it should not be allowed to almost die out and the bars to become bare. As much fire should be pushed against the bridge as possible, and the cleaning then effected.

Clean Fire-Bars are essential to economical combustion. The spaces between the bars should be maintained free from clinkers and ashes. When the air-spaces are clogged sufficient air cannot pass through the bed of fuel to effect complete combustion, resulting in waste of fuel and reduction of the steam-producing capacity of the boiler.

The Draught should be carefully regulated to the nature of the fuel, so as to obtain the greatest heat from the fire without the admission of more air than is necessary for efficient combustion and the prevention of smoke. A roaring draught, which forces the air through the fire-grate like a hurricane, lowers the temperature of the furnace, prevents economical combustion, and carries much heat to waste to the chimney.

Mechanical Stokers.—To burn small bituminous coal economically, with the production of the least quantity of smoke, it is necessary to deliver it on the fire in small equal charges at regular intervals. This is best effected by mechanical firing, hand-firing being imperfect owing to its irregularity.

With a well arranged self-feeding furnace the supply of coal may be readily adjusted to the required rate of combustion, and the coal may be delivered continuously in regular charges evenly distributed over the fire.

The employment of a mechanical stoker dispenses with the frequent opening of the furnace-door necessary in hand-firing, and the consequent inrush of cold air to the flues is prevented.

With a proper supply of air to the fuel, mechanical firing is theoretically conducive to an even furnace-temperature, economical combustion, and a uniform supply of steam. It is, however, difficult in many cases in practice to obtain conditions favourable to economical combustion with mechanical firing, and it may be less economical in the quantity of fuel used than skilful hand-firing. But mechanical stokers generally permit the employment of cheaper kinds of coal than it is expedient to use for hand-firing.

Vicars' Mechanical Stoker is shown in Fig. 9. The hopper A is filled with small coal, which falls into the boxes B. The boxes are fitted with self-acting plungers, which push the coal alternately into the furnace and on to the dead-plate, whence it is delivered to the fire-grate. The fire-bars travel, and carry the fire very slowly for a short distance along the furnace-tube. Any unconsumed fuel which reaches the end of the grate-bars, with the clinkers and ashes, are discharged over the ends of the grate-bars into the bottom of the furnace-tube.

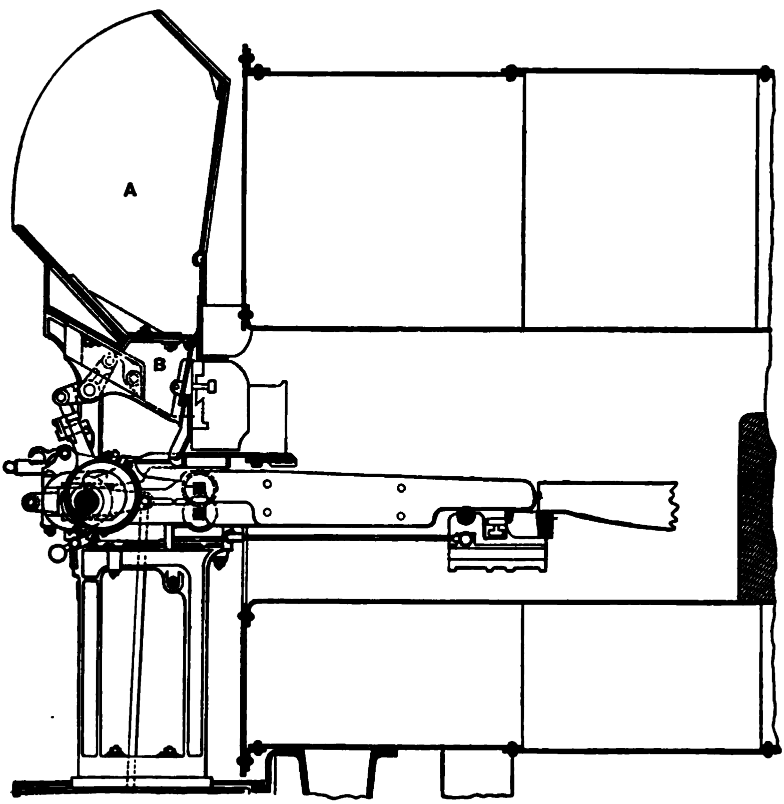


Fig. 9.—Vicars' mechanical stoker.

The following are the results of the test of a Lancashire boiler of 7 feet 5 inches diameter and 30 feet long, with furnace-tubes 2 feet 9 inches diameter, fitted with this mechanical stoker, compared with the results of a test of the same boiler with ordinary hand-firing.

System of stoking	Vicars' stoker .	Hand firing.
Duration of trial	10 hours .	10 hours.
Designation of coal	Bituminous slack .	Welsh small.

Area of fire-grate, total	22 feet	33 feet.
Coal consumed " "	2 tons 13 cwt.	3 tons 4 cwt.
Coal consumed per hour	5·3 cwt.	6·4 cwt.
Water evaporated, total	5,330 gallons	5,020 gallons.
" per hour	533 gallons	502 gallons.
" per pound of fuel	8·98 pounds.	7 pounds.
Price of fuel	13*s*. per ton	15*s*. per ton.
Cost of fuel per 1,000 gallons evaporated	6*s*. 5·5*d*.	9*s*. 4·75*d*.

Henderson's Mechanical Stoker is shown in Fig. 10. The coal is placed in a hopper and is broken by revolving crushers. The pulverised coal drops on to horizontal fans. The fans are actuated by frictional pulleys attached to the driving shaft. The furnace is fitted with a moving-grate. One half of the number of grate-bars move up and down vertically for the purpose of breaking the clinker and keeping the fire open, while the remainder of the bars travel backwards and forwards horizontally, and carry the clinker and ashes to the back of the furnace, where it is discharged over the back-end of the fire-bars into the bottom of the furnace-tube.

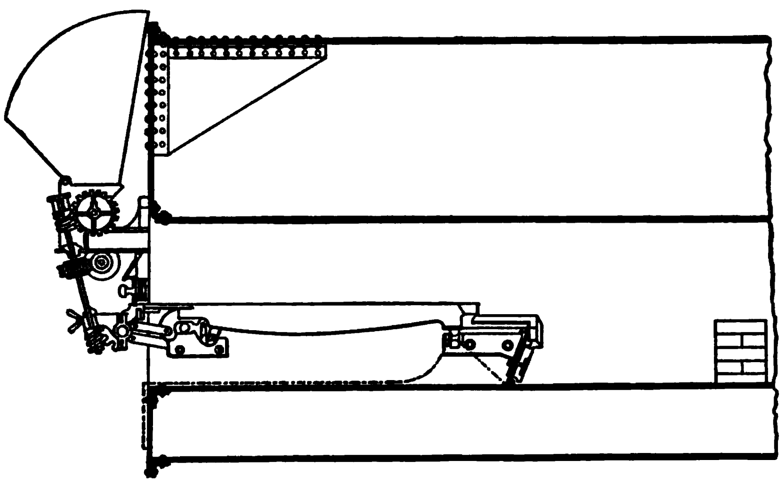


Fig. 10.—Henderson's mechanical stoker.

The following are the results of a test of a Lancashire boiler of 7 feet 6 inches diameter and 28 feet long, fitted with this mechanical stoker, compared with the results of the same boiler with ordinary hand-firing. Rough small coal was used in both tests.

System of stoking	Hand-firing	Henderson's stoker.
Duration of trial	48 hours	48 hours.
Coal consumed in lbs.	22582	23263.
Coal consumed per square foot of fire-grate surface per hour	13·24	19·19.

Proportion of ashes	13·70	8·70.
Water evaporated total in gallons	14733	19468.
Water evaporated per hour in gallons	307	405.
Water evaporated per lb. of fuel in lbs.	6·85.	9·13.
Temperature of feed-water in degrees Fahr.	165	134.
Value of coal per ton	3*s*. 6*d*.	3*s*. 6*d*.
Cost of evaporating 1000 gallons of water from 62° Fahr.	2*s*. 5·6*d*.	1*s*. 10·8*d*.

The results of these tests show an increase of duty of 33·3 per cent. by mechanical stoking, and a diminution of 23 per cent. in the cost of evaporation.

The Rate of Combustion in the furnace of a steam boiler is expressed by the number of pounds of fuel burnt on each square foot of fire-grate surface per hour. It varies with the intensity of the draught and the combustibility of the fuel, and ranges from 5 to 15 lbs. for anthracite coals and from 4 to 26 lbs. for bituminous coals in different types of boilers with natural draught. In locomotives with steam-blast in the chimney, the rate of combustion is on an average from 45 to 85 lbs. in this country, and in others it is from 60 to 180 lbs. In boilers having combustion with forced draught in the furnace, the rate of combustion is from 30 to 160 lbs. of coal per square foot of fire-grate surface per hour.

With high rates of combustion it is necessary to frequently clean and rouse the fire, in order to prevent the furnace becoming choked with clinker and ash, and to provide free access of air to the fuel.

For each type of boiler a certain rate of combustion produces a maximum economical effect, and higher rates only result in increasing the quantity of the products of combustion without increasing the effect. In locomotives, the greatest effect is generally obtained from a combustion of about 65 lbs., and the limit to economical combustion is 85 lbs. of coal per square foot of fire-grate per hour.

Heat Utilized in the Production of Steam in a Steam-Boiler.—The heat actually expended in the evaporation of water to steam, or the heat actually absorbed by the water-heating surfaces per pound of coal fired, is very much less than that theoretically developed by the combustion of the coal. The difference between the total heat evolved and that available for work is due principally to the following sources of loss, viz.:—

Initial-heating of the fuel and air for combustion.

Heating air in excess of that required for combustion.

Heat lost in displacing the atmosphere by the products of combustion.

Heat lost by radiation; and by cooling of the boiler by contact with cold air.

Heat lost in unconsumed fuel and ashes.

Some coals radiate more heat in the furnace than others. Many coals leave more or less tarry deposit on the water-heating surfaces, which reduces the economic evaporative effect. Hence, the proportion of the total heat developed by the combustion of coal which in practice can be applied to the heating-surfaces of a steam-boiler and expended in evaporating water to steam varies considerably.

The steam-producing capacity of coals has been frequently tested, and Table 21 contains the average results of a number of trials of different kinds of best steam-coal in internally-fired steam-boilers, under ordinary working conditions, with natural draught.

TABLE 21.—PROPORTION OF THE TOTAL HEAT THEORETICALLY DEVELOPED BY THE COMBUSTION OF COAL, EXPENDED, ON AN AVERAGE, IN PRACTICE IN EVAPORATING WATER TO STEAM IN INTERNALLY-FIRED STEAM-BOILERS WITH NATURAL DRAUGHT.

Description of Coal.	Proportion of the Total Heat of Coal, on an average, applied to the Heating-Surfaces of Steam Boilers and absorbed by the water.	
	Coal in Lumps, per cent.	Rough Slack, per cent.
Welsh steam-coal, best . . .	73	68
Newcastle steam-coal, best . .	68	62
Lancashire steam-coal, best . .	65	60
Derbyshire steam-coal, best . .	62	54
Yorkshire steam-coal, best . .	58	50
Anthracite coal, best . . .	52	65

All coals, for steam-purposes, yield more heat when used in lumps of moderate size than in the form of slack, except anthracite, which burns with difficulty in lumps, but burns more freely when in small pieces.

Culm, or comminuted anthracite coal, has an efficiency of about 75 per cent. of that of anthracite coal.

The total heat developed by good ordinary Welsh coal is, from Table 20, 14,826 units per pound, and its evaporative power is 15·3 lbs. of water, but it appears from Table 21 that only 73 per cent. of the total heat would, on an average, be expended in the production of steam in a well-arranged steam boiler under ordinary working conditions with natural draught, or $14826 \times \cdot 73 = 10823$ units per pound of coal fired, equal to an evaporative power of $10823 \div 966 = 11 \cdot 2$ pounds of water, from and at 212° Fahr., per pound of coal fired in the boiler.

Heat-Energy of Combustion.—The amount of energy stored in fuel and liberated by combustion is found by multiplying the number of heat-units developed by the complete combustion of the fuel by the mechanical equivalent of each unit of heat, equal 772 foot lbs. of work. For instance, the heat-energy or force developed by 1 lb. of carbon in burning to carbonic acid is equal to 14500 units × 772 = 11194000 foot pounds of work.

The heat-energy yielded by the combustion of 1 lb. of carbon per hour is = 11194000 foot pounds ÷ (33000 foot pounds per min. × 60 mins.) = 5·65 horse-power; so that about one-sixth of a pound of carbon is theoretically capable of liberating by complete combustion an amount of heat-energy equivalent in mechanical energy to 1 horse-power per hour.

The heat-energy yielded by the complete combustion of coal of average quality, containing about 80 per cent. of carbon and having a calorific power of 1296 units per pound of coal, is equal 1296 × 772 = 10000000 foot pounds of work per pound of coal consumed, and = 10000000 ÷ (33000 × 60) = say, 5 horse-power theoretically developed per pound of coal consumed per hour; representing a consumption of 1 ÷ 5 = ·2 lb. of coal per indicated horse-power per hour.

This result has not yet been attained in practice, as the best engines, on an average, only perform about one-tenth of that duty, and the worst engines only average one-thirteenth of the theoretical duty. It will be seen that, theoretically, one horse-power should be developed by the heat yielded by the complete combustion of 3·2 ounces, or less than a quarter of a pound of coal per hour.

The actual power developed by the combustion of coal in practice is very much less than the theoretical quantity, as, owing to imperfect combustion and inefficiency of the heating-surfaces of steam boilers, it is not possible to utilize all the heat-energy in the coal. The highest duty obtainable in practice is probably one horse-power per pound of coal consumed per hour. This has been very nearly attained with quadruple expansion engines, which, in some cases, develop one indicated horse-power with a combustion of $1\frac{1}{8}$ lb. of Welsh coal per hour.

In the most economical modern engines the consumption of coal per indicated horse-power per hour, under ordinary working conditions, averages as follows:—

	lbs.
Quadruple expansion surface-condensing engines	1·25
Triple expansion surface-condensing engines	1·50
Double expansion engines, condensing	2·00
Corliss engines	2·12
Double expansion engines, non-condensing	2·50
Simple engines	2·75

The consumption of coal by these types of engines is, however, frequently considerably greater than this.

The Maximum Temperature of Combustion may be determined by dividing the calorific value of the fuel by the heat-capacity of the gaseous products of combustion. The specific heat of the products of combustion at a temperature of 570°, which is the average temperature of the gases in the chimney of well-arranged boilers with natural draught, averages ·265 according to Table 1, page 5; but it is usual to assume the specific heat of the gases in ordinary calculations to be the same as that of the air, or ·238.

The temperature resulting from the combustion of one pound of carbon burning to carbonic acid, supported by 2·667 lbs. of oxygen supplied by 12 pounds of atmospheric air at 62° Fahr., may be calculated as follows:—

1 lb. of carbon + 12 lbs. of air produces 13 lbs. of gases. The absolute temperature of the air is 461° + 62° = 523° Fahr.

The temperature resulting from combustion is =

$$\frac{\text{14500 units, heat of combustion of 1 lb. of carbon}}{\text{13 lbs. of gases} \times \text{·238 specific heat of air}} + 523° = 5210 \text{ Fahr.}$$

absolute temperature.

Efficiency of Combustion, or the proportion of the heat evolved which can be realised in a furnace, is expressed by the following formula, in which the higher temperature is that of the furnace, and the lower temperature is that of the escaping gases:—

Efficiency of combustion, or proportion of useful effect =

$$\frac{\text{Higher absolute temperature} - \text{Lower absolute temperature}}{\text{Higher absolute temperature.}}$$

Applying this rule to the result of the previous example, and assuming the gases to enter the chimney at an absolute temperature = 552° + 461° = 1013° Fahr., with natural draught. The efficiency is,

$$= \frac{5210 - 1013}{5210} = ·80,$$

that is, the heat realised is only 80 per cent. of that supplied, showing a loss of 20 per cent. of the heat evolved.

The Temperature of the Furnace of a Steam-Boiler with Natural Draught may be calculated in a similar way to the above. Assuming that 1 lb. of good coal burning to carbonic acid, supported by oxygen supplied by 24 lbs. of air at 62° Fahr., that being the quantity of atmospheric air frequently required in practice with natural draught per pound of coal, developes 14300 units of heat.

Then 1 lb. of coal + 24 lbs. of air produces 25 pounds of gases, and the absolute temperature of the air is = 461 + 62 = 523° Fahr.

The temperature of the furnace resulting from combustion is:—

$$\frac{\text{14300 units of heat of combustion of 1 lb. of coal}}{\text{25 lbs. of gases} \times \text{·238 specific heat of air}} + 523° = 2926°$$

Fahr. absolute temperature.

Assuming the gases to enter the chimney at 552° + 461 = 1013 absolute temperature.

The efficiency of the combustion is,

$$= \frac{2926^\circ - 1013^\circ}{2926} = \cdot 66,$$

showing a loss of one-third the quantity of heat evolved.

This is for perfect combustion, but if an allowance of 10 per cent. be made for imperfect combustion, the available quantity of heat would only be 14300 − 1430 = 12870 units per pound of coal, resulting in an absolute temperature in the furnace of 2686° Fahr., and an efficiency of combustion of 63 per cent.: showing a loss of 37 per cent. of the heat evolved.

The Temperature of a Furnace with Forced Draught, assuming perfect combustion, with coal developing 14300 units of heat per pound, and using 18 lbs. of air per pound of coal at a temperature of 62° Fahr., may be found as follows :—

1 lb. of coal + 18 lbs. of air = 19 lbs. of gases, and the absolute temperature of the air is 62 + 461 = 523° Fahr. Assuming that by well-arranged absorbing or heating-surfaces of the boiler and feed-water heater the gases enter the chimney at 400° + 461° = 861° Fahr. absolute temperature.

Then the temperature of the furnace resulting from combustion is,

$$= \frac{14300 \text{ units of heat}}{19 \text{ lbs. of gases} \times \cdot 238} + 523^\circ = 3686^\circ \text{ Fahr.}$$

the absolute temperature. And the efficiency of the combustion is,

$$= \frac{3686^\circ - 861^\circ}{3686^\circ} = \cdot 76:$$

showing a loss of 27 per cent. of the heat evolved.

Or it may be put in this form :—

The quantity of the heat required to raise the contents of the furnace of a steam-boiler one degree in temperature for each pound of coal consumed is = 1 lb. of coal + 24 lbs. of air = 25 lbs. × ·238 specific heat = 5·95 units with natural draught; and 1 lb. + 18 lbs. of air = 19 lbs. × ·238 = 4·522 units with forced draught.

Coal developing 14300 units of heat per pound of coal, will raise the temperature of 1 lb. = 14300 ÷ 5·95 = 2403° Fahr. with natural draught: and = 14300 ÷ 4·522 = 3163° Fahr. with forced draught. Then with air of 523° Fahr. absolute temperature. The absolute temperature of the furnace is = 2403 + 523° = 2926° Fahr. with natural draught; and = 3163° + 523 = 3686 with forced draught.

To obtain the Highest Efficiency of Combustion, or the maximum economical effect from the fuel, it is necessary to have :—

The highest possible temperature in the furnace.

The lowest possible temperature in the chimney.

The smallest possible quantity of air supply that will ensure complete combustion.

As complete combustion as possible of the fuel-gases in the furnace, or before they traverse the remainder of the absorbing, or heating-surfaces, of the boiler.

Complete combustion, resulting in the conversion of all the carbon to carbonic acid, and all the hydrogen to water, may generally be obtained from the admission of from 33 to 50 per cent. more air to the furnace than is theoretically necessary to supply the quantity of oxygen required for perfect combustion, or not exceeding 12 lbs. + 6 = 18 lbs. of air per lb. of coal.

Complete combustion can only be obtained from a moderate coal consumption per square foot of fire-grate surface; when combustion is urged by hard firing, a considerable quantity of fuel is lost in the form of carbonic oxide escaping unconsumed.

Average Temperature of the Furnaces of Steam-Boilers.—The temperature of the furnaces of steam-boilers is less in practice than that theoretically due to the calorific power of the fuel, owing to imperfect combustion, and the cooling effect of the absorbing surfaces and the air entering the furnace. It may be assumed that the maximum temperature of the products of combustion of good coal, with complete combustion, with natural draught, at the instant of their formation, is 2477° Fahr. above that of the atmosphere, or with air at $62° = 62° + 2477° = 2539°$ Fahr. This is equal to an absolute temperature of $2539 + 461° = 3000°$ Fahr.

Assuming that one-third of the heat developed is absorbed by the water-heating surface of the fire-box or furnace, then the temperature of the products of combustion at the fire-bridge of a circular furnace, or at the entrance to the tubes of a locomotive boiler is $= 2539 \times \frac{2}{3} = 1693°$ Fahr., or, say, in round numbers, 1700° Fahr. $= 1700 + 461 = 2161°$ Fahr. absolute temperature.

As pyrometers are seldom reliable, the temperature of the furnace of a steam-boiler cannot accurately be ascertained, but it may be determined approximately in three different ways, viz:—

By the heat imparted to a piece of iron embedded in the glowing fuel: by melting a piece of metal of known melting point in the fire: by the colour of the fire.

To ascertain the temperature of a furnace by the heat imparted to a piece of wrought-iron. An iron-ring should be embedded in the fire and allowed to remain until uniformly heated to the same temperature as that of the fuel, and then quenched in a given weight of water.

The rise of temperature of the water will enable the temperature of the fire to be calculated by the following rule, which assumes the specific heat of wrought-iron to be one-ninth that of water:—

Let T = the temperature of the water produced by quenching the iron.
t = the original temperature of the cooling-water.
W = the weight of the cooling-water in pounds.

w = the weight of the wrought-iron ring in lbs.
F = the required temperature of the furnace.

$$F = \frac{(T - t) \times W \times 9}{w} + T$$

Example. A ring of wrought-iron weighing 18 lbs. was embedded in the fire of a Cornish boiler, and when uniformly heated was quenched in 50 lbs. of water at 62° Fahr., thereby raising the temperature of the water to 150° Fahr. Required the temperature of the furnace?

Then $\frac{(150^\circ - 62^\circ) \times 50 \text{ lbs. of water} \times 9}{18 \text{ lbs. weight of wrought-iron}} + 150^\circ = 2350^\circ$ Fahr. the approximate temperature of the furnace.

The temperature of the furnace of a steam-boiler may be determined approximately by melting a piece of metal, of known melting point, in the fire. The melting points of metals are given in the following Table:—

TABLE 22.—MELTING POINTS OF METALS.

Metal.	Melts at Fahrenheit.	Metal.	Melts at Fahrenheit.
	°		°
Wrought-iron becomes fluid at	4000	Copper	2050
Wrought-iron begins to melt at	2910	Brass	1650
Mild-steel boiler-plates	3100	Aluminium	1300
Platinum	3080	Antimony	810
Mild-steel castings	2930	Zinc	773
Shear-steel	2740	Lead	620
Tool-steel	2550	Bismuth	507
Cast-iron, grey	2190	Tin	446
Cast-iron, white	2010	Cadmium	442

The temperature of the furnace of a steam-boiler may also be determined approximately by the colour of the fire. The colour-temperatures of furnaces, obtained from the experiments of Pouillet, are given in the following Table:—

TABLE 23.—COLOUR-TEMPERATURES OF FURNACES.

Appearance of the Fire.	Temperature Fahrenheit.	Appearance of the Fire.	Temperature Fahrenheit.
	°		°
Red, just visible	977	Orange, deep	2010
Red, dull	1290	Orange, clear	2190
Red, cherry dull	1470	White heat	2370
Red, full	1650	White, bright	2550
Red, clear	1830	White, dazzling	2730

Products of Combustion.—The gases forming the products of perfect combustion are very bad absorbers and bad radiators of heat. The volume of the gases proceeding from combustion depends upon their temperature. Oxygen combining with carbon forms carbonic acid without change of volume of the oxygen, therefore the volume of gas resulting from the combustion of carbon is the same as that of the air entering the furnace, except that it is expanded to the volume corresponding to its increased temperature.

The air may be supposed to enter the fire at 62° Fahr., at which temperature the weights and volumes of gases and vapours are as given in the following Table :—

TABLE 24.—WEIGHT AND VOLUME OF GASES AND VAPOURS AT 62° FAHR. UNDER AN ATMOSPHERIC PRESSURE OF 30 INCHES OF MERCURY.

Description of Gas.	Weight per Cubic foot. lbs.	Volume of one pound in Cubic feet.
Hydrogen gas	·005264	189·73
Coal-gas	·033300	30·00
Vapour of water . . .	·047398	21·00
Carbonic oxide gas . .	·073632	13·65
Nitrogen gas	·073795	13·55
Atmospheric air . . .	·076098	13·14
Oxygen gas	·084133	11·90
Carbonic acid gas . . .	·116365	8·60

The Volume of the Gaseous Products of Combustion of Fuel containing hydrogen may be calculated as follows. Coal, for instance, containing ·05 hydrogen per pound, combines with ·05 × 8 lbs. of oxygen = ·4 lb. of oxygen, to form ·05 + ·4 = ·45 lb. of water. The volume of the vapour of water at 62° Fahr. is, from Table 24, = 21 cubic feet per pound, and ·45 lb. of water will produce ·45 × 21 = 9·45 cubic feet of vapour at 62° Fahr. The air may be assumed to enter the fire at 62° Fahr., = an absolute temperature of 62° + 461° = 523° Fahr., and enter the chimney at 560° Fahr., = an absolute temperature of 560° + 461° = 1021° Fahr.

The quantity of air required for combustion is frequently 300 cubic feet per pound of coal.

Then, the volume of the gases produced at 62° is = 300 + 9·45 = 309·45 cubic feet per pound of coal, and the volume of the gaseous products of combustion in the chimney is =

$$\frac{309{\cdot}45 \text{ cubic feet of gases} \times 1021^\circ \text{ final absolute temperature}}{523 \text{ initial absolute temperature of the air}} = 605 \text{ cubic}$$

feet per pound of coal consumed.

The gaseous products of combustion cannot be heated by radiant heat. Radiant heat can only be communicated to gases by causing them to traverse surfaces which have previously absorbed radiant heat.

Displacement of the Atmosphere by Smoke.—The heat expended by the products of combustion in overcoming the pressure of the atmosphere, with natural draught, may be found by multiplying the increase of volume due to the expansion of the gases by elevation of temperature by the pressure of the atmosphere in pounds per square foot, and dividing the product by 772.

For instance, in the previous example the increase of volume from elevation of temperature is 605 cubic feet, the final volume, — 309·45 cubic feet, the initial volume, = 295·55 cubic feet. Then, the work of, or heat expended in, displacing the atmosphere is =

$$\frac{295{\cdot}55 \text{ cubic feet, increase of volume,} \times 144 \text{ square inches} \times 14{\cdot}7 \text{ lbs.}}{772 \text{ units of heat}} =$$

810 units of heat per pound of coal consumed.

The Velocity of the Gaseous Products of Combustion, V, in feet per second, may be found by the following rule:—

$$V = \frac{\text{Weight of fuel in lbs.} \times \text{volume of gases in cubic feet}}{\text{Time in seconds} \times \text{area in square feet of tube-opening}}.$$

Suppose, for instance, that the gases due to combustion enter the fire-box end of the tubes of a steam-boiler at 2926° Fahr., absolute temperature, and leave the smoke-box end of the tubes at 1013° Fahr., absolute temperature. The area of opening through the tubes is 1·04 square feet; consumption of coal per hour 43 lbs.; air used per pound of coal 24 lbs., with natural draught, its absolute temperature being 523° Fahr.

Then 1 lb. of coal + 24 lbs. of air = 25 lbs. × 13·14 volume per cubic foot = 329 cubic feet of gases.

The volume of the hot gases at the fire-box end of the tubes is =

$$\frac{329 \text{ cubic feet} \times 2926^\circ}{523^\circ} = 1840 \text{ cubic feet.}$$

The velocity of the hot gases at the fire-box end of the tubes is =

$$\frac{43 \text{ lbs. of coal} \times 1840 \text{ cubic feet of gases}}{60 \text{ seconds} \times 60 \text{ mins.} \times 1{\cdot}04 \text{ square feet}} = 21{\cdot}13 \text{ feet per second.}$$

The volume of the hot gases at the smoke-box end of the tubes is =

$$\frac{329 \text{ cubic feet} \times 1013^\circ}{523^\circ} = 638 \text{ cubic feet.}$$

The velocity of the hot gases at the smoke-box end of the tubes is =

$$\frac{43 \text{ lbs. of coal} \times 638 \text{ cubic feet of gases}}{60 \text{ seconds} \times 60 \text{ mins.} \times 1{\cdot}04 \text{ square feet}} = 7{\cdot}33 \text{ feet per second.}$$

The velocity of the fuel-gases varies through different tubes, being greatest through the top row and least through the bottom row of tubes.

Weight of the Gases disengaged by Combustion.—The specific heat, specific gravity, and weight per cubic foot of the gases forming the products of combustion are given in the following Table :—

TABLE 25.—SPECIFIC HEAT, SPECIFIC GRAVITY AND WEIGHT PER CUBIC FOOT OF GASES, AT 32° FAHR., UNDER THE PRESSURE OF ONE ATMOSPHERE, OR 29·9 INCHES OF MERCURY.

Gas.	Symbol.	Specific Heat.	Specific Gravity.	Weight of one Cubic Foot. lb.
Atmospheric air	N_2O	·2380	1·0000	·08072
Carbonic acid	CO_2	·2164	1·5290	·12344
Carbonic oxide	CO	·2460	·9674	·07810
Nitrogen	N	·2440	·9736	·07859

The Composition of the Gaseous Products of Combustion varies considerably. It is important to analyse the fuel-gases in boiler-tests to determine the quantity of fuel wasted by imperfect combustion, and the quantity of air supplied to the fuel. By this means it may be ascertained whether there has been loss of fuel from the admission of a greater quantity of air to the furnace than is necessary to effect proper combustion, or from the admission of a smaller quantity of air and the escape of carbonic oxide unconsumed.

It may be useful to give as a representative example, an analysis of gases from the combustion of coal, obtained during the test of Lancashire boilers, with an explanation of the method of calculating the volumes, weights, and heat-capacities of the constituents.*

The gases were collected from the main flues between the boilers and a feed-water heater, or economiser, and their average composition was as follows :—

Carbonic-acid gas	10·35
Carbonic oxide	·25
Nitrogen and other gases	51·42
Air	37·98
	100·00

Or more shortly :—

Products of combustion	62·02
Air in excess	37·98
	100·00

The volume of the carbonic acid and carbonic oxide is calculated as follows :—

Let $x + y$ be the total weight of carbon in one pound of coal: x the weight converted into carbonic acid, and y the weight converted into carbonic oxide. Then, 1 lb. of carbon combines with 2·66 lbs. of oxygen and forms 3·66 lbs. of carbonic acid; and the weight of carbonic acid per pound of coal is $3·66 \times x$ lbs. Also one cubic foot of carbonic acid at 32° Fahr.

* The Author is indebted for these data and calculations to the report of a trial of Steam-Boilers by Mr. Michael Longridge, Chief Engineer of the Engine and Boiler Insurance Co., Limited, Manchester.

under a pressure of 29·9 inches of mercury weighs ·122 lb. Therefore, the volume of carbonic acid from one pound of coal at 32° Fahr., and under 29·9 inches of mercury is :—

$$\frac{3·66}{·122} \times x \text{ cubic feet} = 30x \text{ cubic feet.}$$

Again 1 lb. of carbon combines with 1·33 lbs. of oxygen, and forms 2·33 lbs. of carbonic oxide; and the weight of carbonic oxide from 1 lb. of coal is $2·33 \times y$ lbs. One cubic foot of carbonic oxide weighs ·078 lbs., and the volume of carbonic oxide from one pound of coal is:—

$$\frac{2·33}{·078} \times y = 30y \text{ cubic feet.}$$

If u and v be the number of volumes of carbonic acid and carbonic oxide in 100 volumes of gas, that is, the volumes given by the analysis, and V = the total volume in cubic feet of the carbonic acid, carbonic oxide, oxygen, and nitrogen, per pound of coal burnt, then

$$30x : V :: u : 100$$
$$\text{and } 30y : V :: v : 100$$
$$\text{or } V = \frac{3000\,(x+y)}{u+v}$$

The weight of carbon burnt was ·655 lb. per pound of coal, and substituting this value for $x + y$, the volume of the carbonic acid, carbonic oxide, air, and nitrogen, per pound of dry coal is :—

$$V = \frac{3000 \times ·655 \text{ lb. carbon}}{10·35 \text{ carbonic acid} + ·25 \text{ carbonic oxide}} = 185·4 \text{ cubic feet, the}$$

total volume of the gases per pound of coal.

The volume of each constituent may be found by rule of three thus :—

Volume of carbonic acid : V : u : 100

and from this volume the weight is found by multiplying the volume by the weight of one cubic foot, as follows:

	Volume of Gas.		Cubic feet.		Weight per cubic foot.		Weight of Gas per lb. of dry coal.
Carbonic acid . . .	185·4 × ·1035	=	19·19	×	·122	=	2.341 lbs.
Carbonic oxide . .	185·4 × ·0025	=	·46	×	·078	=	·036 lb.
Nitrogen	185·4 × ·5142	=	95·33	×	·078	=	7·436 lbs.
Air	185·4 × ·3798	=	70·41	×	·081	=	5·703 lbs.
			185·39				15·516 lbs.

In addition to these gases there is the steam resulting from the combustion of the hydrogen from the moisture in the coal; and the vapour in the air, as follows :—

Steam from hydrogen = ·0404 hydrogen × 9	=	·364 lb.
Vapour in air	=	·250 lb.
Also, steam from water mixed with the coal	=	·012 lb.
Making the total weight of gas	=	16·142 lbs.

The heat-capacity of the gases, or the quantity of heat required to raise the temperature of the gases one degree Fahr. per pound of dry coal, is calculated as follows:—

	Weight.		Specific heat.		Heat capacity.	
Carbonic acid	2·341 lbs.	×	·216	=	·506	
Carbonic oxide.	·036 lb.	×	·246	=	·009	
Nitrogen and other gases . . .	7·436 lbs.	×	·244	=	1·814	
Steam	·364 lb.	×	·481	=	·175	
Products of combustion				=	2·504	units
Air in excess	5·703 lbs.	×	·238	=	1·357	"
Vapour in air	·250 lb.	×	·481	=	·120	"
Steam from water in coal	·012 lb.	×	·481	=	·006	"
					3·987	units

The heat passing up the chimney is calculated by multiplying the heat capacity of the gases, given above, by the excess of temperature above 32°. The temperature of the gases leaving the economiser was 332° Fahr., and the excess of temperature is 332 − 32 = 300° Fahr., then—

Products of combustion	= 2·504 × 300 =	751 units
Excess of air	1·357 × 300 =	407 "
Vapour in air	·120 × 300 =	36 "

Each pound of coal contained ·012 lb. of water mixed with the coal which was put into the furnace and evaporated under atmospheric pressure, and the steam was superheated to 332° Fahr.; the heat carried off by the steam was, therefore:—

$$\cdot012\,(1179 - 32) + \cdot012 \times \cdot481 \times (332 - 212) = 14 \text{ "}$$

$$1208 \text{ "}$$

the heat carried away in the products of combustion to the chimney. The heat lost by imperfect combustion is calculated as follows:—

When one pound of carbon is burnt to carbonic acid 14545 units of heat are set free, but when the same quantity is burnt to carbonic oxide only 4451 units are liberated. Hence, the heat lost by every pound of carbon converted into carbonic oxide is = 14545 − 4451 = 10094 units. The total volume of the gases at 32° Fahr. is 185·4 cubic feet, and the weight of carbon converted into carbonic oxide per pound of dry coal was =

$$\frac{185\cdot4 \text{ cubic feet} \times \cdot25 \text{ carbonic oxide}}{100 \times 30} = \cdot0154 \text{ lb.}$$

Therefore, the heat lost by imperfect combustion in this case was = 10094 × ·0154 = 155 thermal units.

To determine the loss due to unburnt carbon, or the heat equivalent of the carbon which fell through the bars or was drawn out of the furnaces unburnt, the weight of carbon is multiplied by its calorific value. The actual weight of clinker and ash drawn out of the furnace per pound of coal was ·18 lb., and the weight shown by analysis of the coal was ·158; the difference, or the weight of unburnt carbon per pound of coal was = ·18 − ·158 = ·022 lb.; and the loss per pound of coal was = 14545 × ·022 = 320 thermal units.

Smoke is the product of imperfect combustion. It is caused by a portion of the particles of carbon of the hydro-carbon gases passing away unconsumed from the fire. These particles are deposited as soot on the surfaces traversed by the fuel-gases. The colour of smoke depends upon the quantity of particles of carbon carried in suspension by the fuel-gases; the greater the quantity of carbon the blacker the smoke. Black smoke radiates much more heat than the less carbon-laden or dark greyish-brown coloured smoke; and light yellow-coloured transparent vapour radiates no heat.

Soft coal produces more smoke than hard coal, and the more inferior the quality of the coal the greater the quantity of smoke produced.

Bituminous coal cannot be burnt without evolving smoke, unless the supply of air to the furnace be regulated to secure sufficient dilution of the hydro-carbon gases distilled from the coal to ensure their combustion. This may be effected with a sufficiently high furnace-temperature for the carbon to combine with the oxygen, by admitting air in numerous small jets above the fire, and in small streams through the solid fuel from below the fire, flowing through suitable spaces between the fire-bars. To burn some coals without smoke, a considerable quantity of air is required above the fire. This may be admitted through numerous slots $\frac{1}{16}$ inch wide, in a grid extending the full height and width of the door of the furnace.

With a well-regulated air-supply the production of smoke is preventable, and the emission of smoke in large quantities is inexcusable, because it is generally due either to unskilful or careless firing, or to defects in the arrangement of the furnace of the boiler.

Efficient mechanical stoking conduces greatly to the prevention of smoke.

Effect of Soot and other Deposits from Fuel-Gases on Heating-Surfaces.— Fuel-gases deposit on the heating-surfaces of steam-boilers soot, tarry matter, sulphuric acid, and ammonial salts. The quantity of soot deposited depends upon the description of fuel used and the intensity of the draught; it increases as the draught diminishes, and is greatest at the point where the fuel-gases leave the boiler. Soot is a very bad conductor of heat, and a coating of soot greatly reduces the evaporative effect of the heating-surfaces of a steam-boiler.

Soot in a dry state does not injure boiler-plates, but the presence of moisture causes the formation of sulphuric acid produced from the sulphur in the coals. Sulphuric acid in a dry state does little or no injury to boiler-plates, but in the presence of moisture it is very corrosive and rapidly pits iron and steel.

SECTION II.

CHIMNEYS FOR STEAM-BOILERS; STEAM-BLAST; FORCED DRAUGHT; FEED-WATER; EFFECT OF HEAT ON WATER; EXPANSION OF WATER BY HEAT; WEIGHT OF WATER AT DIFFERENT TEMPERATURES; CONVECTION; CIRCULATION; EVAPORATION; PROPERTIES OF SATURATED STEAM; EVAPORATIVE POWER OF BOILERS; PRIMING; ETC.

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Chimneys for Steam-Boilers.—The function of a chimney is to provide sufficient draught to effect the combustion of a certain amount of fuel on the fire-grate of a steam-boiler in a given time, and to carry off the noxious products of combustion. The height of a chimney should be sufficient not only to provide the required intensity of draught for the efficient combustion of fuel, but also to carry the products of combustion to such a distance above the surrounding property as not to be a nuisance.

Air when heated expands and decreases in weight as it increases in volume, consequently the air and fuel-gases carried with it up a chimney are considerably lighter than the atmospheric air outside.

The source of power of the draught of a chimney is the difference of weight of the column of heated gases inside the chimney, and of a column of air of equal height outside the chimney. The excess of pressure of the external air over that in the chimney causes air to flow into the chimney through every available opening above and below the fire-grate, and results in the efflux of the products of combustion at a velocity corresponding to the difference of weight of the two columns, or head, subject to a deduction for the resistance due to friction.

Each pound of coal consumed yields from 10 to 40 lbs. of gas. The average weight, at 32° Fahr., of the gases produced from the coal usually burnt in the furnaces of steam-boilers, exclusive of air, is 11 lbs. per pound of dry coal burnt, the volume of which varies with the temperature.

The Weight of Gases at different temperatures varies considerably, as may be seen from the following Table of the weight of dry air.

TABLE 26.—VOLUME AND WEIGHT OF DRY AIR AT DIFFERENT TEMPERATURES, UNDER A CONSTANT ATMOSPHERIC PRESSURE OF 29·92 INCHES OF MERCURY, THE VOLUME AT 32° FAHR. BEING 1.

Temperature, Degrees Fahrenheit.	Volume.	Weight of a Cubic Foot in Pounds.	Temperature, Degrees Fahrenheit.	Volume.	Weight of a Cubic Foot in Pounds.
°			°		
0	·935	·0864	500	1·954	·0413
12	·960	·0842	552	2·056	·0385
22	·980	·0824	600	2·150	·0376
32	1·000	·0807	650	2·260	·0357
42	1·020	·0791	700	2·362	·0338
52	1·041	·0776	750	2·465	·0328
62	1·061	·0761	800	2·566	·0315
72	1·082	·0747	850	2·668	·0303
82	1·102	·0733	900	2·770	·0292
92	1·122	·0720	950	2·871	·0281
102	1·143	·0707	1000	2·974	·0268
112	1·163	·0694	1100	3·177	·0254
122	1·184	·0682	1200	3·381	·0239
132	1·204	·0671	1300	3·584	·0225
142	1·224	·0659	1400	3·788	·0213
152	1·245	·0649	1500	3·993	·0202
162	1·265	·0638	1600	4·196	·0192
172	1·285	·0628	1700	4·402	·0183
182	1·306	·0618	1800	4·605	·0175
192	1·326	·0609	1900	4·808	·0168
202	1·347	·0600	2000	5·012	·0161
212	1·367	·0591	2100	5·217	·0155
230	1·404	·0575	2200	5·420	·0149
250	1·444	·0559	2300	5·625	·0142
275	1·495	·0540	2400	5·827	·0138
300	1·546	·0522	2500	6·032	·0133
325	1·597	·0506	2600	6·236	·0130
350	1·648	·0490	2700	6·440	·0125
375	1·689	·0477	2800	6·644	·0121
400	1·750	·0461	2900	6·847	·0118
450	1·852	·0436	3000	7·051	·0114

Draught-Power of a Chimney.—The draught-power of a chimney is independent of the internal area, and depends upon the height. The efficiency of a chimney is greatest when the volume of the fuel-gases in the chimney is about double that of the external air. There are then two columns of air of the height of the chimney, one being one-half the weight of the other. If the air outside the chimney be of the standard temperature of 62° Fahr., then it will be seen from Table 26 that the gases should leave the chimney at 552° Fahr. to have a weight equal to one-half that of the external air. If the height of the chimney above the fire-grate be, say, 90 feet high, then the gases will have a velocity due to a head of 90 ÷ 2 = 45 feet.

The draught-power of this chimney, expressed in inches of water, may be calculated as follows:—The weight of a cubic foot of air at 62° Fahr. is, from Table 26, = ·0761 lb., and at 552° = ·0385. The weight of a cubic foot of water at 62° Fahr. is = 62·355 lbs. Therefore, the weight of the air is = 62·355 ÷ ·0761 = 820 times as light as water, and at 552° it is 62·355 ÷ 0385 = 1635 times as light as water. A column of air at 62° Fahr. 90 feet, or 90 × 12 = 1080 inches high, is equal to 1080 ÷ 820 = 1·317 inch of water, and at 552° it is = 1080 ÷ 1635 = ·66 inch of water, or a difference of 1·317 — ·66 = ·657 inch of water, being the draught-power of this chimney.

The draught-power of chimneys of different heights, calculated in this way, is given in the following Table:—

TABLE 27.—DRAUGHT-POWER OF CHIMNEYS HAVING A TEMPERATURE OF 552° FAHR. INTERNALLY, AND 62° FAHR. EXTERNALLY.

Height of Chimney above the Fire-grate in feet.	Draught-power in inches of water.	Height of Chimney above the Fire-grate in feet.	Draught-power in inches of water.
10	·073	130	·948
20	·146	140	1·029
25	·182	150	1·095
30	·219	160	1·167
35	·256	170	1·240
40	·291	180	1·313
50	·364	190	1·386
60	·437	200	1·459
70	·512	225	1·641
80	·583	250	1·825
90	·657	275	2·006
100	·729	300	2·189
110	·802	350	2·553
120	·875	400	2·918

The draught-power in inches of water multiplied by ·03608 = the pressure of the air in pounds per square inch.

It will be seen from the above table that each 10 feet in height of a chimney gives a draught-power of a little more than $\frac{1}{13}$ inch of water.

When the internal temperature of the chimney is not ascertained, the draught-power of the chimney of a well-arranged boiler may be calculated approximately by the following Rule:—

Draught-power in inches of water, approximately = Height of chimney in feet × ·0073.

For instance, the draught-power to be obtained from a chimney 100 feet high is = 100 × ·0073 = ·73 inch of water.

The volume of fuel-gases in cubic feet passing up the chimney may be calculated from the analysis of the coal or of the fuel-gases as explained on pages 59 and 62.

Fuel Expended in Producing Draught in Chimneys.—The draught-power of a chimney is obtained at considerable expense of fuel. To obtain a good draught the column of hot gases inside the chimney requires to be of so high a temperature that from 20 to 30 per cent. of the heat of combustion of coal is expended in producing the draught. The heat carried off by the gases may be found by multiplying the weight of the products of combustion by the difference in temperature of the gases inside the chimney and the external air, and by the specific heat of air.

For instance, if 23 pounds of air are used per pound of coal, there will be 1lb. of coal + 23 lbs. of air = 24 lbs. of gases, and if the difference of temperature of the hot gases inside, and the air outside, the chimney be 500° Fahr., then the heat carried off by the gases is 24 lbs. × 500 degrees × ·238 specific heat = 2856 units. Taking the heat of combustion of one pound of coal at 14300 units, then (2856 units × 100) ÷ 14,300 = 20 per cent. of the total heat developed by combustion is absorbed in producing the draught.

Draught Required for the Combustion of different kinds of Fuels.—The strength of draught required for the efficient combustion of fuel in the furnace of a steam-boiler with well-arranged flues depends principally upon the nature of the fuel, the thickness of the fire, and the efficiency of the air supply to the fuel. Small caking coal generally requires at least double the strength of draught that is necessary for effecting the combustion of good ordinary coal, such as steam-coal.

TABLE 28.—DRAUGHT-POWER REQUIRED FOR THE EFFICIENT COMBUSTION OF VARIOUS KINDS OF FUELS IN THE FURNACES OF STEAM-BOILERS.

Description of Fuel.	Draught-Power of Chimney in inches of Water.	Description of Fuel.	Draught-Power of Chimney in inches of Water.
Straw	·20	Slack, very small	·7 to 1·1
Wood	·30	Coal-dust	·8 to 1·1
Sawdust.	·35	Semi-Anthracite Coal	·9 to 1·2
Peat, light.	·40	Mixture of Breeze and Slack	·1 to 1·3
Peat, heavy	·50	Anthracite, round	1·2 to 1·4
Sawdust mixed with small coal	·60	Mixture of Breeze and Coal-dust.	1·2 to 1·5
Steam-Coal, round.	·4 to ·7	Anthracite-slack	1·3 to 1·8
Slack, ordinary	·6 to ·9		

The draught-power, expressed in inches of water, required in practice for different kinds of moderately dry fuel is, on an average, as given in Table 28.

In burning small coal in the furnace of a steam-boiler having a chimney with a less draught than half an inch of water, it is difficult to maintain a brisk fire without continually stirring it, which results in waste of fuel and the production of smoke.

Height of Factory-Chimneys.—The height of chimney in feet required to produce sufficient draught for the efficient combustion of different kinds of fuels, may be determined by multiplying the draught-power in the previous table by 137. For instance, the least height of chimney desirable for a steam-boiler burning good ordinary, or bituminous, slack is from Table 28, $= \cdot 6 \times 137 = 83$ feet. For burning anthracite-slack the height should not be less than $= 1\cdot 3 \times 137 = 178\cdot 1$, or say 180 feet. The minimum height of chimneys permissible in towns is generally fixed by local bye-laws.

Height of Chimneys required for various rates of Combustion.—In boilers with natural draught it is seldom expedient to burn more than from 20 to 26 pounds of coal per square foot of fire-grate surface per hour. With artificial draught more than 200 lbs. of coal has been fired per square foot of fire-grate surface per hour. In locomotive-boilers as much as 180 lbs. of coal have been fired per square foot of fire-grate surface per hour. But the draught necessary to effect such high rates of combustion is so powerful that it is liable to carry a quantity of coal from the furnace in the form of cinder, and all the coal may not be burnt.

TABLE 29.—HEIGHT OF CHIMNEYS REQUIRED TO EFFECT DIFFERENT RATES OF COMBUSTION OF COAL.

Height of Chimney above the Fire-Grate in feet.	Weight of Coal that can be burnt per square foot of Fire-Grate per Hour in pounds.	Height of Chimney above the Fire-Grate in feet.	Weight of Coal that can be burnt per square foot of Fire-Grate per Hour in pounds.
10	5	110	24
20	8	120	27
25	10	130	30
30	12	140	34
35	13	150	40
40	14	180	50
50	16	200	60
60	17	225	70
70	18	250	80
80	19	300	90
90	20	350	100
100	22	400	112

The largest quantity of coal that can be completely consumed with any draught is from 84 to 100 lbs. per square foot of fire-grate surface per hour.

The height of chimney necessary for various rates of combustion of coal is, in a general way, as given in Table 29.

The draught-power in inches of water of chimneys of these heights is given in Table 27.

Velocity of the Draught or Air-current in a Chimney.—The theoretical velocity of the gases in a chimney may be calculated by the following formula:—

Let V = the velocity of the air-current in feet per second.

g = the velocity acquired in one second by a body falling from a state of rest in a space devoid of air = 32·2.

h = the height of the chimney in feet above the fire-grate.

W = the weight of the atmospheric air outside the chimney.

w = the weight of the hot air inside the chimney, which may be taken from Table 26 page 68.

$$V = \sqrt[2]{2\,g\,h\,\frac{W - w}{W}}.$$

Example: Required the velocity of the current of hot air in a chimney, 100 feet high, the temperature inside the chimney being 552° Fahr., and that of the air outside the chimney 62° Fahr.

Then the weight of air at 552° Fahr. is, from Table 26, = ·0385 lb., and at 62° Fahr. = ·0761 lb., and ·0761 − ·0385 = ·0376 ÷ ·0761 = ·494 × 100 feet × 32·2 × 2 = $\sqrt[2]{3182}$ = 56 feet per second, the velocity of the current of hot air.

The Actual Velocity of the Gases in a Chimney is very much less than the theoretical velocity on account of the resistance to the entrance of air through the fire-grate and coal, and the friction of the flues of a steam-boiler, which have a tendency to choke the draught. As the flow of gases is retarded by bends, angles, and narrow passages, the velocity of the current is considerably influenced by the shape, size, and condition of the flues.

The average or mean velocity through the whole height of the heated gases in a chimney may be found by the following formula, deduced from experiments with a number of factory-chimneys of Lancashire boilers with well-arranged flues.

To find the velocity of the hot gases in factory-chimneys connected to boilers of the Cornish and Lancashire types with flues of not exceeding 150 feet in length in circuit from the furnace to the bottom of the chimney.

Let V = the average velocity of the hot gases in the chimney in feet per second, or the mean speed of the gases through the entire height of the chimney.

T = the temperature in degrees Fahr. of the hot gases inside the chimney.

t = the temperature in degrees Fahr. of the atmospheric air on the outside of the chimney.

H = the height of the chimney in feet above the fire-grate.

$$\text{Then, } V = \frac{(T - t) \times 8\sqrt{H}}{T \times 3 \cdot 3}.$$

Example: Required the mean velocity of the hot gases in a factory-chimney of 100 feet in height, connected to the flues of a Lancashire boiler, the temperature being 552° Fahr. internally, and 62° Fahr. externally. Then, the square root of 100 = 10, and 552° − 62° = 490°, and

$$\frac{490° \times 8 \times 10}{552° \times 3 \cdot 3} = 21 \cdot 5 \text{ feet per second,}$$

the mean velocity of the gases in this chimney.

The least resistance to the passage of the gases, and the least cooling surface is presented to the gases, when the cross-section of a chimney is circular.

Size of Chimneys for Factory Steam-boilers.—The height of chimney should be measured from the top of the fire-grate of the boiler. The height of chimneys of factory-boilers in towns and populous districts should not be less than ninety feet.

When several steam-boilers have a chimney in common, its internal temperature is generally higher and more uniform, and the draught is steadier than is obtained with the chimney of a single boiler. The area of a chimney common to a range of steam-boilers may therefore be less than the product of the area required for a single boiler by the number of boilers. A less area is also permissible because all the boilers of a range are seldom worked at their full capacity.

For factory-boilers with flues not exceeding 150 feet in length in circuit from the fire-grate to the bottom of the chimney, the area of the chimney may be found by the following formula deduced from practice with well-arranged boilers.

Let F = the area of the fire-grate surface of the boiler or boilers in square feet.

W = the weight of coal in pounds consumed per square foot of fire-grate surface per hour.

C = a constant varying with the number of boilers discharging the products of combustion into one chimney.

H = the height of the top of the chimney above the top of the fire-grate in feet.

A = the internal area of the cross-section of the top of the chimney of a factory-boiler in square feet.

$$\text{Then } A = \frac{F \times W \times C}{\sqrt{H}}.$$

In which C = ·100 for a chimney for one steam-boiler.

C = ·085 for a chimney for a range of from 2 to 6 boilers.

C = ·075 for a chimney for a range of from 7 to 11 boilers.

C = ·065 for a chimney for a range of 12 or more boilers.

Example: Required the internal area of the top of a chimney for two Lancashire steam-boilers having a total area of fire-grate of 66 square feet, and a coal-consumption of 18 pounds per square foot of fire-grate surface per hour. Height of chimney above the fire-grate, 100 feet.

Then $\frac{66 \text{ square feet} \times 18 \text{ lbs.} \times \cdot 085}{\sqrt[3]{100} \text{ feet.}} = 10 \cdot 098$ square feet, the internal cross-sectional area of the top of this chimney.

Rate of Combustion for a given size of Factory-Chimney.—The coal-consumption in pounds per square foot of fire-grate surface per hour, W, suitable for a given height of chimney may be found by the following formula, in which the notation is the same as that of the previous formula:—

$$W = \frac{A \times \sqrt[3]{H}}{F \times C}.$$

Example: Required the coal-consumption suitable for the boiler with chimney described in the previous example.

Then $\frac{10 \cdot 098 \text{ square feet area of chimney} \times \sqrt[3]{100} \text{ feet}}{66 \text{ square feet of fire-grate area} \times \cdot 085} = 18$ lbs. of coal per square foot of fire-grate surface per hour.

The height of chimney suitable for a given coal-consumption may be found by the following formula, with the above notation:—

$$H = \left(\frac{F \times W \times C}{A}\right)^3$$

Taking, for example, the data from the previous examples, the height of chimney is =

$\left(\frac{66 \text{ square feet of fire-grate area} \times 18 \text{ lbs.} \times \cdot 085}{10 \cdot 098 \text{ square feet of area of chimney}}\right)^3 = 100$ feet, the height of the chimney above the top of the fire-grate.

Area of Factory-Chimney for Average Rates of Combustion.—The internal area of the cross-section of the top of the chimney of a factory steam-boiler, with flues not exceeding 150 feet in length of circuit from the fire-grate to the bottom of the chimney, when the consumption of coal is not determined, may, in a general way, be found by the following formula:—

Internal area of the cross-section of the top of a factory-chimney in square feet for the average rate of combustion =

$$\frac{\text{Area of fire-grate surface in square feet} \times C}{\sqrt[3]{(\text{height of chimney in feet})}}.$$

In which $C = 1 \cdot 82$ for a chimney for one steam-boiler.
$C = 1 \cdot 52$ for a chimney for a range of from 2 to 6 boilers.
$C = 1 \cdot 34$ for a chimney for a range of from 7 to 11 boilers.
$C = 1 \cdot 17$ for a chimney for a range of 12 or more boilers.

Example: Required the internal area at the top of a chimney, of 121 feet in height above the top of the fire-grate, for four steam-boilers, having a total area of fire-grate surface of 125 square feet.

$$\text{Then } \frac{125 \text{ square feet of fire-grate surface} \times 1{\cdot}52}{\sqrt{121} \text{ feet height of chimney}} = 17{\cdot}27 \text{ square feet of}$$

area of cross-section of the top of the chimney.

The height of the chimney of a factory steam-boiler, when the consumption of coal is not determined, may be found by the following formula, in which the notation is the same as that of the previous formula :—

Height of chimney in feet above the top of the fire-grate =

$$\left(\frac{\text{Area of fire-grate surface in square feet} \times C}{\text{area of chimney at the top in square feet}}\right)^2.$$

Taking, for example, the data from the previous example, the height of the chimney is =

$$\left(\frac{125 \text{ square feet of fire-grate area} \times 1{\cdot}52}{17{\cdot}27 \text{ square feet of area of chimney}}\right)^2 = 121 \text{ feet, the height of the}$$

chimney above the top of the fire-grate.

The internal diameter of the top of a round chimney may be found by dividing the area found by the above rule by ·7854, and extracting the square root of the quotient. The side of the square of a square chimney may be found by extracting the square root of the area of chimney.

A round chimney gives about 2 per cent. better draught than a square chimney of the same sectional area and height, but a square chimney costs less to build than a round chimney.

Main Flues of Chimneys for Factory Steam-boilers.—The velocity at which the products of combustion flow through a flue, all other things being equal, varies directly as the area of the flue. If the area be doubled the velocity of the gases is halved, and so on.

The main flue should be of larger area than the chimney, to provide for the reduction of area due to accumulation of soot. The area of the cross-section of the main flue between the boiler and the chimney may be from one-third to one-fifth the area of the fire-grate surface. The main flue is best made circular, as this form offers the least resistance to the flow of the gases. It should be as short and direct as possible, and have no sudden changes of cross-section, sharp bends, or currents entering at right angles. The internal surfaces of the flue should be as smooth as possible.

The flues from the boiler should have all the bends formed with a large radius, to facilitate the flow of the gases.

The current of the fuel-gases from one set of flues should not cross the direction of other currents. If several currents of gases moving in different directions meet in a common passage, partitions should be provided to maintain the currents separate until they have assumed the same direction, otherwise the current moving with the greatest velocity will retard, and may obliterate, the other currents.

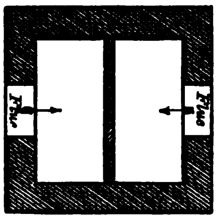


Fig. 11.—Section of a chimney between two ues.

When flues enter a chimney at opposite sides, a partition should be provided, equal in depth or height to at least twice the depth of the flue, as shown in Fig. 11.

Construction of Chimneys for Factory-Steam-Boilers.—A chimney should be constructed on a dry and firm foundation, to prevent unequal settling, tending to crack the walls. The brick and stone should be laid in good mortar, and care taken to prevent the formation of air leak-holes in the walls of the chimney and flues. The inside surfaces of the flues and chimney should be as smooth as possible, to oppose the least resistance to the currents.

The design of chimneys varies considerably. A frequently used plain design is shown in Fig. 12, and ornamental designs are shown in Figs. 13 and 14.

Large chimneys should be provided with an inner perpendicular shell or lining, 4½ inches thick, independent of the outer walls, and having a space at the back filled with sand, as shown in Fig. 15. This arrangement reduces the loss of heat by radiation to a minimum, and prevents excessive unequal expansion and contraction of the outer walls, which is liable to occur, as the difference of the internal and external temperature is frequently as high as 600° Fahr.

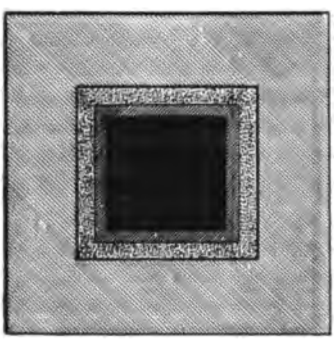


Fig. 15.—Section of a chimney with inner shell.

The thickness of the walls of factory-chimneys up to 150 feet in height is frequently 28 inches thick from the base up to about one-fourth the height, 23 inches thick for the next one-fourth of the height, 18 inches thick for the following one-fourth of the height, and 9 inches for the remainder or top portion. The inside area of the base of the chimney is formed considerably larger than that of the summit, to obtain stability of structure. The area-ratio of the bottom to the top varies considerably in practice.

The internal area-ratio of a number of well-designed factory-chimneys, of from 80 to 120 feet in height, averaged from 1·75 to 2·15, and of from 135 to 180 feet in height, from 2·13 to 3·25.

The proportions of a number of chimneys from practice are given in Table 30, page 78:—

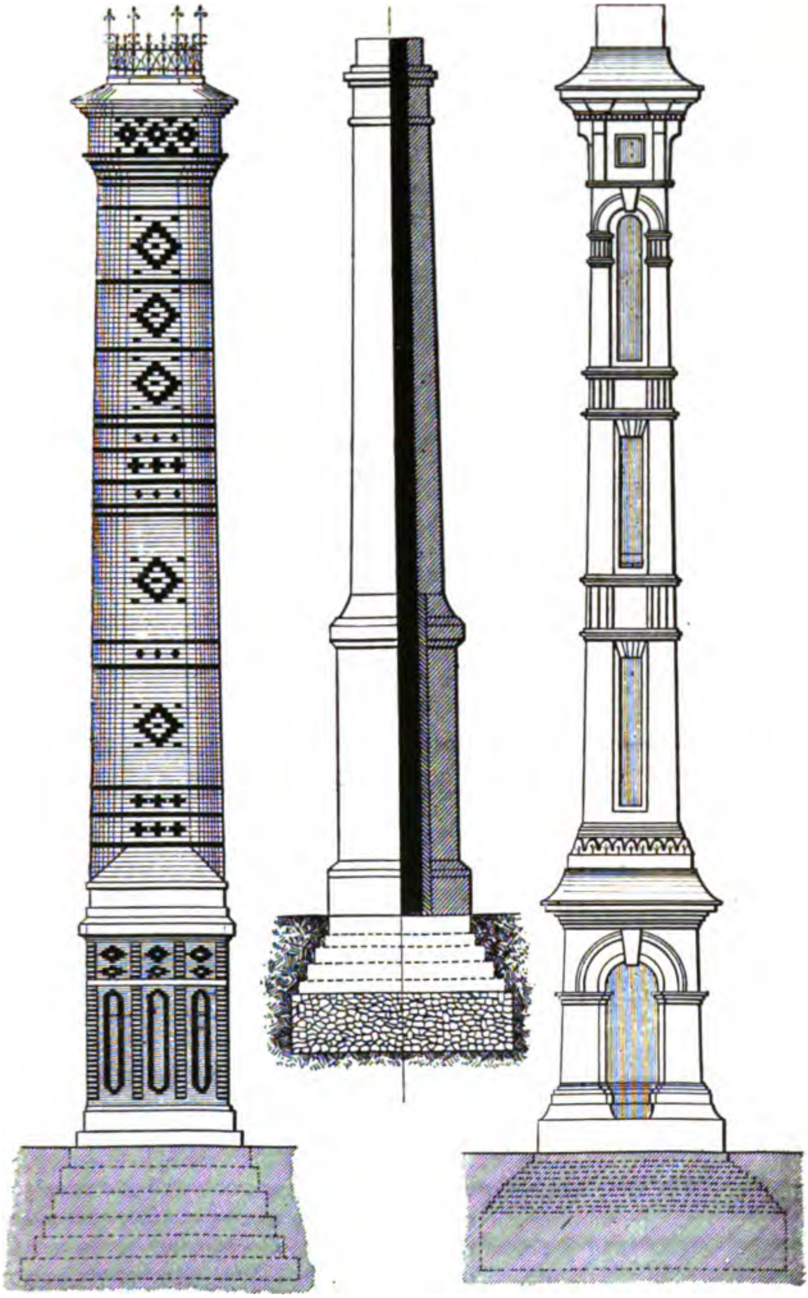


Fig. 13. Fig. 12. Fig. 14.

Figs. 12-14.—Plain and ornamental chimneys for factory steam-boilers.

TABLE 30.—PROPORTIONS OF CHIMNEYS FOR FACTORY-STEAM-BOILERS, COLLATED FROM PRACTICE.

Height of Chimney above the ground in Feet.	Internal Dimensions. Size of Base at the Ground-line. Feet	Inches.	Size of Top. Feet.	Inches.	Internal area-ratio of bottom to top.	Thickness of Walls. Thickness at base, in inches, at ground line.	Thickness at the top, in inches.
40	2	6	1	9 Sq.	2·04	18	9
60	2	11	2	0 Sq.	2·12	18	9
70	3	4	2	3 Sq.	2·13	23	9
80	3	8	2	6 Sq.	2·18	28	9
90	4	0	2	9 Sq.	2·27	28	9
100	4	8	3	0 Diam.	2·40	28	9
110	4	10	3	3 Diam.	2·33	28	9
120	5	6	3	6 Diam.	2·40	28	9
135	6	0	4	0 Diam.	2·30	28	9
150	4	6	3	0 Diam.	2·25	28	14
155	6	0	4	6 Diam.	1·78	56	14
160	9	0	5	0 Sq.	3·24	36	14
170	7	6	5	0 Diam.	2·25	36	14
180	6	4	4	6 Diam.	2·00	54	14
200	5	3	3	6 Diam.	2·28	36	14
225	16	0	6	6 Sq.	4·00	36	14
250	19	0	13	0 Diam.	2·13	40	14
300	14	0	9	0 Diam.	2·42	48	14
450	21	6	10	2 Diam.	4·35	59	14

Factory-chimneys have in some cases been constructed of concrete, but it is liable to be cracked by heat, and it is not a suitable material for this purpose.

Stability of Factory-Chimneys.—The stability of a well-built chimney to resist being overturned by the force of wind may be considered to depend only upon its weight, and to be independent of the tenacity of the mortar. The inclination of the sides of the chimney, or the batter, is so small that its influence may be neglected in calculating the force of the wind on factory-chimneys. The pressure of the wind is greater on a square than on a round chimney of equal width and height.

The maximum pressure of the wind may be assumed to be 56 pounds per square foot on a square chimney, and the pressure on other forms of section is as follows:—

					lbs.	lbs. per sq. ft.
Pressure of wind on a chimney		of square section	=	1·00	= 56 × 1·00	= 56
do.	do.	of hexagonal „	=	·75	= 56 × ·75	= 42
do.	do.	of octagonal „	=	·65	= 56 × ·65	= 36·4
do.	do.	of circular „	=	·55	= 56 × ·55	= 30·8

Weight of Factory-Chimneys.—The weight of a chimney of a given

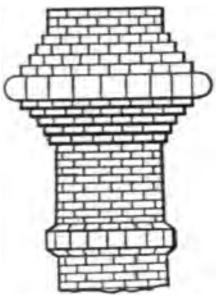


Fig. 16.—Chimney-cap.

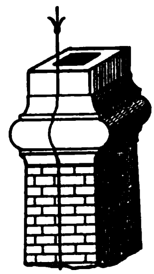


Fig. 17.—Chimney-cap

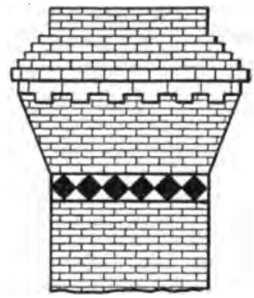
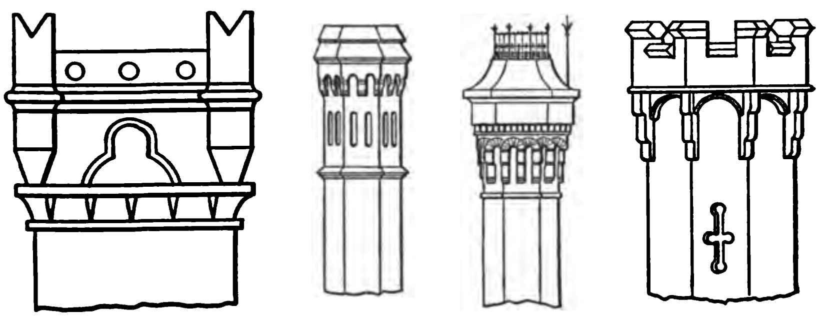


Fig. 18.—Chimney-cap.



Figs. 19–22.—Ornamental caps of chimneys.

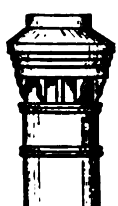


Fig. 23. Chimney-cap.

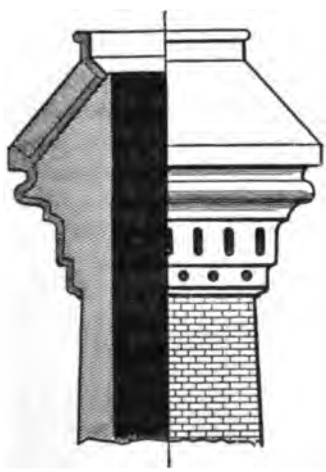


Fig. 24.—Cast-iron cap of chimney.

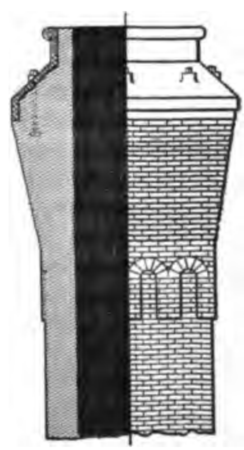


Fig. 25.—Cast-iron cap of chimney.

Figs. 16-25.—Plain and ornamental designs in brick, stone, and cast-iron, for the caps of chimneys for factory-steam-boilers.

height and width necessary to withstand the pressure of the wind may be found by the following formula:—

Let B = the external width of the chimney at the base in feet.
b = the mean external width of the chimney in feet.
H = the height of the chimney from the base in feet.
P = the pressure of the wind per square foot of surface, varying with the form of the cross-section of the chimney as given previously.
W = the total weight of the chimney in pounds necessary to withstand the pressure of the wind.

$$W = \frac{H^2 \times b}{B} \times P.$$

Example: Required the weight of a factory-chimney, necessary to withstand wind-pressure, of square cross-section, 120 feet in height, 12 feet in width at the base, and 10 feet in mean width.

$$\text{Then } \frac{120 \times 120 \text{ feet high} \times 10 \text{ feet mean width}}{12 \text{ feet width of base}} \times 56 = 672000 \text{ lbs.},$$

the total weight of the chimney necessary to withstand the wind-pressure.

The weight of brickwork averages 112 lbs. per cubic foot, and the mean thickness of the chimney should be =

$$\frac{672000\text{, the weight of the chimney}}{120 \text{ feet high} \times 10 \text{ feet mean width} \times 112 \text{ lbs.} \times 4 \text{ sides}} = 1{\cdot}25 \text{ feet},$$

or $1{\cdot}25 \times 12 = 15$ inches, the mean thickness of each side of the chimney, required to resist the pressure of the wind.

The weight of a round chimney of the same proportions, to resist the same wind-pressure, only requires to be = 672000 lbs. × ·55 = 369600 lbs.

Caps of Factory-Chimneys.—The caps or tops of chimneys are generally formed of ornamental stonework, brickwork, or cast-iron castings. Several designs for chimney-caps are shown in Figs. 16–25. The cap of the chimney shown in Fig. 24 is an iron-casting made in segments and bolted together. The top portion of the casting is formed with a trough, which is filled with brickwork and finished with a coat of cement. Another form of cast-iron cap for a chimney is shown in Fig. 25. It is formed in segments, bolted together with inside-lugs.

Wrought-iron Chimneys.—Chimneys formed of belts of wrought-iron plates from $\frac{1}{4}$ to $\frac{3}{8}$ inches thick, with lap-joints riveted together, are sometimes used for temporary work. A wrought-iron chimney is shown in Fig. 26. It is provided with a fire-brick lining, 9 inches thick, for one-half its height, and a stock-brick lining $4\frac{1}{2}$ inches thick for the remainder of its height. The bottom of the chimney is riveted to a cast-iron base-plate, bolted on a foundation of brickwork. The top of the chimney is formed with an ornamental cap of cast-iron riveted to the plates. To resist the action of the wind, steel-wire-guy-ropes may be attached to lugs riveted to the second or third belt of plates from the top of the chimney.

When the diameter of an iron-chimney is too small to permit it being conveniently lined with brickwork, it may be lined with earthenware pipes rammed behind with clay.

Lightning Conductors.—A lightning conductor, formed either of a tape of copper 1½ inches wide and ⅛ inch thick, or of a ⅜ inch copper wire-rope, should be fixed with fastenings spaced about 6 feet apart on the outside of a factory chimney. The bottom end of the conductor should run into damp earth a distance of not less than 7 yards, or it may terminate in a well or water-pipe. When the end of the conductor cannot terminate in permanently damp earth, it should be placed in a pit filled with coke and small pieces of cork, provided with inlets for the percolation of rain-water.

Chimneys for Marine Return-Tube Boilers.—The area of the cross-section of the funnel or chimney of a return-tube boiler may be from one-third to one-fifth the area of the fire-grate surface. The maximum height of a funnel above the fire-grate is 95 feet, and the average height is from 60 to 70 feet.

Chimneys for Vertical Boilers.—The area of the cross-section of chimneys of vertical cross-tube boilers, and of vertical multitubular boilers, varies in practice from one-tenth to one-sixteenth the area of the fire-grate.

Chimneys for Portable-Engine Boilers.—The area of the cross-section of the chimney of a portable-engine boiler of the locomotive type should be from one-tenth to one-twelfth the area of the fire-grate.

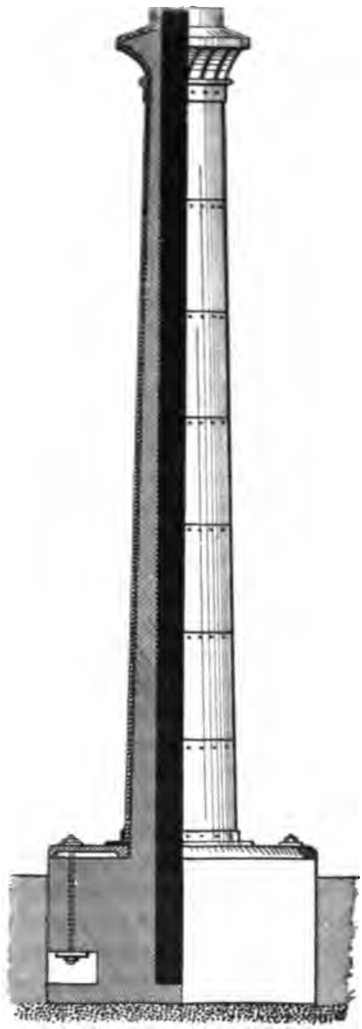


Fig. 26.—Factory chimney constructed of wrought-iron plates.

Chimneys for Locomotive-Engine Boilers.—The area of the cross-section of the chimney of a locomotive-engine boiler should be from one-eleventh to one-twelfth the area of the fire-grate. The height of the chimney from the top of the rails is from 13 feet to 13 feet 4 inches.

Steam-blast in Chimneys.—The exhaust steam-blast is a very efficient means of improving the draught of a chimney. It drives the column of gas before it in the chimney and causes a partial vacuum, resulting in air being forced through the fire and tubes, and conducing to rapid combustion of the fuel. When exhaust-steam is discharged into the chimney of a multitubular boiler, the nozzle should be level with the bottom of the top row of the tubes.

The area of the nozzle may be found by the following formula:—

$$\text{Area of blast-nozzle of exhaust-pipe in square inches} = \frac{\text{Area of fire-grate in square inches,}}{C}$$

in which C = 130 for boilers of locomotive engines.

C = 300 for boilers, of the locomotive type, of portable engines.

For instance, the exhaust-nozzle of a locomotive boiler, having an area of fire-grate of 20 square feet, should be $= (20 \text{ square feet} \times 144) \div 130 =$ 22·15 square inches in area:

$$\text{and} = \sqrt[2]{\frac{22{\cdot}15}{{\cdot}7854}} = 5{\cdot}29 \text{ inches diameter.}$$

Exhaust-steam-blast in the chimney is only suitable for metal chimneys. Steam should not be discharged into a chimney of brick or stone, because it rapidly disintegrates the mortar and injures the walls of the chimney.

Effect of Steam-blast on Evaporation.—The effect of exhaust-steam-blast on evaporation may be measured by comparing the average consumption of coal with natural draught and with steam-blast. For instance, if the consumption of coal in a boiler with natural draught be taken at 20 pounds per square foot of fire-grate surface per hour, and that of a locomotive boiler with exhaust-steam-blast in the chimney at 80 pounds per square foot of fire-grate surface per hour: then the ratio of the effect of exhaust steam-blast to the effect of natural draught is = 80 pounds ÷ 20 pounds = 4. And if the quantity of water evaporated per pound of coal be the same in each case, the rate of evaporation is four times as great with exhaust-steam-blast as with natural draught.

Artificial or Forced Draught for Steam-Boilers.—The object of forcing the draught of the furnaces of steam-boilers by mechanical means, such as by fans discharging air under the fire-grate, or both above and below the grate, is to obtain more rapid generation of steam than can be obtained with the available natural draught. Forced draught is usually effected in marine boilers, either by forcing air into the furnace, or by closing the stokehold and filling it with air of a greater pressure than that of the atmosphere.

Artificial draught can be readily adjusted to effect the combustion of different kinds of fuels at different rates of combustion. It permits efficient combustion of fuel of inferior quality, and enables a steady supply of steam to be maintained, independent of climate and weather. It enables the supply of air to be properly distributed to the fuel in the furnace to effect economical combustion.

The supply of air above the fuel can be readily adjusted to effect combustion of the gases evolved by the fuel, and the supply of air below the fuel can be regulated to effect the combustion of the solid portion of the fuel, and the movement of the hot gases can be readily controlled.

Effect of Forced Draught.—The application of forced draught to a furnace affords a means of obtaining a higher rate of combustion of fuel per square foot of fire-grate surface per hour than is conveniently obtainable with natural draught. The rate of combustion obtained in practice varies with the intensity of the draught from 30 to 200 lbs. of coal per square foot of fire-grate surface per hour. A moderate rate of forced combustion is from 35 to 50 lbs. of coal per square foot of fire-grate surface per hour.

The greater rapidity of combustion with forced draught enables a smaller fire-grate to be used for the development of equal power than is necessary for natural draught.

More complete combustion, giving a higher temperature, may be obtained in a furnace with forced than with natural draught. The heating surfaces of the boiler are also more efficient, because there is a greater difference in the temperatures of the water-surface and fire-surface of the plates forming the heating surfaces. As the rate of transfer of heat varies as the difference in the temperature of the water on one side of the plate and that of the fuel gases on the other side, the greater this difference the greater the amount of heat which will pass through a unit of heating surface in a given time.

The higher rate of evaporation obtained with forced draught permits the use of smaller boilers for engines of a given indicated horse-power, than are necessary with natural draught.

Economy of Forced Draught.—The economy that may be obtained by combustion with forced draught in a steam-boiler is due to the increased rate of combustion and the increased efficiency of the heating-surfaces produced by it, resulting in increased boiler-power. The increase of power obtained depends principally upon the quantity of air brought in intimate contact with the fuel in a given time, but the power of a boiler may generally be increased from 40 to 100 per cent. by the application of well-arranged forced draught.

It is difficult to increase the power of a boiler by forcing the draught without increasing the ratio of consumption of fuel per unit of evaporation. Economy can only be effected when the quantity of air brought into intimate contact with the fuel is less in weight per pound of fuel consumed than is obtained in combustion with natural draught. To prevent waste of heat it is necessary that the heating-surfaces of the boiler be so arranged as to absorb the greater amount of heat generated in a given time by the increased rate of combustion. If these conditions do not exist, the ratio of consumption of fuel to water evaporated invariably increases with the use of forced draught.

A thick fire is necessary for economical combustion with forced draught. It should not, in a general way, be less than 10 inches thick, and it should not be allowed to burn down to a less thickness than 7 inches before stoking. A thin fire causes loss from the entrance through the fuel of an excessive supply of air. The stronger the draught the thicker must the fire be. The height between the top of the fire and the crown of the furnace should not be less than 10 inches, but rather greater.

Power Required to Drive Fans for Forcing Combustion.—The power required to drive a fan for forcing the draught in the furnace of a steam-boiler, may be found by the following formula :—

Let P = the pressure of the air delivered by the fan in pounds per square foot.

V = the volume of air at 32° Fahr. in cubic feet used per pound of fuel.

W = the weight of fuel in pounds burnt per square foot of fire-grate surface per minute.

A = the area of the fire-grate in square feet.

T = the absolute temperature of the air entering the fan in degrees Fahr.

C = the coefficient of the efficiency of the fan, which varies in practice from ·2 to ·5.

Then, the indicated horse-power required to drive a fan =

$$\frac{P \times V \times W \times A \times T}{33000 \times (461^\circ + 32^\circ) \times C}$$

The pressure of the air in pounds per square foot, is found by multiplying the pressure in inches of water by 5·196.

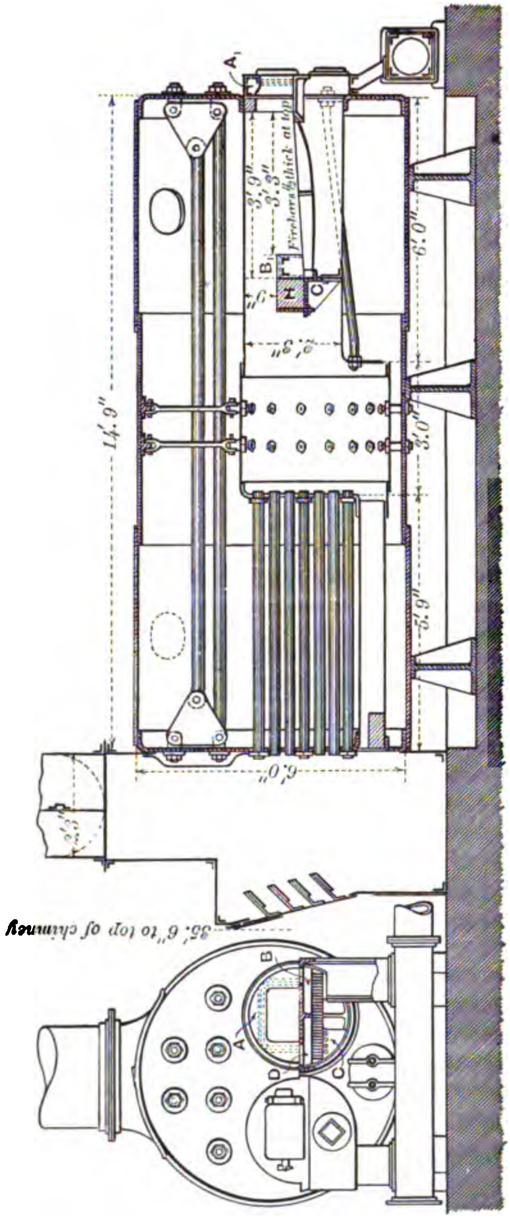
Example: Required the indicated horse-power of an engine to drive a fan to deliver air at a temperature of 69° Fahr., at a pressure of 3 inches of water: weight of coal burnt per square foot of fire grate per hour, 84 lbs.: area of fire-grate 50 square feet: air allowed for combustion 200 cubic feet per pound of coal.

Then the pressure of the air is = 3 inches × 5·196 = 15·588 lbs. per square foot: the coal burnt per square foot of fire-grate per minute is = 84 ÷ 60 = 1·4 lbs.: the absolute pressure of the air entering the fan is = 69° + 461° = 530° Fahr. The efficiency of the fan may be taken at ·5,

and $\dfrac{15{\cdot}588 \text{ lbs.} \times 200 \times 1{\cdot}4 \text{ lbs.} \times 50 \times 530^\circ}{33000 \times 493^\circ \times {\cdot}5} = 14{\cdot}21$ indicated horse-power.

If 22 indicated horse-power were developed per square foot of fire-grate surface per hour, then the power of the boiler will be = 22 × 50 = 1100 indicated horse-power, and the power absorbed in driving the fan is = (14·21 × 100) ÷ 1100 = 1·3 per cent. of the total power developed.

Heating-Surface of Steam-Boilers worked with Forced Draught.—With an efficient system of forced draught, one indicated horse-power may generally be economically developed from two square feet of the heating-surface of marine return-tube boilers fired with good coal. The proportion of heating-surface to fire-grate surface may be 45 to 1. Small tubes are more effective with forced draught than large tubes, and the smoke-tubes may be at least one-fourth less in diameter than the tubes used for natural draught.



Figs. 27 and 28.—Experimental boiler.

When thus proportioned, a boiler worked with efficient forced draught to supply steam for engines of 1000 indicated horse-power, would have $1000 \times 2 = 2000$ square feet of total heating surface, and $2000 \div 45 = 45$ square feet of fire-grate surface; and $1000 \div 45 = 22{\cdot}22$ indicated horse-power would be developed per square foot of fire-grate surface per hour.

A considerable saving of fuel may be effected by the employment of well-arranged forced draught in marine return-tube boilers, and greater power may be obtained with less size or number of boilers than with boilers having combustion with natural draught.

Experiments on Combustion with Natural and Forced Draught in the Furnace of a Steam Boiler.—Numerous evaporative experiments with natural and forced draught were made by Mr. W. G. Spence, with the boiler shown in figs. 27 and 28.* The heating-surface of the boiler is 445·9 square feet: area over the fire-bridges, 2·3 square feet: area through the tubes, 3 square feet: water-surface, 80·5 square feet: steam space, 107·25 cubic feet. There are 2 furnace-tubes 27 inches diameter, and 88 smoke tubes 2¾ inches external diameter.

Taking first the natural draught series of experiments, the firing was maintained as similar as possible in all the experiments, and the effect on the efficiency of combustion of varying the rate of admission of air above the fire-grates was noted. For this purpose each furnace-front was fitted with 44 air-holes, A, each hole being tapped ⅝ inch in diameter, so that by inserting screw-plugs, the number of open holes could be regulated or closed. The bottom part of each backbridge was fitted with 7 holes ¾ inch in diameter, C, the whole or any number of which could be stopped up by the insertion of bolts.

In the front of the fire-bridge, H, in each furnace, a box, B, was placed; these, being open at the bottom, and having each 24 holes ⅝ inch diameter in the top, allowed air to rise direct from the ash-pit to the coal-gas as it passed. These boxes were only put in place in the trials where they are shown as having been open in the last column of the following table; in the other trials they were removed, and replaced with fire-brick. Each ash-pit was closed with a sheet-iron mouthpiece.

The results of four trials with natural draught are given in the following Table.

TABLE 31.—RESULTS OF EXPERIMENTS WITH A STEAM-BOILER HAVING COMBUSTION WITH NATURAL DRAUGHT.

Length of fire-grates in inches . .	39	39	39	39
Ditto air-openings in grate-bars in inches	35	35	35	35
Area of fire-grates in square feet . .	14·625	14·625	14·625	14·625
Sq. ft. of air-opening through grate-bars.	6·08	6·08	6·08	7·05
Square feet of solid grate-bar surface .	8·545	8·545	8·545	7·575
Ratio of opening through fire-grates to total area of the fire-grate . .	$\frac{1}{2·4}$	$\frac{1}{2·4}$	$\frac{1}{2·4}$	$\frac{1}{2·08}$
Number of $\frac{9}{16}$-inch diameter holes round the fire-doors open above the fire-grate	0	16	88	88
Number of ¾-inch diameter holes open in the bottom of back bridge . .	0	13	13	13

* See a paper read by Mr. Spence before the North-East Coast Institution of Engineers and Shipbuilders.

TABLE 31 *continued.*—RESULTS OF EXPERIMENTS WITH A STEAM-BOILER HAVING COMBUSTION WITH NATURAL DRAUGHT.

Number of $\frac{5}{8}$-inch diameter holes open in the top of the bridge above the grates	0	0	0	48
Total square inches of air-opening direct to gas, above the fire-grate	0	9·7	27·56	42·25
Square-inch opening direct to gas / Square-inch opening through fire-grates	$\frac{0}{875}$	$\frac{9·7}{875}$	$\frac{27·56}{875}$	$\frac{42·25}{1015}$
Ratio of opening direct to gas to opening through the fire-grates	$\frac{0}{875}$	$\frac{1}{90·2}$	$\frac{1}{31·7}$	$\frac{1}{24}$
Air used per pound of coal in cubic feet	162·3	224·6	243	243·3
Ditto ditto pounds	12·25	16·83	18·17	18·46
Temperature of the air entering the ash-pits in degrees Fahr.	69	71	73	62
Highest temperature in the uptake in degrees Fahr.	857	810	818	737
Lowest temperature in the uptake in degrees Fahr.	725	707	722	657
Duration of emission of dense black smoke after firing in minutes	3½	3	½	0
Total duration of the emission of smoke after firing in minutes	8	7	5¼	4
Coal consumed per hour in pounds	256·6	272·7	271·3	257
Ditto square foot of fire-grate per hour in pounds	17·54	18·64	18·55	17·39
Percentage of ashes	1·22	1·48	2·1	2·23
Square feet of heating-surface per pound of coal consumed per hour	1·73	1·635	1·64	1·74
Pressure of steam in the boiler in pounds per square inch above the atmosphere	55	55	55	55
Water evaporated per hour in pounds from and at 212° Fahr.	2354	2581	2710	2637
Water evaporated per square foot of heating surface in pounds per hour, from and at 212° Fahr.	5·27	5·78	6·07	5·91
Water evaporated per square foot of fire-grate in pounds per hour, from and at 212° Fahr.	160·9	176·4	185·3	180·4
Water evaporated per pound of coal in pounds, from and at 212° Fahr.	9·16	9·46	10·01	10·25
Efficiency, calorific value of coal being 14·1 pounds	·65	·671	·71	·727

It was concluded from these experiments on combustion with natural draught, that it is impossible to obtain satisfactory combustion in furnaces having the whole air-supply drawn up from the ash-pit through the fire only.

That, with any arrangement of fire-grate similar to that used in these experiments, a collective area of not less than from one-twentieth to one-twenty-fourth of the open space between the fire-bars, should be provided for the admission of air direct to the gases. That this area should be made up from holes of $\frac{5}{8}$ inch diameter, or not exceeding $\frac{3}{4}$ inch diameter, spread over the furnace-front, door, and bottom of backbridge.

That, even with this provision for air-supply, with the thickness of fire necessary in practice, it is difficult to obtain a rate of supply of air exceeding 20 lbs. of air per pound of coal, and that the air supply is more liable to be deficient than excessive.

That, with a sufficient and properly distributed supply of air direct to the gases, the same rate of evaporation can be obtained from a boiler as with the supply of air passing up through the fire only. That the consumption of coal will be reduced about 10 per cent., and the smoke will be diminished 50 per cent. in intensity and duration.

Forced Draught, Experiments.—The forced draught experiments by Mr. Spence were made with the same boiler that was used for the natural draught experiments, but an alteration was made in the fire-grate. The level of the fire-grate was lowered to 5 inches below the centre-line of

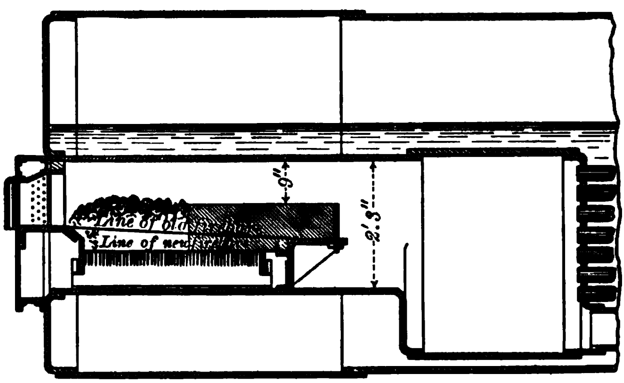


Fig. 29.—Furnace of boiler used in experiments on forced draughts.

the furnace-tube, as shown in fig. 29, in order to obtain a thicker fire, and the length of the fire-grate was reduced. The fire-bars for conveyance were placed across the furnace, they were $\frac{3}{8}$ inch thick, with air-spaces one-sixteenth inch wide, forming a total open space between the bars of one-seventh the total area of the fire-grate, instead of $\frac{1}{2 \cdot 4}$ as used for natural draught.

The results of three experiments on combustion with forced draught with cold-air, and of one experiment with hot-air, are given in the following Table.

TABLE 32.—RESULTS OF EXPERIMENTS WITH A STEAM-BOILER HAVING COMBUSTION WITH FORCED DRAUGHT.

Particulars.	Forced Draught with Cold Air.			Forced Draught with Hot Air.
Length of fire-grates in inches	21	17	21	17
Area of fire-grate in square feet	7·218	5·83	7·218	5·83
Sq. ft. of air-opening through grate-bars	1·03	·83	1·03	·83
Square feet of solid grate-bar surface	6·188	5	6·188	5
Ratio of opening through fire-grates to total area of the fire-grate	$\frac{1}{7}$	$\frac{1}{7}$	$\frac{1}{7}$	$\frac{1}{7}$
Number of $\frac{9}{16}$-inch diameter holes round the fire-doors open above the fire-grate	0	48	58	88
Air-opening direct to gas, in square inches	0	11·9	14·38	21·8
Square-inch opening direct to gas	0	11·9	14·38	21·8
Square-inch opening through fire-grates	148·3	119·5	148·3	119·5
Ratio of opening direct to gas to opening through the fire-grates	0	$\frac{1}{10}$	$\frac{1}{10·3}$	$\frac{1}{5·48}$
Pressure of air above the fire-grate in inches of water	·875	1·66	·937	1·025
Pressure of air in the ash-pits	·4	·66	·4	·438
Air used per pound of coal in cubic feet	224·6	280·9	266·7	351
Air used per pound of coal in pounds	17·1	21·86	20·3	19·9
Temperature of the air entering the ash-pits in degrees Fahr.	62	49	62	242
Highest temperature of the uptake in degrees Fahr.	787	780	813	698
Lowest temperature of the uptake in degrees Fahr.	725	690	745	614
Duration of emission of dense black smoke after firing in minutes	3½	0	0	0
Total duration of the emission of smoke after firing in minutes	8	3½	4	3½
Coal consumed per hour in pounds	285·5	277·2	267·1	227·5
Ditto per square foot of fire-grate per hour, in pounds	39·5	47·5	37	39
Square feet of heating-surface per pound of coal consumed per hour	1·56	1·6	1·66	1·96
Pressure of steam in the boiler in pounds per square inch above the atmosphere	55	55	55	55
Water evaporated per hour, in pounds, from and at 212° Fahr.	2688	2873	2853	2517
Water evaporated per square foot of heating-surface, in pounds per hour, from and at 212° Fahr.	6·02	6·44	6·395	5·64

TABLE 32 *continued.*—RESULTS OF EXPERIMENTS WITH A STEAM-BOILER HAVING COMBUSTION WITH FORCED DRAUGHT.

Particulars.	Forced Draught with Cold Air.			Forced Draught with Hot Air.
Water evaporated per square foot of fire-grate, in pounds per hour, from and at 212° Fahr.	372·3	492·7	395·3	431·8
Water evaporated per pound of coal in pounds, from and at 212° Fahr.	9·41	10·35	10·67	11·06
Efficiency, calorific value of coal being 14·1 pounds	·667	·734	·757	·784

In the trial, giving the results stated in the first column of the above table, no air was admitted direct to the gases, the water evaporated per pound of coal was 9·41 pounds, and the efficiency is ·667. The results obtained from the natural draught experiment, with no air admitted direct to the gases, were from Table 31, = an evaporation of 9·16 pounds of water per pound of coal and an efficiency of ·65, showing a gain of:—

$$\frac{\cdot667 - \cdot65 \times 100}{\cdot65} =$$

2·6 per cent. from the use of forced draught.

In the trial giving the results stated in the third column of the above table, air was admitted direct to the gases through openings equal a little greater than one-tenth of the total open space between the grate-bars, and 10·67 pounds of water were evaporated per pound of coal, the efficiency being ·757. Comparing this with the best results obtained with natural draught, given in the fourth column of Table 31, when 10·25 pounds of water were evaporated per pound of coal and an efficiency obtained = ·727. It shows

$$\frac{\cdot757 - \cdot727 \times 100}{\cdot727} = 4\cdot12 \text{ per cent. in favour of forced combustion.}$$

In the forced draught trial with hot air, the results of which are given in the fourth column of the above Table, the water evaporated per pound of coal was 11·06 pounds and the efficiency ·784.

Comparing this with the highest efficiency obtained with forced draught it shows an increase in efficiency,

$$= \frac{\cdot784 - \cdot757 \times 100}{\cdot757} = 3\cdot55 \text{ per cent.}$$

due to the employment of hot air.

It was concluded from these experiments on combustion with forced draught, that a considerable area for air direct to the gases must be supplied to ensure efficient combustion and reduce smoke.

That, in designing a fire-grate for forced draught, the ratio of open air-space between the fire-bars to total fire-grate surface should be much less than is necessary for natural draught.

That a moderate air-pressure of about ·35 inch of water in the ash-pit, giving a rate of combustion of about 35 pounds of coal per square foot of fire-grate per hour, is more economical than greater air pressure with increased rates of combustion.

That, by the use of moderate forced draught, a higher efficiency of combustion is obtainable than by using natural draught only.

Various Systems of Forced Draught.—Several systems of forced draught have been applied to marine-boilers, some of which may be briefly described as follows:—

Howden's System of Forced Draught.—In Howden's system of forced draught, the air supplied to the furnace is heated by passing through a series of tubes placed in the path of the escaping products of combustion, thus utilising waste heat. The ash-pit is closed and the hot air is supplied to the fire both above and below the fire-grate. The pressure of the air in the ash-pit is from $\frac{5}{8}$ to 1 inch of water, according to the nature of the coal used.

With this system of forced draught from 20 to 26 indicated horse-power per hour have been developed per square foot of fire-grate surface of the boiler, with a coal consumption of 1·25 pounds per indicated horse-power per hour.

Comparing this with 12 indicated horse-power per square foot of fire-grate surface per hour, the average maximum power attainable in marine boilers with natural draught, the increase of power obtainable with this system of forced draught is,

$$= \frac{26 - 12 \times 100}{12} = 116 \text{ per cent.}$$

on the power with natural draught.

Ferrando System of Forced Draught.—In the Ferrando system of forced draught the fire-bars are placed transversely in the furnace, they are narrow, being $\frac{7}{16}$ inch thick, and the air-spaces between them are $\frac{1}{16}$ inch wide. The ash-pit is closed and the air required for combustion is supplied at a pressure equivalent to a head of $\frac{3}{4}$ inch of water. This system of forced draught is very efficient for burning small and inferior kinds of coal.

Fothergill's System of Forced Draught.—In Fothergill's system of forced draught, air is supplied to the furnace and also at the back of the combustion-chamber to assist the combustion of the fuel-gases. The ash-pit is closed, and air is forced into it at a pressure of ·5 to ·7 inch of water. The advantage claimed for this system of forced draught is that it will effect a saving of about 20 per cent. in fuel.

Closed-stokehold System of Forced Draught.—In this system of forced draught the stokeholds are closed and filled with air by fans, all the air being compelled to pass through the furnaces of the boilers. The following figures shewing the indicated horse-power per square foot of fire-grate surface have been obtained with this system of forced combustion :—

Air-pressure in stokehold in inches of water	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$
Indicated horse-power developed per square foot of fire-grate surface per hour	15	16	17	18	19	20	23

Comparing the highest power in this table with the average maximum power attainable in marine boilers with natural draught, the increase of power obtainable with this system of forced combustion is :—

$$= \frac{23 - 12 \times 100}{12}$$

= 92 per cent. on the power obtained with natural draught.

Forced Combustion by Induced Draught.—In Martin's system of induced draught, fans are placed in the base of the funnel of a marine-boiler, by means of which the air is drawn through the furnace and tubes of the boiler. Some experiments were made with this system of induced draught on a locomotive type of boiler, used as a stationary boiler, the results of which are as follows :—

Area of fire-grate in square feet	16·32	19·8	19·80
Coal consumed per square foot of fire-grate in lbs	51·50	39·0	40·50
Water evaporated per pound of coal from and at 212° Fahr. in pounds	11·75	10·7	11·49
Pounds of water evaporated per square foot of heating surface from and at 212° Fahr. in pounds	10·73	9·0	10·38
Indicated horse-power per hour per square foot of fire-grate	25·73	17·87	19·94
Indicated horse-power of the boiler	420	354	395

These experiments show that a higher power may be obtained from a boiler having combustion with induced draught on this principle than can be obtained with natural draught.

Feed-water for Steam-boilers.—Water is never found absolutely pure in nature, because, being a powerful solvent, it becomes more or less contaminated by foreign substances. The purest water that can be found in a natural state is obtained by melting snow collected in clean vessels at a great distance from houses, far beyond the reach of smoke. Rain-water collected in the same manner is also very pure. The only impurities contained are those absorbed by the snow or rain in falling through the atmosphere.

All varieties of water, rising in the form of springs or flowing on the surface of the earth, contain impurities or mineral salts which the water has dissolved from substances with which it has been brought in contact, which render the water more or less hard.

Soap dissolves easily and lathers freely in soft water, but decomposes and makes little or no lather in hard water.

Composition of Water.—Water is a chemical combination of two gases, oxygen and hydrogen, in the following proportions:—

	Weight.	Weight.	By Dulong.
H 2	2	11·111	11·1
O	16	88·888	88·9
H 2 O	18	100	100

The proportion of these constituents is always precisely the same in pure water, namely, eight parts by weight of oxygen and one of hydrogen. Water is a neutral compound, and when pure is absolutely free from acidity and alkalinity.

Water absorbs air and gases in different proportions. It greedily absorbs carbonic-acid gas, and all kinds of water in a natural state contain a small quantity of air and carbonic-acid gas which they have absorbed. These may be driven off by boiling, but if the boiled water be again exposed to the atmosphere it will absorb air and gases again. One sample of 98 cubic inches of water from a spring contained two cubic inches of air, which consisted of 10 per cent. of carbonic acid, and 88 per cent. of oxygen and nitrogen.

Water obtained from freshly-fallen snow contains little or no air. Rain-water generally contains at least from three to four per cent. of carbonic-acid gas. Water containing air and carbonic-acid gas corrodes metals.

Rain-Water is the purest of all natural water when not contaminated during or at its fall. Its purity depends upon the atmosphere of the locality in which it falls, and the state of the surfaces which collect it. It contains in solution such gaseous substances and organic or other matter floating in the air, as it may have dissolved in falling through the atmosphere. From its purity its solvent power is greater than that of other natural waters. The respiration of all animals, decaying dead animal and vegetable matter, and the combustion of fuel, continually evolve acid and sulphurous gases into the atmosphere. The gases arising from decomposition are chiefly carbonic-acid, nitrous, and nitric acid, chlorine and ammonia. These are all soluble in water, and are freely absorbed by mists and showers. Rain-water should be collected and stored in districts where there is a scarcity of water, but it should be purified before being used. Rain-water frequently contains so many impurities, that when used as feed-water for boilers, it rapidly corrodes them.

River-water is composed partly of spring-water, and partly of rain-water or surface-water. River-water is of variable composition, but it is generally soft, and is next in purity to rain-water. It generally contains more organic matter than spring-water, and frequently has a considerable quantity of matter, such as clay or soil, mechanically suspended.

Water from Springs.—Waters from springs hold in solution every soluble mineral contained in the rocks or earth through which they flow. Therefore spring-water frequently contains many impurities and salts in solution. The salts most frequently found in water from springs are sulphate of lime, carbonate of lime, sulphate of magnesia, chloride of sodium, alum, and salts of iron. These salts consist of an acid united to a substance. For instance, sulphate of lime consists of sulphuric-acid and lime, carbonate of lime of carbonic-acid and lime, and sulphate of magnesia of sulphuric-acid and magnesia. Water is termed hard or soft according as the salts are present in a greater or less quantity. Sulphate of lime is the most general cause of the hardness of ordinary spring-water.

Water containing carbonates of lime and magnesia is temporarily hard, and may be softened by boiling. Water containing sulphates of lime and magnesia is permanently hard. These minerals are released or separated from the water when it is heated to a temperature not less than 90° degrees above the boiling point of fresh water.

Water may be purified of foreign substances held in mechanical suspension by filtration, and of mineral substances held in solution by heating it for some time to a high temperature. The minerals may also be removed by chemical treatment. Water for steam-boilers should be as pure as it is possible to obtain it, or as it is practicable to make it, by removal of the mineral substances before it enters the boiler.

Weight of Pure Water.—The weight of pure water at a temperature of 62° Fahr. is usually taken at 62·355 lbs. per cubic foot, and at ·03608 lb. or 252·595 grains per cubic inch. It has, however, been accurately determined that, one cubic inch of distilled water freed from air, and then weighed against brass-weights in air at a temperature of 62° Fahr., the barometer being at 30 inches, weighs 252·286 grains.

Sea-Water.—The composition of sea-water varies in different localities. The water of the English Channel contains less salts than that of mid-ocean. The composition of sea-water from the English Channel, near Hastings, is approximately as follows:—

Water	96·44	parts.
Sodium chloride	2·74	"
Potassium chloride	·08	"
Magnesium chloride	·37	"
Magnesium sulphate	·23	"
Calcium sulphate	·14	"
	100·00	

The quantity of salts held in solution in sea-water varies considerably. The Atlantic Ocean contains 4 per cent., the Mediterranean $3\frac{1}{2}$ per cent., the German Ocean 3 per cent., the Caspian Sea $1\frac{1}{2}$ per cent.

Sea-water, in a general way, contains about $\frac{1}{33}$ part of its own weight in salt. Each $\frac{1}{33}$ part of salt is termed a degree of salt, and it increases the boiling point of fresh water to the extent of 1·2° Fahr. Each degree of salt represents five ounces of salt per gallon of sea-water. For instance, the boiling point of sea-water of 2 degrees of saltness is $= 1{\cdot}2 \times 2 = 2{\cdot}4 + 212° = 214{\cdot}4°$ Fahr., and the water contains 5 ounces $\times$ 2 = 10 ounces of salt per gallon. One gallon of sea-water at 62° Fahr. weighs 10·25 pounds.

Effect of Heat on Water.—Heat is most effective when applied at the lowest part of the liquid to be heated. There is no change of temperature in liquids under ordinary conditions without causing a displacement of particles. If heat be applied under a boiler containing water, the molecular motion of the heated plates is transmitted to the water in contact with them, and the particles of water near the bottom of the boiler being heated first, they expand, become specifically lighter and ascend. Colder particles immediately occupy their place, become heated, and ascend in their turn. In this way a current is established, the heated particles continually rising up through the centre of the mass of water, and colder particles descending at the sides.

The heat is not conducted from particle to particle without displacement, as in the case of solids, but each particle as fast as it receives a fresh accession of heat starts off with it and conveys it to a distance, displacing other and colder particles in its progress. Therefore there are two distinctly separate currents moving in opposite directions. The upward self-rising current consists of a mixture of steam-bubbles and hot water; the downward current consists of cooler water only. These currents are called convection-currents; they rise at the hottest parts and descend at the cooler parts of a boiler.

It is essential to provide ample area in the water-spaces of steam-boilers for both ascending and descending currents, to allow complete separation of the currents, without which circulation cannot take place. When steam has to escape through a narrow channel, such, for instance, as from tubes hanging down into the fire, having their lower ends closed, or pendant water-tubes, it is necessary to provide separate passages for the ascending and descending currents to prevent collision of the currents, which might cause the water to be lifted from the heating-surfaces, and result in overheating and priming in a steam-boiler. This is effected by the introduction of an internal circulating tube into the pendant tube, through which the descending current of cooler water flows undisturbed by the rising steam in the ascending current of water in the outer annular space.

When heat is applied to the side of a liquid, as in the case of an internally heated vertical tube, surrounded by water, say a tube of a vertical steam-boiler, the bubbles of steam have a tendency to cling to the tube, and the steam generated at the lower end of the tube creeps up the surface and tends to keep the water from contact with the metal, and if the heat be very intense, it may lead to overheating and distortion of the tube.

Expansion of Water by Heat.—The weight of a gallon of water is frequently taken at 10 pounds irrespective of temperature, but this leads to serious errors in calculating the weight of hot feed-water. The application of heat to water causes it to expand, and the volume increases with each degree of rise of temperature above the point of maximum density, which is 7·1° above the freezing point, or 39·1 Fahr.

The volume and weight of water at different temperatures, compared with water at 32° Fahr., are given approximately in the following Table, which is useful for calculating the weight of hot feed-water for steam-boilers up to 210° Fahr.

TABLE 33.—EXPANSION AND APPROXIMATE WEIGHT OF PURE WATER AT DIFFERENT TEMPERATURES BETWEEN 32° AND 210° FAHR.

Temperature of Water in Degrees Fahr.	Volume Compared with that of Water at 32° Fahr.	Weight Compared with that of Water at 32° Fahr.	Weight of One Cubic Foot in Pounds.	Weight of One Gallon in Pounds.
deg.				
32	1·00000	1·00000	62·418	10·010
39	·99988	1·00010	62·425	10·011
40	·99988	1·00010	62·425	10·011
50	1·00012	·99983	62·409	10·008
60	1·00075	·99924	62·372	10·005
62	1·00100	·99898	62·355	10·000
70	1·00159	·99830	62·311	9·993
80	1·00298	·99700	62·231	9·980
90	1·00457	·99540	62·133	9·964
100	1·00640	·99362	62·022	9·947
110	1·00887	·99117	61·865	9·922
120	1·01136	·98873	61·716	9·896
130	1·01388	·98628	61·562	9·872
140	1·01687	·98342	61·380	9·843
150	1·01986	·98047	61·200	9·815
160	1·02337	·97713	60·990	9·780
170	1·02686	·97378	60·780	9·747
180	1·03094	·97003	60·546	9·710
190	1·03495	·96630	60·312	9·670
200	1·03883	·96254	60·080	9·634
210	1·04310	·95870	59·847	9·602

The volume and weight of distilled water at different temperatures above 212° Fahr., and the corresponding pressure of steam, are given in the following Table by Mr. R. H. Buel, in which the volume is compared with the volume at the temperature of maximum density, 39·1° Fahr. By means of this Table the volume and weight of water in a steam-boiler may be accurately calculated.

TABLE 34.—VOLUME AND WEIGHT OF DISTILLED WATER AT DIFFERENT TEMPERATURES ABOVE 212° FAHR.

Temperature of Water in Degrees Fahr.	Corresponding Pressure of Steam above the Atmosphere in pounds per square inch.	Relative Volume compared with the Volume at the Temperature of Maximum Density.	Weight of One Cubic Foot in pounds.	Weight of One Gallon in pounds.
°				
212	0	1·0434	59·828	9·595
215·4	1	1·0449	59·741	9·581
218·6	2	1·0464	59·659	9·567
221·6	3	1·0477	59·580	9·555
224·5	4	1·0491	59·503	9·542
227·2	5	1·0504	59·431	9·531
229·8	6	1·0516	59·361	9·520
232·3	7	1·0528	59·293	9·509
234·7	8	1·0540	59·227	9·498
237·1	9	1·0552	59·161	9·487
239·4	10	1·0563	59·097	9·477
241·6	11	1·0574	59·035	9·467
243·7	12	1·0585	58·975	9·458
247·8	14	1·0606	58·858	9·438
251·6	16	1·0626	58·748	9·421
255·2	18	1·0645	58·643	9·404
258·6	20	1·0663	58·541	9·388
266·5	25	1·0707	58·304	9·350
273·9	30	1·0748	58·078	9·314
280·6	35	1·0787	57·869	9·280
286·5	40	1·0823	57·680	9·250
292·1	45	1·0856	57·501	9·221
297·5	50	1·0890	57·325	9·193
302·5	55	1·0921	57·162	9·167
307·2	60	1·0951	57·003	9·141
311·6	65	1·0980	56·855	9·117
315·9	70	1·1008	56·709	9·094
319·8	75	1·1034	56·571	9·072
323·8	80	1·1061	56·437	9·051
328·1	85	1·1086	56·311	9·030
331·0	90	1·1110	56·186	9·010
334·4	95	1·1134	56·065	8·991
337·7	100	1·1157	55·949	8·972
340·8	105	1·1179	55·837	8·955
343·9	110	1·1202	55·727	8·937
346·9	115	1·1223	55·620	8·920
349·8	120	1·1244	55·515	8·903
352·5	125	1·1265	55·412	8·886
355·4	130	1·1286	55·311	8·870
358·0	135	1·1306	55·214	8·854
360·7	140	1·1326	55·116	8·839

TABLE 34 *continued.*—VOLUME AND WEIGHT OF DISTILLED WATER AT DIFFERENT TEMPERATURES ABOVE 212° FAHR.

Temperature of Water in Degrees Fahr.	Corresponding Pressure of Steam above the Atmosphere in pounds per square inch.	Relative Volume compared with the Volume at the Temperature of Maximum Density.	Weight of One Cubic Foot in pounds.	Weight of One Gallon in pounds.
°				
363·2	145	1·1345	55·023	8·824
365·7	150	1·1364	54·931	8·809
368·1	155	1·1383	54·841	8·795
370·5	160	1·1401	54·752	8·781
372·8	165	1·1419	54·668	8·761
375·0	170	1·1436	54·584	8·753
377·2	175	1·1454	54·503	8·740
379·3	180	1·1471	54·422	8·727
381·4	185	1·1487	54·342	8·715
383·5	190	1·1504	54·262	8·702
385·7	195	1·1522	54·182	8·689
387·7	200	1·1538	54·102	8·676
389·7	205	1·1555	54·026	8·664
391·7	210	1·1571	53·941	8·652
393·5	215	1·1586	53·876	8·640
395·5	220	1·1602	53·803	8·628
397·3	225	1·1618	53·733	8·617
399·1	230	1·1633	53·664	8·606
400·9	235	1·1648	53·596	8·595
402·6	240	1·1662	53·529	8·584
404·2	245	1·1676	53·462	8·574
406·0	250	1·1691	53·396	8·563

Convection is the power possessed by fluids of conveying heat acquired at one place to another place. Convection is caused by currents both in air and water. Smoke ascends the chimney, and ventilation is caused by the same principle. Gaseous bodies, from the great mobility of their particles, are the most rapid conveyers, although they are the slowest conductors of heat. Any body hotter than the air heats it and sets it in motion in an upward current, which may be seen rising from highly heated bodies, and the particles which rise are immediately replaced by the influx of other particles from every side. The slightest difference in temperature is sufficient to produce these effects, hence the rapidity with which air reduces solid bodies to its own temperature.

A body colder than the air, such as a lump of ice, produces an opposite action; it cools the air in contact with it, which, becoming denser, descends in a continual stream, supplied by an influx of air from all sides to the ice, until the whole is melted.

The propagation of heat in a gaseous mass is effected by means of the ascending and descending currents formed in it, precisely as in the case of liquids.

Heat is distributed by convection through the water in a boiler. The heat imparted to the outside of the plate from the furnace is conducted through the plate, and the water in contact with it absorbs heat, expands, and rises from it, and colder water immediately descends and occupies its place. Water has great heat-absorbing capacity and will absorb the most intense heat, without the furnace-plates becoming overheated, when it is in contact with, and freely circulates over clean metal, forming the water-heating surfaces. Heat can only be distributed in liquids by convection-currents, which convey the heat from its source to a point at a higher level.

Stagnant water is incapable of distributing heat. Water, being a bad conductor of heat, can only be warmed very slowly by conduction. Owing to the low conducting power of water, the application of heat to its upper surface is of no effect in heating the mass of water beneath. To effect the efficient distribution of heat in water, it is necessary to have free circulation.

When heat is applied to the surface of water it is not diffused by convection, but creeps very slowly downwards by conduction. An inflammable liquid floating on water may be burnt without raising the temperature of the water one degree, whilst developing sufficient heat to evaporate the whole mass if applied underneath.

Water placed underneath a hot tube, on being heated and becoming lighter cannot ascend, but clings to the surface above it, and diffuses little or no heat downwards. For instance, in Cornish and Lancashire-boilers the water underneath the furnace-tubes is frequently comparatively cold some time after the steam has been raised. It will be seen from this that the lower halves of horizontal flue-tubes are incapable of transmitting much heat to water, and that the under portions of the internal flue-tubes of boilers are almost of no value for raising steam.

Circulation of Water in a Steam-Boiler, is the natural flow of water, caused by difference of density of the water, due partly to difference of temperature, but principally to the action of steam-bubbles which lighten the water. The upward self-rising current of steam-bubbles from the heating surfaces is accompanied by a downward current of water to supply its place, which causes the particles of water to continually change their position, and flow or circulate over the water-heating surfaces. Hence, the action of circulation is due to gravity.

Heat can only be effectively abstracted by liquids from heated surfaces by circulation of the liquid, and rapid circulation is essential to rapid abstraction of heat. When circulation is perfect, the steam-bubbles are detached from the heating surfaces as fast as they are formed, and the maximum economical effect is obtained from the fuel.

Circulation is a more important factor in the distribution of heat, than either the nature or thickness of the metal through which the heat is transmitted. The quantity of heat which can be transmitted to water in a given time, is only limited by the rate at which it can be carried away from the heating surfaces by the convection-currents.

The rate of circulation depends upon the rapidity of the liberation of the steam-bubbles from the heating-surfaces, and the rate of the convection-currents. Circulation is assisted by agitation of the liquid, by stirring, or shaking, which probably partly accounts for the high evaporative efficiency of many locomotive boilers.

As the efficiency of a steam-boiler depends greatly upon the efficiency of its circulation, the heating-surfaces should be arranged to facilitate the movement of the convection-currents, and promote free circulation. This is effected in the best manner when the heat is applied underneath the water to be heated, and when the shape and position of the heating-surfaces facilitate the free escape of the heated water in its upward current, and the return of the cooler water in its downward current.

Free circulation is essential to economy and safety in a steam-boiler. When circulation is impeded overheating may take place. It is essential to secure unimpeded circulation at the furnace-crowns of steam-boilers, by providing ample space for the steam to rise without obstruction to the steam-space. When tubes are arranged over a furnace-crown, ample space between the tubes should be provided for the escape of the rising steam.

The efficiency of the circulation of tubular boilers depends greatly upon the tubes being properly arranged to allow free escape of the steam from, and access of water to, their surfaces.

Defective Circulation.—When steam is generated in confined spaces from which it cannot rise freely, it is liable to drive the water before it, and leave the heating-surfaces dry, resulting in priming and deteriorated or burnt metal-surfaces. Steam has a natural tendency to cling to heating-surfaces, and when water-spaces are so cramped that the ascending and descending currents cannot flow separately, circulation cannot take place, and the water is put into a state of perturbation. Therefore as the steam cannot rise through the superincumbent water as fast as it is generated, it accumulates on the heating-surfaces, forming a film between the heated plate and the water, and causes inflation of the water, which rises to a greater height in the boiler, than that due to the natural position of the water and steam. In such cases the quantity of water indicated by the water-gauge is greater than the actual quantity in the boiler.

Defective circulation may cause a portion of the heating-surface to become so highly heated as to repel the water, and become covered with a layer of spheroidal fluid, on which the mass of water will float without contact with the metal.

The Spheroidal Condition of Water, is that state observed when water is applied to a highly heated metal-plate. The water is repelled by the radiant heat and does not wet the plate, but rolls on its surface in liquid drops or flattened globules, buoyed up by, or resting on cushions of, their own vapour. The tendency of the water to become spheroidal is promoted by the presence of a coating of greasy or other deposit on the heating surfaces, which prevents contact of the water with the hot plates, and the water, covering the hot surfaces without wetting them, is maintained in a spheroidal state.

When a plate is overheated, a spheroidal layer of water, several inches thick, may be repelled and constantly supported. The temperature of the spheroidal layer is always lower than its boiling-point. It may be assumed that the temperature of a furnace-plate from which water has been repelled, will, within one minute's time become at least 1100° Fahr. and the strength of the plate will have become seriously impaired.

Vaporisation is the process by which a liquid passes into the gaseous state; or is converted into an elastic vapour capable of being compressed into less space by external pressure, and resisting the force which compressed it. Vaporisation commences the moment heat is applied to a liquid: the escape of the vapour is termed evaporation. The term vapour is generally applied to evaporation without the application of heat. Vapour produced by the direct application of heat to water is termed steam.

Latent Heat of Vaporisation is the number of units of heat absorbed by one pound of a liquid in passing to vapour. The heat thus absorbed is not sensible to the thermometer. It may be restored by condensing the vapour, or changing it to a liquid.

Ebullition is the formation in a liquid of bubbles of vapour, whose internal pressure is equal to the external pressure upon them. Hence, a liquid is in ebullition when it gives off vapour of the same tension as the atmosphere above it. When heat is applied to water, it is absorbed until the water reaches the temperature of ebullition, or the temperature whose corresponding maximum pressure is equal to the pressure on the surface of the liquid, when steam will rise from the water, and the temperature will remain stationary until the whole of the water is evaporated.

The Boiling-point of a Liquid is the temperature at which ebullition and evaporation commences. It is always the same for the same liquid under the same circumstances. The pressure exerted on the surface of a liquid being opposed to the generation and disengagement of the globules of vapour in proportion to its intensity, it follows that there are as many different boiling-points for the same liquid as there may be different pressures on its surface. When the pressure is withdrawn from the surface of a liquid by means of an air-pump, water boils at 70° Fahr., and produces steam of a pressure of about ·33 lb. per square inch. The boiling-point of water varies with the pressure of the atmosphere approximately as follows:—

Barometer Inches of Mercury.	Boiling Point in Degrees Fahr.	Barometer Inches of Mercury.	Boiling Point in Degrees Fahr.
	°		°
27	206·9	29½	211·2
27½	207·8	30	212·0
28	208·7	30¼	212·4
28½	209.5	30½	212·8
29	210·4	31	213·6

Pure water, at the level of the sea, under the pressure of the atmosphere, which is 14·7 lbs. per square inch, boils at 212° Fahr. The higher the elevation above that level, the more is the atmospheric pressure diminished and the boiling point reduced. Hence, ebullition is facilitated on high mountains, and retarded at low levels, or at the bottom of deep mines. The boiling points of a number of liquids, in open vessels, under ordinary atmospheric pressure are given in the following Table :—

TABLE 35.—BOILING POINTS OF LIQUIDS UNDER THE PRESSURE OF ONE ATMOSPHERE.

Liquid.	Temperature Fahrenheit.	Liquid.	Temperature Fahrenheit.
	°		°
Sulphuric ether	100	Sulphuric acid, 1·3 s. g.	240
Sulphuret of carbon	118	Naphtha	307
Ammonia	139	Oil of turpentine	317
Chloroform	140	Mineral oil	436
Wood spirit	150	Phosphorus	555
Alcohol	173	Olive oil	564
Benzine	177	Sulphur	570
Pure water	212	Linseed oil	596
Sea-water, ordinary	213·2	Grease, dense	620
Nitric acid	221	Sulphuric acid, 1·85 s.g.	621
Muriatic acid	223	Mercury	662

Boiling Points of Saturated Solutions of Salt.—Foreign substances in suspension in water, as lime, mud, or sand, not chemically combined with it, do not influence the position of the boiling point. But it is sensibly affected by substances in combination, or in solution. Substances more volatile than water such as ether and alcohol, when combined with water, lower the boiling point. Salts, on the contrary, raise the boiling point. The boiling points of water saturated with a number of salts which combine chemically with water, according to the experiments of Legrand, are given in Table 36.

The vapour produced at the surface of saline solutions is that of pure water, that is, the steam of salt water contains no salt. The temperature of the steam formed under the pressure of one atmosphere, is invariably 212° Fahr., although the temperature of the solution may be much higher. For instance the temperature of ebullition of a saturated solution of common salt is, from the following Table, 227·2° Fahr., but the temperature of steam produced from its surface is only 212° Fahr.

TABLE 36.—BOILING POINTS OF SATURATED SOLUTIONS OF VARIOUS SALTS UNDER THE PRESSURE OF ONE ATMOSPHERE.

Saturated Solution.	Boiling Point. Fahr.	Weight of Salt in 100 lbs. of water.	Saturated Solution.	Boiling Point. Fahr.	Weight of Salt in 100 lbs. of water.
	°	lbs.		°	lbs.
Chlorate of potash .	219·6	61·5	Nitrate of potash . .	240·6	335·1
Carbonate of soda .	220·3	48·5	Chloride of stronium	244·2	117·5
Phosphate of soda .	221·9	113·3	Nitrate of soda. .	249·8	224·8
Chloride of potassium	227·0	59·4	Acetate of soda . .	255·9	209·0
Common salt . .	227·2	41·2	Carbonate of potash .	275·0	205·0
Hydro-chlorate of ammonia . . .	237·6	88·9	Nitrate of lime . .	303·9	362·2
			Acetate of potash . .	336·2	798·2
Neutral tartrate of potash . . .	238·4	296·2	Chloride of calcium .	355·1	325·0
			Nitrate of ammonia .	356·0	unlitd.

Evaporation of Water to Steam.—When water is heated to the boiling point, the temperature remains stationary, and all subsequent additions of heat become latent and are carried off in the form of steam. If a given weight of water be evaporated to steam in a perfectly steam-tight vessel, the weight of steam produced will be exactly the same as that of the water from which it was produced, that is, one pound of water will generate exactly one pound of steam.

One pound of water at a temperature of 212° Fahr. in passing to steam of the same temperature absorbs as much heat as would have raised the temperature of the water 966° Fahr., if it had not become latent. Therefore, a unit of evaporation is the evaporation of one pound of water at 212° Fahr. to steam of the same temperature, and it is equal to 966 thermal units. The heat required to raise one pound of water from 0° to 212° Fahr., and evaporate it to steam of the same temperature is = 966° + 212° = 1178 units.

The quantity of heat required to evaporate a given quantity of water in a steam-boiler depends upon the temperature of the feed-water. If the temperature at which water is supplied to the boiler be represented by t, then, the total heat, H, required to evaporate 1 lb. of water to steam is:—

$$H = 1178 - t.$$

The quantity of heat required to generate steam increases slightly with the pressure, but the increment is so small that, for practical purposes, t is sufficiently accurate to assume that, the total heat required to generate steam is the same for all pressures. Hence, the cost of fuel for the generation of steam is practically the same for high-pressure steam as for low-pressure steam.

The rate of evaporation varies with the rate of combustion of the fuel.

Total Heat of Steam.—The total heat of steam increases with the temperature at the rate of ·305 of a thermal unit for each degree of temperature above 212° Fahr. The total heat of 1 lb. of steam at 212° Fahr. is 1146·6, and the total heat of steam at any given temperature, T, may be found by this rule :—

Total heat of steam at temperature T =

$$1146{\cdot}6 + [{\cdot}305 \times (T - 212^\circ)].$$

For instance, the temperature of steam of a pressure of 100 lbs. per square is = 337·7° Fahr., and the total heat of steam of that pressure is $= 337{\cdot}7 - 212 = 125{\cdot}7 \times {\cdot}305 = 38{\cdot}3 + 1146{\cdot}6 = 1184{\cdot}9$ units per pound of steam.

Heat Expended in the Generation of Steam.—In evaporating one pound of pure water from a temperature of 32° Fahr. to steam of 212° Fahr., or of atmospheric pressure, the water is heated through a range of 180°, and the volume is slightly increased, some heat, or about one half a unit, being absorbed in producing the expansion. The further application of heat expands the water into steam and the heat is expended in two ways, viz.—First, in overcoming the mutual attraction of the molecules of the water for each other, called the internal latent heat, and amounting to 893·8 thermal units per pound. Secondly, in overcoming the force which the surrounding medium opposes to the expansion of water into steam, called the external latent heat, and amounting to 72·2 thermal units. The sum of these two quantities = 893·8 + 72·2 is = 966 thermal units, the total heat of evaporation.

If the water had been evaporated at a greater pressure than that of the atmosphere, the water would be more highly expanded than at atmospheric pressure, and the work of separating the molecules of the water would be less and the internal latent heat would be diminished, while the pressure of evaporation being greater, the work of overcoming this pressure, and consequently the external latent heat, would be increased.

The Properties of Saturated Steam, or the pressures, volumes and quantities of heat, corresponding to different temperatures, are given in Table 37 by Mr. R. H. Buel, in which the pressure of the steam is given in lbs. per square inch above the atmosphere, or as shown by the steam-gauge, which is the most convenient way for boiler-practice.

Equivalent Evaporation.—As feed-water is supplied at various temperatures to steam-boilers, the results of the evaporative tests of boilers are usually reduced to a standard evaporation of from and at 212° Fahr. The total heat of steam is usually reckoned from 32° Fahr. If the total heat per pound of steam from 32° Fahr. be represented by H, and the weight of water actually evaporated per pound of fuel by W, then the factor of evaporation,

TABLE 37.—PROPERTIES OF SATURATED STEAM FOR PRESSURES IN POUNDS PER SQUARE INCH ABOVE THE ATMOSPHERE, OR FOR BOILER-PRESSURES AS SHOWN BY THE STEAM-GAUGE.

Pressure above the Atmosphere.		Temperature of Steam, Fahrenheit degrees.	Heat of Evaporation per pound of Steam.						Weight of Steam, in Pounds per Cubic Foot.	Volume of Steam in Cubic Feet per Pound.	Relative Volume of Steam, compared with Distilled Water at Temperature of Maximum Density.
			British Thermal Units.				Total heat of Evaporation above 32° Fahrenheit.				
			In Water, above 32° Fahrenheit.	In Steam, — Latent Heat of Evaporation.							
Pounds per Square Inch.	Inches of Mercury at 32° Fahrenheit.			Internal.	External.	Total.	British Thermal Units.	British Units of Evaporation.			
0	0.00	212.0	180.5	893.9	72.2	966.1	1146.6	1.187	.03793	26.37	1646
1	2.04	215.4	183.9	891.3	72.4	963.7	1147.6	1.188	.04035	24.78	1547
2	4.08	218.6	187.2	888.8	72.6	961.4	1148.6	1.189	.04276	23.39	1460
3	6.11	221.6	190.3	886.4	72.8	959.2	1149.5	1.190	.04516	22.14	1382
4	8.15	224.5	193.2	884.1	73.1	957.2	1150.4	1.191	.04756	21.03	1313
5	10.19	227.2	195.9	882.0	73.3	955.3	1151.2	1.192	.04995	20.02	1250
6	12.23	229.8	198.5	879.9	73.6	953.5	1152.0	1.193	.05233	19.11	1193
7	14.26	232.3	201.1	877.9	73.8	951.7	1152.8	1.194	.05470	18.29	1141
8	16.30	234.7	203.5	876.0	74.0	950.0	1153.5	1.194	.05706	17.53	1094
9	18.34	237.1	205.8	874.1	74.3	948.4	1154.2	1.195	.05942	16.83	1051
10	20.38	239.4	208.1	872.3	74.5	946.8	1154.9	1.195	.06177	16.19	1011
11	22.41	241.6	210.3	870.6	74.7	945.3	1155.6	1.196	.06411	15.60	973.7
12	24.45	243.7	212.5	868.9	74.9	943.8	1156.3	1.196	.06645	15.05	939.4
13	26.49	245.8	214.5	867.3	75.1	942.4	1156.9	1.197	.06878	14.54	907.6
14	28.53	247.8	216.5	865.7	75.3	941.0	1157.5	1.197	.07111	14.06	877.9
15	30.56	249.8	218.5	864.2	75.4	939.6	1158.1	1.198	.07344	13.62	850.0
16	32.60	251.6	220.5	862.7	75.5	938.2	1158.7	1.198	.07577	13.20	823.9
17	34.64	253.4	222.4	861.2	75.7	937.9	1159.3	1.199	.07809	12.81	799.4
18	36.68	255.2	224.2	859.8	75.8	935.6	1159.8	1.200	.08041	12.44	776.3
19	38.71	257.0	226.0	858.4	75.9	934.3	1160.3	1.200	.08272	12.09	754.7
20	40.75	258.6	227.7	857.1	76.0	933.1	1160.8	1.201	.08503	11.76	734.2
21	42.79	260.3	229.4	855.8	76.2	932.0	1161.4	1.201	.08734	11.45	714.7
22	44.83	261.9	231.1	854.5	76.3	930.8	1161.9	1.202	.08965	11.16	696.3
23	46.87	263.5	232.8	853.2	76.4	929.6	1162.4	1.202	.09195	10.88	678.9
24	48.90	265.0	234.4	851.9	76.5	928.4	1162.8	1.203	.09424	10.61	662.4
25	50.94	266.5	235.9	850.8	76.6	927.4	1163.3	1.203	.09653	10.36	646.7
26	52.68	268.0	237.5	849.6	76.7	926.3	1163.8	1.204	.09882	10.12	631.7
27	55.02	269.5	238.9	848.5	76.8	925.3	1164.2	1.204	.10110	9.891	617.5
28	57.05	271.0	240.3	847.3	77.0	924.3	1164.6	1.205	.10337	9.674	603.6
29	59.09	272.5	241.8	846.2	77.1	923.3	1165.1	1.205	.10564	9.466	590.9
30	61.13	273.9	243.2	845.1	77.2	922.3	1165.5	1.206	.10790	9.268	576.6
31	63.17	275.3	244.6	844.1	77.2	921.3	1165.9	1.206	.11016	9.078	566.7
32	65.20	276.6	245.9	843.0	77.3	920.3	1166.2	1.206	.11242	8.895	555.3
33	67.24	277.9	247.2	842.0	77.4	919.4	1166.6	1.207	.11468	8.720	544.3
34	69.28	279.3	248.5	841.0	77.5	918.5	1167.0	1.207	.11694	8.551	533.8
35	71.32	280.6	249.8	840.0	77.6	917.6	1167.4	1.208	.11920	8.389	523.7
36	73.35	281.8	251.1	839.0	77.7	916.7	1167.8	1.208	.12146	8.233	514.0
37	75.39	283.0	252.3	838.1	77.8	915.9	1168.2	1.209	.12371	8.083	504.6
38	77.43	284.2	253.6	837.1	77.9	914.0	1168.6	1.209	.12596	7.939	495.6
39	79.47	285.4	254.8	836.2	78.0	914.2	1169.0	1.209	.12821	7.800	486.9

TABLE 37 *continued.*—PROPERTIES OF SATURATED STEAM FOR PRESSURES IN POUNDS PER SQUARE INCH ABOVE THE ATMOSPHERE, OR FOR BOILER-PRESSURES AS SHOWN BY THE STEAM-GAUGE.

Pressure above the Atmosphere.		Temperature of Steam, Fahrenheit degrees.	Heat of Evaporation per pound of Steam.						Weight of Steam, in Pounds per Cubic Foot.	Volume of Steam in Cubic Feet per Pound.	Relative Volume of Steam, compared with Distilled Water at Temperature of Maximum Density.
			British Thermal Units.				Total heat of Evaporation above 32° Fahrenheit.				
				In Steam, — Latent Heat of Evaporation.							
Pounds per Square Inch.	Inches of Mercury at 32° Fahrenheit.		In Water, above 32° Fahrenheit.	Internal.	External.	Total.	British Thermal Units.	British Units of Evaporation.			
40	81·50	286·5	256·0	835·2	78·1	913·3	1169·3	1·210	·13046	7·665	478·5
41	83·54	287·7	257·2	834·3	78·2	912·5	1169·7	1·210	·13270	7·535	470·4
42	85·58	288·8	258·4	833·4	78·2	911·6	1170·0	1·211	·13494	7·409	462·6
43	87·62	289·9	259·6	832·5	78·2	910·7	1170·3	1·211	·13718	7·287	455·1
44	89·65	291·0	260·8	831·6	78·2	909·8	1170·6	1·211	·13942	7·169	447·8
45	91·69	292·1	261·9	830·7	78·3	909·0	1170·9	1·212	·14165	7·056	440·7
46	93·73	293·2	263·0	829·9	78·4	908·3	1171·3	1·212	·14388	6·946	433·9
47	95·77	294·2	264·1	829·1	78·4	907·5	1171·6	1·212	·14611	6·841	427·3
48	97·80	295·3	265·2	828·3	78·5	906·8	1172·0	1·213	·14834	6·741	420·9
49	99·74	296·4	266·2	827·5	78·6	906·1	1172·3	1·213	·15057	6·641	414·6
50	101·88	297·5	267·3	826·7	78·7	905·4	1172·7	1·214	·15280	6·545	408·5
51	103·92	298·6	268·4	825·9	78·8	904·7	1173·1	1·214	·15509	6·451	402·7
52	105·96	299·6	269·4	825·1	78·9	904·0	1173·4	1·214	·15724	6·360	397·0
53	107·99	300·6	270·5	824·3	78·9	903·2	1173·7	1·215	·15946	6·271	391·5
54	100·03	301·5	271·5	823·5	79·0	902·5	1174·0	1·215	·16167	6·185	386·1
55	112·07	302·5	272·5	822·8	79·0	901·8	1174·3	1·215	·16388	6·102	380·9
56	114·11	303·4	273·5	822·0	79·1	901·1	1174·6	1·216	·16609	6·021	375·9
57	116·14	304·3	274·4	821·3	79·2	900·5	1174·9	1·216	·16830	5·942	370·9
58	118·18	305·3	275·4	820·6	79·2	899·8	1175·2	1·216	·17051	5·875	366·1
59	120·22	306·3	276·4	819·9	79·2	899·1	1175·5	1·217	·17272	5·790	361·4
60	122·26	307·2	277·4	819·1	79·3	898·4	1175·8	1·217	·17493	5·717	356·9
61	124·29	308·1	278·3	818·4	79·4	897·8	1176·1	1·217	·17713	5·646	352·4
62	126·33	309·0	279·2	817·7	79·5	897·2	1176·4	1·217	·17933	5·577	348·1
63	128·37	309·9	280·1	817·0	79·6	896·6	1176·7	1·218	·18153	5·509	343·9
64	130·41	310·7	281·0	816·3	79·6	895·9	1176·9	1·218	·18373	5·443	339·8
65	132·44	311·6	281·9	815·6	79·7	895·3	1177·2	1·218	·18593	5·378	335·7
66	134·48	312·5	282·8	815·0	79·7	894·7	1177·5	1·219	·18813	5·315	331·8
67	136·52	313·4	283·7	814·3	79·8	894·1	1177·8	1·219	·19033	5·254	328·0
68	138·56	314·2	284·6	813·6	79·8	893·4	1178·0	1·219	·19253	5·194	324·2
69	140·59	315·0	285·4	812·9	79·9	892·8	1178·2	1·219	·19473	5·135	320·6
70	142·63	315·9	286·2	812·3	80·0	892·3	1178·5	1·220	·19692	5·078	317·0
71	144·67	316·7	287·0	811·7	80·0	891·7	1178·7	1·220	·19911	5·022	313·5
72	146·71	317·5	287·8	811·1	80·1	891·2	1179·0	1·220	·20130	4·968	310·1
73	148·74	318·3	288·6	810·5	80·1	890·6	1179·2	1·220	·20348	4·915	306·8
74	150·78	319·1	289·4	809·9	80·2	890·1	1179·5	1·220	·20566	4·863	303·5
75	152·82	319·8	290·2	809·2	80·3	889·5	1179·7	1·221	·20784	4·811	300·3
76	154·86	320·6	291·0	808·6	80·3	888·9	1179·9	1·221	·21002	4·761	297·2
77	156·89	321·4	291·8	808·0	80·4	888·4	1180·2	1·221	·21220	4·712	294·2
78	158·93	322·2	292·6	807·4	80·4	887·8	1180·4	1·221	·21438	4·664	291·2
79	160·97	323·0	293·3	806·8	80·5	887·3	1180·6	1·221	·21656	4·617	288·3

TABLE 37 *continued.*—PROPERTIES OF SATURATED STEAM FOR PRESSURES IN POUNDS PER SQUARE INCH ABOVE THE ATMOSPHERE, OR FOR BOILER-PRESSURES AS SHOWN BY THE STEAM-GAUGE.

Pressure above the Atmosphere.		Temperature of Steam, Fahrenheit degrees.	Heat of Evaporation per pound of Steam. British Thermal Units.				Total heat of Evaporation above 32° Fahrenheit.		Weight of Steam, in Pounds per Cubic Foot.	Volume of Steam in Cubic Feet per Pound.	Relative Volume of Steam, compared with Distilled Water at Temperature of Maximum Density.
				In Steam, — Latent Heat of Evaporation.							
Pounds per Square Inch.	Inches of Mercury at 32° Fahrenheit.		In Water, above 32° Fahrenheit.	Internal.	External.	Total.	British Thermal Units.	British Units of Evaporation.			
80	163.01	323.8	294.1	806.2	80.5	886.7	1180.8	1.222	.21874	4.572	285.4
81	165.05	324.5	294.8	805.6	80.6	886.2	1181.0	1.222	.22092	4.527	282.6
82	167.08	325.2	295.5	805.1	80.6	885.7	1181.2	1.222	.22309	4.483	279.8
83	169.12	325.9	296.2	804.5	80.7	885.2	1181.4	1.222	.22526	4.439	277.1
84	171.16	326.7	297.0	804.0	80.7	884.7	1181.7	1.222	.22743	4.397	274.5
85	173.20	327.4	297.7	803.4	80.8	884.2	1181.9	1.223	.22960	4.355	271.9
86	175.23	328.1	298.5	802.8	80.9	883.6	1182.1	1.223	.23177	4.315	269.3
87	177.27	328.8	299.2	802.2	80.9	883.1	1182.3	1.223	.23394	4.275	266.8
88	179.31	329.5	299.6	801.7	80.9	882.6	1182.5	1.223	.23611	4.235	264.4
89	181.35	330.3	300.7	801.1	80.9	882.0	1182.7	1.224	.23828	4.197	262.0
90	183.38	331.0	301.4	800.5	81.0	881.5	1182.9	1.224	.24045	4.159	259.6
91	185.42	331.7	302.1	800.0	81.0	881.0	1183.1	1.224	.24262	4.122	257.3
92	187.46	332.4	302.8	799.5	81.0	880.5	1183.3	1.224	.24479	4.085	255.0
93	189.50	333.0	303.5	798.9	81.1	880.0	1183.5	1.224	.24696	4.049	252.8
94	191.53	333.7	304.2	798.4	81.1	879.5	1183.7	1.224	.24913	4.014	250.6
95	193.57	334.4	304.9	797.9	81.1	879.0	1183.9	1.225	.25130	3.979	248.4
96	195.61	335.0	305.6	797.3	81.2	878.5	1184.1	1.225	.25347	3.945	246.3
97	197.65	335.7	306.3	796.8	81.2	878.0	1184.3	1.225	.25564	3.912	244.2
98	199.68	336.4	307.0	796.3	81.2	877.5	1184.5	1.225	.25781	3.879	242.1
99	201.72	337.1	307.7	795.8	81.2	877.0	1184.7	1.225	.25997	3.847	240.1
100	203.76	337.7	308.3	795.3	81.3	876.5	1184.9	1.226	.26213	3.815	238.2
101	205.80	338.4	309.0	794.8	81.3	876.1	1185.1	1.226	.26428	3.784	236.2
102	207.83	339.0	309.6	794.3	81.4	875.7	1185.3	1.226	.26643	3.753	234.3
103	209.87	339.6	310.3	793.8	81.4	875.2	1185.5	1.226	.26858	3.723	232.4
104	211.91	340.2	310.9	793.3	81.5	874.8	1185.7	1.227	.27072	3.694	230.6
105	213.95	340.8	311.6	792.8	81.5	874.3	1185.9	1.227	.27287	3.665	228.8
106	215.99	341.4	312.2	792.4	81.5	873.9	1186.1	1.227	.27502	3.636	227.0
107	218.02	342.0	312.9	791.9	81.5	873.4	1186.3	1.227	.27716	3.608	225.2
108	220.06	342.6	313.6	791.4	81.5	872.9	1186.5	1.227	.27931	3.580	223.5
109	221.10	343.3	314.2	790.9	81.6	872.5	1186.7	1.228	.28145	3.553	221.8
110	224.14	343.9	314.9	790.4	81.6	872.0	1186.9	1.228	.28359	3.526	220.1
111	226.17	344.5	315.5	790.0	81.6	871.6	1187.1	1.228	.28573	3.500	218.5
112	228.21	345.1	316.1	789.5	81.7	871.2	1187.3	1.228	.28787	3.474	216.9
113	230.25	345.7	316.7	789.0	81.7	870.7	1187.4	1.228	.29002	3.448	215.3
114	232.29	346.3	317.3	788.6	81.7	870.3	1187.6	1.229	.29216	3.423	213.7
115	234.32	346.9	318.0	788.1	81.7	869.8	1187.8	1.229	.29431	3.398	212.1
116	236.36	347.5	318.6	787.6	81.8	869.4	1188.0	1.229	.29645	3.373	210.6
117	238.40	348.1	319.2	787.2	81.8	869.0	1188.2	1.229	.29859	3.349	209.1
118	240.44	348.6	319.8	786.7	81.8	868.5	1188.3	1.230	.30074	3.325	207.6
119	242.47	349.2	320.4	786.2	81.9	868.1	1188.5	1.230	.30288	3.301	206.1

TABLE 37 *continued.*—PROPERTIES OF SATURATED STEAM FOR PRESSURES IN POUNDS PER SQUARE INCH ABOVE THE ATMOSPHERE, OR FOR BOILER-PRESSURES AS SHOWN BY THE STEAM-GAUGE.

Pressure above the Atmosphere.		Temperature of Steam, Fahrenheit degrees.	Heat of Evaporation per pound of Steam.						Weight of Steam, in Pounds per Cubic Foot.	Volume of Steam in Cubic Feet per Pound.	Relative Volume of Steam compared with Distilled Water at Temperature of Maximum Density.
			British Thermal Units.				Total Heat of Evaporation above 32° Fahrenheit.				
				In Steam, — Latent Heat of Evaporation.							
Pounds per Square Inch.	Inches of Mercury at 32° Fahrenheit.		In Water, above 32° Fahrenheit.	Internal.	External.	Total.	British Thermal Units.	British Units of Evaporation.			
120	244·51	349·8	321·0	785·8	81·9	867·7	1188·7	1·230	·30503	3·278	204·7
121	246·55	350·3	321·6	785·3	81·9	867·2	1188·8	1·230	·30717	3·255	203·2
122	248·59	350·9	322·2	784·9	81·9	866·8	1189·0	1·230	·30931	3·233	201·8
123	250·62	351·4	322·7	784·4	82·0	866·4	1189·1	1·231	·31145	3·211	200·4
124	252·66	351·9	323·2	784·0	82·0	866·0	1189·2	1·231	·31358	3·189	199·1
125	254·70	352·5	323·8	783·6	82·0	865·6	1189·4	1·231	·31571	3·168	197·7
126	256·74	353·1	324·4	783·1	82·1	865·2	1189·6	1·231	·31784	3·146	196·4
127	258·77	353·7	325·0	782·7	82·1	864·8	1189·8	1·231	·31997	3·125	195·1
128	260·81	354·3	325·6	782·3	82·1	864·4	1190·0	1·232	·32210	3·105	193·8
129	262·85	354·8	326·1	781·9	82·1	864·0	1190·1	1·232	·32423	3·084	192·5
130	264·89	355·4	326·7	781·4	82·2	863·6	1190·3	1·232	·32636	3·064	191·3
131	266·92	355·9	327·2	781·0	82·2	863·2	1190·4	1·232	·32849	3 044	190·0
132	268·96	356·4	327·8	780·6	82·2	862·8	1190·6	1·233	·33062	3·025	188·8
133	271·00	357·0	328·4	780·1	82·3	862·4	1190·8	1·233	·33275	3·005	187·6
134	273·04	357·5	328·9	779·7	82·3	862·0	1190·9	1·233	·33488	2·986	186·4
135	275·08	358·0	329·4	779·3	82·4	861·7	1191·1	1·233	·33701	2·967	185·2
136	277·11	358·5	329·9	778·9	82·4	861·3	1191·2	1·233	·33914	2·949	184·1
137	279·15	359·1	330·4	778·6	82·4	861·0	1191·4	1·234	·34126	2·930	182·9
138	281·19	359·6	330·9	778·2	82·4	860·6	1191·5	1·234	·34338	2·911	181·8
139	283·23	360·1	331·5	777·8	82·4	860·2	1191·7	1·234	·34550	2·893	180·7
140	285·26	360·7	332·0	777·4	82·5	859·9	1191·9	1·234	·34762	2·877	179·6
141	287·30	361·2	332·5	777·0	82·5	859·5	1192·0	1·234	·34974	2 859	178·5
142	289·34	361·7	333·1	776·6	82·5	859·1	1192·2	1·234	·35186	2·842	177·4
143	291·38	362·2	333·6	776·2	82·5	858·7	1192·3	1·234	·35398	2·825	176·4
144	293·41	362·7	334·1	775·8	82·6	858·4	1192·5	1·234	·35610	2·808	175·3
145	295·45	363·2	334·6	775·4	82·7	858·1	1192·7	1·235	·35822	2·792	174·3
146	297·49	363·7	335·2	775·0	82·7	857·7	1192·9	1·235	·36034	2·775	173·2
147	299·53	364·2	335·7	774·6	82·7	857·3	1193·0	1·235	·36246	2·759	172·2
148	301·56	364·7	336·2	774·2	82·8	857·0	1193·2	1·235	·36458	2·743	171·2
149	303·60	365·2	336·7	773·9	82·8	856·7	1193·4	1·235	·36670	2·727	170·2
150	305·64	365·7	337·2	773·5	82·8	856·3	1193·5	1·235	·36881	2·711	169·3
151	307·68	366·1	337·7	773·1	82·8	855·9	1193·6	1·235	·37093	2·696	168·3
152	309·71	366·6	338·2	772·7	82·8	855·5	1193·7	1·235	·37305	2·681	167·3
153	311·75	367·1	338·7	772·3	82·9	855·2	1193·9	1·235	·37517	2·666	166·4
154	313·79	367·6	339·2	771·9	82·9	854·8	1194·0	1·235	·37729	2·651	165·5
155	315·83	368·1	339·7	771·6	82·9	854·5	1194·2	1·235	·37941	2·636	164·5
156	317·86	368·6	340·2	771·2	82·9	854·1	1194·3	1·236	·38153	2·621	163·6
157	319·90	369·1	340·7	770·8	82·9	853·7	1194·4	1·236	·38365	2·607	162·7
158	319·94	369·6	341·2	770·4	83·0	853·4	1194·6	1·236	·38576	2·592	161·8
159	323·98	370·1	341·7	770·1	83·0	853·1	1194·8	1·236	·38787	2·578	160·9

TABLE 37 *continued.*—PROPERTIES OF SATURATED STEAM FOR PRESSURES IN POUNDS PER SQUARE INCH ABOVE THE ATMOSPHERE, OR FOR BOILER-PRESSURES AS SHOWN BY THE STEAM-GAUGE.

Pressure above the Atmosphere.		Temperature of Steam, Fahrenheit degrees.	Heat of Evaporation per pound of Steam.						Weight of Steam, in Pounds per Cubic Foot.	Volume of Steam in Cubic Feet per Pound.	Relative Volume of Steam, compared with Distilled Water at Temperature of Maximum Density.
			British Thermal Units.				Total heat of Evaporation above 32° Fahrenheit.				
				In Steam, — Latent Heat of Evaporation.							
Pounds per Square Inch.	Inches of Mercury at 32° Fahrenheit.		In Water, above 32° Fahrenheit.	Internal.	External.	Total.	British Thermal Units.	British Units of Evaporation.			
160	326.01	370.5	342.2	769.7	83.0	852.7	1194.9	1.236	.38998	2.564	160.1
161	328.05	371.0	342.7	769.3	83.0	852.3	1195.0	1.236	.39209	2.550	159.2
162	330.09	371.4	343.1	769.0	83.0	852.0	1195.1	1.237	.39420	2.537	158.4
163	332.13	371.9	343.6	768.7	83.0	851.7	1195.3	1.237	.39631	2.523	157.5
164	334.17	372.3	344.0	768.3	83.1	851.4	1195.4	1.237	.39842	2.513	156.7
165	336.20	372.8	344.5	768.0	83.1	851.1	1195.6	1.237	.40053	2.497	155.0
166	338.24	373.2	345.0	767.6	83.1	850.7	1195.7	1.237	.40264	2.484	155.2
167	340.28	373.6	345.5	767.2	83.1	850.3	1195.8	1.238	.40475	2.471	154.2
168	342.32	374.1	346.0	766.9	83.1	850.0	1196.0	1.238	.40686	2.458	153.4
169	344.35	374.5	346.4	766.5	83.2	849.9	1196.1	1.238	.40897	2.445	152.6
170	346.39	375.0	346.9	766.2	83.2	849.4	1196.3	1.238	.41108	2.433	151.9
171	348.43	375.4	347.3	765.9	83.2	849.1	1196.4	1.238	.41319	2.420	151.1
172	350.47	375.9	347.8	765.6	83.2	848.8	1196.6	1.238	.41530	2.408	150.3
173	352.50	376.3	348.2	765.2	83.3	848.5	1196.7	1.238	.41741	2.396	149.6
174	354.54	376.7	348.6	764.9	83.3	848.2	1196.8	1.239	.41952	2.384	148.8
175	356.58	377.2	349.1	764.6	83.3	847.9	1197.0	1.239	.42163	2.372	148.1
176	358.62	377.6	349.6	764.2	83.3	847.5	1197.1	1.239	.42373	2.360	147.3
177	360.65	378.1	350.1	763.8	83.3	847.1	1197.2	1.239	.42583	2.348	146.6
178	362.69	378.5	350.5	763.5	83.3	846.8	1197.3	1.239	.42793	2.337	145.9
179	364.73	378.9	350.9	763.2	83.3	846.5	1197.4	1.239	.43003	2.326	145.2
180	366.77	379.3	351.4	762.8	83.4	846.2	1197.6	1.239	.43213	2.314	144.5
181	368.80	379.7	351.8	762.5	83.4	845.9	1197.7	1.240	.43424	2.303	143.8
182	370.84	380.2	352.3	762.2	83.4	845.6	1197.9	1.240	.43634	2.292	143.1
183	372.88	380.6	352.7	761.9	83.4	845.3	1198.0	1.240	.43844	2.281	142.4
184	374.92	381.0	353.2	761.5	83.4	844.9	1198.1	1.240	.44054	2.270	141.7
185	376.95	381.4	353.7	761.2	83.4	844.6	1198.3	1.240	.44264	2.259	141.0
186	378.99	381.8	354.1	760.9	83.4	844.3	1198.4	1.240	.44474	2.249	140.4
187	381.03	382.2	354.5	760.5	83.5	844.0	1198.5	1.241	.44684	2.238	139.7
188	383.07	382.7	355.0	760.2	83.5	843.7	1198.7	1.241	.44894	2.228	139.1
189	385.11	383.1	355.4	759.9	83.5	843.4	1198.8	1.241	.45104	2.217	138.4
190	387.14	383.5	355.8	759.6	83.5	843.1	1198.9	1.241	.45314	2.207	137.8
191	389.18	384.0	356.3	759.3	83.5	842.8	1199.1	1.241	.45524	2.197	137.1
192	391.22	384.4	356.7	759.0	83.5	842.5	1199.2	1.241	.45734	2.187	136.5
193	393.26	384.8	357.1	758.7	83.5	842.2	1199.3	1.241	.45944	2.177	135.9
194	395.29	385.2	357.5	758.3	83.6	841.9	1199.4	1.241	.46154	2.167	135.3
195	397.33	385.7	358.0	758.0	83.6	841.6	1199.6	1.241	.46364	2.157	134.6
196	399.37	386.1	358.4	757.7	83.6	841.3	1199.7	1.242	.46574	2.147	134.0
197	401.41	386.5	358.8	757.4	83.6	841.0	1199.8	1.242	.46783	2.138	133.4
198	403.44	386.9	359.2	757.1	83.6	840.7	1199.9	1.242	.46992	2.128	132.8
199	405.48	387.3	359.6	756.8	83.7	840.5	1200.1	1.242	.47210	2.119	132.3

TABLE 37 *continued.*—PROPERTIES OF SATURATED STEAM FOR PRESSURES IN POUNDS PER SQUARE INCH ABOVE THE ATMOSPHERE, OR FOR BOILER-PRESSURES AS SHOWN BY THE STEAM-GAUGE.

Pressure above the Atmosphere.		Temperature of Steam, Fahrenheit degrees.	Heat of Evaporation per pound of Steam. British Thermal Units.				Total heat of Evaporation above 32° Fahrenheit.		Weight of Steam, in Pounds per Cubic Foot.	Volume of Steam in Cubic Feet per Pound.	Relative Volume of Steam, compared with Distilled Water at Temperature of Maximum Density.
Pounds per Square Inch.	Inches of Mercury at 32° Fahrenheit.		In Water, above 32° Fahrenheit.	In Steam, — Latent Heat of Evaporation. Internal.	External.	Total.	British Thermal Units.	British Units of Evaporation.			
200	407·52	387·7	360·0	756·5	83·7	840·2	1200·2	1·242	·47410	2·109	131·7
201	409·56	388·1	360·4	756·2	83·7	839·9	1200·3	1·242	·47620	2·100	131·1
202	411·59	388·5	360·8	755·9	83·7	839·6	1200·4	1·242	·47829	2·091	130·5
203	413·63	388·9	361·2	755·6	83·7	839·3	1200·5	1·243	·48038	2·082	130·0
204	415·67	389·3	361·7	755·3	83·7	839·0	1200·7	1·243	·48247	2·073	129·4
205	417·71	389·7	362·1	754·9	83·8	838·7	1200·8	1·243	·48456	2·064	128·8
206	419·74	390·1	362·5	754·6	83·8	838·4	1200·9	1·243	·48665	2·055	128·3
207	421·78	390·5	362·9	754·3	83·8	838·1	1201·0	1·243	·48874	2·046	127·7
208	423·82	390·9	363·3	754·0	83·8	837·8	1201·1	1·243	·49083	2·037	127·2
209	425·86	391·3	363·7	753·7	83·8	837·5	1201·2	1·244	·49292	2·029	126·6
210	427·89	391·7	364·1	753·4	83·9	837·3	1201·4	1·244	·49501	2·020	126·1
211	429·93	392·1	364·5	753·1	83·9	837·0	1201·5	1·244	·49710	2·012	125·6
212	431·97	392·5	364·9	752·8	83·9	836·7	1201·6	1·244	·49919	2·003	125·1
213	434·01	392·8	365·3	752·5	83·9	836·4	1201·7	1·244	·50128	1·995	124·5
214	436·04	393·2	365·7	752·2	83·9	836·1	1201·8	1·244	·50337	1·987	124·0
215	438·08	393·5	366·1	751·9	83·9	835·8	1201·9	1·245	·50546	1·978	123·5
216	440·12	393·9	366·5	751·6	83·9	835·5	1202·0	1·245	·50755	1·970	123·0
217	442·16	394·3	366·9	751·3	84·0	835·3	1202·2	1·245	·50964	1·962	122·5
218	444·20	394·7	367·3	751·0	84.0	835·0	1202·3	1·245	·51173	1·954	122·0
219	446·23	395·1	367·7	750·7	84·0	834·7	1202·4	1·245	·51382	1·946	121·4
220	448·27	395·5	368·0	750·5	84·0	834·5	1202·5	1·245	·51590	1·938	121·0
221	450·31	395·9	368·4	750·2	84·1	834·3	1202·7	1·245	·51799	1·931	120·5
222	452·35	396·3	368·8	749·9	84·1	834·0	1202·8	1·245	·52008	1·923	120·0
223	454·38	396·7	369·2	749·6	84·1	833·7	1202·9	1·245	·52217	1·915	119·6
224	456·42	397·0	369·6	749·3	84·1	833·4	1203·0	1·245	·52426	1·908	119·1
225	458·46	397·3	369·9	749·1	84·1	833·2	1203·1	1·245	·52635	1·900	118·6
226	460·50	397·7	370·3	748·8	84·1	832·9	1203·2	1·246	·52843	1·892	118·1
227	462·53	398·0	370·7	748·5	84·1	832·6	1203·3	1·246	·53051	1·885	117·7
228	464·57	398·4	371·1	748·2	84·1	832·3	1203·4	1·246	·53259	1·878	117·2
229	466·61	398·8	371·4	748·0	84·1	832·1	1203·5	1·246	·53467	1·870	116·8
230	468·65	399·1	371·8	747·7	84·1	831·8	1203·6	1·246	·53675	1·863	116·3
231	470·68	399·5	372·2	747·4	84·2	831·6	1203·8	1·246	·53884	1·856	115·9
232	472·71	399·8	372·6	747·1	84·2	831·3	1203·4	1·246	·54092	1·849	115·4
233	474·76	400·2	373·0	746·8	84·2	831·0	1204·0	1·246	·54300	1·842	115·0
234	476·80	400·6	373·4	746·5	84·2	830·7	1204·1	1·246	·54508	1·835	114·5
235	478·83	400·9	373·8	746·2	84·2	830·4	1204·2	1·247	·54717	1·828	114·1
236	480·87	401·3	374·2	745·9	84·2	830·1	1204·3	1·247	·54925	1·821	113·7
237	482·91	401·6	374·5	745·6	84·3	829·9	1204·4	1·247	·55133	1·814	113·2
238	484·95	401·9	374·8	745·4	84·3	829·7	1204·5	1·247	·55341	1·807	112·8
239	486·98	402·3	375·2	745·1	84·3	829·4	1204·6	1·247	·55549	1·800	112·4

TABLE 37 *continued.*—PROPERTIES OF SATURATED STEAM FOR PRESSURES IN POUNDS PER SQUARE INCH ABOVE THE ATMOSPHERE, OR FOR BOILER-PRESSURES AS SHOWN BY THE STEAM-GAUGE.

Pressure above the Atmosphere.		Temperature of Steam, Fahrenheit degrees.	Heat of Evaporation per pound of Steam. British Thermal Units.				Total heat of Evaporation above 32° Fahrenheit.		Weight of Steam, in Pounds per Cubic Foot.	Volume of Steam in Cubic Feet per Pound.	Relative Volume of Steam, compared with Distilled Water at Temperature of Maximum Density.
				In Steam,—Latent Heat of Evaporation.							
Pounds per Square Inch.	Inches of Mercury at 32° Fahrenheit.		In Water, above 32° Fahrenheit.	Internal.	External.	Total.	British Thermal Units.	British Units of Evaporation.			
240	489·02	402·6	375·5	744·9	84·3	829·2	1204·7	1·247	·55757	1·794	112·0
241	491·06	402·9	375·9	744·6	84·4	829·0	1204·9	1·247	·55965	1·787	111·5
242	493·10	403·2	376·3	744·3	84·4	828·7	1205·0	1·247	·56173	1·780	111·1
243	495·13	403·5	376·6	744·1	84·4	828·5	1205·1	1·247	·56381	1·774	110·7
244	497·17	403·9	377·0	743·8	84·4	828·2	1205·2	1·247	·56589	1·767	110·3
245	499·21	404·2	377·8	743·6	84·4	828·0	1205·3	1·247	·56797	1·761	109·9
246	501·25	404·6	377·7	743·3	84·4	827·7	1205·4	1·247	·57005	1·754	109·5
247	503·29	405·9	378·1	743·0	84·4	827·4	1205·5	1·247	·57212	1·748	109·1
248	505·32	405·3	378·4	742·7	84·5	827·2	1205·6	1·248	·57419	1·742	108·7
249	507·36	405·7	378·8	742·4	84·5	826·9	1205·7	1·248	·57626	1·735	108·3
250	509·40	406·0	379·1	742·2	84·5	826·7	1205·8	1·248	·57833	1·729	107·9

and the equivalent evaporation, or weight of water evaporated as from and at 212° Fahr., may be found by the following formulæ:—

$$\text{Factor of evaporation} = \frac{(\text{H} + 32^\circ) - \text{temperature of feed-water}}{966^\circ}.$$

$$\text{Equivalent evaporation} = \text{W} \times \frac{(\text{H} + 32^\circ) - \text{temperature of feed-water}}{966^\circ}.$$

Example: Steam is generated at 165 pounds per square inch absolute pressure, 8 pounds of water are evaporated per pound of coal from feed-water supplied to the boiler at 106° Fahr. Required the equivalent weight of water evaporated as from and at 212° Fahr.

Then, the total heat of steam of 165 pounds per square inch absolute pressure from 32° Fahr. is 1194 units: and

$$\frac{(1194^\circ + 32^\circ) - 106^\circ}{966^\circ} = \frac{1120}{966} = 1 \cdot 159, \text{ the factor of evapo-}$$

ration: and 8 pounds × 1·159 = 9·272 pounds, the equivalent weight of water evaporated as from and at 212° Fahr.

When the equivalent weight of water is to be evaporated as from a less temperature than 212° Fahr., it may be found by this formula:—

$$W \times \frac{(H + 32^\circ) - \text{temperature of feed-water}}{1178 - \text{temperature of feed-water}}$$

the notation being the same as that in the previous formulæ.

The Evaporative Power of Different Types of Steam-Boilers varies considerably, although the heat evolved by the combustion of a pound of coal is practically the same in all kinds of boilers with equal strength of draught. The quantity of water evaporated in pounds per pound of coal consumed, from and at 212° Fahr., in different types of boilers averages as follows:—

Plain cylindrical boilers	evaporate	from	5 to 8	lbs. of water	per lb. of coal.
Vertical boilers	"	"	5 to 10	"	"
Water-tube boilers	"	"	5 to 11	"	"
Cornish boilers	"	"	6 to 11	"	"
Lancashire boilers	"	"	6½ to 12	"	"
Marine return-tube boilers	"	"	7 to 12	"	"
Multitubular boilers	"	"	8 to 12	"	"
Galloway boilers	"	"	9 to 12½	"	"
Locomotive boilers	"	"	8 to 13	"	"

The difference in the evaporative economy of the boilers is principally due to the difference in the arrangements of the heating-surfaces, some favouring rapid absorption of heat and free circulation of water more than others. Good stationary boilers with clean heating-surfaces average from 9 to 10 pounds of water per pound of coal, and good locomotive boilers 10 pounds of water per pound of coal. It may be affirmed that a good boiler of any type with clean heating-surfaces, should evaporate 10 pounds of water per pound of coal.

The flues and smoke-tubes of steam-boilers, under ordinary working conditions, are generally coated with soot, which reduces the efficiency of the heating-surfaces so much, that the maximum evaporation in daily work of internally fired steam-boilers is frequently only 8 pounds of water per pound of coal, and that of externally fired boilers only 5 pounds of water per pound of coal.

The evaporative power of a number of boilers of different types is given at pages 353—357.

The Evaporative Power of a Square Foot of Fire-grate Surface of a Steam-boiler with a well-arranged furnace, depends upon the evaporative power of the coal, and the strength of the draught.

The consumption of coal in stationary-engine boilers with natural draught, ranges from 12 to 24 pounds per square foot of fire-grate surface per hour. It is seldom less than 15 pounds, and as each pound of coal should evaporate at least 8 pounds of water, the evaporative power of a square-foot of fire-grate surface is, on this basis, $= 15 \times 8 = 120$ pounds of water per hour, $= 120 \div 60$ minutes $= 2$ pounds of water evaporated per minute.

In Lancashire boilers burning slack, the rate of consumption with easy firing is frequently 18 pounds per square foot of fire-grate surface per hour, and $8\frac{1}{2}$ pounds of water are frequently evaporated per pound of slack; the evaporative power of a square foot of fire-grate surface is $= 18 \times 8\frac{1}{2}$ lbs. $=$ 153 pounds of water per hour, and $= 153 \div 60 = 2{\cdot}216$ lbs. of water evaporated per minute.

The consumption of coal in marine return-tube boilers, with natural draught, when burning South Wales coal in ordinary work at full power, is frequently 16 lbs. per square foot of fire-grate surface per hour, and if 9 lbs. of water are evaporated per pound of coal, it is equal to an evaporation of $16 \times 9 = 144$ lbs. of water per square foot of fire-grate surface per hour; and $= 144 \div 60$ minutes $= 2{\cdot}4$ lbs. of water evaporated per minute. It seldom exceeds 20 pounds per square-foot of fire-grate surface per hour, and if 9 pounds of water are evaporated per pound of coal, it is equal to an evaporation of $20 \times 9 = 180$ pounds of water per square-foot of fire-grate surface per hour, $= 180 \div 60$ minutes $= 3$ pounds of water evaporated per minute.

With the best arranged boilers with natural draught, 10 pounds of water may be evaporated under ordinary working conditions per pound of best steam-coal, and more than 12 pounds of water have been evaporated under conditions favourable to economical evaporation. With a very strong natural draught, as much as 26 pounds of coal may be economically burnt per square-foot of fire-grate surface per hour, and the evaporation may be $= 26 \times 10 = 260$ pounds of water per square foot of fire-grate surface per hour, $= 260 \div 60$ minutes $= 4{\cdot}33$ pounds of water per square foot of fire-grate surface per minute.

The maximum quantity of water that can be economically evaporated in a steam-boiler with natural draught, is 26 pounds of coal $\times$ 12 pounds of water $= 312$ pounds per hour, or $312 \div 60$ minutes $= 5{\cdot}2$ pounds of water per square foot of fire-grate surface per minute.

The boiler of a locomotive engine has the draught assisted by exhaust-steam-blast in the chimney. A passenger-locomotive-boiler having an area of fire-grate surface of $17\frac{1}{2}$ square feet, and burning 28 pounds of coal per mile at a speed of 50 miles per hour, will consume,

$$\frac{28 \text{ lbs. of coal} \times 50 \text{ miles per hour}}{17{\cdot}5 \text{ square feet of fire-grate}} = 80 \text{ pounds of coal per square foot}$$

of fire-grate surface per hour.

Each pound of coal will evaporate, say, $10\frac{1}{2}$ lbs. of water per hour, and the evaporative power of a square foot of fire-grate surface of a passenger-locomotive-boiler, on this basis, is $= 80$ lbs. of coal $\times$ $10{\cdot}5$ lbs. of water $=$ 840 lbs. of water per square foot of fire-grate surface per hour, and $=$ 840 lbs. $\div$ 60 minutes $= 14$ lbs. of water per square foot of fire-grate surface per minute.

The rate of evaporation of passenger-locomotive boilers is consequently

840 lbs. ÷ 120 lbs. = 7 times as great as the average rate of evaporation in stationary boilers, and 840 lbs. ÷ 312 lbs. = 2·7 times as great as the maximum economical rate of evaporation obtainable with natural draught, per square foot of fire-grate surface per hour.

The boiler of a locomotive goods-engine having an area of fire-grate surface of 17 square feet, and burning 34 lbs. of coal per mile at a speed of 20 miles an hour, will consume,

$$\frac{34 \text{ lbs. of coal} \times 20 \text{ miles an hour}}{17 \text{ square feet of fire-grate}} = 40 \text{ lbs. of coal per square foot of}$$

fire-grate surface per hour. Each pound of coal will evaporate, say, 10 lbs. of water per hour, and the evaporative power of a square foot of fire-grate surface of a goods locomotive-boiler, on this basis, is = 40 lbs. of coal × 10 lbs. of water = 400 lbs. of water per square foot of fire-grate surface per hour, and = 400 lbs. ÷ 60 minutes = 6·66 lbs. of water per square foot of fire-grate surface per minute.

Steam-Producing Capacity of Different Types of Boilers per Square Foot of Fire-Grate Surface.—The preceding calculations show that the production of steam varies considerably in different types of boilers. Taking the average of a number of evaporative tests of good boilers of different types when burning good coal, the rates of evaporation under ordinary working conditions are usually as follows:—

			Pounds of water evaporated per square foot of fire-grate surface per hour.
Vertical cross-tube boilers	have a	maximum average evaporation of	90
„ „ „	„	mean „ „ „	50
Vertical tubular boilers . .	„	maximum „ „ „	110
„ „ „ . . .	„	mean „ „ „	60
Plain cylindrical boilers . .	„	maximum „ „ „	140
„ „ „ . . .	„	mean „ „ „	80
Cornish boilers . . .	„	maximum „ „ „	190
„ „	„	mean „ „ „	140
Portable boilers, of locomotive type	„	maximum „ „ „	200
„ „ „ „	„	mean „ „ „	115
Water-tube boilers of various kinds	„	maximum „ „ „	210
„ „ „	„	mean „ „ „	140
Externally fired cylindrical multitubular boilers . .	„	maximum „ „ „	215
„ „ „	„	mean „ „ „	155
Lancashire boilers . . .	„	maximum „ „ „	220
„ „	„	mean „ „ „	160
The Galloway boiler . . .	„	maximum „ „ „	240
„ „ „ . . .	„	mean „ „ „	170
Internally fired cylindrical multitubular boilers . . .	„	maximum „ „ „	250
„ „ „	„	mean „ „ „	180

		Pounds of water evaporated per square foot of fire-grate surface per hour.
Boilers of modified locomotive type with natural draught	have a maximum average evaporation of	300
	„ mean „ „ „	200
Boilers of locomotive goods-engine	„ maximum „ „ „	500
	„ mean „ „ „	400
Boilers of locomotive passenger-engines . . .	„ maximum „ „ „	900
	„ mean „ „ „	600

The evaporative performances of these boilers are with normal rates of combustion, and all with natural draught, except those of the locomotive type, which have steam-blast in the chimney.

TABLE 38.—EVAPORATIVE PERFORMANCE OF A MARINE RETURN-TUBE STEAM-BOILER, WITH DIFFERENT RATES OF COMBUSTION.

Pounds of Anthracite Consumed per Hour per Square Foot of Fire-grate Surface.	ECONOMICAL EVAPORATION.		Temperature in Degrees Fahr. of the Products of Combustion when leaving the Boiler.	Weight of Steam furnished by the Boiler in equal time, expressed proportionally.	Weight of Steam furnished by equal weights of Anthracite, expressed proportionally.	Weight and Bulk of Anthracite required to furnish equal Weights of Steam, expressed proportionally.
	Pounds of Water Evaporated under Atmospheric Pressure from 212° Fahr. by 1 lb. of Anthracite.	Per-centage of the total Heat developed by the Combustion utilised evaporatively				
6	10·49	84·4	445	1·000	1·000	1·000
7	10·44	84·1	455	1·161	·995	1·004
8	10·35	83·6	473	1·312	·986	1·013
9	10·23	82·3	497	1·462	·975	1·025
10	10·05	80·8	532	1·596	·958	1·043
11	9·81	78·9	580	1·714	·935	1·069
12	9·53	76·7	636	1·817	·908	1·100
13	9·21	74·1	699	1·902	·878	1·138
14	8·87	71·3	767	1·973	·845	1·182
15	8·52	68·5	837	2·030	·812	1·231
16	8·21	66·0	898	2·087	·782	1·277
17	7·95	63·9	950	2·147	·758	1·319
18	7·70	61·9	999	2·202	·734	1·362
19	7·48	60·2	1043	2·258	·713	1·402
20	7·32	58·9	1075	2·326	·697	1·433
21	7·16	57·6	1107	2·389	·682	1·465
22	7·04	56·6	1131	2·460	·671	1·490
23	6·92	55·7	1154	2·528	·659	1·515
24	6·82	54·8	1174	2·600	·650	1·538

The Economical Evaporative Efficiency of Steam-Boilers is considerably affected by the rate of combustion. This is clearly shown in Table 38, which contains the results of experiments by Mr. Isherwood, with a steam-boiler having a ratio of heating-surface to fire-grate surface of 25 to 1, and an area through the tubes equal one-eighth of the fire-grate surface. The temperature of the products of combustion, at the moment of their formation, was assumed to be 2469° Fahr. above that of the atmosphere, which was 68° Fahr.

It will be seen from the results of this boiler-test, that, as the rate of combustion increased the economical evaporation decreased, the weight of steam supplied in equal time increased, and the temperature of the products of combustion on leaving the boiler increased rapidly.

Priming of Steam-Boilers.—When water is carried with the steam from the evaporating surface, a boiler is said to prime. Priming is caused by the resistance to the escape of the steam-bubbles opposed by the friction of the water on their surfaces, as they rise through the water to the steam-space. It is increased by the presence of impurities in the water, because they tend to cause the retention of the steam-bubbles, which results in inflation of the water and more or less violent agitation of its surface, and the ejection of spray to the steam-space.

Priming is also caused by a sudden withdrawal, or rapid rush, of steam from the boiler, by violent local ebullition, and by a wide difference of pressure in different parts of a boiler, which produces currents of spray. Priming may be caused by insufficient evaporative-surface and deficient steam-space. Priming always exists where circulation is defective or impeded, and the steam cannot escape freely from the water-heating surfaces.

When intense heat impinges upon the side of a water-spaced compartment it is liable to cause priming, and when it impinges upon both sides of the compartment, the water becomes so greatly agitated by steam-bubbles clinging to the surfaces as to cause the ejection of much spray, resulting in more or less severe priming.

To prevent priming, it is essential to provide efficient circulation with ample evaporative-surface and steam-space in a boiler, and to withdraw the steam quietly; and the boiler should be large enough to do its work without forced firing. The tendency of a boiler to prime decreases as the pressure of the steam increases.

Oil has sometimes been injected into boilers to prevent priming. Mineral-oil is the most effective for this purpose, but it should be of the purest and best quality, and its vaporising-point should be higher than the temperature of the steam. Common mineral-oil contains much bituminous matter, and all animal and vegetable oils contain resinous matter, and they are consequently dangerous in steam-boilers. They leave a deposit which mixes with sediment from the water and forms an impermeable coating on the heating-surfaces, and causes overheating, and bulged furnace-crowns.

Foaming.—Priming differs from foaming, which is a dirty frothy condition of the surface of the water. It is caused by scum, arising from impurities held principally in suspension in the water, and floating on its surface. Water in a pure state can not produce foam. Less moisture is produced in steam from foaming than from priming. The remedy for foaming is frequent blowing-off at the surface of the water.

SECTION III.

WATER-HEATING-SURFACES OF STEAM-BOILERS; TRANSMISSION OF HEAT; SMOKE-TUBES; EVAPORATIVE POWER AND EFFICIENCY OF BOILERS; WATER-CAPACITY AND STEAM-CAPACITY OF BOILERS; FIRE-GRATES, FIRE-BRIDGES AND FIRE-BARS; POWER OF BOILERS; CYLINDRICAL SHELLS AND FURNACE-TUBES OF BOILERS, ETC.

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The Requirements of a Good Steam-Boiler may be briefly stated as follows:—

Excellency of design, for securing safety and economy in working.

Ample size for the development of the required power with easy firing.

Simplicity of construction, with all parts easy of access for thorough internal and external examination, cleaning, and repairs, also admitting of ready renewal, but not liable to frequent repairs.

Durability of construction, and freedom from excessive wear and tear.

Excessive strength, with freedom from excessive deteriorating strains, such as result from unequal expansion and contraction.

Sufficient elasticity to permit expansion by heat.

Ample heating-surface, formed to facilitate circulation, and arranged in the best position for the efficient absorption of the available radiant heat, and also for the extraction of the greatest possible quantity of heat from the products of combustion; and having all parts readily accessible for cleaning or scaling.

A flow of hot fuel-gases well distributed over the heating-surfaces, and of sufficient duration to permit efficient absorption of the heat.

Efficient natural circulation of the water, with the greatest possible uniformity of temperature throughout the boiler.

Ample steam-space and evaporative-surface to effect tranquil release of the steam from the water, and secure an ample and steady supply of dry steam, of uniform quality.

A furnace arranged in the best manner to effect the most complete combustion of the fuel, and the development of the greatest possible quantity of heat.

Fire-grate surface properly proportioned to the rate of combustion of the fuel and to the area of the heating-surface.

Ample furnace-space above the fire to permit development of flame, and obtain the greatest possible amount of radiant heat.

A combustion-chamber, or sufficient space to effect mixture of air and fuel-gases, and complete their combustion. The combustion-space provided in multitubular boilers should be sufficient to permit complete combustion of the fuel-gases before they reach the smoke-tubes.

It should have no joints or seams of plates exposed to direct impingement of flame.

No boiler which does not reasonably fulfil these requirements can be considered a highly efficient, or practically perfect, steam-generator.

It is desirable to provide an excess of boiler-power. When a boiler has no margin of power, it may be said to be too small for its work, and more or less forced firing will be required, resulting in extravagant consumption of fuel.

A boiler should be capable of economically developing its reputed power with easy firing, moderately strong draught, and fuel of average quality. It should also be capable of developing in an emergency, when maximum economy is not of importance, at least one-third more than its reputed power.

Water-Heating Surfaces of Steam-Boilers.—The plates forming the water-heating surfaces of a boiler absorb heat, communicated from the fuel by radiation and contact, and transmit it to the water. A sound plate in continual contact with water on one side, never acquires a temperature sufficiently high to sensibly diminish its strength.

The evaporative efficiency of a steam-boiler depends greatly upon the efficiency of its heat-absorbing surfaces and the activity of the circulation of the water over them. It is therefore essential to give the heating-surfaces the best form and position for permitting free escape of heated water and steam from them, and facilitate rapid absorption and transmission of heat.

In order to determine the best shape or form of heating-surfaces, an experiment was made with a cubical metal-box submerged in water and heated from within; and it was found that steam was generated from its upper surface more than twice as fast per unit of area than from its vertical sides, and that the bottom surface generated no steam. These differences are owing to the difficulty with which steam separates from a vertical surface to give place to fresh charges of water, and to the impossibility of its escape from an inverted surface. The box was also placed in an inclined position, when the elevated side permitted the steam to escape much more readily, and the rate of evaporation was increased; while on the depressed side the steam hung so sluggishly as to cause the metal to become overheated.

It appears from this and other experiments that, when a fire is enclosed by water-space sides, the sides of the fire-box should be inclined instead of vertical, in order to permit the free escape of steam-bubbles as fast as they are formed, and that surfaces beneath the fire are of no value as heating-surfaces.

Thickness and Conductive Value of Metal for Water-Heating Surfaces.—The thickness of the plates forming the heating-surfaces of steam-boilers, within the limits of ordinary practice, has very little effect on their evaporative efficiency. In an experiment by Mr. Isherwood, made with cylinders of different kinds of metal, of $\frac{1}{8}$, $\frac{1}{4}$, and $\frac{3}{8}$-inch thick, having water inside, and heat applied uniformly on the outside, it was found that, all other things being equal, the weight of water vaporised in a given time was in the direct ratio of the difference of the temperatures inside and outside the metal; and that the weight of water vaporised in a given time was not affected by the thickness of the metal.

The quantity of water evaporated under an atmospheric pressure of 29·92 inches of mercury, by a difference of temperature of one degree Fahr. between the inside and the outside of the metal cylinders, and the absolute heat-conducting power of the metals employed in the experiment, are given in the following Table:—

TABLE 39.—RESULTS OF EXPERIMENTS ON THE EVAPORATIVE-POWER, AND HEAT-TRANSMISSIVE POWER OF METALS.

Description of Metal.	Thermal Conductivity in terms of fractions of a lb. of water of 212°, vaporised under atmospheric pressure.	Thermal conductivity in terms of Heat-units transmitted per Hour, through one square foot of material by difference of Temperature of 1° Fahrenheit.	Relative Thermal Conductivity.
Copper . . .	·665365	642·543	1·000000
Brass . .	·576610	556·832	·866607
Wrought-iron .	·386895	373·625	·581478
Cast-iron . .	·326956	315·741	·491393

In these experiments the surface of the metals was clean and bright, a condition not obtainable in practice with steam-boilers, as the surfaces are always tarnished, or coated with a slight scale even in their best condition.

It has been proved as the results of experiments, and practice, that when after a few days work the heating-surfaces of a steam-boiler become coated with a slight skin of scale, however thin, there is no difference in the evaporative efficiency of copper, brass, iron, or steel, and that these metals all possess the same heat-transmissive-power.

The Steaming-Capacity of the Furnace-Tubes of Steam-Boilers increases slightly as the thickness diminishes below $\frac{3}{8}$ inch, but the benefit disappears as the plates become coated with incrustation. The results of many experiments prove that, for all thickness of plates not exceeding $\frac{7}{16}$ inch thick, the evaporative effect is practically the same, but above that thickness the evaporative efficiency decreases with the thickness of the plate.

It is frequently considered desirable to limit the thickness of furnace-tubes to $\frac{1}{2}$ inch, but when they are constructed of homogeneous plates free from laminations, thicker plates than this may be used without much affecting the transmission of heat. Thick plates suffer more injury from unequal expansion than thin plates, the injury being directly proportional to the thickness. Therefore it is essential in using thick furnace-plates to make ample provision for the accommodation of expansion; and there should be free circulation of the water.

Furnace-tubes of plates $\frac{5}{8}$ inch and $\frac{3}{4}$ inch have been employed, and found to work satisfactorily, and it is a practical question whether the thickness may be further increased. The effect of thickness in raising the temperature of the interior of plates forming water-heating surfaces has not been determined, but it is probable that when the circulation of the water is efficient, the interior of plates of any thickness within reasonable limits, or say, not exceeding $1\frac{1}{8}$ inches in thickness, would be very little higher in temperature than that of the water in the boiler, and there would be no sensible diminution in the strength of the plates.

The transmission of heat is obstructed by a break in the continuation of a plate such as is produced by laminations, blisters, and imperfections in the contact of two plates forming a seam, because these defects break the continuity of the conduction of heat. The plates of heating-surfaces should, therefore, be of homogeneous structure, and as riveted seams are liable to obstruct the transmission of heat more or less, they should not be exposed to the impact of fierce heat or flame.

Temperature of the Evaporative-Surface, or Water-Side of Plates forming Heating-Surfaces.—In an experiment by Mr. Hirsch, it was found that the difference between the temperature of the external surface of the plate and the temperature of the water on the other side of the plate, rises progressively as the rate of evaporation per square foot of heating-surface per hour increases. The difference at a rate of evaporation of 41 pounds of water per square foot of heating-surface per hour was 212° Fahr., and it did not exceed 302° Fahr. when the evaporation was as high as 61 pounds of water per square foot of fire-grate surface per hour. It was found that even with very high rates of evaporation the temperatures of the external surface of the plate did not exceed 536° Fahr.

Evaporative Value of Heating-Surface.—It is necessary to provide sufficient heating-surface in a steam-boiler to absorb the heat evolved by the fuel and transmit it to the water. When the heating-surface is insufficient to absorb the heat, the gaseous products of combustion escape to the chimney at a high temperature, and the maximum heating-effect cannot be obtained from the fuel. The evaporative value of heating-surfaces varies considerably in different classes of boilers, and also in the same class of boiler.

In a careful experiment, made to determine the evaporative value of the heating-surface of a small cylindrical multitubular steam-boiler, with natural draught, it was found that the greatest quantity of water that could be evaporated under ordinary working conditions, from cold feed water, was 6 pounds per square foot of total heating-surface per hour with a moderate draught, and 7 pounds with a strong draught. Therefore it may be affirmed that one cubic foot of water, or 62·4 pounds $\div$ 7 pounds per square foot $=$ 8·915, or say, in round numbers, 9 square feet of the heating-surface of this type of boiler will transmit the heat required to evaporate one cubic foot of water per hour.

Each unit of heating-surface does not possess the same heat-transmissive value, because the rate of evaporation is most rapid near the fire, and decreases with the distance from the fire. It may be 21 pounds of water per square foot of heating-surface per hour from the surface immediately over the fire, and only 1 pound per square foot of heating-surface per hour from the surface at the point furthest from the fire, or where the gases leave the boiler.

Although the rate of evaporation varies greatly for different portions of the heating-surface, it is usual to measure the evaporation by the average rate of evaporation per square foot of total heating-surface per hour.

Heating-Surface of Fire-Boxes of Boilers of the Locomotive Type.—The capacity of the fire-box depends upon the quantity and character of the coal burnt. The more bituminous the coal the greater is the area required in the fire-box above the top of the fuel. Anthracite coal gives up its heat principally upon the fire-grate and evolves only a small quantity of combustible gases; it therefore requires much less space in the fire-box above the fuel than bituminous coal which evolves a large quantity of combustible gases. These gases unite with oxygen before combustion takes place, and considerable space is required above the fuel for their mixture and combustion.

Radiant heat is the most effective for heating purposes, and the heating-surfaces of the fire-box absorb radiant heat from the incandescent fuel on the fire-grate and from the flames. It has consequently a much higher evaporative efficiency than any other portion of the heating-surface. The quantity of radiant heat evolved from coal is equal to about fifty per cent. of the total heat due to its combustion. The fire-box should have sufficient capacity, or area above the top of the fuel, to admit of free development of flame and the production of the greatest quantity of radiant heat, and also to ensure efficient combustion of the fuel-gases before they enter the tubes, which is essential to economical combustion.

When gases and solid particles of fuel enter the tubes unconsumed they are practically wasted. Solid particles of fuel are prevented from being drawn into the tubes, and combustion of the fuel-gases is promoted, by the employment of a brick-arch in the fire-box, which absorbs radiant heat and gives it up to the gases as they ascend from the fire, thus assisting their ignition and giving time for their combustion, by hindering and prolonging their passage from the bed of fuel to the tubes. The gases are deflected by the brick-arch towards the fire-hole, where they meet air admitted through the deflector scoop at the fire-door.

The brick-arch furnishes the necessary igniting temperature for promoting the combustion of the mixture of air and gases. The space between the brick-arch and the tubes forms a small combustion-chamber, which assists the development of flame and conduces to efficient combustion, but the run of the gases is much too small, before entering the tubes of the boiler, to effect complete combustion of the gases before they leave the fire-box. By lengthening the fire-box and removing the tube-plate a good distance from the fire, the tube-ends are protected from destructive heat, and a greater portion of the heat is absorbed before entering the tubes.

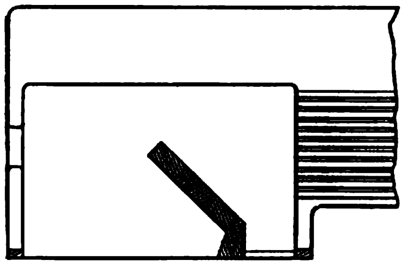


Fig. 30.—Fire-box of boiler.

Modified Fire-Boxes.—A modified form of locomotive fire-box is shown in Fig. 30. It is shallow, and of sufficient length to permit the employment of an efficient combustion-chamber, which is formed by a brick arch.

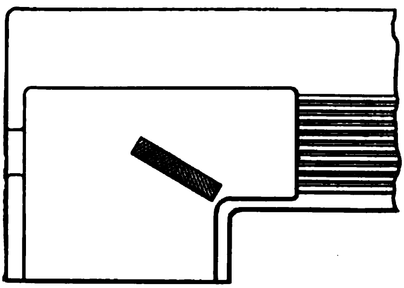


Fig. 31.—Fire-box of boiler.

Another form of fire-box with combustion chamber is shown in Fig. 31. The fire-box is carried a short distance into the barrel of the boiler. Air may be supplied above the fire through hollow stays in the sides of the fire-box, the total area of aperture of which should be equal to about one-fiftieth of the area of the fire-grate surface.

These arrangements facilitate free development of flame and favour economical combustion. A large area of crown-plate of the fire-box is presented for the absorption of radiant heat, which conduces to evaporative efficiency.

The Evaporative Value of a Square Foot of Fire-Box Heating-Surface of a Steam-Boiler has been determined approximately by experiment. In an experiment with a locomotive boiler, it was found that, one square foot of fire-box heating-surface evaporated 16·67 lbs. of water per hour. This is a much lower result than that obtained in other experiments. In the experiment described on page 127 the highest result was 36·9 lbs. of water evaporated per hour.

In experiments with four different locomotive boilers, the results obtained were respectively 19·40 lbs.; 20·64 lbs.; 21·96 lbs.; and 23·12 lbs. of water evaporated per square foot of fire-box heating-surface per hour: this gives an average of 21·28 lbs. Therefore it may be assumed that, one square foot of the heating-surface of the fire-box of a boiler of the locomotive type, will evaporate on an average 21 lbs. of water per hour.

In an experiment with the boiler of a portable-engine, of the locomotive type, one square foot of fire-box heating-surface evaporated 16·85 lbs. of water per hour.

Water-Space of Fire-Boxes.—The width of the water-space surrounding fire-boxes and combustion-chambers, depends principally upon the weight of fuel burnt per hour on the fire-grate, the larger the quantity of fuel burnt the larger should be the water-space. In order to secure efficient evaporation and durability of the plates, the width of water-spaces should never be less even in the smallest boilers than $3\frac{1}{2}$ inches at the bottom; and the width at the top of the water-space should be at least one-half greater than that of the bottom, in order to permit free movement of the convection currents and obtain free circulation and efficient evaporation. This proportion refers to boilers using good water; when the feed-water is impure, or largely charged with foreign matter, the water-space should not be less than five inches in any case.

Heating-Surface of the Smoke-Tubes of Steam-Boilers.—The greatest proportion of heating-surface is presented to a volume of gas when it is divided into small streams by passing through small tubes. The smoke-tubes should be placed at sufficient distance above the fire to permit the greatest possible efficiency of combustion of the fuel-gases before they enter the tubes. Efficient combustion of gases cannot be effected in small tubes of from $1\frac{1}{2}$ to 4 inches in diameter, such as boiler-tubes, of a greater length than that equal to 15 times the internal diameter of the tube. Beyond that point the flame is attenuated and feeble and is liable to become extinguished, because the comparatively cold portion of the tubes may cool the gases below the temperature of ignition. Where combustion is continued beyond that distance, it is very imperfect and results in loss from the escape of unconsumed fuel to the chimney.

The evaporative power of tube-heating-surface varies considerably. In the case of two locomotive boilers, one evaporated 6·8 lbs. and the other 9·6 lbs. of water per square foot of the whole heating-surface of the tubes per hour. Taking the higher value and assuming that each square foot of the fire-box heating-surface evaporated 21 lbs. per hour, and that the ratio of the fire-box surface to the tube-surface was as 1 to 3. Then the evaporation of this boiler would be $1 \times 21 = 21$ lbs. by the fire-box, and $3 \times 9{\cdot}6 = 28{\cdot}6$ lbs. by the tubes, or a total of $21 + 28{\cdot}6 = 49{\cdot}6$ pounds of water evaporated from each square foot of total heating-surface per hour. Showing that the heating-surface of the tubes evaporated $(28{\cdot}6 \times 100) \div 49{\cdot}6 = 58$ per cent. of the total evaporation.

It is probable that the whole evaporation from the tube-surface of multitubular boilers seldom exceeds 8 lbs. of water per square foot of tube-surface per hour, and in many cases it is only about 6 pounds.

The relative value of the fire-box heating-surface and the tube heating-surface of steam-boilers cannot be accurately determined, because it varies with the intensity of the draught. With a light draught the fire-box absorbs the largest quantity of heat, and the smallest quantity passes through the tubes; and less heat is absorbed by the fire-box and a greater quantity passes through the tubes as the draught is increased.

In an experiment with the boiler of a locomotive engine used as a stationary boiler the fire-box heating-surface evaporated twenty times as much water as the tube heating-surface, with natural draught; but when the draught was increased artificially, the fire-box heating-surface only evaporated seven and one-half times as much as the tube heating-surface.

In another experiment with a locomotive boiler, 1 square foot of fire-box heating-surface evaporated as much water as 4 square feet of tube heating-surface.

In another experiment with a locomotive boiler, 1 square foot of fire-box heating-surface evaporated as much water as 2·7 square feet of tube heating-surface.

In an experiment with the boiler, of a portable engine, 1 square foot of fire-box heating-surface evaporated as much water as 3·65 square feet of tube heating-surface.

It may be assumed that when the sides of the fire-box are inclined, or taper to facilitate evaporation, 1 square foot of the fire-box heating-surface of a locomotive boiler will on an average evaporate as much water to steam as $3\frac{1}{2}$ square feet of tube heating-surface. If the sides of the fire-box are vertical, it may be assumed that on an average, 1 square foot of fire-box heating-surface is equal in evaporative power to 3 square feet of tube heating-surface of a locomotive boiler.

Relative Evaporative Power of different Portions of the Length of Boiler-Tubes.—Many experiments have been made to determine the relative evaporative value of different portions of the length of boiler-tubes. The following are the particulars of a few of such experiments:—

In an experiment with a multitubular boiler, the barrel was divided by partitions into 5 lengths or sections each 12 inches long, when the following quantities of water were evaporated from the sections of the tubes:—

Section of tubes . . .	1	2	3	4	5
Water evaporated in lbs. .	65	29	16	13	10

The first section of tubes was next the fire-box.

In an experiment with a locomotive boiler, the evaporation from the first 6 inches in length of the tubes was found to be the same as that of the fire-box, and the first 6 inches in length did more work than the remainder of the length of the tubes.

In an experiment with a multitubular boiler, the tubes were divided into six sections, and the percentage of evaporative duty performed by each section was as follows:—

Section of tubes, commencing at the furnace-end . .	1	2	3	4	5	6
Percentage of the evaporation performed by each section .	47	23	14	8	5	3

In an experiment with an externally fired plain cylindrical boiler, divided into three sections of equal length, with a coal consumption of 100 lbs. per hour, 4·67 lbs. of water were evaporated from 58° Fahr. per lb. of coal, with a heating-surface of 15 square feet.

The water evaporated by each section was as follows:—

Section of boiler, commencing at the furnace-end . . .	1	2	3
Percentage of evaporation performed by each section . . .	65	24	11

In an experiment with the boiler of a locomotive engine, the boiler was divided into five compartments. The first compartment consisted of the fire-box, including three inches in length of the tubes; the four tube-sections were each 3·02 feet long. When the boiler was fired with coke the quantity of water evaporated per square foot of heating-surface per hour in the first or fire-box compartment was 24·5 lbs., and in the first section of the tubes 8·72 lbs. When fired with block-fuel or briquettes of coal, the quantity of water evaporated per square foot of heating-surface per hour was 36·9 lbs. in the first or fire-box compartment, and 11·44 lbs. in the first tube-section.

The results of the test are given in the following Table:—

TABLE 40.—SHOWING THE PERCENTAGE OF EVAPORATION PERFORMED BY EACH SECTION OF THE BOILER IN A TEST OF A LOCOMOTIVE BOILER.

Vacuum in the Smoke-Box in Inches of Water.	Sections of Boiler-Barrel.				Total of Tubes.	Total of Fire-Box.
	No. 1.	No. 2.	No. 3.	No. 4.		
	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.
·78	15	7	3	2	30	70
1·56	17	9	4	3	35	64
2·34	18	7	6	3	36	64
3·12	15	8	5	3	32	67
3·90	21	9	6	4	41	58

Average relative Evaporative Value of different Portions of the Heating-Surfaces of different kinds of Steam-Boilers.—The results of experiments on the value of different portions of the heating-surfaces of steam-boilers vary considerably. It may be assumed that the rate of evaporation from the heating-surface of boiler-tubes diminishes one-half at each one-third in length of the tube. On this basis, the evaporative value of different portions of the heating-surface of a locomotive boiler, expressed in terms of the total heating-surface, is approximately as follows:—Heating-surface of the fire-box 65 per cent.; of the first one-third in length of the tubes, next to the fire, 20 per cent.; of the second one-third in length of the tubes, 10 per cent.; of the last one-third in length of the tubes, 5 per cent. Although the latter portion of the tubes has little evaporative value, it is effective in raising the temperature of the feed-water to the point necessary for the rapid generation of steam from it by the more effective portions of the heating-surface.

In a marine return tube-boiler the combustion-chamber allows the gases to become properly mixed and facilitates their combustion; it also permits the development of flame, which radiates a considerable quantity of heat. Hence, the temperature of the products of combustion are practically the same in the combustion-chamber as in the furnace-tube: and the heating-surfaces of the furnace-tube and the combustion-chamber perform about 60 per cent. of the evaporation, and the heating-surfaces of the tubes perform 40 per cent.

In Lancashire and Cornish boilers the heating-surfaces of the furnace-tubes and Galloway-tubes perform nearly all the evaporation, the external heating-surfaces in the flues principally doing duty as water-heaters.

In multitubular boilers with horizontal tubes, the bottom rows of tubes are of little value as heating-surfaces, because the bulk of the gases rise to the top of the fire-box or combustion-chamber, and escape through the top rows of the tubes, the temperature in which is frequently 25 per cent. greater than that in the bottom rows of tubes. This difference of temperature may, however, be partly due to the top rows of tubes being surrounded by steam-bubbles evaporated from the lower tubes, while the bottom rows of tubes are surrounded by water.

Heat-transmissive Capacity of the Heating-surfaces of Steam-Boilers.—In the experiment with the plain cylindrical boiler, described on page 127, 100 lbs. of coal, × 4·67 lbs. of water evaporated, × ·65 the efficiency of evaporation of the first section, = 303 lbs. of water were evaporated per hour, from a heating-surface of 15 square feet = 303 ÷ 15 = 20·2 lbs. of water evaporated per square foot of heating-surface per hour. Taking the heat required to evaporate 1 lb. of water at 1178° — 58° Fahr., the temperature of the feed-water, = 1120 units, then each square foot of heating-surface transmitted 1120 units × 20·2 lbs. of water evaporated = 22624 units of heat per hour.

In the experiment on the locomotive boiler described at page 127, the water evaporated with coke was = 24·5 lbs. per square foot of heating-surface, and if the temperature of the feed-water was 60° Fahr., then the heat required to evaporate one pound of water was = 1178 — 60 = 1118 units; and each square foot transmitted 24·5 lbs. × 1118 units = 27391 units of heat per hour. As 36·9 lbs. of water were evaporated per square foot of heating-surface with coal, each square foot of heating-surface transmitted 1118 units × 36·9 lbs. = 41255 units of heat per hour.

Peclet found that the bottom surface of a boiler, exposed to intense heat from a furnace, evaporated about 20 lbs. of water per square foot of heating-surface per hour; this is equal to 1118 units × 20 = 22360 units of heat transmitted per square foot of heating-surface per hour.

Assuming the temperature of the furnace of the boiler to be 2400° Fahr., the heat-transmissive capacity of one square foot of heating-surface, or the number of units transmitted, from the hot gases outside the heating-surface to the water inside the boiler, per hour per square foot of heating-surface for each degree of difference of temperature is, taking the quantity of heat transmitted in the experiment with the plain cylindrical boiler previously described, as follows:—

$$\text{Heat transmitted} = \frac{22624 \text{ units of heat per hour}}{2400^\circ \text{ Fahr. temperature of furnace}} = 9{\cdot}42 \text{ units of}$$

heat per hour, by each square foot of surface for each degree of difference of temperature.

In the experiment with the locomotive boiler, above described, the heat transmitted was = 27391 ÷ 2400° Fahr. = 11·41 units per hour with coke, and = 41255 units ÷ 2400° = 17·19 units per hour with coal.

In Peclet's experiment 22360 units ÷ 2400° Fahr. = 9·31 units of heat were transmitted per hour.

In an experiment with Thorneycroft's torpedo-boat boiler, 18 lbs. of water were evaporated per square foot of heating-surface per hour from feed-water at 56° Fahr., under an air-pressure of 6 inches of water in the stokehold. The heat required to evaporate one pound of water was = 1178 − 56° = 1122 units: and each square foot of heating-surface transmitted, 18 lbs. × 1122 = 20196 units of heat per hour. The heat transmitted, assuming a furnace-temperature of 2400° Fahr., was = 20196 units ÷ 2400 = 8·41 units per hour by each square foot of surface for each degree of difference of temperature.

In the experiment with an evaporator described on pages 339 and 340, 140·23 lbs. of water were evaporated per square foot of heating-surface per hour, and 1224 units of heat were transmitted per square foot of heating-surface per hour for each degree of difference of temperature.

It will be seen from these experiments that the rate of evaporation varies considerably.

The Rate of Transmission of Heat by water-heating surfaces is proportional to the area exposed to the heat, the time of contact, and to the difference between the temperature due to combustion on the outside, and that of the water on the inside of the metal plates. It is nearly inversely proportional to the thickness of metal when more than ⅜ inch thick, and depends upon the state of the heating-surfaces as regards being clean or encrusted on the water-side, and clean or coated with soot on the fire-side.

The transmissive efficiency varies with the efficiency of the circulation of the water on one side of the plate, and that of the gases on the other side of the plate.

The rate of transmission of heat varies in different kinds of boilers. It may in a general way be estimated approximately by the following formula, which applies to well-arranged boilers with clean heating-surfaces:—

Let R = the average rate of transmission of heat in units per hour per square foot of heating-surface for each degree of difference of temperature.
C = a constant varying for different types of boilers as follows:—

C = ·5 for Cornish boilers.	C = ·7 for marine return tube-boilers.
C = ·6 for Lancashire boilers.	C = ·8 for locomotive boilers.

Then $R = \sqrt[3]{\text{(working pressure of steam)}} \times C$.

For instance, the rate of transmission through the plates of a Lancashire boiler having clean heating-surfaces, with a working pressure of steam of 100 lbs. per square inch is approximately,

$$= \sqrt[3]{100} \text{ lbs. pressure per square inch} \times {\cdot}6 = 10 \times {\cdot}6 = 6$$

units of heat per square foot of heating-surface per one degree of difference of temperature per hour.

When the heating-surfaces are only moderately clean, multiply the result obtained by this rule by ·8.

The Quantity of Heat transmitted at a given rate is proportional to the area of the heating-surface. A steam-boiler with well-arranged heating-surfaces, when burning coal which deposits little or no soot or non-conducting coating on the heating-surfaces, will on an average transmit through its effective water heating-surfaces 73 per cent. of the total quantity of heat theoretically due to the complete combustion of coal in its furnace.

On this basis the rate of transmission of heat by the heating-surfaces of a steam-boiler may be calculated by the following formula:—

Let C = the calorific power of one lb. of the fuel.
p = the percentage of heat available for transmission, which in well-arranged boiler averages, ·73.
W = the total weight of coal burnt per hour.
T = the absolute temperature of the gases produced.
t = the absolute temperature of the water in the boiler.
H = the heating-surface of the boiler in square feet.

Then the number of units of heat, U, absorbed per square foot of heating-surface per one degree of difference of temperature per hour is:—

$$U = \frac{C \times p \times W}{(T - t) \times H}.$$

Example: Required the rate of transmission of heat by the heating-surfaces of the boiler of a portable steam-engine, which consumed 225 lbs. of coal in five hours, calorific power of the coal = 14300 units per lb.;

temperature of the water in the boiler 349° Fahr. = 349 + 461 = 810° Fahr. absolute temperature. Absolute temperature of the furnace = 2820° Fahr.; heating-surface, 230 square feet.

Then 225 lbs. of coal ÷ 5 hours = 45 lbs. of coal burnt per hour,

and $\frac{14300 \text{ units} \times \cdot 73 \times 45 \text{ lbs. of coal}}{(2820 - 810) \times 230 \text{ square feet}} = 1{\cdot}016$ units of heat absorbed per square foot of heating-surface per one degree of difference of temperature per hour.

This is the average rate of transmission for the whole of the heating-surface. It is greatest from the fire-box heating-surface, and the rate decreases as the gases fall in temperature on their way to the chimney.

The Area of Heating-surface required to transmit a given number of units of heat may be found by the following formula, in which the notation is the same as that in the previous formula:—

$$\text{Area of heating-surface in square feet} = \frac{C \times p \times W}{(T - t) \times U}.$$

Example: Required the area of the heating-surface of a steam-boiler to transmit 5 units of heat per square foot of heating-surface, per one degree of difference of temperature per hour; calorific power of the coal 14500 units per lb.; consumption of coal 200 lbs. per hour; absolute temperature of the furnace 2961° Fahr.; absolute temperature of the water in the boiler 825° Fahr.

Then $\frac{14500 \text{ units} \times \cdot 73 \times 200 \text{ lbs. of coal}}{(2961^\circ - 825^\circ) \times 5 \text{ units}} = 208$ square feet of heating-surface are required for this boiler.

The area of heating-surface required for the efficient transfer of heat in a steam-boiler depends principally upon the form and position of the heating-surfaces. It should be sufficient to effect the reduction of the gaseous products of combustion to the lowest practicable temperature before they are discharged into the chimney. The least extent of heating-surface is permissible where it is presented to the fuel-gases in the best manner for the rapid and efficient absorption of heat, or at right angles to the current of the hot gases, so that the gases may impinge on the heating-surfaces.

The steam-producing efficiency of a boiler depends upon the efficiency of its heating-surface in absorbing heat, and favouring free and rapid circulation of the water; therefore, the area of heating-surface, necessary for a given evaporation, varies according to its arrangement in different types of boilers.

The Area of Total Heating-Surface required in practice for different kinds of steam-boilers burning good coal, with easy firing, all with natural draught, except locomotive and portable-engine boilers, which have exhaust steam-blast in the chimney, may be found approximately by the following formula:—

Area of heating-surface in square feet required for a given evaporation =

$$\frac{\text{Pounds of water evaporated per hour} \times C}{\text{Total heat of the working-pressure of the steam.}}$$

In which C is a constant varying for each type of boiler, as follows:—

Locomotive boilers	. C = 90	Externally-fired plain cylindrical boilers . . .	C = 280
Marine return-tube boilers	C = 180	Portable-engine boilers of the locomotive type	. C = 530
Lancashire boilers . .	. C = 205		
Cornish boilers . .	. C = 220		

Example 1: Required the total heating-surface of a Lancashire boiler, to evaporate 5000 lbs. of water per hour, to steam of 95 lbs. per square inch absolute pressure.

Then the total heat of steam of 95 lbs. per square inch absolute pressure is 1181·5 units per lb., and,

$$\frac{5000 \text{ lbs. of water} \times 205 \text{ constant}}{1181{\cdot}5 \text{ units per lb. of steam}} = 870 \text{ square feet, the total heating-}$$

surface required for this steam-boiler.

Example 2: Required the total heating-surface of a portable-engine boiler, of the locomotive type, to evaporate 500 lbs. of water per hour to steam of 150 lbs. per square inch absolute pressure.

Then, the total heat of steam of 150 lbs. per square absolute pressure is 1192 units per lb., and,

$$\frac{500 \text{ lbs. of water} \times 530 \text{ constant}}{1192 \text{ units per lb.}} = 223 \text{ square feet,}$$

the heating-surface required for this boiler.

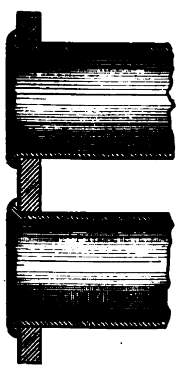


Fig. 32.—Section of smoke-tubes.

Smoke-Tubes of Steam-Boilers.—Boiler-tubes, or smoke-tubes, of multitubular boilers, shown in Fig. 32, are of wrought-iron, steel, copper, and brass.

Draught-Area of Smoke-Tubes.—The area of aperture of boiler-tubes, or smoke-tubes of multitubular boilers, that is, the area through which the products of combustion pass, is generally proportioned to the area of the fire-grate. The smaller the diameter of the tubes, the greater the heating-surface obtainable in a given space. Small tubes are liable to become choked with ashes and soot, and are not suitable for coal which yields a tarry deposit, because it coats the tubes to such an extent as to sensibly diminish their area, and being a non-conductor of heat it reduces their evaporative efficiency. The larger the tube, the greater the distance travelled by the flame before extinction.

Excessive area through the smoke-tubes produces a low velocity of the gaseous products of combustion and a bad draught, and may cause extravagant consumption of fuel. It generally results in a heavy deposit of soot in the tubes, and in the products of combustion passing through only a few of the tubes instead of the whole of them.

Deficient area through the tubes produces a high velocity of the fuel-gases, which may pass through the tubes so rapidly as not to allow sufficient time for the absorption of the heat, causing much loss of fuel.

The cross-sectional area of the smoke-tubes of steam-boilers with natural draught should not be greater than two-ninths, or less than one-eighth of the area of the fire-grate; one-fifth is a common proportion.

When forced draught is employed, the area through the tubes may be from one-tenth to one-twelfth the area of the fire-grate, according to the strength of the draught.

The heat of gases in passing through tubes is diffused to the tube-surface by radiation. The heat is supposed to be concentrated in the axis of the tube, and passes from the centre to the circumference with very great velocity.

Area through the Smoke-Tubes of Steam-Boilers of Various Types.—The area through the tubes of a steam-boiler, or the area of aperture of the tubes, is = (internal diameter of tube)^2 × ·7854 × number of tubes.

The following are good proportions for the aperture-area of smoke-tubes.—

Area through the tubes of locomotive boilers in square inches = area of fire-grate in square feet × 19.

Area through the tubes of portable-engine boilers of the locomotive type in square inches = area of fire-grate in square feet × 26.

Area through the tubes of marine return-tube boilers with natural draught, including the area of aperture of the stay-tubes, in square inches = area of fire-grate in square feet × 28.

Area through the tubes of horizontal internally-fired cylindrical multitubular boilers with natural draught, including the area of aperture of the stay-tubes, in square inches = area of fire-grate in square feet × 40.

Length of Boiler Smoke-Tubes.—The length of the smoke-tubes of multitubular steam-boilers depends principally upon the strength of the available draught. The ratio of the length to the diameter of the smoke-tubes of boilers with natural draught is limited by the necessity of not presenting excessive resistance to the passages of the fuel-gases.

The length of the smoke-tubes may be as great as that equal to 40 times the internal diameter of the tubes for boilers with a strong natural draught, but it should not exceed 30 times the internal diameter for a moderately strong natural draught.

The length of the tubes of locomotive and other boilers with a steam-blast in the chimney may be as great as that equal to 120 the internal diameter of the tubes.

The length of the tubes should be sufficient to absorb the heat of the fuel-gases, and prevent more heat from being carried through them than is unavoidable. The stronger the draught the longer may be the run of the gases through the boiler.

When the tubes are deficient in length, the fuel-gases pass through them before their available heat has become absorbed by the surfaces, resulting in an excessively high temperature in the chimney and waste of fuel.

The length of run of fuel-gases through small tubes, such as are used for smoke-tubes, necessary for efficient absorption of heat is 15 feet. The quantity of water evaporated per lb. of coal decreases at an average rate of 2 per cent. for each foot of decrease, in length of the tubes from 15 feet to 10 feet. Below that point the evaporative efficiency decreases irregularly but rapidly as the length decreases. But the rapidity of the generation of the steam, or the rate of evaporation, decreases at an average rate of 1 per cent. for each foot of increase of length from 10 feet to 15 feet. Hence, a boiler having smoke-tubes 15 feet long will not generate so much steam in a given time, but will be more economical in evaporation than one having tubes 10 feet long.

Although the fire-box end of smoke-tubes has the greatest evaporative efficiency as previously explained, it is sufficiently accurate for most practical purposes, for tubes within the limits of 8 to 13 feet in length, to assume that evaporative economy is not influenced by the length of smoke-tubes; and also that so long as there is sufficient surface to absorb the available heat and transmit it to the water, the evaporative effect is the same whether the surface is contained in short smoke-tubes of large diameter, or in long tubes of small diameter.

Diameter and Length of Smoke-Tubes of Steam-Boilers of Various Types.—The following are good proportions for the diameter and length of smoke-tubes for several kinds of boilers:—

Internal diameter in inches of the tubes of locomotive boilers = length of tube in inches divided by 90.

Internal diameter in inches of the tubes of portable-engine boilers of the locomotive type = length of tubes in inches divided by 47.

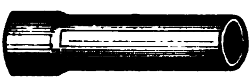


Fig. 33.—Smoke-tube with swelled end.

Internal diameter in inches of the tubes of marine return-tube boilers with natural draught = length of tube in inches divided by 27.

Length of the tube in inches of locomotive boilers = internal diameter of tube in inches multiplied by 90.

Length of the tubes in inches of portable-engine boilers of the locomotive type = internal diameter of the tube in inches multiplied by 47.

Length of the tubes in inches of marine return-tube boilers with natural draught = internal diameter of tube in inches multiplied by 27.

The smoke-tubes of steam-boilers are frequently enlarged slightly at one end, as shown in Fig. 33, to facilitate their withdrawal.

Heating-Surface of Smoke-Tubes.—The whole external surface of the smoke-tubes of steam-boilers is generally included in calculating their heating-surface. The heating-surface of smoke-tubes may be found by this *Rule*:—

$$\text{Heating-surface of one tube in square feet} = \frac{\text{External diameter of tube in inches} \times 3{\cdot}1416 \times \text{length of tube in inches}}{144}.$$

For instance, the heating-surface of a smoke-tube of $3\frac{1}{4}$ inches external diameter, and 6 feet 9 inches long, is $= 3{\cdot}25 \times 3{\cdot}1416 = 10{\cdot}21$, the circumference of the tube $\times$ 81 inches length of tube $= 827{\cdot}01$ square inches $\div\ 144 = 5{\cdot}743$ square feet.

The heating-surface of smoke-tubes per foot in length is given in the following Table:—

TABLE 41.—HEATING-SURFACE OF THE SMOKE-TUBES OF STEAM-BOILERS, IN SQUARE FEET PER FOOT OF THE LENGTH OF THE TUBE.

External Diameter of Tube in inches.	Heating Surface per Foot in Length in Square Feet.	External Diameter of Tube in inches.	Heating Surface per Foot in Length in Square Feet.
1	·2618	3	·7853
$1\frac{1}{8}$	·2945	$3\frac{1}{8}$	·8181
$1\frac{1}{4}$	·3272	$3\frac{1}{4}$	·8508
$1\frac{3}{8}$	·3600	$3\frac{3}{8}$	·8835
$1\frac{1}{2}$	·3926	$3\frac{1}{2}$	·9163
$1\frac{5}{8}$	·4254	$3\frac{5}{8}$	·9490
$1\frac{3}{4}$	·4580	$3\frac{3}{4}$	·9817
$1\frac{7}{8}$	·4909	$3\frac{7}{8}$	1·0144
2	·5236	4	1·0472
$2\frac{1}{8}$	·5563	$4\frac{1}{4}$	1·1126
$2\frac{1}{4}$	·5890	$4\frac{1}{2}$	1·1781
$2\frac{3}{8}$	·6217	$4\frac{3}{4}$	1·2362
$2\frac{1}{2}$	·6545	5	1·3680
$2\frac{5}{8}$	·6872	6	1·5708
$2\frac{3}{4}$	·7200	7	1·8326
$2\frac{7}{8}$	·7540	8	2·0944

The heating-surface of a tube from this Table, multiplied by the length of tube in feet and by the number of tubes, will give the heating-surface of the smoke-tubes of a steam-boiler.

Height of Smoke-Tubes of Marine Return-Tube Boilers.—The top of the top row of the smoke-tubes of marine return-tube boilers should not be placed nearer to the top of the shell than at a distance equal to one-third the internal diameter of the shell of the boiler, but the distance is preferably made equal to the diameter of the shell in inches $\times$ ·38, in order to obtain ample depth of water above the smoke-tubes, and also sufficient height for the steam to rise tranquilly without priming.

The Space between the Smoke-Tubes of multitubular boilers should be sufficient to permit free circulation of the water, and allow the steam to rise freely from their surfaces. It may be equal to one-half the external diameter of the tube, and should not in any case be less than equal to the external diameter of tube × ·36.

The tubes should not be nearer to the shell of a boiler than 3½ inches, and they should be arranged in horizontal and vertical rows as shown in Fig. 34 to facilitate the flow of the convection currents. They should not be spaced zigzag, as shown in Fig. 35, as this arrangement impedes free circulation and encourages sediment to lodge on the upper surface of the tubes.

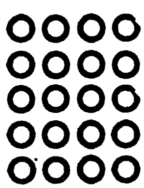


Fig. 34.—Section of smoke-tubes.

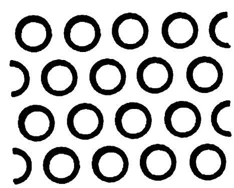


Fig. 35.—Section of smoke-tubes.

The tubes should be widely spaced in boilers using bad feed-water. When the tubes are placed very closely together, if the water be hard, scale is liable to bridge the space between the tubes and stop the circulation of the water. The steaming capacity of boilers is frequently impaired by retarded evaporation due to crowded tubes.

Boiler-Tubes are injured by bad water and inferior coal. The wear of smoke-tubes is principally due to the attrition of cinders and small particles of coal drawn through the tubes by the draught. The tubes wear very irregularly. The upper rows of tubes do the most work, as the levity of the fuel-gases naturally induces their escape through the upper tubes. The top row of tubes frequently wears about 20 per cent. thinner than the bottom row. With natural draught the most of the fuel-gases are drawn through the tubes nearest to the chimney, but with artificial draught all the tubes should be equally effective.

The Thickness of Smoke-Tubes of from 1½ to 4½ inches diameter for steam-boilers may be found by the following rule, in which the working-pressure is that above the atmosphere.

Thickness of the smoke-tubes of steam-boilers, in inches =

$$\frac{\text{External diameter} \times \text{working pressure of steam}}{C}.$$

Where C is a constant; C = 4000 for mild-steel tubes.
C = 3000 for wrought-iron tubes.
C = 2500 for brass tubes.
C = 2000 for copper tubes.

Example 1. Required the thickness of wrought-iron smoke-tubes 3½ inches external diameter, for a marine return-tube boiler having a working-pressure of 150 lbs. per square inch.

$$\text{Then } \frac{3{\cdot}25 \text{ inches diameter} \times 150 \text{ lbs. working-pressure}}{3000} = {\cdot}163 \text{ inch,}$$

or nearly 8 BWG thick.

Example 2. Required the thickness of the copper-tubes of a locomotive boiler, for a working-pressure of steam of 160 lbs. per square inch, the external diameter of the tubes being $1\frac{5}{8}$ inch.

$$\text{Then } \frac{1{\cdot}625 \text{ inches diameter} \times 160 \text{ lbs. pressure}}{2000} = {\cdot}13 \text{ inch. or a little}$$

more than $\frac{1}{8}$ inch thick, or slightly over 11 BWG in thickness.

Smoke-tubes of brass and copper should be solid drawn.

Smoke-tubes are fixed in their places by expanding and beading over the ends, and a ferrule of steel or copper is driven into the fire-box-end of the tube to assist in securing the tubes, and to protect their ends from the fiercest action of the flames. The effect of beading the tubes is to give them great holding power, and to make each tube an effective stay for the tube-plates.

Expansion of Smoke-Tubes by Heat.—The linear expansion of smoke-tubes is considerable. It may be computed from the Table on page 16. For boilers with working-pressures of from 150 to 200 lbs. per square inch, the approximate expansion of the smoke-tubes may be simply calculated by multiplying the length of the tubes in inches by one of the following numbers:—

Brass smoke-tubes . .	·0040	Mild-steel smoke-tubes .	·0026
Copper smoke-tubes . .	·0036	Wrought-iron smoke-tubes .	·0025

For instance, in a Locomotive-boiler working with steam of 160 lbs. per square inch, having brass tubes 10 feet 8 inches long, the tubes will expand, 128 inches × ·004 = ·512, or a little more than $\frac{1}{2}$ inch. This shows that it is necessary to provide ample elasticity in the tube-plates of multitubular boilers to accommodate the expansion of the tubes.

Tests for Smoke-Tubes of Wrought-Iron and Steel.—The tensile strength of the metal of which boiler smoke-tubes are composed should not be less than 25 tons per square inch for wrought-iron tubes, or less than 27 tons per square inch for mild-steel tubes. The ductility of which, measured by elongation, should not be less than 15 per cent. Each tube should be subjected to an internal hydraulic test of 500 lbs. per square inch.

Tests for Brass and Copper Smoke-Tubes.—A test-piece, cut from a tube of brass or copper, 30 inches long, annealed and filled with rosin, should withstand being doubled until the extremities touch each other, without showing defects. A piece of tube 30 inches long, not annealed, filled with rosin, and placed on supports 20 inches apart, should withstand bending to a deflection of 3 inches without showing defects. A piece of tube 4 inches long, annealed and sawn lengthways, should withstand being doubled inside out without showing defects. Each tube should be subjected to an internal hydraulic pressure of 400 lbs. per square inch.

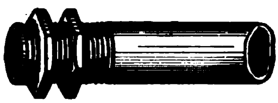


Fig. 36.—Stay-tube.

Brass smoke-tubes are frequently composed of 70 parts of copper and 30 parts of zinc, impurities not exceeding 1 per cent.

Stay-Tubes, shown in Fig. 36, are employed to support the tube-plates of marine return-tube boilers, they are generally of the same external diameter as the smoke-tubes, but are of greater thickness. They are screwed with a fine thread into one or both tube-plates and secured by nuts. One end of the stay-tube is frequently either thickened or expanded.

The thickness of stay-tubes of the same external diameter as smoke-tubes, may be found by the following rule, in which the working-pressure of the steam is that above the atmosphere.

Thickness of stay-tubes of steam-boilers in inches =

$$\frac{\text{External diameter} \times \text{Working-pressure of steam}}{C}.$$

Where C is a constant; C = 1700 for mild-steel stay-tubes.
C = 1300 for wrought-iron stay-tubes.

Example: Required the thickness of wrought-iron stay-tubes $3\frac{1}{4}$ inches external diameter, for a marine return-tube boiler having a working-pressure of 150 lbs. per square inch.

$$\text{Then } \frac{3{\cdot}25 \text{ inches} \times 150 \text{ lbs. pressure}}{1300} = {\cdot}375, \text{ or } \tfrac{3}{8} \text{ inch thick.}$$

The number of stay-tubes employed in marine return-tube boilers is generally equal to from one-third to one-fourth the number of the smoke-tubes. The total sectional area of stay-tubes may be computed by the rule on page 239.

Relative Value of various forms of the Heating-Surface of Steam-boilers.—As the steam-generating power of different forms of heating-surface varies considerably, it is essential to economical evaporation to distribute the heating-surfaces in the best manner to effect rapid and efficient absorption of the heat. The heating-surfaces should be arranged so as to intercept the radiant heat from the fuel as effectively as possible. The heating-surfaces of boilers having the furnace placed inside the boiler, can be most favourably arranged for receiving the greatest effect from radiant heat. The greater the extent of surface exposed to the direct rays of heat from the fuel the greater the evaporative efficiency. The proper distribution of the heating-surfaces is a more important factor in economical evaporation than the amount of the heating-surface.

The best form of heating-surface, as regards evaporative efficiency, is a flat horizontal surface above the fire, or parallel to the surface of the fire, such as the top or crown-plate of a locomotive fire-box. Heating-surface concave to the fire, such as the crown of a circular furnace, is nearly as efficient as a flat surface, and it promotes circulation by facilitating the descent of cooler water to replace the ascending current. The best position for any form of heating-surface is at right angles to the current of the hot fuel-gases, where it receives the full benefit of radiant heat from the fuel.

The relative value of different forms of heating-surface, compared with flat horizontal surface above the fire, is as follows:—

One square foot of flat horizontal surface above the fire, such as the crown-plate of the fire-box of the boiler of a locomotive-engine	= 1·00
One square foot of circular surface above, and concave to, the fire, such as the crown-plates of the circular furnace of an internally fired boiler	= ·95
One square foot of circular surface above, and convex to, the fire, such as the furnace-plates of an externally fired plain cylindrical or egg-ended boiler	= ·90
One square foot of flat surface at right angles to the current of gases, exposed to direct impingement of flame, such as the fire-box tube-plate of a locomotive boiler	= ·80
One square foot of water-tube surface at right angles to the current of hot gases, such as that portion of the surface of a Galloway tube which faces the fire	= ·70
One square foot of sloping surface at the side of, and inclined towards, the fire, such as the sides of a fire-box when inclined sufficiently to facilitate evaporation	= ·65
One square foot of vertical surface at the side of the fire, such as the sides of a fire-box when vertical	= ·50
One square foot of the surface of the tubes of a locomotive boiler, contained in a length not exceeding 3 feet from the fire-box tube-plate	= ·30

Horizontal surfaces below the fire and the under portions of internally heated tubes, have practically no evaporative value, and cannot be considered as effective heating-surface, therefore the lower half of a furnace-tube below the grate-bars should not be included in calculating the heating-surface of a steam-boiler.

Effective Heating-Surface.—The bottom of internally-heated tubes for the evaporation of water, such as boiler-tubes, is not effective in absorbing and transmitting heat, because the steam-bubbles cannot escape freely from its surface. If sufficient motion were imparted to the water to sweep the steam-bubbles from the bottom surface as fast as they formed, all the surface of the tube would be equally effective, but as this condition cannot be obtained with natural circulation, only that portion of the tube which covers the gases, or the upper semi-diameter, is of any practical value in evaporating water to steam.

If the whole surface of a tube be divided into four portions, and the efficiency of the top portion taken as 1, and that of the bottom portion as 0, the value of the side portions will be intermediate between that of the top and bottom portions, consequently the efficiency of each of the sides will be $\frac{1}{2}$, and the average of the whole circumference is $(1 + \cdot 5 + \cdot 5 + 0) \div 4 = \frac{1}{2}$, showing that only one-half the quantity of heat is utilized by the whole surface of the tube, which would be utilized if all the surface were equally as effective as the top surface. Hence the area of effective heating-surface of a boiler-tube or smoke-tube is only one-half the total area of its surface.

The Evaporative Power of a Steam-Boiler can only be accurately determined by experiment, because it depends on variable conditions, and every portion of the heating-surface is not equally effective.

Evaporative efficiency depends principally upon the proportion of the heating-surface to the quantity of coal burnt per hour. A small proportion of heating-surface to the quantity of coal burnt results in waste-heat passing into the chimney. The evaporative value of the heating-surface of a steam-boiler with free circulation of the water, and fired with good coal, when its arrangement and proportions are within the limits of, and according to, modern practice in boiler-making, may be calculated approximately by the following formula :—

Let H = the total heating-surface of the steam-boiler in square feet.
W = the weight of coal in pounds burnt on the fire-grate of the boiler in one hour.
P = the theoretical calorific power of the coal burnt, or the total heat of combustion of one pound of dry coal, in pounds of water evaporated from and at 212° Fahr.
E = the estimated evaporative power of the boiler, in pounds of water, evaporated per pound of fuel from and at 212° Fahr.

$$E = \frac{H}{W \times C + H} \times P.$$

In which C is a constant, deduced from experiments with good boilers, which varies for each class of steam-boiler as follows :—

C = ·40 for Galloway boilers.
C = ·42 for locomotive boilers, burning coal.
C = ·45 for Lancashire boilers.
C = ·50 for marine return-tube boilers.
C = ·65 for Cornish boilers.
C = ·90 for externally-fired plain cylindrical boilers.
C = 1·60 for boilers of portable engines of the ordinary locomotive type.

All these constants refer to boilers with natural draught, except the locomotive and portable-engine boilers which have exhaust-steam blast in the chimney, applied in the usual manner.

Example 1.—Required the evaporative power of a Lancashire boiler which burnt 540 lbs. of coal on the fire-grates in one hour; the theoretical evaporative value of 1 lb. of the coal from and at 212° Fahr. is 14 lbs. of water; the total heating-surface of the boiler is 920 square feet?

Then 540 lbs. of coal × ·45 constant = 243 + 920 square feet = 1163, and 920 ÷ 1163 = ·79 × 14 lbs. = 11·06 lbs. of water evaporated by the boiler per lb. of coal from and at 212° Fahr., the estimated evaporative power of this boiler.

Example 2.—Required the evaporative power of a portable-engine boiler which consumed 34 lbs of coal on its fire-grate in one hour, the theoretical evaporative power of 1 lb. of the coal from and at 212° Fahr. is 15·4 lbs.; the total heating-surface of the boiler is 216 square feet?

Then 34 lbs. of coal × 1·6 constant = 54·4 + 216 = 270·4 and 216 ÷ 270·4 = ·8 × 15·3 lbs. = 12·24 lbs. of water evaporated by the boiler from and at 212° per lb. of coal, the estimated evaporative power of this boiler.

Efficiency of the Heating-Surface of Steam-Boilers.—The efficiency of the heating-surface of a steam-boiler is the ratio of the quantity of the heat transmitted to the total quantity available for transmission. The evaporative efficiency of steam-boilers may be calculated approximately by the following formula, in which the notation is the same as that in the previous formula:—

Estimated evaporative efficiency of the heating-surface of a steam-boiler =

$$\frac{H}{W \times C + H}.$$

Example.—Required the efficiency of a marine return-tube steam-boiler which consumed 300 pounds of coal on its fire-grate in one hour; the total heating-surface of the boiler is 435 square feet.

Then 300 lbs. of coal × ·50 constant = 150 lbs. + 435 square feet = 585, and 435 ÷ 585 = ·74, the estimated efficiency of this boiler.

These rules give the effect or value of the heating-surface of steam-boilers frequently obtained in practice under ordinary working conditions with ordinary draught.

When air is forced under pressure into the furnace of the boiler by any efficient arrangement of forced draught the evaporation may be increased from 20 to 50 per cent.

Efficiency of Steam-boilers.—The efficiency of a steam-boiler is represented by the following expression:—

$$\text{Efficiency of steam-boiler} = \frac{\text{Heat absorbed by the water}}{\text{Heat developed by the fuel}}.$$

Example.—Required the efficiency of a boiler which evaporated $8\frac{1}{2}$ pounds of water per pound of coal from feed-water at 102° Fahr.; temperature of the steam in the boiler 367° Fahr.; calorific value of the coal used 14,500 units per pound?

Then the total heat of the steam is $= (367^\circ \times \cdot305) + 1082 = 1194$ units per pound.

The factor of evaporation is =

$$\frac{(1194 \text{ units} + 32) - 102^\circ \text{ temperature of feed-water}}{966} = 1\cdot163.$$

The equivalent weight of water evaporated per pound of coal from and at 212° Fahr. is $= 8\cdot5$ lbs. of water $\times\ 1\cdot163 = 9\cdot885$ pounds.

The equivalent amount of heat utilised per pound of coal is $= 9\cdot885$ lbs. of water $\times$ 966 units = 9549 Thermal units.

The efficiency of the boiler is =

$$\frac{9549 \text{ units of heat absorbed by the water}}{14500 \text{ units of heat developed by the coal}} = \cdot658, \text{ or say 66 per cent.},$$

showing that the heat utilized in this boiler is only 66 per cent. of the calorific value of the coal used.

The Efficiency of Steam-boilers of various kinds under ordinary working-conditions may be estimated approximately by the following formula, applicable in a general way to all types of internally-fired boilers.

Let R = the ratio of the heating-surface to the coal consumption, or the number of square feet of heating-surface of the boiler per pound of coal consumed per hour.

$$\text{Efficiency of a steam-boiler} = \frac{R}{R + \cdot7}.$$

Example.—Required the efficiency of a steam-boiler consuming 550 pounds of coal per hour, having a total heating-surface of 1100 square feet?

Then the ratio of the heating surface to the coal consumption is $1100 \div 550 = 2$, and $\dfrac{2}{2 + \cdot7} = \cdot74$, the efficiency of the boiler, or the percentage of the available heat actually utilized in the production of steam.

When an economiser, or feed-water heater, is placed between the boiler and the chimney, heated by the products of combustion, its heating-surface may be included with that of the boiler and the efficiency of the boiler and economiser may be estimated by this rule.

The Area of Heating-surface for a Given Efficiency and given ratio of heating-surface to coal-consumption may be calculated approximately by adapting the previous formula as follows:—

Area of heating-surface of a steam-boiler in square feet =

$(R + \cdot7) \times$ efficiency $\times$ coal consumption.

Example.—Required the heating-surface of a steam-boiler necessary to secure an efficiency of 77 per cent. with a consumption of 600 pounds of coal per hour, allowing 2·25 square feet of heating-surface per pound of coal burnt per hour?

Then, $(2{\cdot}25 + {\cdot}7) \times {\cdot}77 \times 600$ lbs. of coal $= 1363$ square feet of heating-surface.

Calculations of the Efficiency of the heating-surface of steam-boilers are frequently made by Rankine's formula, which is as follows:—

$$\text{Efficiency of heating-surface} = \frac{E'}{E} = \frac{B \times S}{S + A\,F}.$$

Where E' = the available, and E = the theoretical evaporative power of one pound of a given kind of fuel in an ordinary boiler, in which S = the total area of heating-surface per square foot of fire-grate surface, including that of the feed-water heater, if one be used; and F = the number of pounds of fuel burnt per square foot of fire-grate per hour. A and B are constants found by experience. A is probably approximately proportionate to the square of the quantity of air in lbs. supplied per pound of fuel. B is a fractional multiplier to allow for miscellaneous losses of heat.

Example.—Required the efficiency of the heating-surface of a steam-boiler with natural draught in which 10 pounds of coal are burnt per square foot of fire-grate surface per hour? The heating-surface is 28 times that of the fire-grate surface, or a ratio of 28 to 1?

Then, taking A as $= {\cdot}5$, and $B = {\cdot}9$, the efficiency is =

$$\frac{28 \times {\cdot}9}{21 + (12 \times {\cdot}5)}$$

$= {\cdot}74$ per cent.

This formula admits of the following adaptations:—

The Area of Heating-surface Required for a Given Efficiency of Heating-surface may be found by the following rules:—

Efficiency of the heating-surfaces of a steam-boiler E=

$$E = \frac{B}{1 + (A \times R)}.$$

In which R represents the ratio of the weight of fuel burnt to the area of the heating-surface, found as follows:—

$$R = \frac{\text{lbs. of fuel burnt per square foot of fire-grate surface per hour}}{\text{No. of square feet of heating-surface per square foot of fire-grate}},$$

$$\text{or } R = \frac{B - E}{E \times A}.$$

The number of times the area of the heating-surface exceeds that of the fire-grate S, is as follows:—

$$S = \frac{\text{weight of fuel burnt per square foot of fire-grate surface per hour}}{R},$$

$$\text{or } S = \frac{\text{lbs. of fuel burnt per square foot of fire-grate surface per hour}}{(B - E) \div (E \times A)}.$$

The values of the constants A and B may in a general way be as follows:—

	Value of A.	Value of B.
Boilers of highest efficiency with natural draught	·5	·95
Boilers of ordinary efficiency with natural draught	·5	·80
Boilers of highest efficiency with forced draught	·3	·97
Boilers of ordinary efficiency with forced daught	·3	·87

As an example of these rules, take the case of a steam-boiler in which the coal-consumption, with natural draught, is 14 pounds per square foot of fire-grate surface per hour, and the heating-surface is 25 times that of the fire-grate surface or a surface-ratio of 25 to 1.

Then, taking values of the constants A and B respectively at ·5 and ·80:—

$$\text{The ratio, R, is} = \frac{14}{25} = \cdot 56,$$

$$\text{or } R = \frac{\cdot 80 - \cdot 625}{\cdot 625 \times \cdot 5} = \cdot 56.$$

The efficiency of the heating-surface, E, is =

$$E = \frac{\cdot 80 \text{ constant}}{1 + (\cdot 5 \times \cdot 56)} = \cdot 625 \text{ per cent.}$$

The number of times the area of the heating-surface exceeds that of the fire-grate is =

$$S = \frac{14 \text{ lbs.}}{\cdot 56} = 25,$$

$$\text{or } S = \frac{14 \text{ lbs. of coal per square foot of fire-grate surface}}{(\cdot 80 - \cdot 625) \div (\cdot 625 \times \cdot 5)} = 25.$$

That is, the ratio of the grate-surface to the heating-surface is as 1 to 25.

The Area of Heating-Surface of a Steam-Boiler required for a given efficiency may be calculated by the previous formula, as shown by the following *example*:—

Example: Required the heating-surface suitable for a steam-boiler to consume 600 lbs. of coal on its fire-grate per hour, at the rate of 16 lbs. per square foot of fire-grate surface per hour, for an efficiency of 80 per cent.

$$\text{Then } \frac{\left(\begin{array}{c}\text{Pounds of fuel burnt per square foot of fire-}\\ \text{grate surface per hour}\end{array}\right)}{\left(\begin{array}{c}\text{Number of square feet of heating-surface}\\ \text{per square foot of fire-grate surface}\end{array}\right)} = \text{ratio}$$

$$\text{and the ratio is} = \frac{1 - \cdot 80}{\cdot 5} = \frac{\cdot 2}{\cdot 5}$$

Showing that the best ratio of heating-surface to the fire-grate surface of this boiler is equal to $\cdot 5 \div \cdot 2 = 2\cdot 5$, or two and one-half times the number of lbs. of coal burnt per square foot of fire-grate surface per hour, or a ratio of 16 lbs. of coal $\times$ 2·5 = 40 to 1. If 20 per cent. be allowed for imperfect combustion, the quantity of coal burnt per hour is $600 \times 1\cdot 2 =$ 720 lbs.; the area of the fire-grate required is $720 \div 16$ lbs. = 45 square feet; and the area of the heating-surface is 45 square feet $\times$ 40 ratio = 1800 square feet.

The Area of Heating-Surface provided in different Types of Steam-Boilers, per indicated horse-power developed by good engines, in regular work, under ordinary working conditions, averages in practice as follows:—

Description of Steam-Boiler.	Square feet of Heating-surface per indicated Horse-power.
Boilers of torpedo-boats, modified locomotive type	1·33
Boilers of yacht engines, locomotive type	1·55
Locomotive-engine boilers	1·4 to 1·85
Locomotive type of boiler, driving stationary engines	1·5 „ 2·00
Marine boilers, modified locomotive type	1·6 „ 2·00
Coil-boilers	1·5 „ 2·25
Marine return-tube boilers	2·6 „ 4·00
do. do. do. for small steamers	3·00
do. do. do. for quadruple-expansion engines	2·6 „ 2·80
do. do. do. average of a large number	3·25
do. do. do. for triple-expansion engines	3·30
do. do. do. for triple-expansion engines of cargo vessels	3·60
do. do. do. for double-expansion engines	4·00
Marine multitubular boilers with tubes in line with furnace	2·7 „ 3·00
Multitubular boilers, average of a number of various kinds	3·50
Lancashire boilers	2·75 „ 4·25
Cornish boilers	4·00 „ 5·50
Plain cylindrical, or egg-ended, boilers	5·00 „ 7·50
Water-tube boilers of various types	5·00 „ 11·60
do. do. for non-condensing engines average	10·00 „ 11·00
do. do. for compound condensing engines average	6·00 „ 8·00
do. do. Babcock and Wilcox	7·00 „ 11·50
do. do. De Naeyer's	7·00 „ 10·00
do. do. Root's	6·50 „ 9·60
do. do. on Root's principle average	6·40 „ 8·00
do. do. Harrison's	6·25 „ 9·50
do. do. with vertical tubes, average	8·00 „ 10·00
Horizontal externally-fired cylindrical multitubular boilers	7·00 „ 11·00
Vertical cross-tube boilers	10·00 „ 12·00

Description of Steam-Boiler.	Square feet of Heating-surface per indicated Horse-power.
Horizontal internally-fired cylindrical multitubular boilers	9·00 to 14·00
Portable-engine boilers, ordinary locomotive type	11·00 „ 16·00
Vertical tubular boilers	14·00 „ 16·00

All these boilers have natural draught, except those of the locomotive type, which have exhaust steam-blast in the chimney.

In marine return-tube boilers having combustion with forced draught, the area of heating-surface provided in practice ranges from 1·75 to 2·5 per square feet indicated horse-power developed by triple-expansion engines. It may, in a general way, be averaged at 2 square feet.

The Quantity of Water per Square Foot of Heating-Surface provided in practice in different types of steam-boilers varies considerably. It averages as follows:—

Description of Steam-Boiler.	Quantity of Water per square foot of Heating-surface. Cubic feet = lbs.
Plain cylindrical, or egg-ended, boilers	1·50 = 93·60
Cornish boilers	·57 = 35·56
Lancashire boilers	·54 = 33·69
Galloway boilers	·47 = 29·32
Marine return-tube boilers	·44 = 27·45
Vertical cross-tube boilers	·24 = 14·97
Stationary horizontal cylindrical multitubular boilers	·20 = 12·48
Vertical tubular boilers	·16 = 9·98
Locomotive-engine boilers	·12 = 7·49
Portable-engine boilers, ordinary locomotive type	·10 = 6·24
Water-tube boilers, Babcock and Wilcock	·15 = 9·80
do. do. average of a number of different kinds	·11 = 6·86
do. do. Root's improved	·13 = 8·12
do. do. De Naeyer's	·096 = 6·00
do. do. on Root's principle, by different makers, average	·08 = 5·00
do. do. Harrison's	·06 = 3·74
Steam fire-engine boilers	·06 = 3·74
Water-tube boilers composed of rows of wrought-iron tubes of 3 inches diameter and under, average	·05 = 3·12

It will be seen that the quantity of water contained in water-tube boilers in proportion to the heating-surface, is very small compared with other types of stationary boilers.

Water-Surface of Steam-Boilers.—The water-surface, or area of the working water-level of steam-boilers, should be sufficiently large to permit tranquil release of the steam, in order to obtain a steady and ample supply of dry steam without inducing priming. When the area of the water-level is deficient, the steam is released more or less violently from the water, which results in priming and a supply of wet steam.

The water-surface at the working-level of a steam-boiler may be reduced as the pressure of the steam is increased, because the steam-bubbles decrease in size as the pressure of the steam is increased. Therefore, engines using high rates of expansion, and consequently steam of very high pressure, such as triple-expansion and quadruple-expansion engines, may be worked with smaller boilers than those using steam of lower pressure.

The area of the working water-level of marine return-tube-boilers required for the evaporation of water to steam of any pressure between 50 and 260 lbs. per square inch, may be found by the following formula, in which the pressure of the steam is that shown by the steam-gauge, or above the atmosphere :—

Water-surface at the working water-level of a marine return-tube-boiler in square feet =

$$\frac{2\cdot6}{\sqrt[3]{\text{working-pressure of boiler}}} \times \text{indicated horse-power of engine.}$$

For boilers of modified locomotive type, use a constant of 2 instead of 2·6.

This Rule may be illustrated by calculating the water-surface required for marine-return-tube-boilers to supply steam for three different types of engines, each of 1000 indicated horse-power, viz., steam of 90 lbs. per square inch for double-expansion engines, of 150 lbs. per square inch for triple-expansion engines, and of 180 lbs. per square inch for quadruple-expansion engines.

Then $\frac{2\cdot6}{\sqrt[3]{90}} = 2\cdot6 \div 9\cdot486 = \cdot274 \times 1000 = 274$ square feet, the area of working water-level required for the boiler to supply steam for the double-expansion engines, and $\frac{2\cdot6}{\sqrt[3]{150}} = 2\cdot6 \div 12\cdot247 = \cdot212 \times 1000 = 212$ square feet, the area of working water-level required for a boiler to supply steam for the triple-expansion engines, and $\frac{2\cdot6}{\sqrt[3]{180}} = 2\cdot6 \div 13\cdot416 = \cdot194 \times 1000 = 194$ square feet, the area of working water-level required for a boiler to supply steam for the quadruple-expansion engines.

The area of water-level would, in each case for convenience of size, be divided between two boilers.

The water-surface of a steam-boiler should be maintained at as nearly uniform a working-level as possible, by proper regulation of the feed-supply, which should be continuous and not intermittent.

Height of the Water-Level of Steam-Boilers.—The working water-level of steam-boilers should be as high as permissible. A high water-level produces a more uniform temperature throughout the boiler and a steadier supply of steam than a low water-level, because the larger volume of water stores more heat, and the temperature of the boiler is less influenced by the cooling effect of the feed-water than it is with the smaller volume of water.

The working water-level of cylindrical steam-boilers is, on an average, placed at a distance from the top of the shell equal to one-fourth the internal diameter of the boiler.

The working water-level of Cornish and Lancashire boilers is generally 9 inches above the crowns of the furnace-tubes, and the low water-line is 4 inches above the furnace-crowns.

The working water-level of marine return-tube-boilers is generally 10 inches above the top of the combustion-chambers.

Capacity of the Steam-Space of Steam-Boilers.—The steam-space of cylindrical steam-boilers should be sufficiently high to permit efficient separation from the steam of water carried with it from the evaporating surface. The capacity of the steam-space is frequently equal to one-fourth the capacity of the shell of the boiler, without deduction for space occupied by the tubes or fire-box.

The steam-space required for a steam-boiler reduces as the steam-pressure increases, because it depends upon the volume of steam, which reduces as the pressure increases. For instance, the volume of steam of 150 lbs. per square inch absolute pressure is three cubic feet per lb., or only one-half that of steam of 69 lbs. per square inch absolute pressure, of which the volume is six cubic feet per lb.

The capacity of the steam-space provided in boilers for different types of engines, averages in practice as follows:—

Description of Steam-boiler.	Capacity of Steam-space per Indicated Horse-power of the Engine Cubic foot.
Steam-boilers for locomotive engines	·06 to ·08
Steam-boilers of modified locomotive type	·25 to ·33
Cylindrical steam-boilers for quadruple-expansion engines	·30 to ·50
do. do. for triple-expansion engines	·40 to ·60
do. do. for double-expansion engines	·50 to ·70
do. do. for simple engines	·60 to ·80
Steam-boilers of locomotive type for simple portable-engines	·70 to ·90

The ratio of the capacity of the steam-space of cylindrical steam-boilers to that of the high-pressure cylinder of double, triple, and quadruple-expansion engines ranges from 50 to 140 to 1, or it is = (diameter of high-pressure cylinder in inches^2 × ·7854 × length of stroke in inches) ÷ 1728, and the quotient × 50 to 140.

The ratio of the capacity of the steam-space of the boiler of a simple locomotive engine to the sum of the capacities of its two cylinders, is, on an average, equal to $6\frac{1}{2}$ to 1.

The capacity of the steam-space is sometimes equal to the volume of steam consumed by the engine during 20 seconds, or one-third of a minute. For instance, the steam-space of a boiler for an engine with a cylinder 23 inches diameter, 42 inches length of stroke, making 65 revolutions per minute, and cutting off steam at ·6 of the stroke, is =

$$\frac{23 \times 23 \text{ inches} \times \cdot 7854 \times 42 \text{ inches} \times \cdot 6 \times 2 \times 65 \text{ revolutions}}{1728} \times \frac{1}{3} =$$

262 cubic feet of steam-space.

It was found in some experiments with marine-boilers having a working-pressure of less than 50 lbs. per square inch, that, when the capacity of the steam-space was only equal to the volume of steam consumed by the engine during twelve seconds, a considerable quantity of water was carried mechanically with the steam, and that when the steam-space contained sufficient steam to run the engine for 15 seconds no water was carried into the cylinders, and that boilers containing steam sufficient to run the engine for 20 seconds never gave trouble from wet steam.

The Capacity of the Steam-space per square foot of the Total Heating Surface of a Boiler, provided in practice for different types of steam-boilers, averages as follows :—

Description of Steam-boiler.	Capacity of Steam-space per square foot of total Heating-surface. Cubic foot.
Harrison's water-tube boiler	·023
Water-tube boilers on Root's principle average	·043
Root's water-tube boiler, improved	·108
De Naeyer's water-tube boilers	·052
Locomotive-engine boilers	·053
Water-tube boilers, average of a number of different kinds	·070
Stationary multitubular boilers, locomotive type.	·083
Marine return-tube boilers for quadruple-expansion engines	·085
Babcock and Wilcox water-tube boilers	·091
Marine return-tube boilers for triple-expansion engines	·115
Marine boilers of modified locomotive type	·135
Marine return-tube boilers for high-speed double-expansion engines	·160
The Galloway boiler	·175
Marine return-tube boilers for low-speed double-expansion engines	·200
Lancashire boilers for pressure of steam up to 200 lbs. per sq. inch	·210
Cornish boilers for pressure of steam above 120 lbs. per sq. inch	·240
Stationary horizontal multitubular cylindrical boilers	·250
Cornish boilers for pressure of steam up to 100 lbs per sq. inch	·260
Lancashire boilers for pressure of steam up to 100 lbs. per sq. inch	·300

When the capacity of the steam-space of a boiler is deficient, it conduces to priming, and results in an unsteady supply of steam containing more or less moisture.

Galloway-Cone-Tubes.—The heating surface is increased, the circulation improved, and the temperature throughout the boiler is rendered more uniform, by placing Galloway-cone-tubes in the furnace-tubes of Cornish and Lancashire boilers, as shown in Fig. 37. The comparatively cool water entering at the bottom of a Galloway-cone-tube from the water-space below it, is heated and partially evaporated in the tube, and, being lighter than the water outside the tube, an upward current of water and steam is created, which flows out at the top of the tube into the water-space above it.

Fig. 37.
Galloway-cone-tubes.

The internal diameter of Galloway-tubes is $10\frac{1}{2}$ inches at the top, and $5\frac{1}{2}$ inches at the bottom; the top flange is 16 inches diameter, and the bottom flange is $10\frac{1}{2}$ inches diameter. For a steam-pressure of 100 lbs. per square inch, a cone-tube $37\frac{3}{8}$ inches long is $\frac{7}{16}$ inch thick, and weighs 136 lbs.; and a cone-tube $34\frac{3}{8}$ inches long is $\frac{3}{8}$ inch thick, and weighs 123 lbs. The sectional area of the top of the cone-tube is $= 86{\cdot}59$ square inches area $\div$ $23{\cdot}76$ square inches area $= 3{\cdot}64$ times as great as that of the bottom. The tubes being so much wider at the top than the bottom there is room for free movement of the convection-currents, and their shape favours circulation. Parallel tubes should not be employed for this purpose.

Galloway-cone-tubes are best placed at the back-end of the furnace-tubes of Cornish and Lancashire boilers, and at a considerable distance from the fire, otherwise they are liable to obstruct free development of flame and impede the combustion of the fuel-gases. When placed too near the fire-bridge, they lower the temperature of the fuel-gases before combustion is complete, which results in a quantity of unconsumed carbon being deposited in the form of soot in the furnace-tubes. For this reason, it is not conducive to economy to crowd furnace-tubes with water-tubes.

Only a few cone-tubes should be employed, and the first tube should not be placed nearer to the fire-bridge than 6 feet when the boiler is fired with coal depositing little soot, or nearer than 10 feet when the coal is of a very smoky nature. The first cone-tube may be placed vertically in a belt of the plates, and the remainder of the tubes placed in consecutive belts, fixed right and left alternately at an angle of 30 degrees.

The evaporative efficiency of Galloway-cone-tubes is greatest where combustion is most complete on the fire-grate, and little unburnt soot-laden fuel-gases pass beyond the fire-bridge. In boilers using coal producing much smoke, the surfaces of the cone-tubes may become so thickly coated with soot as to render them of little effect in increasing the evaporative efficiency of a boiler.

Fire-Grate of Steam-Boilers.—The function of a fire-grate is to support the fuel and provide access of air to it to effect combustion. The length of the fire-grate is in some cases as great as 7 feet, but it is preferably limited to 6 feet, that being the greatest length of fire that can be readily worked by a stoker. Grates of these lengths cannot, however, be efficiently fired, and in order to enable the fire to be properly stoked to effect economical combustion, the length of the fire-grate should not in ordinary cases exceed 4 feet 6 inches, or 5 feet as a maximum in special cases.

As a short fire-grate is more economical than a long one, the grate should not be longer than necessary to burn the required quantity of fuel per hour. The width of the fire-grate is limited by the diameter of the furnace-tube in internally-fired boilers, and by the diameter of the shell in externally-fired boilers.

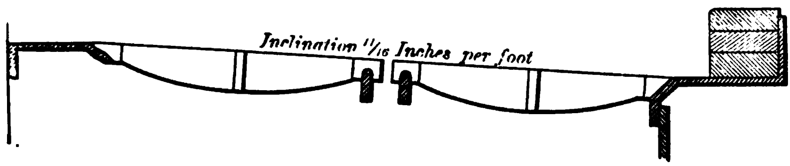


Fig. 38.—Fire-grate of a steam-boiler.

Sufficient space should be provided above the fire-grate to allow the fire to be easily worked, and to secure proper mixture of the gases, and also to present as much surface as possible for absorbing radiant heat from the fuel. The higher the rate of combustion the greater should the combustion space be. The fire-grate of internally-fired boilers is frequently inclined, or sloped downwards from the front to the back, as shown in Fig. 38, at the rate of from $\frac{1}{2}$ to $1\frac{1}{2}$ inches per foot in length of the grate. This arrangement facilitates the stoking of the back of the fire, and gives more room in the furnace than when the fire-grate is level.

Area of Fire-Grate Surface of Steam-Boilers.—The area of fire-grate surface required for a steam-boiler depends upon the extent of the heating-surface and the strength of the draught; and it is inseparably connected with the character and quality of the fuel used. Small coal requires a large area of grate-surface, and very small coal and coal-refuse may be successfully burnt as fuel for steam-boilers if a large single fire-grate be provided. The size of the fire-grate should be proportioned to the nature and size of the pieces of fuel, and the quantity of clinker and ash yielded by its combustion. To burn a given quantity of coal, a larger fire-grate is required with a light draught than with a strong draught. If the area of fire-grate be excessive, it results in a slow draught through the fuel.

The area of fire-grate is determined principally by the quantity of fuel to be burnt on each square foot of fire-grate surface per hour. For instance, to burn 420 lbs. of coal per hour, at the rate of 14 lbs. per square foot of fire-grate surface per hour, requires $420 \div 14 = 30$ square feet of fire-grate surface.

The area of fire-grate is frequently limited by practical considerations, but the fire-grate and heating-surfaces should be so proportioned to each other that the products of combustion leave the boiler at the lowest practicable temperature.

When the area of the fire-grate is excessive in proportion to the heating-surface, a proper rate of combustion cannot be maintained without the escape of waste heat to the chimney, the heating-surface being insufficient to absorb and transmit it to the water.

Fire-grates are frequently made too large for economical combustion. In one case, a fire-grate having a ratio of 1 to 25 feet of heating-surface was altered to a ratio of 1 to 37, which resulted in reducing the temperature of the escaping fuel-gases considerably, and a saving of 12 per cent. in fuel was effected with the same rate of combustion, without impairing the evaporative efficiency of the boiler.

The Area of Fire-Grate required for the combustion, at a given rate, of a given quantity of coal per indicated horse-power of an engine per hour, may be calculated by dividing the coal-consumption per indicated horse-power per hour by the rate of combustion.

For instance, if 16 lbs. of coal be consumed per square foot of fire-grate surface per hour, and $1\frac{1}{2}$ lbs. of coal be consumed per indicated horse-power per hour, the area of fire-grate surface required for the boiler of an engine of 500 indicated horse-power, is $= 1{\cdot}5$ lbs. $\div\ 16 = {\cdot}0937 \times 500 = 46{\cdot}85$ square feet.

The area of fire-grate required for various rates of combustion and consumption is given in Table 42.

The Area of Fire-Grate Surface provided in Steam-Boilers per Indicated Horse-Power developed by good steam-engines, averages in practice as follows:—

Description of Boiler.	Square foot of Fire-grate surface per Indicated Horse-power.
Boilers of locomotive-engines	·021 to ·034
Boilers of yacht-engines, locomotive type	·035 to ·040
Marine-boilers of modified locomotive type	·050 to ·070
Marine return-tube boilers of armour-plated warships	·050 to ·070
Coil-boilers	·050 to ·083
Marine return-tube boilers of Atlantic-racing steamers with triple-expansion-engines	·083
Marine return-tube boilers of navy dispatch boats	·065 to ·095
Marine return-tube boilers for triple-expansion engines	·080 to ·100
Boilers of torpedo boats, modified locomotive type	·100 to ·125
Marine return-tube boilers for double-expansion engines	·090 to ·140
Marine return-tube boilers, average of a large number of boilers	·120
Lancashire boilers	·100 to ·165
Water-tube boilers of various kinds	·130 to ·230
Water-tube boilers for non-condensing engines, average of a number of different kinds	·220
Water-tube boilers for double-expansion condensing engines, average of a number of different kinds	·160

Description of Boiler.	Square foot of Fire-grate surface per Indicated Horse-power.
Cornish boilers	·130 to ·210
Plain cylindrical or egg-ended boilers	·150 to ·240
Horizontal internally-fired cylindrical multitubular boilers	·200 to ·270
Portable-engine boilers, locomotive type	·250 to ·280
Vertical tubular-boilers	·300 to ·510

All these boilers have natural draught, except those of the locomotive type, which have exhaust steam-blast in the chimney.

TABLE 42.—AREA OF FIRE-GRATE SURFACE REQUIRED FOR A STEAM-BOILER PER INDICATED HORSE-POWER OF THE ENGINE FOR DIFFERENT RATES OF COMBUSTION AND CONSUMPTION.

Rate of Combustion per square foot of Fire-Grate Surface per Hour in pounds.	Coal Consumed per Indicated Horse-power of the Engine per Hour.					
	Pounds. $1\frac{1}{4}$	Pounds. $1\frac{1}{2}$	Pounds. $1\frac{3}{4}$	Pounds. 2	Pounds. $2\frac{1}{4}$	Pounds. $2\frac{1}{2}$
	Square Foot of Fire-grate Surface required per Indicated Horse-power of the Engine for a Steam Boiler with Natural Draughts.					
	Square foot.	Square foot.	Square foot.	Square foot.	Square foot.	Square foot.
12	·1040	·1250	·1458	·1666	·1874	·2082
13	·0960	·1152	·1346	·1538	·1730	·1922
14	·0892	·1071	·1250	·1428	·1607	·1785
15	·0833	·1000	·1166	·1333	·1500	·1666
16	·0781	·0937	·1093	·1250	·1406	·1562
17	·0735	·0882	·1029	·1176	·1323	·1500
18	·0694	·0833	·0972	·1111	·1250	·1333
19	·0657	·0789	·0921	·1052	·1183	·1316
20	·0625	·0750	·0875	·1000	·1125	·1250
21	·0595	·0714	·0833	·0952	·1071	·1190
22	·0522	·0681	·0795	·0909	·1022	·1136
23	·0543	·0652	·0760	·0869	·0978	·1087
24	·0520	·0625	·0729	·0833	·0937	·1041
25	·0500	·0600	·0700	·0800	·0900	·1000
26	·0480	·0576	·0673	·0769	·0865	·0961
	For Steam Boilers with Forced Draught.					
30	·0416	·0500	·0583	·0666	·0750	·0833
40	·0312	·0375	·0437	·0500	·0562	·0625
50	·0250	·0300	·0350	·0400	·0450	·0500
60	·0208	·0250	.0291	·0333	·0375	·0416
70	·0178	·0214	·0250	·0285	·0321	·0357
80	·0156	·0187	·0218	·0250	·0281	·0312
90	·0139	·0166	·0194	·0222	·0250	·0277
100	·0125	·0150	·0175	·0200	·0225	·0250

Power of Fire-Grates.—It will be seen from the Table on pages 152 and 153 that the power developed per square foot of fire-grate surface varies considerably in different types of boilers with natural draught. The power developed by return-tube boilers of warships is = 1 square foot of fire-grate ÷ ·05 = 20 indicated horse-power per square foot of fire-grate surface per hour.

The average power developed on voyage by a large number of marine return-tube boilers was = 1 square foot of fire-grate ÷ ·12 = 8·34 indicated horse-power per square foot of fire-grate surface per hour.

Marine return-tube boilers of triple-expansion engines develop at least one indicated horse-power from $\frac{1}{10}$ square foot of fire-grate = 1 square foot ÷ ·10 = 10 indicated horse-power per square foot of fire-grate, and require 1000 ÷ 10 = 100 square feet of fire-grate surface for the development of 1000 indicated horse-power per hour, in constant work on voyage.

Return-tube boilers of Atlantic racing steamers develop, 1 square foot ÷ ·083 = 12 indicated horse-power per square foot of fire-grate surface, which is probably the highest power obtainable in constant work in this type of boiler with natural draught and a funnel of ordinary height.

Lancashire-boilers frequently develop under ordinary working conditions 1 square foot ÷ ·11 = 9 indicated horse-power per square foot of fire-grate surface per hour, and require for the development of 300 indicated horse-power per hour, 300 ÷ 9 = 33 square feet of fire-grate surface.

In marine return-tube boilers having combustion with forced draught, the area of fire-grate surface per indicated horse-power is from ·038 to ·065 square foot. It probably averages ·045 square foot.

Marine return-tube boilers having combustion with well-arranged forced draught develop, 1 square foot ÷ ·045 = 22·2 indicated horse-power per square foot of fire-grate surface per hour, and require for the development of 1000 horse-power per hour only 1000 ÷ 22·2 = 45 square feet of fire-grate surface.

The Area of Fire-Grate Required for a Given Evaporation in a Steam-Boiler may be found by dividing the weight of steam used by an engine by the given evaporation in pounds per square foot of fire-grate surface per hour. If, for instance, an engine of 250 indicated horse-power uses 20 lbs. of steam per indicated horse-power per hour, and the consumption of coal per square foot of fire-grate surface per hour is required to be 18 lbs., and $8\frac{1}{2}$ lbs. of water are to be evaporated per lb. of coal.

Then, the evaporation is = 18 × 8·5 = 153 lbs. of water per square foot of fire-grate surface per hour. The weight of steam used by the engine is = 250 indicated horse-power × 20 lbs. of steam = 5000 lbs. per hour: and 5000 lbs. ÷ 153 lbs. = 32·69 square feet of fire-grate surface are required for this boiler.

The Ratio of Fire-Grate Surface to Heating-Surface varies considerably in different types of steam-boilers, because certain proportions are necessary to obtain the maximum useful effect. The heating-surface per square foot of fire-grate area ranges in different types of steam-boilers from 25 to 70 square feet.

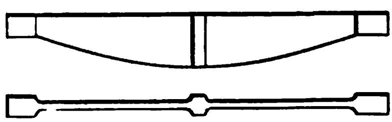
The proportion of fire-grate surface to heating-surface averages as follows in practice:—

Description of Boiler.	Ratio of Fire-grate surface to Heating surface.
Marine return-tube boilers	$\frac{1}{25}$ to $\frac{1}{35}$
Marine return-tube boilers, average of a large number of boilers	$\frac{1}{30}$
Lancashire-boilers	$\frac{1}{26}$ to $\frac{1}{38}$
Lancashire-boilers, average of a large number of boilers	$\frac{1}{28}$
Cornish-boilers	$\frac{1}{27}$ to $\frac{1}{34}$
Boilers of modified locomotive type	$\frac{1}{30}$ to $\frac{1}{34}$
Boilers of yacht-engines, locomotive type	$\frac{1}{40}$ to $\frac{1}{46}$
Horizontal internally fired cylindrical multitubular boilers	$\frac{1}{45}$ to $\frac{1}{50}$
Portable engine-boilers, locomotive type	$\frac{1}{35}$ to $\frac{1}{70}$
Water-tube-boilers	$\frac{1}{34}$ to $\frac{1}{65}$
Water-tube boilers, average of a number of different kinds	$\frac{1}{45}$
Locomotive-engine-boilers	$\frac{1}{60}$ to $\frac{1}{70}$

All these boilers have natural draught in the chimney, except those of the locomotive type in which the exhaust-steam is discharged into the chimney.

In marine return-tube boilers having combustion with forced draught, the ratio of fire-grate surface to heating-surface is from $\frac{1}{45}$ to $\frac{1}{50}$.

Fire-Bars of Steam-Boilers.—Fire-bars, or grate-bars of steam-boilers, shown in Figs. 39 and 40, are usually made of cast-iron, but castings of steel, and wrought-iron bars, are employed in some cases. Thin and deep bars stand the fire better, and warp less, than thick and narrow bars. Thin bars should always be used for high rates of combustion, as they are less liable to burn than thick bars.



Figs. 39 & 40.—Fire-bars of steam-boilers.

Cast-iron fire-bars are generally from $\frac{5}{8}$ inch to 1 inch thick on the top, from $\frac{5}{16}$ inch to $\frac{5}{8}$ inch thick at the bottom, and from 3 to 5 inches deep at the middle of their length. In order to maintain the same space between the bars when they become worn, they are formed parallel for a depth of $\frac{3}{4}$ inch from the top and then taper downwards. The side-bars should be formed to fit closely to the sides of the furnace-tube, to prevent the admission of air at the sides of the fire. Ribs, or distance-pieces, are placed at the end and middle of the bars to form an air-space between them.

The width of air-space depends upon the nature of the fuel and the available draught. With natural draught an air-space of from $\frac{3}{8}$ inch to $\frac{1}{2}$ inch in width is generally sufficient to supply air to the fuel on the bars, and keep them sufficiently cool to prevent excessive twisting and deterioration. Coals which cake much, and yield the largest quantity of clinker and ash, require the widest air-space between the bars. For burning small coal or slack of average size and quality, an air-space equal to one-third the area of the fire-grate is a good proportion.

With forced draught, the thickness of the fire-bars is generally from $\frac{3}{8}$ to $\frac{9}{16}$ inch, and the width of the air-space between them is from $\frac{1}{16}$ to $\frac{1}{4}$ inch, according to the nature of the coal and the strength of the draught. A dead-plate at least 2 inches wide should be fitted closely to the furnace-tube at each side of the fire-grate, to prevent the admission of air at the sides of the fire, and the development of a blow-pipe-action of intense local heat on the plates at this point.

The length of fire-bars should not, for convenience of handling, much exceed 3 feet. The length of the fire-grate of stationary boilers is frequently formed of three short fire-bars. The top of the bar should be grooved for the reception of fine ashes, which prevent the fusion of clinkers to the bar. It is necessary to allow sufficient space for the bars to expand, both lengthways and sideways, otherwise they are liable to warp and bend. The bars should slope downwards to the fire-bridge, at the rate of at least $\frac{1}{2}$ inch per foot in length when convenient, to obtain as much room as possible in the furnace above the fire for combustion of the fuel-gases.

Cast-iron fire-bars for Cornish and Lancashire boilers are frequently three in a line, each 2 feet long, 2 inches deep at the ends, 4 inches deep at the middle, and $\frac{1}{2}$ inch thick at the top. The weight of one of these bars is about $6\frac{1}{2}$ lbs. About 270 bars are used for a 7 feet diameter Lancashire boiler, and about 330 for a 7 feet 6 inches diameter Lancashire boiler.

Cast-iron fire-bars for marine return-tube boilers are frequently $\frac{7}{8}$ inch thick at the top, and $\frac{1}{2}$ inch thick at the bottom, and the depth at the middle is from 3 to $4\frac{1}{2}$ inches. A fire-bar 29 inches long weighs 20 lbs.; 34 inches long, 24 lbs.; 36 inches long, 26 lbs.; 42 inches long, 30 lbs.; and 48 inches long, 36 lbs. The distance between the bars is frequently $\frac{3}{8}$ inch. The side-bars of corrugated furnace-tubes have flanges fitting into the corrugations.

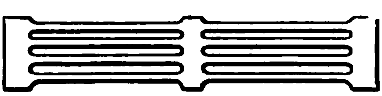


Fig. 41.—Fire-bars of steam-boilers.

Coals containing much sulphur rapidly destroy fire-bars. When bad coal is used, the fire-bars are rendered most durable by casting four or five bars together, as shown in Fig. 41, instead of using separate bars.

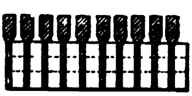


Fig. 42.—Section of wrought-iron fire-bars.

When coal is used which developes very intense heat, the fire-bars should be covered with either chalk or limestone to protect them from burning.

Wrought-Iron Fire-Bars for steam-boilers can be made lighter than cast-iron bars. They are stronger and do not fuse and burn away so rapidly as cast-iron. They are best made in sets of six or more single bars riveted together as shown in Fig. 42. The air-space is formed by washers through which the rivets pass. When constructed in this way they may be as thin as $\frac{5}{16}$ inch on the top, with an air-space $\frac{3}{16}$ inch wide, or less, if required.

Tubular Fire-Bars.—In burning anthracite in a steam-boiler, fire-bars are employed formed of water-tubes, arranged in sets of three. An ordinary fire-bar is placed between each set of tubular-bars, which can be removed when cleaning the fires.

Fire-Bridges of Steam-Boilers.—The fire-bridge at the back of the grate is employed to prevent the fire from being pushed over the grate-bars. It is composed of fire-brick laid on a wall in externally fired boilers, and laid on a casting fixed in the furnace-tube of internally fired boilers.

Fire-bridges are frequently made considerably higher than necessary for keeping the fire on the grate, with the object of assisting combustion by retarding the escape of the fuel-gases and promoting their admixture with air. But a fire-bridge does not conduce to economy of combustion and it should therefore be made as low as possible.

The effect of a high bridge is to bring the products of combustion in contact with the metal, and cool the gases before combustion is complete, and so contract the flame as to impede combustion. The velocity of the gases is greatest with a high bridge, and diminishes as the height is reduced. The bridge should be low enough not to interfere with the development of flame, and permit the flame to run the longest possible distance before being extinguished. A high bridge is liable to cause overheating with sedimentary feed-water.

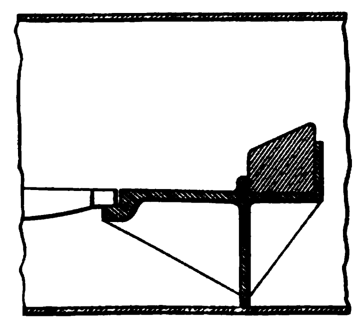


Fig. 43.—Fire-bridge of a steam-boiler.

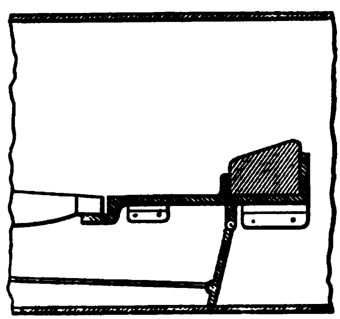


Fig. 44.—Fire-bridge of a steam-boiler.

An excessively high bridge impedes the draught, and is liable to cause destructive impingement of flame against the furnace plates, and rapid cooling of the gases, resulting in loss of radiant heat, emission of smoke, and waste of fuel. Its tendency being to almost extinguish the flames, causing long streams of black smoke to form at the ends of the flames, which are deposited as soot the instant the heat of that portion of the gases which has been consumed is absorbed by the heating-surfaces.

A fire-bridge for a steam-boiler is shown in Fig. 43. It is of cast-iron, formed with a trough at the back in which blocks of fire-clay, 9 inches wide, are fitted. A dead-plate, from 10 to 12 inches wide, is placed in front of the bridge on to which the fire is pushed when cleaning it. This arrangement greatly facilitates rapid and efficient cleaning of the fires.

Another fire-bridge is shown in Fig. 44. It is provided with a door fitting the underneath portion of the tube and hinged at the top, for the purpose of readily cleaning the bottom of the furnace-tube.

In double-ended marine return-tube boilers, with two opposite furnace-tubes discharging into a common combustion-chamber, the bridge should be built a good height in the combustion-chamber to prevent the draught in one furnace obliterating that in the other.

A fire-bridge is frequently simply formed by a wall of brick built in the furnace-tube, but it has the objections of concealing defects, and absorbing moisture from leaking seams, causing corrosive wasting of the plates in contact with the brickwork.

Space above the Fire-Bridge.—The space provided above the bridge for the escape of the fuel-gases from the furnace, depends upon the rate of combustion. It may generally be from $\frac{1}{6}$ to $\frac{1}{8}$ the area of the fire-grate. When a fire-bridge is employed higher than necessary to guard the fuel, its

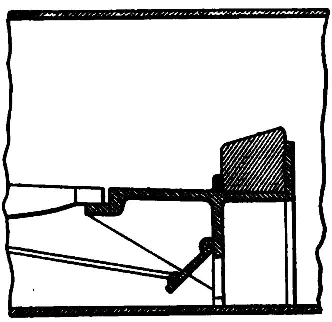


Fig. 45.—Split fire-bridge.

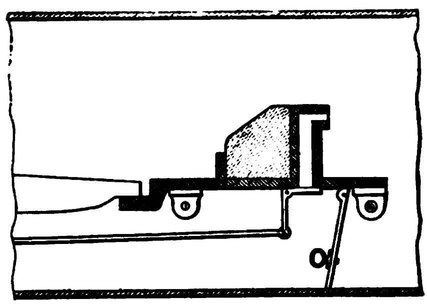


Fig. 46.—Split fire-bridge.

height should be proportioned to the nature of the fuel, the strength of the draught, and the thickness of the fire. The distance between the top of the bridge and the crown of the furnace-tube, measured on its vertical centre-line, should not be less than = the internal diameter of furnace-tube × ·3.

For instance, the fire-bridge of a furnace-tube of 33 inches internal diameter, should not be placed nearer to its crown than 33 × ·3 = 9·9, or say 10 inches. The thickness or width of the bridge should not be greater than 9 inches.

Split Bridges.—The fire-bridge of a steam-boiler is sometimes split, or formed hollow, for the purpose of admitting air at the back of the fire to promote combustion and prevent the emission of smoke. A split-bridge is shown in Fig. 45. It consists of a box-casting, forming an air-chamber, with a perforated air-plate at the back, and having a door at the front, underneath the fire-bars, by which air is admitted to the box. The air in passing through the perforated plate becomes subdivided into minute jets, which effect the oxygenation of partly consumed gases passing from the furnace, promote their combustion, and prevent the production of smoke. The perforations are $\frac{3}{8}$ inch diameter and their aggregate area is 4 inches per square foot of fire-grate surface.

A simpler form of split-bridge, which does not interfere with the cleaning of the furnace-tube, for supplying air behind the bridge is shown in Fig. 46. Two different arrangements of box-shaped split-bridges are shown in Figs. 47 and 48.

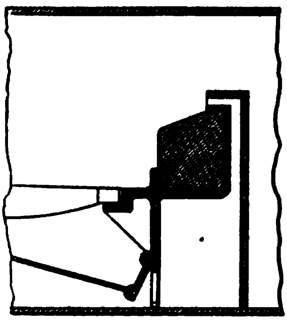


Fig. 47.—Split fire-bridge.

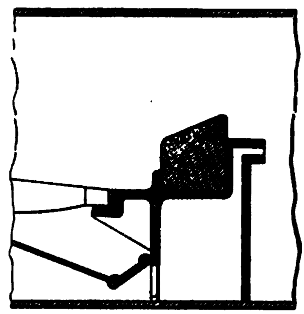


Fig. 48.—Split fire-bridge.

A split-bridge is useful for assisting imperfect combustion, and the prevention of smoke, especially in the case of a defective draught, or when the fires are unduly pushed, but as the air can only have a trifling amount of heat imparted to it in the chamber, it has a cooling effect on the flues, and therefore, it is preferable to provide an ample supply of air from the front of the furnace above the fire, and through the grate-bars from below the fire.

Water-Bridges.—The fire-bridge of a steam-boiler is sometimes of metal, forming a water-space compartment communicating with the boiler. The objections to this form of bridge are that, being so near the burning fuel, it chills the gases before combustion is complete, and as the steam-bubbles can not escape except laterally, a film of steam is liable to accumulate between the water and the metal and cause overheating, resulting in burnt and fractured metal at the front and top of the bridge.

Steam-Generating Power of Steam-Boilers.—The steam-generating power of a boiler depends upon the efficiency of its circulation, the arrangement and extent of its heating-surface, the proportion of the fire-grate surface to the heating-surface, the strength of the draught, and the rate of combustion of the fuel. The power of a steam-boiler may be calculated in terms either of its evaporative capacity, or of its horse-power.

The evaporative capacity of a boiler represents the quantity of water it will convert into steam per hour. It may be ascertained by testing the boiler; or it may be calculated approximately from the average rates of fuel-consumption and evaporation under ordinary working conditions. It may be assumed that with a good draught, 20 lbs. of coal can be readily consumed per square foot of fire-grate surface per hour, and at least 8 lbs. of water may be evaporated per lb. of coal of moderately good quality. On this basis, the evaporative capacity of a well arranged boiler having 33 square feet of fire-grate surface is = 33 square feet × 20 lbs. of coal × 8 lbs. of water = 5280 lbs. of water per hour.

Taking the steam-consumption of a good compound condensing-engine at 15 lbs. per indicated horse-power per hour, and allowing a margin of 50 per cent., or say 8 lbs., in order to admit of the power being obtained with easy firing, then 15 + 8 = 23 lbs. of water are required per indicated horse-power per hour, and the power of this boiler is = 5280 lbs. evaporative capacity ÷ 23 lbs. = 230 indicated horse-power.

Power developed by a Boiler of a given Evaporative Capacity.—The size of engine for which a steam-boiler of a given evaporative capacity is suitable, may be found by the following *Rule*:—

Indicated horse-power of engine the boiler is suitable for =
Evaporative capacity of boiler × C.

In which C is a constant varying for different types of engines as follows:—

C = ·025 for simple non-condensing engines.
C = ·050 for double-expansion condensing engines.
C = ·062 for triple-expansion surface-condensing engines.
C = ·070 for quadruple-expansion surface-condensing engines.

For instance, a boiler having an evaporative capacity of 7000 lbs. of water per hour will supply steam for a simple non-condensing engine of 7000 lbs. × ·025 = 175 indicated horse-power; or for a double-expansion engine of 7000 × ·05 = 350 indicated horse-power; or, for a triple-expansion engine of 7000 × ·062 = 434 indicated horse-power; or, for a quadruple-expansion engine of 7000 × ·07 = 490 indicated horse-power. This *Rule* gives the power of engines the boilers will on an average readily supply steam for, but greater power may be obtained under favourable conditions.

Horse-Power of Steam-Boilers.—The term horse-power is frequently applied to steam-boilers. It is based on the generation of a sufficient quantity of steam, for the development of one horse-power in a steam-engine. But, as the quantity of steam required for this purpose depends upon the type and perfection of the engine, it is necessary to determine what should constitute a boiler horse-power, and to define some unit expressing the quantity of steam generated in a boiler, from and at a given standard temperature, which shall develop one horse-power on an average, when used by a fairly good engine.

A unit of this kind is desirable in order that boilers may be designed to develop a given horse-power with a given consumption of fuel. This unit should vary, for boilers supplying steam to different kinds of engines, according to the lowest average weight of steam required in practice to develop one horse-power by the best engines of each type, because, a given evaporation produces widely different results in different types of engines.

The Nominal Horse-Power of Steam-Boilers is frequently measured by the evaporation of one cubic foot of water, or 62·4 lbs., to steam per hour, but this is much too small a standard for modern steam-boilers. One nominal horse-power of a modern steam-boiler may be measured by the evaporation of 45 lbs. of water, from and at 212° Fahr. per hour.

Theoretical Quantity of Water and Coal required for the Development of One Indicated Horse-power.—The quantity of water and coal theoretically required by a steam-boiler for the development of one indicated horse-power may be calculated as follows:—

One indicated horse-power is equivalent to,

$$\frac{33000 \text{ foot pounds}}{772 \text{ foot pounds}} =$$

42·746 thermal units per minute, and = 42·746 × 60 minutes = 2564·76 thermal units per hour.

The heat required to evaporate 1 lb. of water from 212° Fahr. to steam of the same temperature, is 966 units, and,

$$\frac{2564{\cdot}76 \text{ thermal units per hour}}{966, \text{ the unit of evaporation}} =$$

2·655 lbs., the quantity of water theoretically required for the development of one indicated horse-power per hour.

If Welsh steam-coal, having a calorific power of, say, 14973 units per lb., be used in the boiler, it will evaporate,

$$\frac{14973 \text{ thermal units}}{966, \text{ the unit of evaporation}} =$$

15·5 lbs. of water to steam, from and at 212° Fahr., per lb. of coal, and the consumption of coal per indicated horse-power per hour is =

$$\frac{2{\cdot}655 \text{ lbs. of water per indicated horse-power per hour}}{15{\cdot}5 \text{ lbs. of water evaporated per lb. of coal}} =$$

·1712 lb. of coal.

Showing that about 2¾ ounces of coal will theoretically develop one indicated horse-power per hour; and 1 ÷ ·1712 lb. = 5·841 indicated horse-power per hour may theoretically be developed by 1 lb. of coal.

These quantities of water and coal are very much less than are obtained in practice, owing to numerous sources of loss of heat in its conversion into work.

Actual Horse-Power of Steam-Boilers.—The steam-generating power of a steam-boiler is determined by the quantity of water at the boiling point, 212° Fahr., it will evaporate to steam of the same temperature with the consumption of a given quantity of fuel per hour, which can only be determined by experiment. The power of a steam-boiler may be conveniently determined by the evaporation required to supply the quantity of steam actually consumed per unit of power by an economical steam engine under ordinary working conditions. Hence, the power of a steam-boiler may be estimated by a unit expressing the quantity of water required to be evaporated to dry steam per hour, from and at 212° Fahr., to supply the quantity of steam necessary for the development of one indicated horse-power by an economical steam engine.

A high-class-condensing engine using steam of very high pressure with large measures of expansion, in cylinders efficiently protected from condensation, should not use more than 10 lbs. of steam per indicated horse-power per hour. Therefore one boiler-horse-power may be expressed by the evaporation of 10 lbs. of water to steam per hour, from and at 212° Fahr. This may be taken as a standard unit of power, and one boiler-horse-power is equivalent to 10 lbs. of water × 966 units of heat per pound, = 9660 thermal units.

But as the quantity of steam consumed per indicated horse-power varies considerably in different types of engines, it is necessary to provide boiler-power considerably in excess of the capacity of the engine to utilize the steam generated: and it is convenient to adopt a standard expressing the quantity of steam consumed by the best engines of different types and allow a margin in each case to ensure the provision of ample boiler-power. For which purpose, it may be affirmed that the power of a steam-boiler should be at least 50 per cent. greater than that of the engine it supplies with steam, in order to obtain economical evaporation and an ample supply of steam. On this basis the power of boilers for different types of engines may be estimated with precision.

Actual Horse-Power of Boilers for Quadruple-Expansion Surface-Condensing Engines.—A good quadruple-expansion surface-condensing engine should not use more than 12 lbs. of steam per indicated horse-power per hour. Allowing a margin of 50 per cent. for the purpose of obtaining a boiler of sufficient capacity to permit the economical generation of an ample supply of steam, the quantity becomes $12 \times 1{\cdot}5 = 18$ lbs. of steam per indicated horse-power per hour. Therefore the actual horse-power of boilers which supply steam to quadruple-expansion surface-condensing engines may be measured by the evaporation of 18 lbs. of water to steam per hour from and at 212° Fahr. This is equivalent to 18 lbs. of water × 966 units of heat per lb. = 11592 thermal units.

Actual Horse-Power of Boilers for Triple-Expansion Surface-Condensing Engines.—The best triple-expansion surface-condensing engines use 15 lbs. of steam per indicated horse-power per hour. Allowing a margin of 50 per cent., the quantity becomes $15 \times 1{\cdot}5 = 22{\cdot}5$, or, say, in round numbers 23 lbs. of steam per indicated horse-power per hour. Hence, the actual horse-power of boilers which supply steam to triple-expansion surface-condensing engines may be measured by the evaporation of 23 lbs. of water to steam per hour, from and at 212° Fahr. This is equivalent to 23 lbs. of water × 966 units of heat per lb. = 22218 thermal units.

For example, the power of two boilers supplying steam to a set of triple-expansion engines, burning together 2423 lbs. of coal per hour, and evaporating $9\frac{1}{2}$ lbs. of water per lb. of coal, from and at 212° Fahr. is, on this basis = 2423 lbs. of coal × $9\frac{1}{2}$ lbs. of water = 23018 lbs. of water evaporated by both boilers per hour, and 23018 ÷ 23 lbs. of water consumed per indicated horse-power per hour = 1000 horse-power, the actual power of these boilers.

Actual Horse-Power of Steam-Boilers for Double-Expansion Surface-Condensing Engines.—The best double-expansion engines use on an average 18 lbs. of steam per indicated horse-power per hour. Allowing a margin of 50 per cent. the quantity becomes $18 \times 1{\cdot}5 = 27$ lbs. of steam per indicated horse-power per hour. The actual horse-power of boilers for double-expansion surface-condensing engines may therefore be measured by the evaporation of 27 lbs. of water to steam per hour, from and at 212° Fahr. This is equivalent to 27 lbs. of water × 966 units of heat per lb. = 26082 thermal units.

Actual Horse-Power of Steam-Boilers for Compound Non-Condensing Engines.—The best compound non-condensing engines use on an average 20 lbs. of steam per indicated horse-power per hour. Allowing a margin of 50 per cent. the quantity becomes $20 \times 1{\cdot}5 = 30$ lbs. of steam per indicated horse-power per hour. Hence, the actual horse-power of boilers for compound or double-expansion non-condensing engines may be measured by the evaporation of 30 lbs. of water to steam per hour, from and at 212° Fahr. This is equivalent to 30 lbs. of water × 966 units of heat per lb. = 28980 thermal units.

Actual Horse-Power of Steam-Boilers for Simple Non-Condensing Engines.—The best non-condensing simple engines use 23 lbs. of steam per indicated horse-power per hour. Allowing a margin of 50 per cent., the quantity becomes $23 \times 1{\cdot}5 = 34\frac{1}{2}$ lbs. of steam per indicated horse-power per hour. Therefore the actual horse-power of boilers for simple non-condensing engines may be measured by the evaporation of $34\frac{1}{2}$ lbs. of water to steam per hour from and at 212° Fahr. This is equivalent to 34·5 lbs. of water × 966 units of heat per lb. = 33327 thermal units.

Water-Consumption by Steam-Engines.—The quantity of water in lbs. per indicated horse-power per hour, actually used by steam-engines under ordinary working conditions, agrees very closely with the result given by the following formula:—

$$\text{Water-consumption per I H P} = \frac{190}{\sqrt[3]{\text{Boiler-pressure above the atmosphere}}}.$$

Example:—Required the consumption of water by a set of triple-expansion engines, supplied with steam by a boiler having a working pressure of 150 lbs. per square inch above the atmosphere.

Then $190 \div \sqrt[3]{150}$ lbs. $= 15\frac{1}{2}$ lbs. of water consumed per indicated horse-power per hour.

Power Developed by the Heating-Surfaces of Steam-Boilers in Practice.—The quantity of heat required to generate steam being practically the same for all pressures, the coal-consumption in a given boiler is constant for all pressures. The maximum weight of steam generated in a given time per square foot of heating-surface is tolerably constant in practice with the best arranged steam-boilers, therefore the heating-surface of an economical boiler may be assumed to bear a definite relation to the power developed by an economical engine.

The maximum power which a good steam-engine is capable of developing constantly, under ordinary working conditions, from a given total area of heating-surface of a good steam-boiler, with natural draught, may be estimated approximately by the following formula, in which the working pressure in lbs. per square inch is that shown by the steam-gauge, or above the atmosphere:—

Estimated indicated horse-power of steam-engines =

$$\frac{\text{Total heating-surface in square feet}}{C} \times \sqrt[3]{\text{Working pressure.}}$$

In which C is a constant for engines using steam of pressures within the limits of ordinary practice with each type of engine; its value being as follows:—

C = 130 for simple non-condensing engines.
C = 120 for compound or double-expansion non-condensing engines.
C = 28 for double expansion surface-condensing engines.
C = 33 for triple-expansion surface-condensing engines.
C = 30 for quadruple expansion surface-condensing engines.

Example 1: Estimate the indicated horse-power capable of being developed by a double expansion surface-condensing engine, supplied with steam by two boilers having an aggregate total heating-surface of 2950 square feet; working pressure 90 lbs. per square inch?

Then $\frac{2950 \text{ square feet}}{28} \times \sqrt[3]{90} = 1000$ indicated horse-power, the power which good engines of this type should develop under ordinary working conditions.

Example 2: Estimate the indicated horse-power capable of being developed by a triple-expansion engine, supplied with steam by two boilers having an aggregate total area of heating-surface of 2700 square feet; working pressure 150 lbs. per square inch?

$$\text{Then } \frac{2700 \text{ square feet}}{33} \times \sqrt[3]{150 \text{ lbs.}} = 1002 \text{ indicated horse-power,}$$

the power which a good engine of this type should develop under ordinary working conditions.

The Total Heating-Surface of a Steam-Boiler required to supply steam to a good engine of given indicated horse-power may be estimated approximately by the converse of the previous formula, as follows:—

Total heating-surface of a steam-boiler in square feet,

$$= \frac{\text{Indicated horse-power of engines} \times C}{\sqrt[3]{\text{boiler-pressure}}}$$

in which the different values of C are the same as in the previous formula.

Example 1: Estimate the total heating-surface of two steam-boilers to supply steam of 85 lbs. per square inch working pressure, to a double expansion surface condensing engine of 800 indicated horse-power.

Then $\frac{800 \times 28}{\sqrt[3]{85}} = 2428$ square feet, the aggregate total area of heating-surface required.

Example 2: Estimate the total heating-surface of two boilers to supply steam of 150 lbs. per square inch working pressure, to a set of triple expansion engines of 1200 indicated horse-power?

Then $\frac{1200 \times 33}{\sqrt[3]{150}} = 3233$ square feet, the aggregate total area of heating-surface required.

Normal Indicated Horse-Power of Steam-Boilers.—The council of the North-East Coast Institution of Engineers and Shipbuilders give the following formula for the power of steam-boilers:—

Let D = the diameter of the low-pressure cylinder of the engine in inches. If there be more than one low-pressure cylinder, let
D^2 = the sum of the squares of the diameters.
S = the length of stroke of the piston in inches.
P = Working pressure of the steam in lbs. per square inch above the atmosphere.
H = Heating-surface of boilers in square feet.

Then for engines driving screw-propellers:—

$$\text{Normal indicated horse-power of steam boiler} = \frac{H \sqrt[3]{P}}{16}$$

$$\text{and } H = \frac{D^2 \sqrt[3]{S}}{3 \cdot 25}.$$

Normal indicated horse power of combined engines and boilers =

$$\frac{(D^2 \sqrt[3]{S} + 3\,H)\sqrt[3]{P}}{100}.$$

For engines driving paddle-wheels:—

Normal indicated horse-power of paddle-engines $= \frac{(D^2 \sqrt[3]{S} + 5\,H)\sqrt[3]{P}}{160}$

$$\text{and } H = \frac{D^2 \sqrt[3]{S}}{5{\cdot}2}.$$

$$\text{Nominal horse-power} = \frac{\text{normal indicated horse-power}}{6}.$$

As an example of the above *Rules*, take the case of a triple-expansion engine of 48 inch stroke, diameter of low-pressure cylinder 72 inches, working pressure of steam 150 lbs. per square inch, for driving a screw-propeller.

Then 72×72 inches $\times \sqrt[3]{48} = 5184 \times 3{\cdot}6342 = 18839 \div 3{\cdot}25 =$ 5797 square feet of heating surface are required for the boilers for this engine: and $18839 + (5797 \times 3) = 18839 + 17391 = 36230 \times \sqrt[3]{150} =$ $36230 \times 5{\cdot}313 = 192490 \div 100 = 1924$ normal indicated horse-power of combined engines and boilers. This amount of heating-surface would be divided between two boilers.

If the power of these boilers be estimated by the *Rule* on page 164, it gives 5797 square feet of heating-surface $\div 33 = 172{\cdot}64 \times \sqrt[3]{150}$ lbs. pressure $= 172{\cdot}64 \times 12{\cdot}247 = 2114$ indicated horse-power, the power that good engines should develop with steam supplied by these boilers.

Different Types of Steam-Boilers.—Steam-boilers may be divided into three classes, viz., externally-fired and internally-fired cylindrical boilers: and water-tube boilers, or those in which water is contained in small tubes having heat applied on the outside. The simplest form of steam generator is the plain cylindrical boiler with hemispherical ends, commonly called an egg-ended boiler. It is externally fired, and not being an economical steam-generator is only now used to a limited extent. The next in simplicity are as follows:—

The cylindrical boiler with single furnace-tube, or Cornish boiler; the cylindrical boiler with two furnace-tubes, or Lancashire boiler; the Galloway boiler with two furnace-tubes communicating with a large tube containing a number of vertical tubes; the marine return-tube boiler; the locomotive boiler; vertical cross-tube boilers; and multitubular boilers. All these boilers are internally-fired.

Internally-fired boilers generally have greater evaporative power and are more economical in fuel than externally-fired boilers, and suffer less injury from strains due to unequal expansion and contraction. When a cylindrical boiler is fired under the bottom, there is considerable difference between the temperatures of the top and bottom of the shell, which results in unequal expansion and contraction, and causes deterioration of strength. The plates are also liable to injury from overheating, caused by the settlement of sediment at the bottom boiler.

Form of Boiler-Shells.—The strongest form of a boiler for resisting internal pressure is that of the sphere, because it is the only shape capable of resisting pressure uniformly throughout its entire surface, and of accommodating itself uniformly to the expansive action of heat, without distortion. The sphere contains the greatest volume with the smallest possible surface, but a spherical vessel presents a small amount of heating-surface to the fire, and is not a convenient form for a steam-boiler. The form of structure which possesses the greatest strength next to the sphere, is the cylindrical form, and it is consequently the shape most extensively used for steam-boilers.

Arrangement of Boiler-Shell Plates.—Cylindrical boiler-shells are composed of plates with riveted seams. Plates of homogeneous structure, being of uniform strength, may be placed either lengthways or crossways round the circumference of the shell, but plates of fibrous material, such as iron, are strongest in the direction in which they were rolled, and the fibre of the plates should run round the circumference of the shell of internally-fired boilers.

In plain cylindrical externally-fired boilers, considerable strain may be thrown on the bottom of the boiler by sudden contraction from an inrush of cold air to the flue, or by the delivery of cold feed-water on to the bottom plates, and when constructed of iron-plates, the furnace-platės should be arranged with the fibre running lengthways of the boiler. This arrangement also facilitates renewal of the plates.

When fibrous plates are employed with the fibre running lengthways of the shell, it is necessary to allow a larger margin of safety than when the plates are arranged with the fibre running circumferentially.

The longitudinal seams joining the ends of the plates forming a belt of a cylindrical boiler-shell, are either lap-jointed, formed by lapping two plate-ends and riveting them together with one or more rows of rivets; or butt-jointed, formed by butting two plate-ends together and covering the joint with a strip of plate on one side, or on both sides, of the shell-plate, and riveting them together with one or more rows of rivets. In order to increase the strength of the shell, the longitudinal seams are not arranged in line from end to end of the boiler, but are placed as far apart as possible, or break joint successively. They should be placed clear of the boiler-seating, and away from the centre-line of the top and bottom of a horizontal boiler.

Circumferential seams, or transverse seams, or ring seams, for joining the belts of plates, add considerable stiffness to the structure; and the width of the belts of plates of long boilers should not exceed 39 inches in order to counteract the bulging of the plates in case of over-pressure.

Effect of Internal Pressure on Cylindrical Boiler-Shells.—The pressure of steam being equal in all directions, the pressure inside the shell of a cylindrical boiler acts uniformly all round its circumference, and tends to restore any departure in shape from a true circle and maintain a perfectly circular form. The pressure against the circumference of the shell may be considered to radiate from the axis as shown in Fig. 49, and tends to enlarge the circumference uniformly, and rupture the shell in lines parallel to the axis, or at the longitudinal seams of a shell formed with riveted joints. The tension of the radial pressure may be regarded to be uniformly distributed through the material in the case of cylinders of ductile material of small thickness compared with their diameter, such as the shells of steam-boilers.

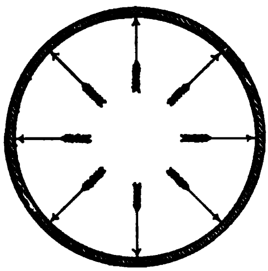


Fig. 49.—Section of boiler-shell.

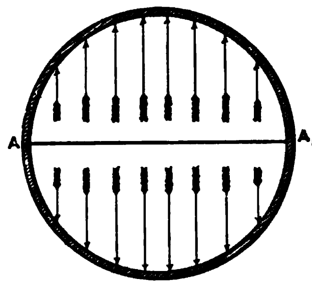


Fig. 50.—Section of boiler-shell.

The force tending to rupture a cylindrical shell longitudinally, or in lines parallel to its axis, may be considered as acting and re-acting in opposite directions to divide the cylinder in halves, or into two parts in a plane drawn through the diameter as shown in Fig. 50, in which the diametrical line A A represents the force and the arrows show the direction in which the force is exerted. The resistance opposed to this force is that due to the area of the metal on two opposite sides of the shell, or at the points A A.

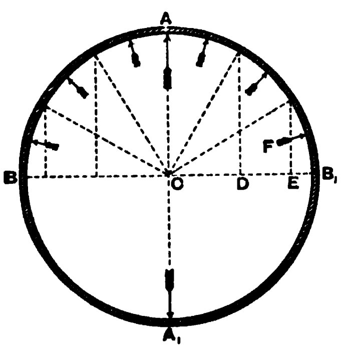


Fig. 51.—Section of boiler-shell.

The shell may be considered to be composed of a number of narrow rings, of a unit's length, say 1 inch, placed side by side, each of which resists the pressure independently of the others. A ring of this description is shown in Fig. 51, and as it is a true circle and supposed to be composed of material of uniform strength throughout, it is equally strong at all points of the circumference, and the same radial force will be required to burst it in one direction as another.

The force tending to burst or rupture this ring through the horizontal centre line B B is exerted upwards and downwards along the vertical centre line A A. On receding right and left from this line the pressure is exerted diagonally with diminishing vertical-effect to produce tension at A A. The force is resisted by the section of the metal at the points B B. If the circumferential line exposed to the force be divided into a number of divisions, say six, by radial lines from the centre C, and vertical lines be drawn from the line B B to the points of inter-section of the radial lines with the circumference, as lines D and E, the length divided off by these vertical lines on the line B B, represents the effect of the area of each division to rupture the boiler in the direction of A. If the component vertical forces be taken at a great number of points in the semi-circumference, it will be found that their sum is equal to the full pressure exerted on a line equal in length to the diameter.

Hence, although the pressure acts upon the whole circumference, its effect in rupturing the boiler is only measured by the diameter of the shell, and the force tending to burst a cylindrical shell in lines parallel to its axis, is equal to the internal diameter × pressure on each unit of surface × by the length of the shell.

Bursting-Pressure of Cylindrical Boiler-Shells.—A cylindrical boiler-shell is strained by internal pressure in two directions, viz., transversely by a circumferential strain due to the pressure tending to burst the shell by enlarging its circumference, as shown by the vertical arrows in Fig. 52; and longitudinally, due to the pressure tending to force the ends asunder as shown by the horizontal arrows.

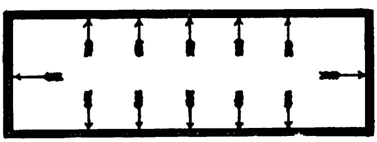


Fig. 52.—Section of boiler-shell.

The strength of the shell in a transverse direction may be ascertained by calculating the strength of a narrow belt of the shell, say a solid ring 1 inch wide. This ring will be exposed to a pressure on a surface equal to its internal diameter, and the pressure will be resisted by a sectional area of metal = the thickness of the shell × 2 × 1 inch wide.

The bursting-pressure of a cylindrical boiler-shell, without seams, in a transverse direction may be found by the following formula:—

Let P = the internal pressure on the shell, or the bursting-pressure in pounds per square inch.

t = the thickness of the shell in inches.

s = the tensile strength of the metal in pounds per square inch.

D = the internal diameter of the boiler-shell in inches.

$$P = \frac{t \times 2 \times s}{D}.$$

Example: Required the bursting-pressure in a transverse direction of the shell of a cylindrical boiler, without seams, of 30 inches diameter, thickness of shell ½ inch, tensile strength of the steel-plates 60,000 lbs. per square inch?

Then, $\frac{\cdot 5 \text{ inch thick} \times 2 \times 60000 \text{ lbs.}}{30 \text{ inches diameter of shell}} = 2000$ lbs. per square inch, the pressure required to burst this boiler-shell in a transverse direction.

The thickness of the shell in inches, t, required to resist the bursting-pressure, for equilibrium of stresses, may be found by this formula:—

$$t = \frac{P \times D}{s \times 2}.$$

Taking the data from the last example, the thickness of the shell,

$$t = \frac{2000 \text{ lbs.} \times 30 \text{ inches diameter}}{60000 \text{ lbs.} \times 2} = \cdot 5 \text{ inch.}$$

Again, $D^2 \times 3{\cdot}1416 \times P = D \times 2 \times 3{\cdot}1416 \times t \times s$.

That is, the strain on the metal is equal to the strength of the metal opposed to the strain.

Applying this to the above data:—

Then 30 inches × 30 inches × 3·1416 × 20000 lbs. = 5654880 lbs.,

And 30 inches × 2 × 3·1416 × ·5 inch × 60000 lbs. = 5654880 lbs.

Showing that the strength of the metal is equal to the pressure, or that the stresses are in equilibrium.

The internal pressure in lbs. per square inch, tending to rupture a cylindrical shell in a transverse direction, may also be calculated from the internal radius of the shell by the following formula:—

$$\text{Bursting pressure} = \frac{\text{Tensile strength of metal} \times \text{thickness in inches}}{\text{Internal radius of shell in inches}}.$$

Thickness of metal in inches required to resist the bursting-pressure for equilibrium of stresses =

$$\frac{\text{Bursting pressure in lbs. per square inch} \times \text{internal radius of shell in inches}}{\text{Tensile strength of metal in lbs. per square inch,}}$$

And bursting-pressure × radius = tensile strength × thickness of metal. Showing that the strain on the metal is equal to the strength of the metal opposed to the strain, or the stresses are in equilibrium.

Strength of a Cylindrical Boiler-Shell in a Longitudinal Direction.—The internal pressure tending to force the ends of a boiler asunder, or to sever the ring-seams, may be calculated as follows:—

Taking the data from the previous example, the sectional area exposed to the pressure is = 30 inches × 30 inches diameter × ·7854 = 706·86 square inches. The resistance of a ring of the shell in a transverse direction is = 30 inches diameter × 3·1416 = 94·248 inches, the circumferential length of the ring × ·5 inch thicknesses × 60000 lbs. = 2827440 lbs., and 2827440 ÷ 706·86 square inches of sectional area of the shell =

40000 lbs. per square inch, the bursting-pressure in the longitudinal direction of the boiler.

This calculation shows that the strength of this boiler in a longitudinal direction is double that in a transverse direction. Hence, in a cylindrical boiler-shell having riveted joints of equal proportions, the ring-seams are twice as strong as the longitudinal seams.

Effect of Internal Flue-Tubes on the Strength of the Shell of a Cylindrical Boiler.—The transverse strength of a boiler-shell is not influenced by internal flue-tubes, but they increase the longitudinal strength of the structure. In calculating the longitudinal strength of a boiler with one or more internal flue-tubes, it is necessary to deduct the area of the flue-tubes from the area of the end of the boiler, and to add the circumference of the tubes to that of the shell, in calculating the strength of the boiler.

Take, for instance, a Cornish boiler, 5 feet internal diameter, with an internal flue-tube of 30 inches diameter, the boiler being composed entirely of ½-inch steel-plates, having a tensile strength of 60000 lbs. per square inch.

The area of the shell is = 60 × 60 inches diameter × ·7854 = 2827·44 square inches.

The area of the flue-tube is = 30 × 30 inches diameter × ·7854 = 706·86 square inches.

The area exposed to the pressure is = 2827·44 − 706·86 = 2120·58 square inches.

The circumference of the shell is = 60 × 3·1416 = 188·496 inches.

The circumference of the flue-tube is = 30 × 3·1416 = 94·248 inches.

Then 188·496 + 94·248 inches = 282·744 inches total circumferential length; and the resistance opposed to internal pressure is = 282·744 inches circumferential length × ·5 inch thickness × 60000 lbs. per square inch tensile strength = 8482320 lbs. Then 8482320 lbs. resistance ÷ 2120·58 square inches of surface exposed to the pressure = 4000 lbs. per square inch, the bursting-pressure in a longitudinal direction.

The bursting-pressure of this shell in a transverse direction is =

$$\frac{60000 \text{ lbs.} \times \cdot 5 \text{ inch thickness}}{30 \text{ inches internal radius}} = 1000 \text{ lbs. per square inch.}$$

Showing that this boiler-shell is 4000 ÷ 1000 lbs. = 4 times as strong in a longitudinal direction as it is in a transverse direction.

When there are several flue-tubes their aggregate surface must be used in calculating the longitudinal strength of the shell.

Bursting-Pressure of Cylindrical Boiler-Shells with Riveted Joints.—The resistance of a cylindrical boiler-shell to internal pressure varies inversely as the diameter: a shell of three feet diameter will bear double the internal pressure of one of 6 feet diameter, the thickness and strength of metal being the same in both cases. The resistance of the plates forming the shell varies directly as their thickness; a shell of 1 inch thickness will bear double the pressure of one of ½-inch thickness, the diameter of the shell and the thickness of metal being the same in both cases. The area of metal opposed to the pressure is equal to the thickness of metal at each side of the shell, multiplied by the length of the shell.

For instance, the thickness of the cylindrical shell shown in Fig. 53 is ½-inch, the length of the shell, or width of the belt of plate, is 1 inch, therefore the thickness of the two sides × by the width $= \cdot 5 \times 2 \times 1$ inch $= 1$ square inch of metal, the sectional area opposed to the internal pressure on the shell.

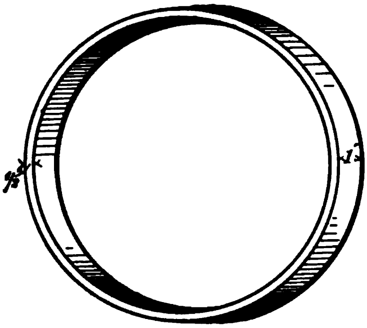


Fig. 53.—Section of boiler-shell.

As boilers are generally composed of belts of plates with riveted seams, it is necessary to take the strength of the riveted joints into consideration in determining the strength of the shell. As the circumferential seams are twice as strong as the longitudinal seams, the strength of a cylindrical boiler-shell to resist internal pressure depends principally upon the strength of the material and the proportions of the longitudinal seams.

The bursting-pressure of cylindrical boiler-shells formed with riveted seams may be found by the following *Rule*:—

Bursting-pressure of a cylindrical boiler-shell in lbs. per square inch =

$$\frac{\left(\begin{array}{c}\text{Tensile strength of plate in lbs.} \times \text{its thickness} \\ \text{in inches} \times \text{percentage of strength of joint}\end{array}\right)}{\text{Internal radius of shell in inches.}}$$

Example: Required the bursting-pressure of a cylindrical boiler-shell, thickness of plate ½-inch, the strength of the riveted joints being 56 per cent. of that of the solid plate; internal diameter of shell 84 inches; and the tensile strength of the steel-plates being 63000 lbs. per square inch?

$$\text{Then } \frac{63000 \text{ lbs.} \times \cdot 5 \text{ inch thickness} \times \cdot 56}{42 \text{ inch internal radius of shell}} = 420 \text{ lbs. per square inch,}$$

the bursting-pressure of the shell.

The thickness of plate in inches of a cylindrical boiler-shell for a given bursting-pressure is =

$$\frac{\text{Bursting-pressure in lbs.} \times \text{internal radius of shell in inches}}{\text{Tensile strength of plate in lbs.} \times \text{percentage of strength of joint}}.$$

Example: Required the thickness of plate of the boiler described in the previous example?

$$\text{Then } \frac{420 \text{ lbs. bursting-pressure} \times 42 \text{ inches radius of shell}}{63000 \text{ lbs. tensile strength} \times \cdot 56} = \cdot 5 \text{ inch,}$$

the thickness of plate required for this bursting-pressure.

The internal diameter of a boiler-shell for a given thickness of plate, and bursting-pressure is =

$$\frac{\text{Tensile strength of metal} \times \text{thickness of plate} \times \text{strength of joint}}{\text{Bursting-pressure in lbs. per square inch}} \times 2.$$

Example: Required the internal diameter of the boiler-shell described in the previous example?

$$\text{Then } \frac{63000 \text{ lbs. tensile strength} \times \frac{1}{2} \text{ inch thick} \times \cdot 56}{420 \text{ lbs. per square inch bursting-pressure}} \times 2 = 84 \text{ inches,}$$

the internal diameter of shell required for that bursting-pressure.

The strength of a steam-boiler depends upon its design, the proportion of its parts, and the quality of the materials and workmanship.

Factor of Safety for Steam-Boilers.—A steam-boiler should possess a large excess of strength, in order to provide for defects in materials and workmanship, which may exist without external indications, and for the reduction of the original strength of the structure due to corrosion, and also to the effect of strains arising from unequal expansion and contraction. A boiler is seldom of equal strength in all parts, and it does not wear uniformly. It is, therefore, necessary to allow a considerable margin between the working-pressure and the bursting-pressure of steam-boilers.

This margin, or the ratio in which the bursting-pressure exceeds the working-pressure, is termed the factor of safety. Boilers of moderately good materials and workmanship may have a factor of safety of six. Boilers of known high-class materials and workmanship, well cared for, but not worked under inspection, may have a factor of safety of five. High-class boilers that are regularly tested, and worked under both favourable conditions and efficient periodical examination, may have a factor of safety of four, but it should not, in any case, be less.

The following additions should be made to the factor of safety for structural defects when they exist, viz., an addition of ·25 if the rivet-holes are not good and fair in the circumferential seams; ·5 if the seams do not break-joint properly; ·5 if the rivet-holes are not good and fair in the longitudinal seams; 1 if the longitudinal seams are single-riveted; and 2 when the quality of the materials and workmanship is doubtful or unsatisfactory, without being positively bad.

Working-Pressure of Steam-Boilers.—The maximum working-pressure of a cylindrical boiler-shell in pounds per square inch may be found by the following *Rule*, in which the strength of the joints of the plates is expressed in percentage of the strength of the solid-plate:—

Maximum working-pressure of boiler-shell =

$$\frac{\text{Strength of plate in lbs.} \times \text{thickness in inches} \times \text{strength of joint}}{\text{Internal radius of shell in inches} \times \text{factor of safety.}}.$$

Example: Required the maximum working-pressure of a steel boiler-shell of 7 feet internal diameter; thickness of shell-plates $\frac{1}{2}$ inch; the strength of the riveted joints is 70 per cent. of that of the solid plate; ultimate tensile strength of the plates 29 tons; factor of safety 6?

$$\text{Then } \frac{29 \text{ tons} \times 2240 \text{ lbs.} \times {\cdot}5 \text{ inch thick} \times {\cdot}70}{42 \text{ inches internal radius of shell} \times 6} = 90{\cdot}22 \text{ lbs.},$$

or say 90 lbs. per square inch working-pressure.

Diameter of Boiler-Shells.—The internal diameter of a boiler-shell in inches for a given working-pressure of steam and thickness of plate, may be found by the following *Rule*:—

Internal diameter of boiler-shell in inches =

$$\frac{\text{Strength of plate in lbs.} \times \text{thickness in inches} \times \text{strength of joint}}{\text{Working-pressure in lbs. per square inch} \times \text{factor of safety}} \times 2.$$

Example: Required the internal diameter of the boiler-shell described in the previous example?

$$\text{Then } \frac{29 \text{ tons} \times 2240 \text{ lbs.} \times {\cdot}5 \text{ inch thick} \times {\cdot}70}{90{\cdot}22 \text{ lbs. working-pressure} \times 6} \times 2 = 84 \text{ inches},$$

the internal diameter of this boiler-shell.

Thickness of Plates of Boiler-Shells.—The thickness of plate in inches of a boiler-shell for a given diameter in inches and working-pressure in lbs. per square inch may be found by this *Rule*:—

Thickness of plates of boiler-shell =

$$\frac{\text{Working-pressure} \times \text{factor of safety} \times \text{internal radius of shell}}{\text{Strength of plate} \times \text{strength of joint}}.$$

Example: Required the thickness of the plates of the boiler-shell described in the previous example?

$$\text{Then } \frac{90{\cdot}22 \text{ lbs.} \times 6 \times 42 \text{ inches radius}}{29 \text{ tons} \times 2240 \text{ lbs.} \times {\cdot}70} = {\cdot}5 \text{ inch, the thickness of}$$

plate required for this boiler-shell.

Furnace-Tubes of Steam-Boilers.—The furnace-tube of a steam-boiler is exposed to a circumferential compressive stress from external pressure, equal in intensity to the tensile stress which an equal pressure would produce if applied inside the tube. Furnace-tubes should be as thin as the working-pressure will permit, because the heat-transmitting power of the plate decreases with the thickness, when it exceeds $\frac{3}{8}$-inch. Thick plates are more liable to blister and become injured by excessive heat than thin ones. Furnace-plates are made either of best Yorkshire iron, or of a mild quality of Siemens or other steel.

The longitudinal seams are best welded, they can then be made nearly perfectly circular, or so as not to deviate more than $\frac{1}{8}$ inch. When they are riveted, butt-joints are preferable, as it is important to obtain a perfectly circular tube, which cannot be obtained with a lap-joint. There should be as few seams as possible, and when lap-jointed their laps should be placed so that the current of the products of combustion does not impinge on the edges of the plate. The longitudinal seams should be placed below the level of the fire-grate.

Sufficient care is not taken in many cases to construct furnace-tubes perfectly circular in form, and tubes both with welded seams and riveted joints with double butt-straps, are frequently found as much out of truth as lap-jointed tubes. When Galloway-tubes are welded in a furnace-tube, they are liable to pull it out of a true circle when contracting in cooling, and may produce a deviation of from $\frac{1}{2}$ inch to $\frac{3}{4}$ inch from a true circle.

Strength of Furnace-Tubes of Steam-Boilers.—The furnace-tubes and flue-tubes of steam-boilers are exposed to external pressure tending to change the form of their transverse sections by bending, wrinkling, or buckling, the plates, and to collapse the tubes. The tendency of external pressure being to aggravate any distortion or departure from a cylindrical form, tubes subject to external pressure should be perfectly circular in shape.

The strain due to external pressure on a tube is theoretically a crushing strain, independent of the length of the tube, but directly proportional to the thickness, and inversely as the diameter; but in practice it is found that the strength of a tube to resist collapse is considerably affected by its length.

Fairbairn made a number of experiments to ascertain the collapsing-strength of tubes, and found that with tubes of the same diameter and the same thickness of metal, the strength to resist collapse was nearly inversely as the length; that with tubes of the same length and thickness, but of different diameters, the strength is nearly inversely as the diameter; and that the strength of tubes of the same length and diameter, but of different thickness, varies as the 2·19th power of the thickness.

From these experiments the following formula was deduced by Fairbairn:

Let P = the collapsing-pressure in lbs. per square inch.
t = the thickness of the tube in inches.
L = the length of the tube in feet.
D = the diameter of the tube in inches.

$$(1) \qquad P = \frac{806300 \times t^{2.19}}{L \times D}.$$

For all practical purposes, it is sufficiently correct to use the square of the thickness instead of the $2 \cdot 19$ power of the thickness, and the formula is simply as follows:—

$$(2) \qquad P = \frac{806300 \times t^2}{L \times D}.$$

Example: Required the collapsing-pressure of a furnace-tube of wrought-iron ½ inch thick, 20 feet long, and 3 feet external diameter?

Then $\frac{806300 \times \cdot 5 \text{ inch} \times \cdot 5 \text{ inch thick}}{20 \text{ feet long} \times 36 \text{ inches diameter}} = 280$ lbs. per square inch, the strength of this tube to resist collapsing pressure.

Later investigations prove that this rule is approximately correct for long furnace-tubes, but it is not suitable for short tubes, and should not be used for tubes shorter than 18 feet.

Mr. George Harrison gives a modification of Fairbairn's rule for determining the working pressure suitable for a furnace-tube, in which a factor of safety is employed varying with the length of the tube, as determined by the following rule:—

$$\text{Factor of safety} = \sqrt[2]{\frac{300}{\text{Length in feet.}}}$$

For instance, the factor of safety for the furnace-tube described in the previous example is $= \sqrt[2]{\frac{300}{20}} = 3 \cdot 872$,

and the working pressure allowable on that furnace-tube is =

$$\frac{280 \text{ lbs. per square inch collapsing-pressure}}{3 \cdot 872, \text{ the factor of safety,}} = 72 \text{ lbs. per square inch.}$$

This rule may be expressed more simply as follows:—

Let D = the diameter of the tube in inches.
L = the length of tube in feet.
t = the thickness of the tube in 32nds of an inch.
Working pressure in lbs. per square inch =

$$45 \cdot 3 \times \frac{t^2}{\sqrt[2]{L} \times D}.$$

Applying this *Rule* to the previous example it gives—

$$45 \cdot 3 \times \frac{16 \times 16}{\sqrt[2]{20 \text{ feet}} \times 36 \text{ inches}} = 72 \text{ lbs. per square inch,}$$

or the same result as in the previous calculation.

The following modification of Fairbairn's *Rule* is sometimes used for calculating the collapsing-pressure of furnace-tubes.

Collapsing-pressure in lbs. per square inch =

$$\frac{375023 \times \text{thickness in inches}^2}{\text{length in feet} \times \text{diameter in inches}}$$

Example: Required the collapsing-pressure of a furnace-tube 20 feet long, 36 inches external diameter, and $\frac{1}{2}$ inch thick?

$$\text{Then } \frac{375023 \times \cdot5 \times \cdot5}{20 \text{ feet long} \times 36 \text{ inches diameter}} = 131 \text{ lbs. per square inch,}$$

the collapsing-pressure of this furnace-tube.

Nystrom's *Rule* for calculating the collapsing-pressure of tubes subjected to external pressure is as follows:—

Let D = the internal diameter of the tube in inches.
L = the length of the tube in feet.
t = the thickness of the tube in inches.

$$\text{Collapsing-pressure in lbs. per square inch} = \frac{200000 \times t^2}{D \times \sqrt[2]{L}}.$$

Example: Required the collapsing-pressure of a furnace-tube of 3 feet external diameter, $\frac{1}{2}$ inch thick, and ten feet long?

$$\text{Then } \frac{200000 \times \cdot5 \times \cdot5 \text{ inch}}{36 \times \sqrt[2]{10}} = 439 \text{ lbs. per square inch,}$$

the collapsing-pressure of this furnace-tube.

Mr. Michael Longridge's formula for finding the collapsing-pressure of furnace-tubes is as follows:—

$$\text{Collapsing-pressure in lbs. per square inch} = \frac{174000 \times t^2}{D \times \sqrt[2]{L}},$$

In which D = the external diameter of the tube in inches.
L = the length of tube in feet.
t = the thickness of the tube in inches.

Example: Required the collapsing-pressure of a furnace-tube of 33 inches diameter, $\frac{3}{8}$ inches thick, and 15 feet long?

$$\text{Then } \frac{174000 \times \cdot375 \times \cdot375 \text{ inch thick}}{33 \text{ inches} \times \sqrt[2]{15} \text{ feet}} = 191 \text{ lbs. per square inch,}$$

the collapsing-pressure of this tube.

Mr. Longridge's *Rule* for the working-pressure of steam-boilers of wrought-iron or mild-steel, expressed in terms of the strength of the furnace-tube, is as follows:—

Let D = the diameter of the tube in inches.
L = the length of the tube in feet.
t = the thickness of the tube in thirty seconds of an inch.

$$\text{Working-pressure of boiler} = \frac{50 \times t^2}{D \times \sqrt[2]{L}}.$$

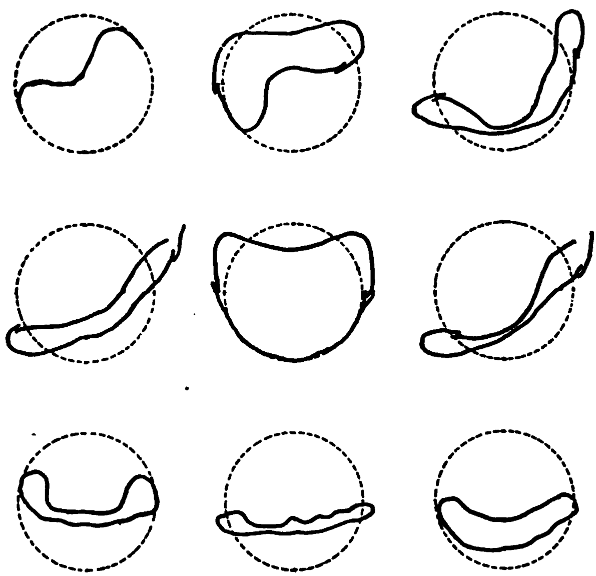
Example: Required the working-pressure for a steam-boiler with the furnace-tube described in the previous example?

Then $\frac{50 \times 12 \times 12}{33 \text{ inches} \times \sqrt[3]{15 \text{ feet}}} = 56$ lbs. per square inch, the working-pressure for this furnace-tube.

For boilers with large circular fire-boxes of, say, about 5 feet diameter, such as are used in donkey-boilers of steam-ships, Mr. Longridge's formula is as follows:—

Working-pressure $= \frac{50 \times t^2}{\sqrt[3]{L} \times D} - \frac{D}{L}$, in which the notation is the same as that in the previous formula.

The Resistance of a Furnace-Tube to change of form under external pressure, decreases as the length of the tube increases, therefore the collapsing-strength of tubes decreases as the length increases.



Figs. 54–62.—Examples of collapsed furnace-tubes.

The furnace-tubes of steam-boilers frequently become distorted by long wear, and as the collapsing-strength depends greatly upon the roundness and stiffness of the tube, the furnace-tubes of old boilers are generally weak. The strength of old furnace-tubes cannot be accurately determined by calculation, because the plates do not wear evenly and the thickness is not uniform, the sections of the tubes are seldom either fairly circular or symmetrical about the diametrical plane of the tubes, and there are generally weak places due to diminution of stiffness from strains caused by unequal expansion and contraction.

Furnace-tubes assume a variety of shapes in collapsing, a few of which are shown in Figs. 54 to 62.

Collapsing-strength of various forms of Furnace-Tubes.—The collapsing-pressure of furnace-tubes of different forms, when composed of plates of good quality, may be found by the various rules given in the following pages, each of which has been deduced from the results of carefully-conducted experimental tests.

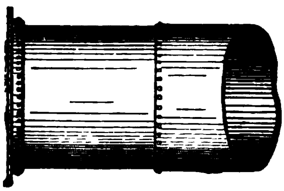


Fig. 63.—Plain furnace-tubes.

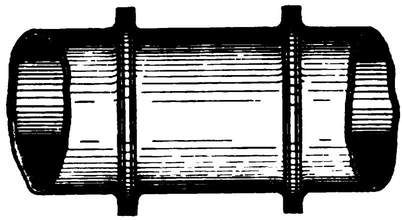


Fig. 64.—Furnace-tube with flanged seams.

The collapsing-pressure of plain furnace-tubes with either lap-jointed circumferential seams, as shown in Fig. 63, or with flanged seams as shown in Fig. 64, may be found by the following formula:—

Let t = the thickness of the tube in thirty seconds of an inch.
D = the external diameter of the tube in inches.
L = the length of the tube in inches, or the length between the flanges of a tube with flanged seams.
C = 600 for wrought-iron plates, and = 660 for mild-steel plates.

Collapsing-pressure of a plain furnace-tube in lbs. per square inch =

$$\frac{t^2 \times C}{D \times \sqrt[3]{L}}.$$

In the case of oval tubes, the diameter of the greatest circle of curvature is to be substituted for D in this formula.

Example 1: Required the collapsing-pressure of a plain furnace-tube of wrought-iron, as shown in Fig. 63, of ten feet in length, 32 inches external diameter, and $\frac{3}{8}$ inch thick?

$$\text{Then } \frac{12 \times 12 \times 600}{32 \text{ inches} \times \sqrt[3]{120} \text{ inches}} = 246 \text{ lbs. per square inch,}$$

the collapsing-pressure of this furnace-tube.

Example 2: Required the collapsing-pressure of the furnace-tube of wrought-iron with flanged seams, shown in Fig. 64. The length between the flanges is 36 inches, external diameter of tube 36 inches, thickness of tube $\frac{7}{16}$ inch?

Then $\dfrac{14 \times 14 \times 600}{36 \text{ inches} \times \sqrt[2]{36} \text{ inches}} = 544$ lbs. per square inch, the collapsing-pressure of this furnace-tube.

Steel possesses more stiffness than wrought-iron, but as the stiffness may be impaired by overheating, and by strains due to unequal expansion and contraction, steel may be reduced in strength to the same level as iron. Therefore the collapsing-pressures of furnace-tubes of wrought-iron and mild-steel are frequently both calculated by the same rule. In that case the value of C will be 600 for both these metals.

The results given by this formula agree very closely with the results of practical tests of plain tubes, as will be seen from the following Table, which contains in the last column a number of collapsing-pressures of furnace-tubes calculated by it, and also, for comparison, collapsing-pressures as found by experiment, and others calculated by Fairbairn's formula, No. 2.

TABLE 43.—COMPARISON OF CALCULATED COLLAPSING-PRESSURES WITH THE RESULTS OBTAINED IN PRACTICAL TESTS OF PLAIN FURNACE-TUBES.

Description.	Dimensions of the Experimental Tubes.			Collapsing-Pressure of Tube in Pounds per square inch.		
	External Diameter in Inches.	Length in Inches.	Thickness in thirty-seconds of an inch.	As found by Fairbairn's Formula (2).	As found by Experiment.	As found by the Author's Formula.
				lbs.	lbs.	lbs.
Tube tested by Fairbairn	7·87	276	5	109	110	114
Ditto, ditto	33·5	360	11	81	99	113
Ditto, ditto	42	420	12	78	97	100
Ditto, ditto	42	300	12	108	127	119
Furnace-tube tested by the Chief-Engineer of United States Navy	54	36	8	311	128	120
Furnace-tube tested at Greenock	38	86	16	740	450	436
Furnace tube-tested by Mr. Knight	36	24	8	700	235	218
Ditto, ditto	36	24	12	1568	468	490
Ditto, ditto	36	48	12	784	390	350
Furnace-tube tested at the works of Messrs. J. Howden & Co., Glasgow	43	23	17	2758	840	842

Corrugated Furnace-Tubes.—Furnace-tubes formed with annular corrugations, as shown in Fig. 65, are much stronger to resist a collapsing-pressure than plain furnace-tubes, and are considerably more elastic. They contain about 25 per cent. more heating-surface than plain tubes, when their surfaces are clean. The objection to these tubes is that, the corrugations are liable to become filled up with scale, when working with feed-water which deposits much sediment.

The collapsing-pressure of corrugated furnace-tubes, with corrugations $1\frac{1}{2}$ inches deep, shown in Fig. 65, may be found by the following formula:—


Fig. 65.—Corrugated furnace-tube.

Let D = the greatest external diameter of the tube in inches.
L = the length of the tube in inches.
t = the thickness of the tube in thirty seconds of an inch.

Collapsing-pressure of a corrugated furnace-tube in lbs. per square inch =

$$\frac{t^2 \times 1200}{D \times \sqrt[3]{L}}.$$

Example: Required the collapsing-pressure of a corrugated furnace-tube, 7 feet long, $\frac{1}{2}$ inch thick, and 38 inches extreme external diameter?

Then $\dfrac{16 \times 16 \times 1200}{38 \text{ inches} \times \sqrt[3]{84} \text{ inches}} = 882$ lbs. per square inch, the collapsing-pressure of this furnace-tube.

Spirally corrugated Furnace-Tubes.—The corrugations of these furnace-tubes are arranged in a spiral direction, shown in Fig. 66, with the object of obtaining greater longitudinal strength than is obtained when the corrugations are placed vertically.

The collapsing-pressure of spirally corrugated furnace-tubes, with corrugations $1\frac{1}{2}$ inches deep, may be calculated from the following formula:—


Fig. 66.—Spirally corrugated furnace-tube.

Let D = the greatest external diameter of the tube in inches.
L = the length of the tube in inches.
t = the thickness of the tube in thirty seconds of an inch.

Collapsing-pressure of spirally corrugated furnace-tubes in lbs. per square inch =

$$\frac{t^2 \times 1100}{D \times \sqrt[3]{L}}.$$

Example: Required the collapsing-pressure of a spirally corrugated furnace-tube 36 inches extreme external diameter, 6 feet 6 inches long, and $\frac{1}{2}$ inch thick?

Then $\frac{16 \times 16 \times 1100}{36 \text{ inches} \times \sqrt{78}} = 886$ lbs. per square inch, the collapsing-pressure of this furnace-tube.

Fig. 67.—Ribbed furnace-tube.

Ribbed Furnace-Tubes.—These furnace-tubes, shown in section in Fig. 67, are intended to combine the advantage of the longitudinal strength of plain furnace-tubes with the great resistance to a collapsing-pressure obtained in corrugated furnace-tubes. The collapsing-pressure of ribbed furnace-tubes, with ribs 9 inches apart, and not less than $1\frac{5}{16}$ inches high above the plain part of the tube, and with grooves under the ribs not more than $\frac{5}{8}$ inch deep, may be found by the following formula:—

Let D = the external diameter of the plain part of the tube in inches.
L = the length of the tube in inches.
t = the thickness of the plain part of the tube in thirty seconds of an inch.

Collapsing-pressure of ribbed and grooved furnace-tubes in lbs. per square inch =

$$\frac{t^2 \times 1350}{D \times \sqrt{L}}.$$

Example: Required the collapsing-pressure of a ribbed and grooved furnace-tube 84 inches long, 38 inches external diameter of plain part of tube, thickness of plain part $\frac{1}{2}$ inch?

Then $\frac{16 \times 16 \times 1350}{38 \text{ inches diameter} \times \sqrt{84}} = 992$ lbs. per square inch, the collapsing-pressure of this furnace-tube.

Compressive Stress on Furnace-Tubes from Collapsing Pressure.—The ultimate crushing or compressive stress to which a tube is subjected from a collapsing pressure may be found by the following formula:—

Let D = the external diameter of the tube in inches.
P = the pressure in pounds per square inch on the tube when it collapsed.
t = the thickness of the tube in inches.

The compressive stress in pounds per square inch of section of the tube =

$$\frac{D \times P}{t \times 2}$$

Example: Required the compressive stress per square inch of section of the tube described in the previous example, when collapse took place?

Then $\frac{38 \times 992}{\cdot 5 \times 2} = 37696$ pounds, the crushing stress on the tube per square inch of section of the tube at the moment it collapsed.

Strengthening Furnace - Tubes. — Plain furnace - tubes may be strengthened by flanged seams and anti-collapse hoop-seams. These seams are sometimes called expansive seams, but they are too rigid to permit expansion, except of a trifling amount, and they are only effective in preventing collapse of the tube. Weak furnace-tubes may be strengthened by anti-collapse angle-rings.

Adamson's Flanged Seam, shown in Fig. 68, consists of external flanges formed on the ends of the belts of the tubes. A caulking-ring is placed between each flange and they are riveted together. This seam is elastic in a longitudinal direction, but it is very rigid circumferentially, and the root of the flange should have a considerable radius, to prevent grooving from expansion and contraction. This radius should not be less than $\frac{3}{4}$ inch, but rather greater.

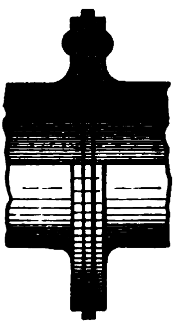


Fig. 68.—Adamson's flanged seam.

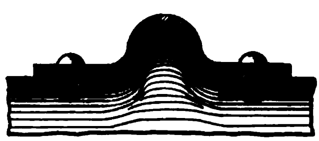


Fig. 69.—Anti-collapse hoop-seam.

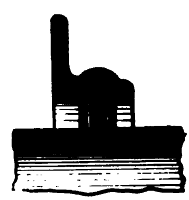


Fig. 70.—Strengthening ring.

Anti-Collapse Hoop-Seams are frequently employed to joint the belts of plates of a furnace-tube, as shown in Fig. 69. They give considerable strength to a tube to resist a collapsing pressure, but have the disadvantage of exposing the rivets of the joint to the fire.

When several furnace-tubes are employed in a steam-boiler, the flanged seams, or hoop-seams, of one tube should be placed sufficiently clear of those of the adjoining tube to prevent the harbourage of deposits of sediment or scale.

Anti-Collapse Angle-Rings, or strengthening rings, for the middle of a belt of plates of a long plain furnace, shown in Fig. 70, are generally formed of 3 inches × 3 inches × $\frac{1}{2}$ angle-iron or angle-steel. A water-space of not less than 1 inch deep is formed by ferules placed under the ring, through which the rivets pass that secure the ring to the furnace-tube. The thickness of the ferules is generally $\frac{3}{16}$ inch, and the pitch of the rivets is about 6 inches.

Rings of tee-section have been employed in some cases for this purpose, but they are very rigid, and inferior to angle-rings, which, being narrower, offer less resistance to the escape of steam-bubbles generated under the ring.

Anti-collapse angle-rings are liable to harbour sediment and scale, and they are troublesome to keep clean. They are only employed to give support to weak furnace-tubes.

Various forms of Furnace-Tubes.—There are numerous special forms of furnace-tubes, a few of which may be briefly described as follows:—

Fig. 71.—Paxman's furnace-tube.

Paxman's Furnace-Tubes, shown in Fig. 71, consist of welded belts, or sections, bell-mouthed at each end. Each section has a large and a small bell-mouth, the small end of one fitting into the large end of the other. The inner plate edges are turned away from the fire, and the rivets are well removed from injurious action of flame. This seam gives stiffness to the tube circumferentially, and favours elasticity longitudinally.

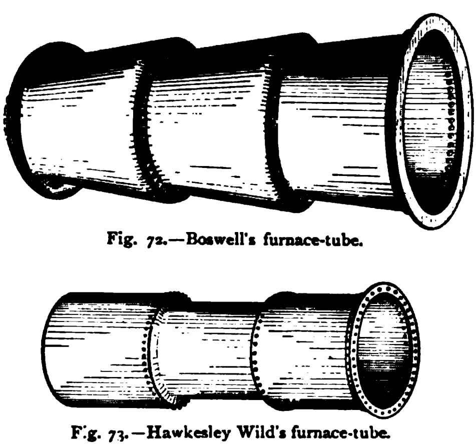


Fig. 72.—Boswell's furnace-tube.

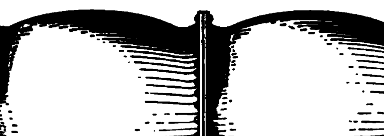


Fig. 73.—Hawkesley Wild's furnace-tube.

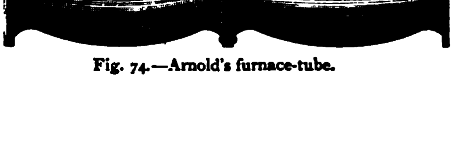


Fig. 74.—Arnold's furnace-tube.

Boswell's Cone-Belt Furnace-Tube, shown in Fig. 72, consists of taper rings or belts, with the small ends flanged outwardly, and the large ends flanged inwardly. The taper is sufficient to allow the whole tube to pass inside the angle-ring on the end-plate of the boiler, so that, in case of repairs, the furnace-tube can be withdrawn from the boiler without removing the front end-plate.

Hawkesley Wild's Furnace-Tube, shown in Fig. 73, is formed of alternate large and small rings or belts of plates. The ends of the small belts are enlarged to fit into the ends of the large belts, and they are riveted together. A certain amount of elasticity is imparted to the tube by the curved enlargement at the ends of the small belts.

Arnold's Furnace-Tube, shown in Fig. 74, consists of rings or belts of plates of convex form, joined with flanged seams. This form is adopted with the view of obtaining circumferential strength.

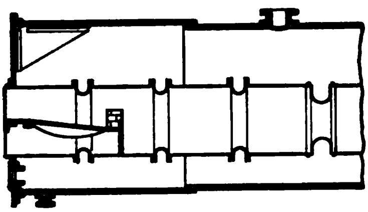


Fig. 75.—Midgeley's furnace-tube.

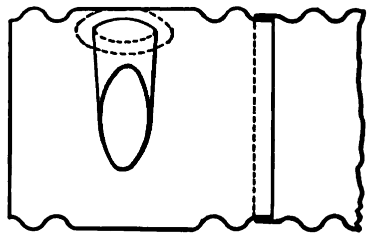


Fig. 76.—Corrugated furnace-tube with plain part for cross-tubes.

Midgeley's Furnace-Tube, shown in Fig. 75, consists of inverted expansion-hoops, riveted between the belts of plates forming the furnace-tube, for the purpose of giving elasticity to the tube; and also of causing the current of the fuel-gases to be broken up by contact with the convex surface of the hoop projecting into the tube.

Special Furnace-Tubes.—A corrugated furnace-tube formed with plain parts spaced suitably for the insertion of Galloway-tubes is shown in Fig. 76. The plain parts are of the same external diameter as that of the corrugations.

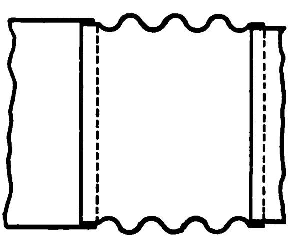


Fig. 77.—Corrugated hoop-seam.

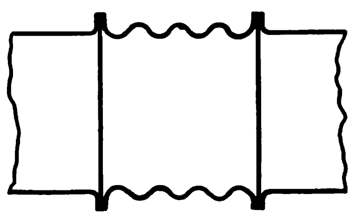


Fig. 78.—Corrugated hoop flanged-seam.

A furnace-tube with corrugated hoop-seams is shown in Fig. 77. A corrugated hoop is inserted between belts of plain plates and joined with riveted lap-joints. A furnace-tube with corrugated hoop-seams formed with flanges to join belts of plain plates is shown in Fig. 78.

Collapsing-Pressure of Cylindrical Fire-Boxes of Vertical Steam-Boilers.—The collapsing pressure of the fire-box of a vertical boiler fitted with cross tubes as shown in Fig. 79, may be estimated by the following formula:—

Let t = the thickness of the fire-box in thirty-seconds of an inch.
D = the external diameter of the fire-box in inches.
H = the height of the fire-box in inches.
C = a constant varying as follows:—

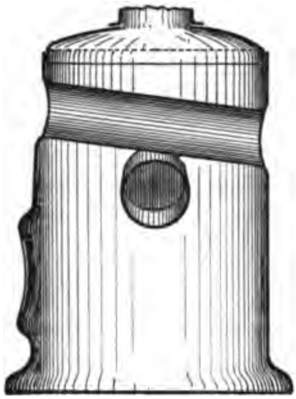


Fig. 79.—Fire-box of vertical boiler.

C = 330 for wrought-iron plates, and = 363 for mild-steel plates, for fire-boxes with single-riveted lap-jointed vertical seams, and punched rivet-holes.

C = 380 for wrought-iron plates, and = 420 for mild-steel plates, for fire-boxes with double-riveted lap-jointed vertical seams, and punched rivet-holes.

C = 410 for wrought-iron plates, and = 450 for mild-steel plates, for fire-boxes with single-riveted lap-jointed vertical seams, or single-riveted single butt-straps, and drilled rivet-holes.

C = 440 for wrought-iron plates, and = 485 for mild-steel plates, for fire-boxes with double-riveted lap-jointed vertical seams, or double-riveted single butt-straps, and drilled rivet-holes.

C = 470 for wrought-iron plates, and = 520 for mild-steel plates, for fire-boxes with single-riveted double butt-jointed vertical seams, and drilled rivet-holes.

C = 490 for wrought-iron plates, and = 540 for mild-steel plates, for fire-boxes with double-riveted double butt-jointed vertical seams, and drilled rivet-holes.

C = 500 for wrought-iron plates, and = 550 for mild-steel plates when the vertical seams are welded.

Collapsing pressure of a vertical circular fire-box fitted with cross-tubes, in pounds per square inch, =

$$\frac{t^2 \times C}{D \times \sqrt[2]{H}}.$$

Example: The fire-box of a vertical boiler is 5 feet in diameter, and 6 feet 6 inches high, and formed of mild-steel plates, $\frac{7}{16}$ inch thick, with double-riveted lap-jointed vertical seams, and drilled rivet-holes. Required the collapsing pressure?

Then, the fire-box is 14 thirty-seconds thick, and,

$$\frac{14 \times 14 \times 485}{60 \text{ inches} \times \sqrt[2]{78} \text{ inches}} = 180 \text{ pounds per square inch,}$$

the estimated collapsing pressure of this fire-box.

A large fire-box of this form is very weak to resist collapsing pressure, and should be strengthened by stays.

SECTION IV.

TESTS OF MATERIALS; STRENGTH AND WEIGHT OF BOILER-PLATES; EFFECT OF TEMPERATURE ON METALS; RIVET-HOLES; RIVETS; RIVETED-JOINTS OF STEAM-BOILERS; CAULKING; ENDS OF CYLINDRICAL SHELLS; STAYS FOR BOILERS, ETC.

which results from finishing the metal at a red-heat and allowing it to cool in the open air. When a tensile stress is applied exceeding the natural elastic limit, a new elastic limit is formed, equal to, and sometimes exceeding, that stress. In this manner a new elastic limit may be formed anywhere up to nearly the tensile strength, showing that little importance can be attached to the elastic limit, unless it is known how it is formed.

The elastic limit may be found to range from about 40 per cent. of the tensile strength of the metal up to nearly its tensile strength, according to the previous manipulation of the metal, or the influence of its chemical composition. The natural elastic limit of a metal may be elevated by cold-rolling, cold-hammering, cold-stretching, and wire-drawing.

The Tensile Strength of a Metal is the resistance offered by it to being pulled asunder. It represents a load, which after being applied has been gradually increased until it produced rupture. The tensile strength of metal is influenced by previous mechanical treatment, and the same treatment which causes an elevation of elastic limit may also cause an elevation of tensile strength.

The tensile strength of steel-plates increases with the amount of work done on them in the process of manufacture. For instance, a piece of mild-steel of $1\frac{1}{2}$ inches thickness, having a tensile strength of 24 tons per square inch, was reduced in thickness by rolling when hot, when the tensile strength varied with the thickness to which the plate was rolled down, as follows:—

Thickness of plate in inches . .	$1\frac{1}{2}$,	$1\frac{1}{4}$,	1,	$\frac{3}{4}$,	$\frac{1}{2}$,	$\frac{3}{8}$,	$\frac{5}{16}$
Tensile strength in tons per square inch	24,	25·3,	26·7,	27·5,	28·6,	30,	31·3

Showing that the effect of rolling down the metal to about one-fifth of its original thickness, was to increase its tenacity by about one-third.

In another case, a piece of mild-steel plate $\frac{3}{4}$-inch thick, having a tensile strength of 28 tons per square inch, and an elongation of 21 per cent. in 8 inches, was rolled down to $\frac{3}{8}$-inch thick, when the tenacity became about 31 tons per square inch, and the elongation was 19 per cent. in 8 inches.

Elongation under Tensile Stress is displayed by a metal between the load at the elastic limit and its tensile strength. The maximum elongation ranges from less than 1 per cent. in cast-iron and hard steel, to about 30 per cent. in soft wrought-iron and mild-steel. Treatment which elevates the elastic limit and tensile strength of a metal, detracts from its power of elongation.

Contraction of Area under Tensile Stress begins after the elastic limit is reached. The section contracts rapidly as elongation proceeds. Contraction of area occurs locally after the maximum load has been passed, and marks the place where rupture is made. The nature of the broken surface is fibrous in ductile iron, silky in ductile steel, and granular in brittle metal.

Test-Requirements of the Governing Authorities for Steel-Plates, Angles and Tee-bars.—The test requirements of the various governing authorities are briefly summarised in the following paragraphs. In each case there is written after the words *Tensile test* the limits of tensile strength between which the material is required to break, together with the minimum elongation per cent. after fracture, which will be accepted as a guarantee of ductility. The parallel length over which the elongation is calculated is also given.

The normal tensile test-piece consists of a shearing, say 20 inches by 2 inches, cut from the plate or bar, and afterwards machined to the form shown in Fig. 80. The part A B is made parallel, and is generally 8 inches long. In view of the great amount of testing which has to be done, uniformity of length is desirable. One and a half inches is a convenient width for the parallel portion.

Fig. 80.—Test-piece.

The bending test-pieces are usually shearings 9 inches by 2 inches or thereabout. The radius of the curve to which the test-piece is required to bend without fracture is usually stated in terms of the thickness of the piece. The tests may consist of either *cold* or *temper* bends. In the first case the shearings are bent in the condition in which they are when taken from the plate; but in making the temper-test the shearing is, before bending, heated to a dull cherry-red, and cooled in water having a temperature of 82° Fahr.

All plates and angles are numbered consecutively in the order of rolling at the steel-works, and this distinguishing number is stamped on the finished plate or bar, and also on the tensile and bending test-pieces. By means of this number, it is possible at any time to trace the behaviour of the plate when tested, its composition, the charge from which it was made, the date of manufacture, dimensions, weight, appropriation-marks and general order number.

British Admiralty Test-Requirements.—*Boiler Plates, Tensile test.*—Internal plates exposed to flame 24 to 26 tons. Shell-plates 26 to 30 tons per square inch, with an elongation of 20 per cent. in a length of 8 inches.

Bending Test.—Strips cut crosswise or lengthwise $1\frac{1}{2}$ inches wide, heated uniformly to a low cherry-red, and cooled in water of 82° Fahr., must stand bending in a press to a curve of which the inner radius is one and a half times the thickness of the steel tested.

The ductility of every plate, and bar, is to be ascertained by the application of one or both of these tests to the shearings, or by bending them cold by the hammer. One plate to be taken for testing from every invoice, provided the number of plates does not exceed fifty. If above that number, one for every additional fifty or portion of fifty. Steel may be r eceived. or rejected, without a trial of every thickness on the invoice.

Board of Trade Test-Requirements.—*Tensile Test.*—Furnace-plates, and plates exposed to flame, 26 to 28 tons per square inch, with 20 per cent. elongation in 10 inches. Shell-plates and plates not exposed to flame, 27 to 32 tons per square inch, with 20 per cent. elongation in 10 inches.

A tensile test is required from every plate. Twenty-five per cent. of these tests are made in presence of a Board of Trade surveyor, the others are made by the steel-makers. The tensile strength and elongation are stamped on every plate. The letters **B.T.** are also stamped on all plates passed by the Board of Trade surveyor.

Bending Test.—From every plate exposed to flame a shearing is taken, heated to a low cherry-red, and cooled in water of 82° Fahr. It must then stand bending to a curve, the inner radius of which is equal to one and a half times the thickness of the plate.

In the case of shell-plates and plates not exposed to flame, a test-piece is taken from each plate and bent cold to a curve, the inner radius of which is equal to one and a half times the thickness of the plate. All bending tests are made in the presence of the surveyor.

Lloyds-Register Test-Requirements.—*Tensile Test.*—Boiler-plates, 26 to 30 tons per square inch, and 20 per cent. elongation in 8 inches. A tensile test is taken by the surveyor from every charge or cast from which the plates are made.

Bending Tests.—A temper-bend is taken from every plate and bar, and tested in the presence of the surveyor. The test-pieces are heated to a low cherry-red, and cooled in water of 82° Fahr. They must then stand bending to a curve, the inner radius of which is one and a half times the thickness of the piece.

All plates passed by Lloyd's surveyors are branded,

Bureau-Veritas Test-Requirements.—*Tensile Test.*—Boiler-plates, and angles, 27 to 32 tons per square inch, with 20 per cent. elongation in 8 inches.

Bending Tests.—Temper-tests are taken from each piece, and bent in the presence of the surveyor. The pieces are heated to a low cherry-red, cooled in water of 82° Fahr., and bent to a curve, the inner radius of which is one and a half times the thickness of the piece.

All plates passed by the Bureau Veritas surveyors are branded,

The above described tests by the several governing authorities refer to materials used in the construction of marine-boilers. They are the only compulsory tests existing for this purpose.

General Physical Tests of Materials.—All materials used in the construction of steam-boilers should be of the best quality of their respective kinds, carefully inspected and subjected to the tests given in the following pages:—

Tensile Strength of Bar-Iron.—All round, flat, or square bars of wrought-iron should be capable of sustaining an ultimate tensile stress of 22 tons, or 49280 lbs., per square inch, with an elastic limit of 25000 lbs. per square inch, and a minimum ductility, measured by elongation, of 20 per cent.

Tensile Strength of Iron-Stays and Bolts.—Wrought-iron stays and bolts should have an ultimate tensile strength of 23 tons, or 51520 lbs., per square inch, with an elongation of not less than 20 per cent.

Tensile Strength of Mild-Steel Stays and Bolts.—Steel for stays, studs, and bolts, should have an ultimate tensile strength of not exceeding 27 tons, or 60480 lbs., per square inch, with an elongation of not less than 20 per cent.

Tensile Strength of Rivets.—Iron-rivets should be of good charcoal-iron, having an ultimate tensile strength of from 24 to 25 tons per square inch, with an elongation of not less than 30 per cent. Steel-rivets should be of soft mild-steel, having an ultimate tensile strength of from 28 to 30 tons per square inch, with an elongation of not less than 30 per cent.

Wrought-Iron Boiler-Plates should be soft, ductile, of good quality, and free from laminations, blisters, and defects. At least one plate of every boiler should be 3 inches longer than the required size, from which a test-strip should be cut and tested. The piece so tested should have an ultimate tensile strength in the case of ordinary plates of 21 tons, or 47040 lbs., per square inch along the grain, and 18 tons, or 40320 lbs., per square inch across the grain: and show a ductility, measured by elongation, of not less than 10 per cent. in 8 inches.

Plates of best Yorkshire iron should have a tensile strength along the grain of not less than 23 tons, or 51520 lbs., per square inch, and 20 tons, or 44800 lbs., per square inch across the grain, with an elongation of not less than 15 per cent. in 8 inches. They should admit of being bent hot to an angle of 125° with the grain, and to an angle of 100° across the grain.

The tensile strength of wrought-iron boiler-plates varies considerably. The following are the results of tests of plates, by different makers, of the ordinary quality used for boiler-shells.

Tensile strength of wrought-iron boiler-plates along the grain:—

20·00	tons per	square inch, or	44800	lbs. per	square inch.
20·60	,,	,,	46144	,,	,,
20·83	,,	,,	46659	,,	,,
21·00	,,	,,	47040	,,	,,
21·16	,,	,,	47398	,,	,,
21·50	,,	,,	48160	,,	,,
22·00	,,	,,	49280	,,	,,
22·24	,,	,,	49818	,,	,,

The tenacity of the plates across the grain, is on an average 10 per cent. less than that along the grain.

Weight of Wrought-Iron Plates.—The weight of a cubic foot of wrought-iron is = 480 lbs.

The weight of one square foot of wrought-iron in lbs. = thickness of plate in sixty-fourths of an inch multiplied by ·625.

The weight of one square foot of wrought-iron in lbs. = thickness of plate in thirty-seconds of an inch multiplied by 1·25.

The weight of one square foot of wrought-iron in lbs. = thickness of plate in sixteenths of an inch multiplied by 2·5.

The weight of one square foot of wrought-iron in lbs. = thickness of plate in eighths of an inch multiplied by 5.

The weight of one square foot of wrought-iron plate of different thickness is given in the following Table:—

TABLE 44.—WEIGHT OF ONE SQUARE FOOT OF WROUGHT-IRON PLATE.

Thickness of Plate.	Weight of 1 Square Foot in lbs.	Thickness of Plate.	Weight of 1 Square Foot in lbs.
Inch.		Inch.	
$\frac{3}{16}$	7·5	$\frac{5}{8}$	25·0
$\frac{1}{4}$	10·0	$\frac{11}{16}$	27·5
$\frac{5}{16}$	12·5	$\frac{3}{4}$	30·0
$\frac{3}{8}$	15·0	$\frac{13}{16}$	32·5
$\frac{7}{16}$	17·5	$\frac{7}{8}$	35·0
$\frac{1}{2}$	20·0	$\frac{15}{16}$	37·5
$\frac{9}{16}$	22·5	1	40·0

The length of a boiler-plate in feet multiplied by its width in feet, and by the weight of a square foot of plate of the required thickness, from the above table, will give the weight of the plate in lbs.

Mild-Steel Boiler-Plates.—Mild-steel is the best material for boiler-plates, being ductile, homogeneous, and free from laminations, but it is a much more delicate material to work than wrought-iron, and it will not bear rough treatment without injury. It should be worked at either a red-heat or quite cold, and not at a blue-heat. In welding, it should be pushed or pressed and not flogged together. During the process of bending a plate, it should be pressed into shape and not driven by blows.

Local hammering of a hot-plate may cause the formation of hard knots of condensed metal, which behave differently in cooling to the surrounding uninjured metal, and develop into cracks when the plate is cold. Plates may fail from the presence of initial strains, resulting from inequality of cooling when the steel is in the process of manufacture, and also from the effects of punching or shearing, and hammer-blows on the cold metal. These injuries may generally be corrected and the plates rejuvenated by annealing. They should be heated to a blood-red heat in a furnace, and allowed to cool gradually. The cooling is best effected by allowing the fire of the furnace to die out, the plates remaining in the furnace until quite cold.

All plates should be soft and ductile, and free from cracked edges and other defects. A test-strip taken from each plate should have a tensile strength of about 28 tons, or say 63000 lbs., per square inch, with an elongation of not less than 20 per cent. in 8 inches. Plates should not be used if the test shows a tensile strength less than 26 tons, or 58240 lbs., per square inch; or greater than 30 tons, or 67200 lbs., per square inch, or if the elongation is greater than 30 per cent.

The elongation may vary with the thickness of the plate as follows:—For plates up to $\frac{3}{8}$ inch thick, the elongation may be 20 per cent.; for plates from $\frac{7}{16}$ to $\frac{3}{4}$ inch thick, 22 per cent.; and for plates above $\frac{3}{4}$ inch thick, 25 per cent.

The tenacity of mild-steel boiler-plates varies considerably.

The following are the results of a number of tests of good plates of the quality generally used for boiler-shells.

Tensile strength of mild-steel boiler-plates:—

26·23	tons, or	58756	lbs. per sq. inch,	with an elongation of	24·5	per cent.
26·88	,,	60012	,,	,,	23·9	,,
27·20	,,	60928	,,	,,	25·4	,,
27·45	,,	61488	,,	,,	24·9	,,
27·70	,,	62048	,,	,,	24·5	,,
28·12	,,	62989	,,	,,	26·6	,,
28·30	,,	63392	,,	,,	25·8	,,
28·57	,,	64000	,,	,,	23·7	,,
28·68	,,	64244	,,	,,	24·1	,,
29·00	,,	64960	,,	,,	23·3	,,

The elongation was taken in a length of 8 inches. The tenacity was the same both lengthways and crossways of the plates.

The reduction of area at the point of fracture of the test-piece varied from 45 to 61 per cent.

It is difficult to measure the reduction of area of a test-piece accurately, and it is unreliable and of little or no value as an indication of the quality of a material.

Analysis of Mild-Steel Boiler-Plates.—The strength of mild-steel is influenced by its chemical composition. The nature and quality of steel depends largely upon the percentage of constituent carbon.

The average analysis of a number of mild-steel boiler-plates by different makers, is given in Table 45 as representative examples of the quality used for boiler-shells.

In addition to these constituents, the plates frequently contain from ·01 to ·03 silicon, and ·05 copper. It is essential to durability that the plates contain the smallest possible amount of phosphorus and sulphur.

TABLE 45.—ANALYSIS AND STRENGTH OF MILD-STEEL BOILER-PLATES.

Chemical Composition.				Tensile Strength.		Elongation in a Length of 8 inches.
Carbon.	Phosphorus.	Sulphur.	Manganese.	In Tons per Square Inch.	In lbs. per Square Inch.	Per Cent.
·170	·041	·040	·58	29·14	65274	27
·166	·037	·040	·56	28·75	64400	26
·161	·038	·034	·57	28·62	64109	26
·149	·034	·042	·57	28·20	63168	23
·158	·035	·037	·55	28·31	63415	26
·155	·036	·035	·53	28·10	62744	25
·148	·032	·030	·52	27·80	62270	24
·160	·039	·024	·47	27·68	62000	28

Analysis of Mild-Steel for Fire-Boxes.—The following is an analysis of good mild-steel for fire-boxes:—

Carbon ·10, phosphorus ·005, sulphur none, manganese ·30, silicon ·014. Its tensile strength is 25 tons, or 56000 lbs., per square inch, with an elongation of 35 per cent. in 8 inches.

Tests of other steel used for fire-boxes show a tensile strength of from 59000 to 68000 lbs. per square inch, with an elongation of a little more than 30 per cent.

Flanging-Steel.—Steel-plates which require to be flanged, should have a tensile strength of from 56000 to 63000 lbs. per square inch, with an elongation of not less than 25 per cent. The steel should be flanged at a good red-heat, and should not be worked after it has cooled to less than a cherry-red heat in daylight. After flanging, the steel should be annealed by heating the whole plate uniformly to a dull-red heat and allowing it to cool slowly.

Bending-Tests for Steel Boiler-Plates.—A piece of boiler-plate, not less in length than twenty times the thickness of the plate, should be capable of being bent double and hammered down upon itself when cold, as shown in Fig. 81, without fracture, when its thickness does not exceed ½ inch. Plates above that thickness should admit of being bent round a mandrel, of a diameter equal to one-and-a-half-times the thickness of the plate, as shown in Fig. 82, without showing signs of distress. If the bent edge shows any marks or roughness, which appear under a magnifying-glass to be a series of incipient cracks, the plate may be considered to be defective.

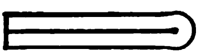


Fig. 81.—Bending-test.

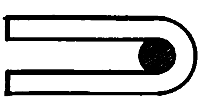


Fig. 82.—Bending-test.

Weight of Mild-Steel Plates.—The weight of a cubic foot of Siemens mild-steel is = 490 lbs.

The weight of one square foot of mild-steel in lbs. = thickness of plate in sixty-fourths of an inch multiplied by ·638.

The weight of one square foot of mild-steel in lbs. = thickness of plate in thirty-seconds of an inch multiplied by 1·276.

The weight of one square foot of mild-steel in pounds = thickness of plate in sixteenths of an inch multiplied by 2·552.

The weight of one square foot of mild-steel = thickness of plate in eighths of an inch multiplied by 5·104.

The weight of a square foot of mild-steel plate of different thickness is given in the following Table :—

TABLE 46.—WEIGHT OF ONE SQUARE FOOT OF MILD-STEEL PLATE.

Thickness of Plate.	Weight of 1 Square Foot in lbs.	Thickness of Plate.	Weight of 1 Square Foot in lbs.
Inch.		Inch.	
$\frac{1}{4}$	10·208	$\frac{15}{16}$	38·280
$\frac{5}{16}$	12·760	1	40·833
$\frac{3}{8}$	15·312	$1\frac{1}{16}$	43·385
$\frac{7}{16}$	17·864	$1\frac{1}{8}$	45·937
$\frac{1}{2}$	20·416	$1\frac{3}{16}$	48·489
$\frac{9}{16}$	22·968	$1\frac{1}{4}$	51·041
$\frac{5}{8}$	25·520	$1\frac{5}{16}$	53·593
$\frac{11}{16}$	28·072	$1\frac{3}{8}$	56·145
$\frac{3}{4}$	30·624	$1\frac{7}{16}$	58·697
$\frac{13}{16}$	33·176	$1\frac{1}{2}$	61·249
$\frac{7}{8}$	35·728	$1\frac{9}{16}$	63·801

Weight of Angle-Bars and Tee-Bars of Wrought-Iron and Mild-Steel.—The weight of angle-bars and tee-bars by different makers, varies considerably. The average weight of these bars may be found by the following *Rules* :—

Weight of 1 foot in length of angle-iron and tee-iron in pounds. *Rule :* Add the outside widths of the flanges in inches together, from the sum subtract the thickness of one flange in inches, multiply the remainder by the thickness, and by 3·36 for iron, and by 3·429 for mild-steel, the product is the weight of 1 foot in length of the bar in pounds. When the flange is taper, the mean thickness is to be measured.

Example : Required the weight of 1 foot in length of angle-iron $3 \times 3 \times \frac{1}{2}$: and also the weight of 1 foot in length of the same size of angle-steel.

Then $(3 + 3) - \cdot 5 = 5\cdot 5 \times \cdot 5 \times 3\cdot 36 = 9\cdot 24$ lbs., the weight of 1 foot in length of the angle-iron.

And $(3 + 3) - \cdot 5 = 5\cdot 5 \times \cdot 5 \times 3\cdot 429 = 9\cdot 43$ lbs., the weight of one foot in length of the angle-steel.

Length of Angle-Bar required to make an Angle-Ring, or Hoop.—The length of bar required to form an angle-ring of a given diameter varies with the temperature at which the bar is worked, and with the method of bending it.

The length of bar required to make angle-hoops of the sizes generally used in boiler work, may in a general way, be determined by the following *Rules*:—

The length of bar required to form an outside angle-hoop, as shown in Fig. 83, may be found by this *Rule*:—

Add the thickness of the bar in inches to the width in inches, multiply the sum by ·45, add the internal diameter of the hoop in inches to the product, and multiply by 3·1416: the product divided by 12 will give the length of the bar in feet required to make the hoop.

In some sections of angle-bars it is more correct to use a multiplier of ·4 with the above rule than ·45, and in others ·5 gives more correct results. Hence, the average multiplier is ·45.

When the angle-bar has flanges of unequal widths, the width of the flange forming the end or base of the ring is to be measured.

Example: Required the length of angle-bar of $3 \times 3 \times \frac{1}{2}$ inch, to form an outside hoop of 5 feet internal diameter?

Then, 3 inches wide + ·5 inch thick = $3·5 \times ·45 = 1·575 + 60$ inches internal diameter = $61·575 \times 3·1416 = 193·44$ inches circumference $\div$ 12 = 16 feet $1\frac{7}{16}$ inches circumference, or the length of bar required to form this hoop.

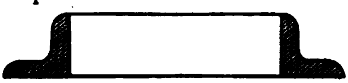


Fig. 83.—Outside angle-hoop.

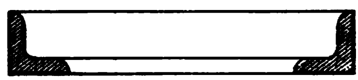


Fig. 84.—Inside angle-hoop.

The length of bar required to form an inside angle-hoop as shown in Fig. 84 may be found by this *Rule*:—

Add the thickness of the bar in inches to the width in inches, multiply the sum by ·45, subtract the product from the outside diameter of the hoop, and multiply by 3·1416: the product divided by 12 will give the length of bar in feet required to form the hoop.

Example: Required the length of angle-bar $3 \times 3 \times \frac{1}{2}$ inch required to form an inside hoop of 6 feet external diameter.

Then, 3 inches wide + ·5 inch thick = $3·5 \times ·45 = 1·575$, and 6 feet $\times$ 12 = 72 inches external diameter − 1·575 = $70·425 \times 3·1416 = 221·247$ inches circumference $\div$ 12 = 18 feet $5\frac{5}{16}$ inches circumference, or the length of bar required to form the hoop.

The length of angle-bar required to form hoops is frequently determined by the following *Rules*:—

To find the length of bar for an outside angle-hoop. *Rule*: To the internal diameter of the hoop add twice the thickness of the root of the angle, multiply the sum by 3·1416, and divide the product by 12; the quotient will be the length of bar in feet.

To find the length of bar for an inside angle-hoop. *Rule*: From the external diameter of the hoop subtract twice the thickness of the root of the angle, multiply the remainder by 3·1416 and divide the product by 12; the quotient will be the length of the bar in feet.

For instance, the length of bar required to form an outside angle-hoop of 2 feet $10\frac{1}{4}$ inches internal diameter, the thickness at the root of the angle being $\frac{7}{8}$ inch, is $= 34{\cdot}25$ inches diameter $+ ({\cdot}875 \times 2) = 36$ inches $\times$ $3{\cdot}1416 \div 12 = 9$ feet $5\frac{1}{16}$ inches.

Length of Bar or Plate to form a Plain Ring or Hoop.—The length in feet of bar or plate to form a plain ring, may be found by adding the thickness of the bar or plate to the internal diameter of the ring in inches, multiplying the sum by 3·1416 and dividing by 12. If the external diameter be given, subtract the thickness of the bar or plate from the diameter, multiply the remainder by 3·1416, and divide by 12; the quotient will be the length of the bar or plate in feet.

Copper for Fire-Boxes.—The fire-boxes of locomotive boilers are generally made of copper, but they are in some cases of mild-steel. Copper possesses great heat-enduring properties and a high conductivity. It is less liable to blister, and suffers less injury from overheating, than mild steel, although it is sooner reduced in tenacity by overheating than the latter metal.

Copper for fire-box-plates, rivets, and stud-stays, should be of best quality, having a tensile strength of not less than 34,000 lbs. per square inch, with an elongation of not less than from 20 to 25 per cent. It should contain $99\frac{1}{2}$ per cent. of pure copper, or only $\frac{1}{2}$ per cent. of impurities.

A test-piece cut from a plate or bar should admit of being bent double without showing any signs of cracking.

The weight of 1 cubic foot of fine copper is = 558 lbs.

The weight of 1 square foot of copper-plate in pounds is = thickness of plate in sixty-fourths of an inch multiplied by ·7265.

The weight of 1 square foot of copper-plate in pounds is = thickness of plate in thirty-seconds of an inch multiplied by 1·453.

The weight of 1 square foot of copper-plate in pounds is = thickness of plate in sixteenths of an inch multiplied by 2·906.

The weight of 1 square foot of copper-plate in pounds is = thickness of plate in eighths of an inch multiplied by 5·812.

The weight of a square foot of copper-plate of different thickness is given in the following Table :—

TABLE 47.—WEIGHT OF ONE SQUARE FOOT OF COPPER-PLATE.

Thickness of Plate.	Weight of 1 Square Foot in lbs.	Thickness of Plate.	Weight of 1 Square Foot in lbs.
Inch.		Inch.	
$\frac{3}{16}$	8·718	$\frac{5}{8}$	29·060
$\frac{1}{4}$	11·624	$\frac{11}{16}$	31·966
$\frac{5}{16}$	14·530	$\frac{3}{4}$	34·872
$\frac{3}{8}$	17·436	$\frac{13}{16}$	37·778
$\frac{7}{16}$	20·342	$\frac{7}{8}$	40·684
$\frac{1}{2}$	23·248	$\frac{15}{16}$	43·590
$\frac{9}{16}$	26·154	1	46·496

Copper-plates for locomotive fire-boxes are generally ½ inch thick.

Shearing-Strength of Metals.—The shearing-strength of a metal is the resistance offered by it to being severed or cut through by detrusion.

The ratio of the shearing-strength to the tensile-strength of metals is on an average as follows:—

	Per cent.	
The shearing-strength of wrought-iron of best quality averages	84	of its tensile strength.
The shearing-strength of wrought-iron of good medium quality averages	80	,, ,,
The shearing-strength of wrought-iron of common quality averages . . .	75	,, ,,
The shearing-strength of mild-steel of best quality averages	82	,, ,,
The shearing-strength of mild-steel of good average quality averages	80	,, ,,
The shearing-strength of mild-steel of hard quality averages	67	,, ,,
The shearing-strength of cast-iron of good quality averages	38	,, ,,

The shearing-strength of metals of a hard and brittle nature is very low and variable.

Strength of Metals at different Temperatures.—The tenacity of metals is considerably diminished at high temperatures. Alloys are very unreliable when heated, as they suffer considerable diminution in strength at moderately high temperatures. It is, therefore, important to know at what temperatures the strength of different metals is seriously impaired.

A series of experiments on the strength of metals at various temperatures were made by the Admiralty, the principal results of which are given in Table 48.

In these experiments, all the mixtures of metals composing the gun-metal suffered a gradual but slight loss of strength and ductility up to a certain temperature, at which a great change took place, the strength being reduced to about one-half the original, and the ductility ceased. At temperatures above this point up to 500° Fahr. there was little further loss of strength. The temperature at which this change took place varied even with the same composition of metal.

The precise temperature at which this great change and loss took place, was found to be about 370° Fahr. in test number 1, and at little over 250° Fahr. in test number 2.

Phosphor-bronze was less affected by temperature than gun-metal, and at 500° Fahr. retained more than two-thirds of its strength and one-third its ductility. Copper was not much affected in strength up to 350°, and the loss of strength at 500° Fahr. was not serious.

TABLE 48.—TENACITY OF VARIOUS METALS AT DIFFERENT TEMPERATURES.

Temperature.	Gun-metal Rods, 1-in. Diameter.											
	1. Copper . 87·75 Tin . . 9·75 Zinc . . 2·5		2. Second Set. Copper . 87·75 Tin . . 9·75 Zinc . . 2·3		3. Copper . . 91 Tin . . 7 Zinc . . 2		4. Copper . . 85 Tin . . 5 Zinc . . 10		5. Copper . . 83 Tin . . 2 Zinc . . 15		6. Copper . 92·5 Tin . 5 Zinc . . 2·5	
	Tensile.	Ductility.	Tensile.	Ductility.	Tensile.	Ductility.	Tensile.	Ductility.	Tensile.	Ductility.	Tensile.	Ductility.
° Fahr.	lb.	Per cwt.	lb.	Per cwt.	lb.	Per cwt.	lb.	Per cwt.	lb.	Per cwt.	lb.	Per cwt.
Atmospheric	535	12·5	575	8·75	525	16	525	21	485	26	560	20
100	505	10	—	—	525	15·5	550	18	480	26	—	—
150	525	11	—	—	525	14	530	19·5	450	25·5	—	—
200	485	10	535	8·75	460	9	523	19	460	26·25	440	11
250	505	10	385	5	255	3	515	16	440	26	360	6
300	500	10	295	·66	265	*nil.*	531	18·25	435	23	255	·66
350	450	8·25	295	*nil.*	—	—	495	17	435	25	—	—
400	245	·75	—	—	260	*nil.*	260	2	435	25	—	—
450	265	*nil.*	—	—	—	—	250	2	152	1·2	—	—
500	250	*nil.*	—	—	275	*nil.*	230	2	152	*nil.*	—	—

Temperature.	Phosphor-bronze Rods 1-in. diameter. Copper . . 92·5 Tin . . 7 Phos. . . ·5		Copper Rods ·72-inch diam.		Cast-iron Rods 1-in. diam. Welsh . . ·25 Scotch . ·25 Scrap . ·5		Wrought-iron. Re-manufactured. Rods ·74-in. diam.		Wrought-iron. Yorkshire-rods ·70-in. diam.		Mild Steel-strips, ·74-in. by ·49-inch.	
	Tensile.	Ductility.	Tensile	Ductility.	Tensile.	Ductility.	Tensile.	Ductility.	Tensile.	Ductility.	Tensile.	Ductility.
° Fahr.	lb.	Per cwt.	lb.	Per cwt.	lb.	Per cwt.	lb.	Per cwt.	lb.	Per cwt.	lb.	Per cwt.
Atmospheric	609	17·5	460	2·5	480	*nil.*	500	22	460	25	555	26
100	614	17	465	2·5	423	,,	530	18·75	460	24·25	—	—
150	610	18	445	4	441	,,	530	15	—	—	—	—
200	605	18	440	5	440	,,	530	15	465	17·25	535	22·5
250	580	15	430	7	470	,,	540	15	—	—	—	—
300	575	12	430	6	490	,,	650	15·5	480	7·5	555	11·25
350	470	7	430	6	470	,,	660	12·5	—	—	—	—
400	424	5	420	6	401	,,	640	12	465	15	490	10·25
450	380	4	415	6	400	,,	680	15	—	—	—	—
500	420	5	390	6	400	,,	680	20	550	13·75	600	10

Wrought-iron increased in strength up to 500° Fahr., but lost slightly in ductility up to 300° Fahr., where it began to increase, and continued up to 500°. The strength of the steel was not affected by temperature up to 500° Fahr., but its ductility was reduced more than one half.

General Effect of Temperature on Metals.—There is considerable variableness in the temperatures at which the strength of metals becomes seriously impaired, even when of the same quality and strength. The effect of temperature may in a general way be averaged as follows:—

Description of metal.	Temperature at which the metal changes, and either suddenly weakens or becomes brittle.	Dangerous temperature, or that at which there is a great reduction in the tenacity of the metal
	°	°
Brass, cast	250 (Fahr.)	350 (Fahr.)
Gun-metal	300	400
Copper	450	800
Mild-steel	560	750
Wrought-iron	600	820
Cast-iron	630	860

A great change takes place in all descriptions of iron and mild-steel at a temperature of about 630° Fahr., and they either become weak or so brittle as to be easily broken by a sudden stress or blow, without bending. All descriptions of gun-metal become materially changed, and more or less rotten or brittle, at a temperature of about 400° Fahr.

Joints or Seams of the Plates of Steam-Boilers.—Joints or seams of cylindrical shells are formed either by welding or riveting.

Welded Joints.—A welded seam makes the neatest form of a joint. A carefully welded joint is frequently considered to be as sound and strong as the solid plate, but this is seldom the case, and the strength of welds is very variable. The strength of a welded seam, however carefully and excellently made, is generally about 5 per cent. less than that of the solid plate. It is frequently only 50 per cent., and probably averages, in a general way, 70 per cent. of the strength of the plate; but a good weld should sustain 80 per cent. of the strength of the solid bar.

The Rivet-holes of Riveted-Joints are made either by punching or drilling. The tenacity of a plate is reduced by punching, but is not affected by drilling, unless the work is badly done and the drill tears instead of cuts the metal.

Punches and Dies, when employed for forming the rivet-holes of the joints of plates, have very little variety. A punch with a plain end or face is shown in Fig. 85. Another form of punch with a centre-point on the face is shown in Fig. 86. The centre-point enables the centres of the holes marked on the plate to be found readily, and ensures greater accuracy of work than is obtainable with a punch having a plain face.

Punches are made slightly larger in diameter at the face than higher up the shank, and the face is generally formed slightly concave to make a clean cut and a smooth hole.

A punch with a helical face is shown in Fig. 87. The spiral cutting-action of this punch is supposed to cause less injury to the metal round the hole than a flat-faced punch.

Fig. 85.
Punch with plain face.

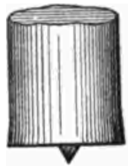


Fig. 86.
Punch with centre-point on the face.

Fig. 87.
Punch with spiral face.

Punched Rivet-Holes.—Punching is a rough and very unsatisfactory method of making rivet-holes in boiler-plates. The tendency of the action of a punch in passing through a metal-plate of a fibrous or seamy nature, is to open the seams or develop laminations around the circumference of the hole formed by it, owing to the metal being thrust off or detruded by the punch, which does not cut the metal. The more fibrous the metal the greater the injury resulting from punching.

Plates of a hard and unyielding nature are liable to have cracks developed round the holes by punching. Plates of a soft and yielding nature suffer the least injury from punching.

The effect of punching the holes for the rivets in a boiler-plate, is to distress and considerably reduce the elasticity of that portion of the metal immediately surrounding the rivet-hole, and diminish the tenacity of the plate. Hence, the finer the pitch or closer the rivet-hole, the greater the loss of strength of a punched plate. The liability to injury from punching increases as the distance of the rivet-hole from the edge of the plate decreases. Plates of a hard and brittle nature suffer most injury from punching.

The reduction of the strength of plates due to punching varies considerably, as it is affected by the form and condition of the punch and die. A bad punch and die may cause great injury to the metal, and develop considerable weakness in a line of rivet-holes. The injury to plates of a soft and ductile nature, such as best Yorkshire iron, by the detrusion of the punch, is generally confined to an annulus round the punched hole of a breadth equal to from one-fourth to one-third the thickness of the plate.

The distressing effect on the metal due to punching, may generally be neutralised by removing an annulus one-eighth inch in width round the rivet-hole with a rhymer.

Resistance of Plates to Punching.—In punching a hole through a plate, the metal outside or round the acting surface or circumference of the punch is broken off or detruded in a similar manner to shearing; a punch may, therefore, be regarded as a shear-blade equal in length to the circumference of the punch. The resistance to punching depends upon the extent of the surface acted upon by the punch, and the nature of the material to be punched. The resistance increases directly as the strength of the plate and the diameter of the punch, and it is considerably affected by the form and condition of the detruding edge of the punch, and by the amount of clearance between the punch and the die.

The stress required to punch wrought-iron was found in some experiments to be as follows: To punch a hole of $\frac{3}{4}$ inch diameter through a thickness of $\frac{5}{8}$ inch, required a pressure of $23\frac{1}{2}$ tons per square inch of detruded sectional area; to punch a hole of 1 inch diameter through a thickness of $\frac{1}{2}$ inch, required a pressure of 22 tons per square inch; and to punch a hole of 1 inch diameter through a thickness of 1 inch, required a pressure of $24\frac{1}{2}$ tons per square inch of detruded sectional area.

It appears from these and other experiments that, the resistance of a plate to punching is practically the same as its resistance to tearing by a tensile strain. The resistance opposed to the punch is the area of metal detruded by it, and the force required to punch a hole through a plate of metal may be found by the following formula: Diameter of hole $\times$ 3·1416 $\times$ thickness of plate $\times$ tensile strength of plate. For instance, to punch a hole of $\frac{3}{4}$ inch diameter through a steel-plate $\frac{3}{8}$ inch thick having a tensile strength of 30 tons per square inch, requires a pressure of ·75 inch diameter $\times$ 3·1416 $\times$ ·375 inch thick $\times$ 30 tons tensile strength = 26·5 tons per square inch.

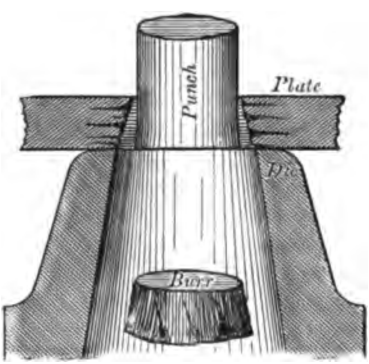


Fig. 88.—Punch and die with excessive clearance.

The compressive strength of a punch of fine quality hardened steel is 100 tons per square inch, or nearly four times as great as the tensile strength of ordinary boiler-plates.

Effect of the Form of a Punch on a Punched-plate.—The loss of strength due to punching the rivet-holes in a plate is considerably influenced by the form and condition of the detruding edge of the punch, and the proportion between the diameter of the punch and that of the hole in the die-block. The loss of strength is greatest when the punch fits the die exactly without clearance, and also when the punch is considerably less than the die, or has excessive clearance. The effect of excessive clearance is shown in Fig. 88.

The extent to which the diameter of the die should exceed that of the punch, depends upon the thickness and nature of the metal to be punched. The diameter of the die may be = diameter of punch $\times$ 1·1 to 1·2, according to the character of the plate to be punched. For rivets $\frac{3}{4}$ inch diameter, for plate not exceeding $\frac{1}{2}$ inch thick, the diameter of the die may be $\frac{51}{64}$ inch. For rivets $\frac{13}{16}$ inch diameter, in boiler-plates not exceeding $\frac{1}{2}$ inch in thickness, the diameter of the die may be $\frac{7}{8}$ inch.

The effect of clearance between the punch and the die is to produce a conical hole in the plate as shown in Fig. 89.

Fig. 89.—Punched hole.

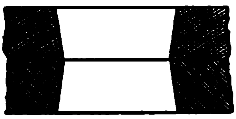


Fig. 90.—Arrangement of plates with punched holes.

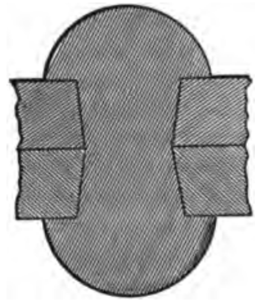


Fig. 91.—Section of riveted plates.

Punched plates are generally arranged with the large ends of the holes outside, as shown in Fig. 90.

The holes become completely filled with the rivets, as shown in Fig. 91, when the riveting is efficiently performed. By this arrangement the tension due to the contraction of the rivet in cooling is distributed through the whole length of the hole, and is not concentrated at the ends, whereby the stress on the rivet-head is considerably relieved. The countersunk outline of the rivet tends to hold the plates together after a rivet-head has been broken off.

Loss of Strength due to Punching Rivet-holes in Boiler-plates.—Many experiments have been made to ascertain the loss of tenacity of a plate by punching the rivet-holes. In some experiments with plates of Yorkshire iron, Mr. Kirkaldy found that by punching the rivet-holes :—

					Lengthways per cent.	Crossways per cent.
Lowmoor-iron plates	·37	inch thick,	lost in	tenacity	—	21·2
,, ,, ,,	·38	,,	,,	,,	17·7	—
,, ,, ,,	·63	,,	,,	,,	—	17·8
Bowling-iron plates	·53	,,	,,	,,	—	13·9
,, ,, ,,	·54	,,	,,	,,	8·6	—
Farnley-iron plates	·40	,,	,,	,,	16·8	—
,, ,, ,,	·41	,,	,,	,,	—	19·8
Taylor's-iron plates	·52	,,	,,	,,	7·7	—
,, ,, ,,	·53	,,	,,	,,	—	12·8
Monkbridge-iron plates	·50	,,	,,	,,	—	16.7
,, ,, ,,	·51	,,	,,	,,	6·7	—

The mean of these experiments is 11·5 per cent. loss of tenacity along the grain of the metal, and 17 per cent. across the grain.

Mr. William Parker of "Lloyd's" found in some experiments that:—

Steel-plates $\frac{1}{4}$ inch thick, lost in tenacity by punching					8 per cent.
Steel-plates $\frac{3}{8}$	,,	,,	,,	,,	18 ,,
Steel-plates $\frac{1}{2}$	,,	,,	,,	,,	26 ,,
Steel-plates $\frac{3}{4}$	,,	,,	,,	,,	33 ,,
Iron-plates $\frac{3}{4}$	,,	,,	,,	,,	18 to 23 per cent.

The effect of increasing the size of the hole in the die of the punch was as follows:—

							Per cent.
Taper of hole in steel-plate $\frac{1}{16}$ inch; loss of tenacity due to punching =							17·8
,,	,,	,,	$\frac{1}{4}$,,	,,	,,	,,	12·3
,,	,,	,,	$\frac{1}{8}$,,	,,	,,	,,	24·5

The plates were from ·675 inch to ·712 inch in thickness. The effect of punching was found not only to reduce the tenacity but also to diminish the elongation before fracture, and to cause the fracture to become crystalline instead of silky. When punched-holes $\frac{7}{8}$ inch diameter were rhymered out to $1\frac{1}{8}$ diameter, the loss of tenacity disappeared and the plates carried as high a stress as drilled plates. Annealing also restores to punched plates the original tenacity.

Punching is not employed for best work. In high-class boilers all the rivet-holes are drilled in place by special tools, and the plates are afterwards separated, and the burrs left by the drill are taken off.

Rivets of Riveted-joints of Steam-boilers.—Rivets are made from bars of tough and ductile quality, cut into pieces of the required length. Each piece required to form a rivet is pressed in dies, which knock one end of it up to form the tail of the rivet. The end of the rivet-shank is taper for half its length, so that it may enter the hole easily. The tails of rivets are generally made either of pan or flat-shape for hand-riveting, and cup-shaped for machine-riveting. The proportion of rivets vary, as there are no standard sizes. The following are good average proportions:—

Tails of Rivets.—A pan-tailed rivet is shown in Fig. 92. The largest diameter of the tail, A, is = diameter of rivet × 1·5; the smallest diameter, B, is = diameter of rivet × 1·3. The depth or thickness of the tail, C, is = diameter of rivet × ·67. The diameter of the end of the rivet, D, is = diameter of rivet × ·93.

A flat-tailed rivet is shown in Fig. 93. The diameter of the tail is = diameter of rivet × 1·5, and the thickness of the tail and diameter of the point is the same as that in Fig. 92.

A snap-tailed, or cup-tailed, rivet is shown in Fig. 94. The diameter of the tail, A, is = diameter of rivet × 1·6, and the depth of the tail, B, is = diameter of rivet × ·65.

A snap-tailed rivet with conical neck to fit a punched hole, is shown in Fig. 95. The diameter of the tail is = diameter of rivet × 1·7, and the thickness of the tail is = diameter of rivet × ·66. The diameter at the top of the cone, A, is = diameter of rivet × 1·125. Depth of cone = one-half the diameter of the rivet.

Fig. 92. Pan-tailed rivet.

Fig. 93. Flat-tailed rivet.

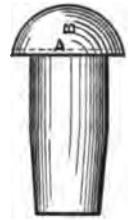


Fig. 94. Snap-tailed rivet.

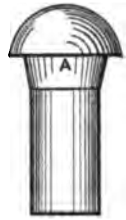


Fig. 95. Snap-tailed rivet with conical neck.

Fig. 96. Countersink-tailed rivet.

A countersunk-tailed rivet is shown in Fig. 96. The diameter of the top of the tail is = diameter of rivet × 1·5.

Heads of Rivets.—The length of shank projecting through the plates to form the head, as shown in Fig. 97, should be equal to the diameter of the rivet for countersunk rivets, and to about $1\frac{1}{4}$ the diameter of rivet for snap and conical heads in hand-riveting, and about $\frac{1}{4}$ inch longer for machine-riveting.

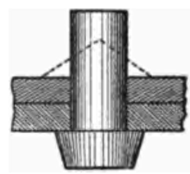


Fig. 97.—Rivet.

A conical-headed rivet is shown in Fig. 98. The pointed head is formed by knocking down the point of the rivet with light hammers. The diameter of the head is = diameter of rivet × 2, and the thickness or height of the head is = diameter of rivet × ·75.

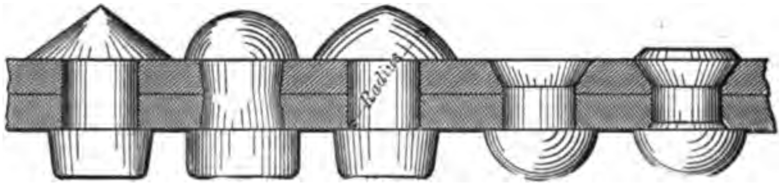


Fig. 98. Fig. 99. Fig. 100. Fig. 101. Fig. 102.

Figs. 98–102.—Various forms of rivet-heads.

The snap-headed, or cup-headed, rivet, shown in Fig. 99, is formed by hammering down the point and finishing it with a cup-shaped die applied by a hammer. The diameter of the head is = diameter of rivet × 1·6, and the thickness of the head is = diameter of rivet × ·63.

A conoidal-headed rivet is shown in Fig. 100. The diameter of the head is = diameter of the rivet × 2, and the height of the head is = diameter of rivet × ·75. The radius of the head is = diameter of rivet × 1·8.

A countersunk-headed rivet, as shown in Fig. 101, is formed by hammering the point into a conical rivet hole. The diameter of the top of the head is

= diameter of rivet × 1·5, and the thickness of the head is = diameter of rivet × ·5 to ·7.

Another countersunk-head rivet is shown in Fig. 102, the head of which projects a little above the plate to increase the strength of the head.

An elliptical countersunk-rivet is shown in Fig. 103. The height of the head above the plate is = diameter of rivet × ·25. The diameter of the head is = diameter of rivet × 1·5.

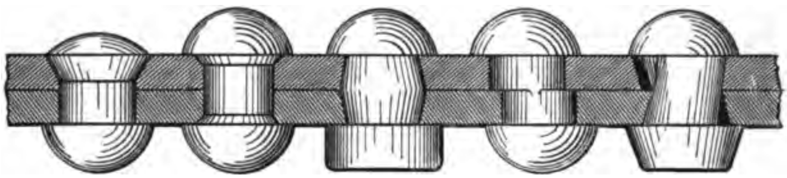


Fig. 103. Fig. 104. Fig. 105. Fig. 106. Fig. 107.

Figs. 103–104.—Different forms of rivets. Figs. 105–107.—Examples of defective rivets.

A rivet with a slight countersink under both the head and tail is shown in Fig. 104. By rounding or slightly countersinking the edges of rivet-holes in this manner, the strength of rivet is increased and the cutting-action of the edge of the hole is diminished.

Defective Rivets.—A rivet assumes a barrel-shape when riveted in plates arranged with the small ends of punched holes outside, as shown in Fig. 105.

When a rivet is riveted to completely fill a partly blind hole, due to inaccurate punching, it assumes the form shown in Fig. 106.

A rivet when inserted in a partly blind hole and hand-riveted, frequently assumes the shape shown in Fig. 107.

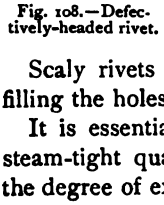


Fig. 108.—Defectively-headed rivet.

A rivet very defectively headed, or formed eccentrically, is shown in Fig. 108.

It is important that the end of the shank should be perfectly square with the shank, and not bevel.

Rivets are frequently injured by being worked when too highly heated. Rivets should be quickly heated in a clean fire, and not allowed to remain a long time in the fire; and they should be quite free from dirt and scales when inserted in the holes.

Scaly rivets are liable to cause leakage, as the scale prevents the rivets filling the holes.

It is essential to good work that the rivets fill the holes, because the steam-tight quality of a well-proportioned joint depends principally upon the degree of excellence of the fit of the rivets in their holes, and also upon the efficiency of the riveting.

When the heads and tails of the rivets are deficient in thickness, they are liable to curl up round the edges and cause leakage.

Tests for Rivets.—Rivets should admit of being bent cold, round a bar equal in diameter to that of the rivet, as shown in Fig. 109. They should bend hot without fracture in the manner shown in Fig. 110. The tail of the rivet should admit of being flattened when hot, without cracking at the edges, until its diameter is equal to $2\frac{1}{2}$ times the diameter of the rivet, as shown in Fig. 111. The tail should admit of being hammered when cold down to $\frac{1}{8}$ inch in thickness, without cracking at the edges. The shank of the rivet should admit of being flattened and punched with a hole equal in diameter to that of the shank, as shown in Fig. 112, without cracking in the vicinity of the hole.

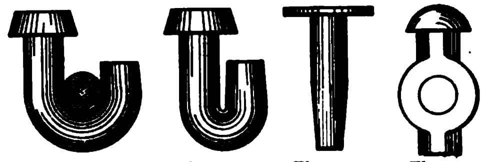


Fig. 109. Fig. 110. Fig. 111. Fig. 112.

Fig. 109–112.—Tests for rivets.

Tensile Strength of Rivets.—The tensile strength of a rivet is frequently considerably reduced in the processes of heating, and hammering down while hot to fill the rivet-hole and form the rivet-head. The tensile strength of the bars from which iron-rivets are made is from 24 to 25 tons per square inch, but the tensile strength of a rivet when riveted in the joint of wrought-iron plates, is frequently only about 22 tons per square inch, showing a loss of $22 \div 25 = \cdot 88$, or 12 per cent., due to forming the rivet-head while hot. The tensile strength of the bars from which steel-rivets are made is from 28 to 30 tons per square inch, but the tensile strength of a rivet when riveted in the joint of steel-plates is frequently only about 23 tons per square inch, showing a loss of $23 \div 30 = \cdot 76$, or 24 per cent., due to riveting hot.

By heating the rivets carefully, and working them at the lowest practicable temperature, they should not suffer diminution of strength, but on the contrary the tensile strength might probably be increased about 10 per cent. by the extra work on the metal due to riveting.

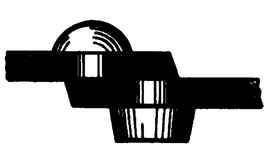


Fig. 113.—Rivet in single-shear.

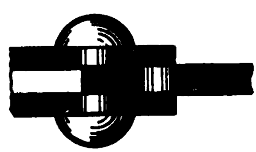


Fig. 114.—Rivet in double-shear.

Rivets in Single and Double-Shear.—When two plates are connected by a rivet, a tensile strain applied to the plates tends to cut the rivet in halves, or to shear it in a single plane, as shown in Fig. 113, and the rivet is in single-shear. The strain required to shear a rivet is proportional to the area sheared, and is independent of the shape of the rivet. When a plate is fixed centrally between two plates by a rivet, a tensile strain applied to the plate tends to cut the rivet into three pieces, or to shear it in two planes, as shown in Fig. 114, and as a double area is exposed to shearing, the rivet is in double-shear.

Shearing Strength of Rivets.—The shearing strength of rivets is generally less than the tensile strength of the plates forming the joint. The shearing strength of iron-rivets varies with the character of the rivet-holes. It was found in some experiments to be as follows:—

			Per square inch.
Shearing strength of iron-rivet		in a counter-sunk drilled hole .	$18\frac{1}{2}$ tons
do.	do.	in ordinary drilled rivet-hole . .	19 „
do.	do.	in an ordinary punched hole .	20 „

Hence, the shearing strength of an iron-rivet is =
(diameter of rivet)^2 × ·7854 × shearing strength, according to the kind of rivet-hole.

Numerous other experiments have been made on the shearing resistance of rivets, the results of a few of which are given in the following Table:—

TABLE 49.—RESULTS OF TESTS OF THE SHEARING-RESISTANCE OF IRON AND STEEL RIVETS OF THE AVERAGE QUALITY USED IN BOILER-WORK, WHEN RIVETED IN DRILLED HOLES.

Shearing-Resistance of Iron-Rivets.		Shearing-Resistance of Mild-Steel Rivets.	
Tons per Sq. Inch.	Lbs. per Sq. Inch.	Tons per Sq. Inch.	Lbs. per Sq. Inch.
17·03	38147	20·91	46839
17·54	39289	21·55	48272
18·59	41642	22·33	50020
20·39	45674	23·03	51588
20·90	46816	23·37	52349
21·37	47868	23·91	53559
22·11	49527	24·12	54029
22·70	50848	24·70	55328

It may be assumed, in a general way, that the shearing-resistance of a rivet is equal to the tensile strength of the plates × ·8. Taking the tensile strength of good wrought-iron boiler-plates at 21 tons per square inch, and that of good mild-steel boiler-plates at 28 tons per square inch. The shearing-strength of wrought-iron rivets in the joints of wrought-iron plates is on this basis = 21 × ·8 = 16·8 tons per square inch, and that of mild-steel rivets in the joints of mild-steel plates = 28 × ·8 = 22·4 tons per square inch: or say, in round numbers, shearing-strength of wrought-iron rivets = 17 tons, or 38080 lbs. per square inch; shearing-strength of mild-steel rivets 23 tons, or 51520 lbs. per square inch.

The resistance per square inch to shearing, is practically the same for rivets either in single or double-shear.

The following *Rules* are also used in practice for determining the shearing-strength of rivets. In which S = the shearing-resistance in tons per square inch, and d = the diameter of the rivet in inches.

Shearing-strength of rivets in ordinary punched holes:—

$$S = \frac{148}{9} \times d^2 \text{ for iron rivets and plates.}$$

$$S = \frac{198}{9} \times d^2 \text{ for steel rivets and plates.}$$

Shearing-strength of rivets in ordinary drilled holes:

$$S = \frac{140}{9} \times d^2 \text{ for iron rivets and plates.}$$

$$S = \frac{188}{9} \times d^2 \text{ for steel rivets and plates.}$$

Shearing-strength of rivets in drilled counter-sunk holes:

$$S = \frac{136}{9} \times d^2 \text{ for iron rivets and plates.}$$

$$S = \frac{183}{9} \times d^2 \text{ for steel rivets and plates.}$$

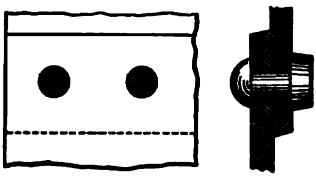
Gripping-Power of Rivets.—When two plates are fastened together by properly proportioned and well-closed rivets, the frictional adhesion of the plates depends upon the longitudinal tension of the rivets. The adhesion of the plates, or their resistance to sliding, per square inch of sectional area of the rivets, is, in a general way, equal to two-ninths of the ultimate tensile strength of the rivet. For instance, the resistance to sliding of plates secured with rivets having an ultimate tensile strength of 28 tons per square inch, is approximately = 28 tons × $\frac{2}{9}$ = 6·22 tons per square inch of the sectional area of the rivets.

Riveted Joints of Steam-Boilers.—The strength of a riveted joint or seam is generally considerably less than that of the solid plate, it is, therefore, essential to adopt that form of joint which will bring the strength of the seams up as nearly as practicable to that of the plate of which the joint is formed, in order to obtain the greatest possible uniformity of strength in the construction of steam-boilers with riveted joints.

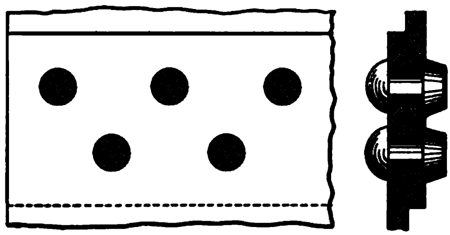
It is necessary to make the rivets of the same material as the plates, to prevent corrosive wasting from galvanic action. That is, iron-rivets should be used for iron-plates, steel-rivets for steel-plates, and copper-rivets for copper plates, thus placing metals in contact which are similar in nature and electric affinities.

Forms of Riveted Joints.—Lap-joints are formed by overlapping one plate on the other, and riveting them together with one or more rows of rivets. Butt-joints are formed by butting the ends of the plates together, and covering the joint with a covering-plate, or butt-strap, on one side, or both sides, of the plate, riveted together by one or more rows of rivets at each end of the plate.

A single-riveted lap-joint is shown in Fig. 115, and in sectional end view in Fig. 116.

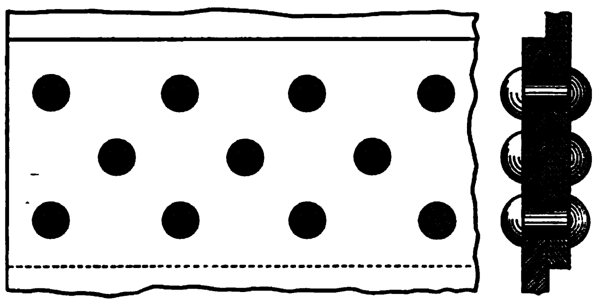


Figs. 115 & 116.—Single-riveted lap-joint.



Figs. 117 & 118.—Double-riveted lap-joint.

A double-riveted lap-joint is shown in Fig. 117, and in sectional end view in Fig. 118.



Figs. 119 & 120.—Treble-riveted lap-joint.

A treble-riveted lap-joint is shown in Fig. 119, and in sectional end view in Fig. 120.

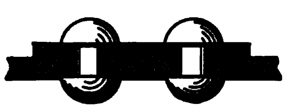


Fig. 121.—Butt-joint with single butt-strap

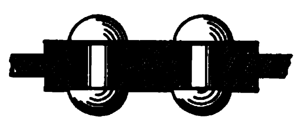


Fig. 122.—Butt-joint with double butt-strap.

A butt-joint with a single butt-strap is shown in section in Fig. 121.

A butt-joint with a double butt-strap is shown in section in Fig. 122.

Arrangement of the Rivets of Riveted Joints.—The rivets are either arranged in rows opposite to each other, as in chain-riveting, or placed so that the rivets in one row divide the spaces between the rivets in the next row, as in zigzag-riveting. In ordinary joints the rows of rivets are evenly pitched.

A double-riveted lap-joint, chain-riveted, is shown in Fig. 123, and a double-riveted lap-joint, zigzag-riveted, is shown in Fig. 124.

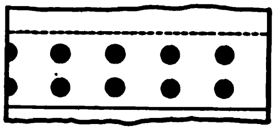


Fig. 123.—Double-riveted lap-joint chain-riveted.

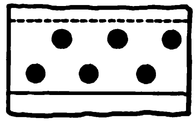


Fig. 124.—Double-riveted lap-joint zigzag-riveted.

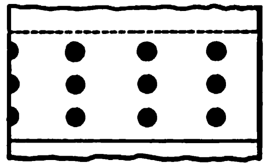


Fig. 125.—Treble-riveted lap-joint chain-riveted.

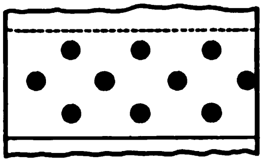


Fig. 126.—Treble-riveted lap-joint zigzag-riveted.

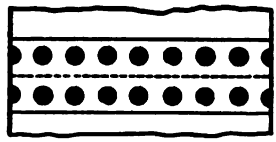


Fig. 127.—Single-riveted butt-joint.

A treble-riveted lap-joint, chain-riveted, is shown in Fig. 125, and a treble-riveted lap-joint, zigzag-riveted, is shown in Fig. 126.

A single-riveted butt-joint, that is, with one row of rivets on each side of the joint, is shown in Fig. 127.

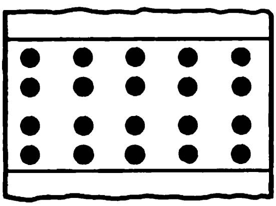


Fig. 128.—Double-riveted butt-joint chain-riveted.

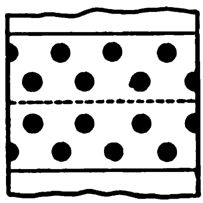


Fig. 129.—Double-riveted butt-joint zigzag riveted.

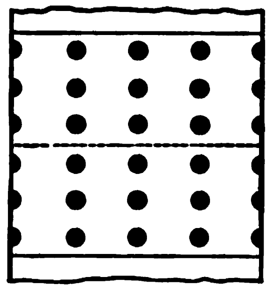


Fig. 130.—Treble-riveted butt-joint chain-riveted.

A double-riveted butt-joint, chain-riveted, is shown in Fig. 128, and a double-riveted butt-joint, zigzag-riveted, is shown in Fig. 129.

A treble-riveted butt-joint, chain-riveted, is shown in Fig. 130, and a treble-riveted butt-joint, zigzag-riveted, is shown in Fig. 131.

In all these examples the rivets are evenly pitched, or spaced alike along each line of the rivets.

Riveted Joints with rows of Unevenly-Pitched Rivets.—Chain-riveted and zigzag-riveted joints, are frequently arranged with every alternate rivet omitted in the outer row and in the outer and inner rows, and they are also arranged with more rivets in the inner than the outer row.

A treble-riveted lap-joint, in which every alternate rivet is omitted in the outer and inner rows of rivets, is shown in Fig. 132, and a slightly different arrangement of the same form of joint is shown in Fig. 133.

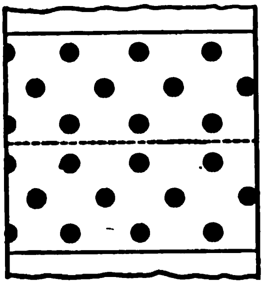


Fig. 131.—Treble-riveted butt-joint zigzag riveted.

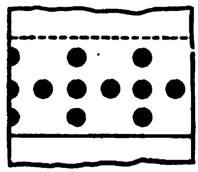


Fig. 132.—Treble-riveted lap-joint.

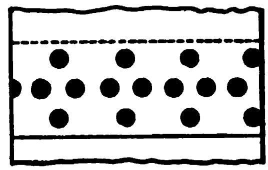


Fig. 133.—Treble-riveted lap-joint.

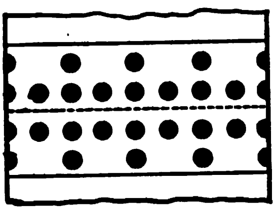


Fig. 134.—Double-riveted butt-joint.

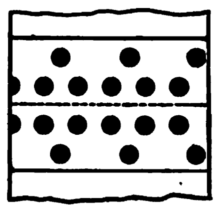


Fig. 135.—Double-riveted butt-joint.

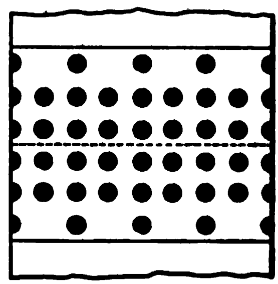


Fig. 136.—Treble-riveted butt-joint.

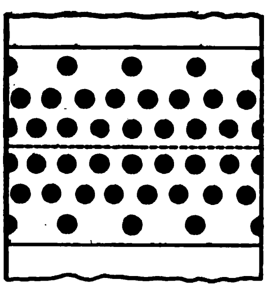


Fig. 137.—Treble-riveted butt-joint.

A double-riveted butt-joint, in which every alternate rivet is omitted in the outer rows, is shown in Fig. 134, and the same form of joint with the outer row of rivets differently spaced is shown in Fig. 135.

A treble-riveted butt-joint, in which every alternate rivet is omitted in the outer row, is shown in Fig. 136, and the same form of joint, with the outer row of rivets differently spaced, is shown in Fig. 137.

Pitch of Rivets.—The maximum strength of a riveted joint is obtained when the percentage of the strength of the seam to that of the solid plate, is equal to the percentage of the strength of the rivets to that of the solid plate.

When the shearing strength of the rivets is equal to the tensile strength of the plates, the maximum strength of a riveted joint, taking all the dimensions in inches, is expressed as follows:—

$$\frac{\text{Pitch of rivets} - (\text{diameter of rivet-hole})}{\text{Pitch of rivets}} =$$

$$\frac{(\text{Diameter of rivet-hole})^2 \times \cdot 7854 \times \text{number of rows of rivets}}{\text{Pitch} \times \text{thickness of plate}}.$$

Pitch of rivets $-$ (diameter of rivet-hole) $=$

$$(\text{diameter of rivet-hole})^2 \times \cdot 7854 \times \frac{\text{number of rows of rivets}}{\text{thickness of plate}}.$$

Pitch of rivets $=$

$$(\text{diameter of rivet-hole})^2 \times \cdot 7854 \times \frac{\text{number of rows of rivets}}{\text{thickness of plates}} +$$

diameter of rivet-hole.

Where there are two rows of rivets, then $\cdot 7854 \times 2 = 1 \cdot 5708$, and for a double-riveted joint, measuring along one line of rivets:

Pitch of rivets $=$

$$1 \cdot 57 \times \frac{(\text{diameter of rivet-hole})^2}{\text{thickness of plate}} + \text{diameter of rivet-hole.}$$

Where there are three rows of rivets then, $\cdot 7854 \times 3 = 2 \cdot 3562$, and for a treble-riveted joint, measuring along one line of rivets,

Pitch of rivets $=$

$$2 \cdot 3562 \times \frac{(\text{diameter of rivet-hole})^2}{\text{thickness of plate}} + \text{diameter of rivet-hole.}$$

These rules apply to all thicknesses of plates and diameters of rivet-holes.

Diagonal Pitch of the Rivets of Riveted Joints.—The diagonal pitch of a double-riveted joint, or the oblique distance between the centre of a rivet in one row to the centre of a rivet in the next row, is generally equal to the straight pitch in inches $\times \cdot 75$ to $\cdot 8$.

For instance, the diagonal pitch of a double-riveted seam of $3\frac{1}{8}$ inches straight pitch is $= 3 \cdot 125 \times \cdot 8 = 2\frac{1}{2}$ inches pitch.

The diagonal pitch of the rivets of a seam having several rows of rivets, all of the same pitch, should not be less than that obtained by this *Rule*:—

Diagonal pitch in inches $=$

$$\frac{(\text{Pitch in inches} \times 6) + (\text{diameter of rivet in inches} \times 4)}{10}.$$

For instance, the diagonal pitch of an ordinary treble-riveted sigzag joint, with rivets of 1 inch diameter, and 4 inches pitch, should not be less

than $= \dfrac{(4 \text{ inches} \times 6) + (1 \text{ inch} \times 4)}{10} = 2 \cdot 8$ inches.

Diameter of the Rivet-Holes.—The diameter of the rivet-holes of wrought-iron plates up to $\frac{5}{8}$ inch thickness, is frequently equal to the thickness of the plate in inches $\times 2 + \frac{1}{16}$ inch.

A rule applicable to all thicknesses of plates of wrought-iron is as follows:—

$$\text{Diameter of rivet-hole in inches} = \frac{\sqrt[3]{(\text{thickness of plate in inches} \times 92)}}{8}.$$

For instance, the diameter of the rivet-holes of plates $\frac{5}{8}$ inch thick is $= \cdot 625 \times 92 = 57 \cdot 5$, and $\sqrt[3]{57 \cdot 5} \div 8 = \cdot 985$ inch, or say 1 inch diameter.

The diameter of rivet-hole found by this rule may be taken to the nearest one-sixteenth of an inch.

Distance of Rivet-Holes from the Edge of the Plate.—The space between the edge of the rivet-hole and the edge of the plate should not be less than the diameter of the rivet-hole, but rather greater.

The seams should have abundant breadth, otherwise they are liable to leak, especially when placed at the bottom of a boiler-shell.

Other rules for rivet-holes are given at pages 218—221.

Pitch of Rivets in Punched Plates.—In determining the pitch of rivets for plates with punched rivet-holes, the larger diameter of the punched hole should be taken as the diameter of the rivet-hole, and the less diameter as the diameter of the rivet.

Pitch of Rivets for the Joints of the Plates of Steam-Boilers.—The pitch of rivets necessary to secure equality of strength of the rivets and the plates between the rivet-holes, when the shearing strength of the rivets is equal to the tensile strength of the plates, may be found by the following *Rules*:—

For single-riveted lap-joints and joints with single-riveted single butt-straps.

Pitch of rivets in inches =

$$\frac{\text{Area of rivet-hole in square inches}}{\text{Thickness of plate in inches}} + \text{diameter of rivet-hole in inches.}$$

Example: Required the pitch of the rivets of a steam-boiler having plates $\frac{3}{8}$ inch thick, with lap-joints, and rivet-holes $\frac{13}{16}$ inch diameter?

$$\text{Then } \frac{\cdot 8125 \times \cdot 8125 \text{ inch} \times \cdot 7854}{\cdot 375 \text{ inch thickness of plate}} + \cdot 8125 = 2 \cdot 572 \text{ inches, the pitch}$$

of rivet required for equality of rivet-section and plate-section.

For double-riveted lap-joints and joints with double-riveted single butt-straps, the sectional area of two rivets is to be allowed for, and:—

Pitch of rivets in inches =

$$\frac{\text{Area of rivet-hole in square inches} \times 2}{\text{Thickness of plate in inches}} + \text{diameter of rivet-hole in inches.}$$

This rule also applies to single-riveted butt-joints with double butt-straps, as the rivets are sheared in two places before rupture.

For a double-riveted joint with double butt-straps:—

Pitch of rivets in inches =

$$\frac{\text{Area of rivet-hole in square inches} \times 4}{\text{Thickness of plate in inches}} + \text{diameter of rivet-hole in inches.}$$

For treble-riveted lap-joints three rivets have to be allowed for, and:—

Pitch of rivets in inches =

$$\frac{\text{Area of rivet-hole in square inches} \times 3}{\text{Thickness of plate in inches.}} + \text{diameter of rivet-hole in inches.}$$

The pitch of the rivets may be proportioned to the strength of the plates and rivets, and the number of rows of rivets by the following *Rule*, in which d = the diameter of rivet-hole in inches:—

Pitch of rivets in inches =

$$\frac{\begin{pmatrix}\text{Area of rivet-hole}\\ \text{in square inches}\end{pmatrix} \times \begin{pmatrix}\text{shearing strength}\\ \text{of rivet}\end{pmatrix} \times \begin{pmatrix}\text{number of rows}\\ \text{of rivets}\end{pmatrix}}{\text{Thickness of plate} \times \text{tensile strength of plate}} + d.$$

Example: Required the pitch of the rivets of a steam-boiler having plates $\frac{1}{2}$ inch thick, with lap-joints, rivet-holes $\frac{7}{8}$ inch diameter, and two rows of rivets. Tensile strength of the plates 50,000 pounds per square inch. Shearing strength of rivets 20 per cent. less than the tensile strength of the plates?

$$\text{Then } \frac{(\cdot875 \times \cdot875 \text{ inch} \times \cdot7854) \times (50000 \times \cdot80) \times 2 \text{ rows}}{\cdot5 \text{ inch thickness of plate} \times 50000 \text{ lbs.}} + \cdot875 =$$

2·8 inches, the pitch of rivets required.

As the rivets of a joint are protected from deterioration while the plates are thinned by wear, the shearing strength of iron-rivets is frequently taken as equal to the tensile strength of iron-plates.

Pitch of Rivets required in a Joint to retain a given Percentage of Plate between the Rivet-Holes.—It is seldom convenient in proportioning the joints of the plates of steam-boilers to maintain equality of section of plates and rivets. The rivet-section is generally in excess of the plate-section in order to secure the plates firmly together, and maintain steam-tight joints.

The pitch of rivets required to retain a certain percentage of plate between the rivet-holes may be found by this *Rule*:—

Pitch of rivets =

$$\frac{\begin{pmatrix}\text{Diameter of rivet-}\\ \text{hole in inches}\end{pmatrix} \times \begin{pmatrix}\text{percentage of plate}\\ \text{between rivet-holes}\end{pmatrix}}{100 - \text{percentage of plate between rivet-holes}} + \text{diam. of rivet-hole in inches.}$$

Example: Required the pitch of the rivets of the joint of the plates of a steam-boiler, with rivet-holes $\frac{3}{4}$ inch diameter, to retain 70 per cent. of plate between the rivet-holes?

$$\text{Then } \frac{\cdot75 \text{ inch diameter} \times 70 \text{ per cent.}}{100 - 70} = 1\cdot75 + \cdot75 = 2\tfrac{1}{2} \text{ inches, the}$$

pitch of rivets required.

The percentage of strength of the plate of the riveted joint as compared with the solid plate, may be found by this *Rule*:—

$$\frac{(\text{Pitch in inches} - \text{diameter of rivet-hole}) \times 100}{\text{Pitch of rivets in inches}}.$$

Example: Required the strength of the joints of the plates of a steam-boiler as compared with the solid plate, the pitch of rivets being $2\frac{7}{8}$ inches, and the diameter of the rivet-holes $\frac{7}{8}$ inch.

$$\text{Then } \frac{2{\cdot}875 - {\cdot}875 \times 100}{2{\cdot}875 \text{ inches pitch}} = 73 \text{ per cent. the strength of the joint.}$$

The percentage of strength of the rivets, when in single-shear, as compared with the solid plate may be found by this *Rule*:—

$$\frac{(\text{Area of rivet} \times \text{number of rows of rivets}) \times 100}{\text{Pitch of rivet in inches} \times \text{thickness of plate in inches}}.$$

Rivets in double-shear may be considered to be $1\frac{3}{4}$ times as strong as when in single-shear. Hence, when rivets are exposed to double-shear, multiply the percentage as found by 1·75.

Example: Required the percentage of the strength of the rivets of the joints of the plates of a steam-boiler as compared with the solid plate, of plates $\frac{1}{2}$ inch thick, having double-riveted lap-joints, with rivets $\frac{3}{4}$ inch diameter and $2\frac{1}{2}$ inch pitch.

$$\text{Then } \frac{{\cdot}75 \times {\cdot}75 \text{ inch} \times {\cdot}7854 \times 2 \text{ rows of rivets} \times 100}{2{\cdot}5 \text{ pitch} \times {\cdot}5 \text{ inch thickness of plate}} = 71 \text{ per cent. the}$$

strength of the rivets compared with the solid plate when in single-shear, and $71 \times 1{\cdot}75 = 124$ per cent. when in double-shear.

In calculating the strength of a boiler the smaller of the two percentages of rivet-section and plate-section is to be taken.

Friction of Plates of Riveted Joints.—The assistance given to a riveted joint by the friction of the rough plates is an uncertain element, and cannot be taken into account in estimating the strength of a joint.

Proportions of the Riveted Joints of Steam-Boilers.—The diameter and pitch of the rivet-holes of the seams, or riveted joints, of the plates of a steam-boiler, depend principally upon the thickness of the plates. The proportions of the joints vary considerably in practice. The following are good general rules for determining the proportions of the riveted joints of the plates of steam-boilers, for working-pressures of steam up to 200 lbs. per square inch, when the materials and workmanship are of the best description.

Proportions of Riveted Joints of the Plates of Steam-Boilers when composed of Wrought-Iron Plates and Iron-Rivets, with rows of rivets of equal pitch.

Single-riveted lap-joints, as shown in Figs. 115 and 116, page 212, with drilled rivet-holes.

Diameter of rivet-hole = thickness of plate in inches × 1·25 + ·28125.

Lap of joint = diameter of rivet-hole in inches × 3.

Distance of the centre line of the row of rivet-holes from the edge of the plate = diameter of rivet-hole in inches × 1·5.

Pitch of the rivet-holes in inches = thickness of the plate in inches × 3 + ·75.

Double-riveted lap-joints zigzag riveted, as shown in Figs. 117 and 118, page 212, with drilled rivet-holes.

Diameter of rivet-hole = thickness of plate in inches × 1·25 + ·28125.

Lap of joint = diameter of rivet-hole in inches × 5.

Distance of the centre-line of each row of rivet-holes from the edge of the plate = diameter of rivet-hole in inches × 1·5.

Pitch of the rivet-holes, measured along the centre-line of a row of rivets = thickness of plate in inches × 4·375 + ·75.

Treble-riveted lap-joints, zigzag riveted, as shown in Figs. 119 and 120, page 212, with drilled rivet-holes.

Diameter of rivet-hole = thickness of plate in inches × 1·4375.

Lap of joint = diameter of rivet-hole in inches × 7.

Distance of the centre-line of each outer row of rivet-holes from the edge of the plate = diameter of rivet-hole in inches × 1·5.

Pitch of the rivet-holes measured along the centre-line of a row of rivets = thickness of plate in inches × 5·75.

Single-riveted butt-joints, with single butt-straps as shown in Fig. 121, with drilled rivet-holes, are proportioned by the same rules as single-riveted lap-joints, as each half of the joint represents a lap-joint. The thickness of a single butt-strap should be = thickness of plate in inches × 1·25.

Single-riveted butt-joints with double butt-straps as shown in Fig. 122, page 212, with drilled rivet-holes.

Diameter of rivet-hole = thickness of plate in inches × 1·25 + ·125.

Width of each butt-strap = diameter of rivet-hole × 6. Thickness of each butt-strap = thickness of plate in inches × ·75.

Distance of the centre-line of the row of rivets from the edge of the plate = diameter of rivet-hole in inches × 1·5. Pitch of the rivet-holes = thickness of plate in inches × 3·25 + ·75.

Double-riveted butt-joints with double butt-straps, zig-zag riveted, with drilled rivet-holes. Diameter of rivet-hole = thickness of plate in inches × 1·25 + ·125.

Width of each butt-strap = diameter of rivet-hole × 11. Thickness of each butt-strap = thickness of plate in inches × ·8. Distance of the centre-line of each row of rivets from the edge of the plate = diameter of rivet-hole × 1·5.

Pitch of the rivet-holes measured along the centre-line of each row of rivets = thickness of plate in inches × 4·75 + ·75.

When the pitch is required in eighths of an inch, the nearest eighth of an inch below the dimension obtained by the rule may be taken.

In Illustration of some of these Rules, the proportions of the riveted joints of the plates of the shell of a Cornish-boiler of wrought-iron plates $\frac{3}{8}$ inch thick, with single-riveted circumferential seams, as shown in Fig. 138, and double-riveted longitudinal seams with lap-joints, as shown in Fig. 139, may be computed as follows:—

The diameter of the drilled rivet-holes is $=$ ·375 inch thickness of plate, $\times$ 1·25 $=$ ·46875 $+$ ·28125 $= \frac{3}{4}$ inch. The distance of the centre of the rivet-holes from the edge of the plate is $=$ ·75 inch diameter of rivet-hole $\times$ 1·5 $= 1\frac{1}{8}$ inches.

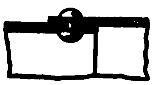


Fig. 138.—Single-riveted circumferential seam.

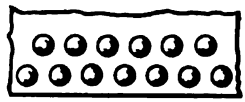


Fig. 139.—Double-riveted longitudinal seam.

The pitch of the rivet-holes of the single-riveted circumferential seams is $=$ ·375 inch thickness of plate $\times$ 3 $=$ 1·125 $+$ ·75 $= 1\frac{7}{8}$ inches. The pitch of the double-riveted longitudinal seams, measured along one line of rivets is $=$ ·375 inch thickness of plate $\times$ 4·375 $+$ ·75 $=$ 1·64 $+$ ·75 $= 2\frac{3}{8}$ inches.

The lap of the single-riveted joint is $=$ ·75 inch diameter of rivet-hole $\times$ 3 $= 2\frac{1}{4}$ inches. The lap of the double-riveted seam is $=$ ·75 inch diameter of rivet-hole $\times$ 5 $= 3\frac{3}{4}$ inches: the centre-line of each row of rivets being ·75 inch diameter of rivet-hole $\times$ 1·5 $= 1\frac{1}{8}$ inch from the edge of the plate.

Proportions of Riveted joints of the plates of Steam-boilers when composed of mild-steel plates and steel-rivets, with rows of rivets of equal pitch.

Single-riveted lap-joints with drilled rivet-holes.

Diameter of rivet-hole $=$ thickness of plate in inches $\times$ 1·25 $+$ ·1875.

Lap of joint $=$ diameter of rivet-hole in inches $\times$ 3.

Distance of the centre-line of the row of rivets from the edge of the plate $=$ diameter of rivet-hole in inches $\times$ 1·5.

Pitch of the rivet-holes in inches $=$ thickness of plate in inches $\times$ 2·5 $+$ ·75.

Double-riveted lap-joints, zigzag riveted, with drilled rivet-holes.

Diameter of rivet-hole $=$ thickness of plate in inches $\times$ 1·25 $+$ ·1875.

Lap of joint $=$ diameter of rivet-hole in inches $\times$ 5.

Distance of the centre-line of each row of rivet-holes from the edge of the plate $=$ Diameter of rivet-hole in inches $\times$ 1·5.

Pitch of the rivet-holes, measured along the centre-line of a row of rivets $=$ thickness of plate in inches $\times$ 3·75 $+$ ·75.

Treble-riveted lap-joint, zigzag riveted with drilled rivet-holes.

Diameter of rivet-hole $=$ thickness of plate in inches $\times$ 1·375.

Lap of joint $=$ diameter of rivet-hole $\times$ 7.

Distance of the centre-line of each outside row of rivet-holes from the edge of the plate $=$ diameter of rivet-hole in inches $\times$ 1·5.

Pitch of the rivet-holes, measured along the centre-line of a row of rivets = thickness of plate in inches × 5·5.

Single-riveted butt-joints, with double butt-straps, and drilled rivet-holes.

Diameter of rivet-hole = thickness of plate in inches × 1·25 + ·0625.

Width of each butt-strap = diameter of rivet-hole × 6.

Thickness each butt-strap = thickness of plate in inches × ·75.

Distance of the centre-line of the row of rivets from the edge of the plate = diameter of rivet-hole in inches × 1·5.

Pitch of the rivet-holes = thickness of plate in inches × 3 + ·75.

Double-riveted butt-joints with double butt-straps zigzag riveted, with drilled rivet-holes.

Diameter of rivet-hole = thickness of plate in inches × 1·25 + ·0625.

Width of each butt-strap = diameter of rivet-hole × 11.

Thickness of each butt-strap = thickness of plate in inches × ·8.

Distance of the centre-line of each row of rivets from the edge of the plate = diameter of rivet-hole in inches × 1·5.

Pitch of the rivet-holes, measured along the centre-line of each row of rivets = thickness of plate in inches × 4·375 + ·75.

In Illustration of some of these Rules, the proportions of the riveted joints of the plates of the shell of a Lancashire boiler of mild-steel plates $\frac{1}{2}$ inch thick, with the circumferential seams single-riveted, and the longitudinal seams double-riveted, with lap-joints as shown in Fig. 140, may be computed as follows:—

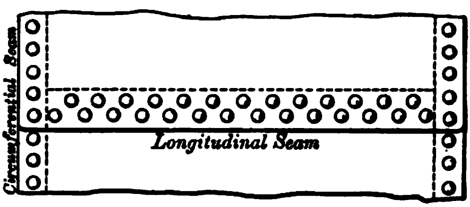


Fig. 140.—Riveted seams of a boiler-shell.

The diameter of the drilled rivet-holes is = ·5 inch thickness of plate × 1·25 + ·1875 = ·8125, or $\frac{13}{16}$ inch.

The distance of the centre of the rivet-holes from the edge of the plate is = ·8125 inch diameter of rivet-hole × 1·5 = 1·21875 = $1\frac{7}{32}$ inches.

The pitch of the rivet-holes of the single-riveted circumferential seams is = ·5 inch thickness of plate × 2·5 + ·75 = 2 inches. The pitch of the rivet-holes of the double-riveted longitudinal seams along one line of rivets is = ·5 inch thickness of plate × 3·75 = 1·875 + ·75 = 2·625 inches, or $2\frac{5}{8}$ inches.

The width of the lap of the single-riveted joint is = ·8125 inch diameter of rivet-hole × 3 = $2\frac{7}{16}$ inches. The width of the lap of the double-riveted joint is = ·8125 inch diameter of rivet-hole × 5 = $4\frac{1}{16}$ inches.

The diameter of the rivet is $\frac{1}{16}$ inch less than that of the rivet-hole, or = ·8125 − ·0625 = ·75, or $\frac{3}{4}$ inch.

Rivets in the end-plates and gusset-stays of Steam-boilers

should be from 5 to 8 per cent. larger in diameter than those of the shell, to provide for additional strains due to expansion, and accidental overstraining due to overheating.

Various Rules of Practice for the Diameter and Pitch of Rivets.—It may be useful to give here a few of the numerous empirical rules by which the diameter and pitch of rivets of riveted joints are determined in practice. They are collected in Tables 50–52: in which d = the diameter of rivet in inches, and t = the thickness of plate in inches.

TABLE 50.—VARIOUS RULES USED IN PRACTICE FOR DETERMINING THE DIAMETER OF THE RIVETS FOR THE JOINTS OF STEAM-BOILERS.

Diameter of Rivet = Thickness of Plate in Inches multiplied by the following numbers:—	Diameter of Rivet = Thickness of Plate in Inches multiplied by the following numbers:—
$d = t \times 1{\cdot}125$	$d = t \times 1{\cdot}80$
$d = t \times 1{\cdot}20$	$d = t \times 1{\cdot}82$
$d = t \times 1{\cdot}22$	$d = t \times 1{\cdot}85$
$d = t \times 1{\cdot}24$	$d = t \times 1{\cdot}88$
$d = t \times 1{\cdot}25$	$d = t \times 1{\cdot}90$
$d = t \times 1{\cdot}27$	$d = t \times 2{\cdot}00$
$d = t \times 1{\cdot}30$	$d = t \times 2{\cdot}125$
$d = t \times 1{\cdot}33$	$d = (t \times 2) - {\cdot}125$
$d = t \times 1{\cdot}36$	$d = (t \times 2) - {\cdot}25$
$d = t \times 1{\cdot}38$	$d = (t \times 2) - {\cdot}3125$
$d = t \times 1{\cdot}40$	$d = (t \times 2) - {\cdot}375$
$d = t \times 1{\cdot}45$	$d = 1{\cdot}875 \times \sqrt[3]{t}$
$d = t \times 1{\cdot}55$	$d = 1{\cdot}125 \times \sqrt[3]{t}$
$d = t \times 1{\cdot}57$	$d = 1{\cdot}100 \times \sqrt[3]{t}$
$d = t \times 1{\cdot}60$	$d = (t \times 1{\cdot}375) + {\cdot}125$
$d = t \times 1{\cdot}66$	$d = (t \times 1{\cdot}3) + {\cdot}20$
$d = t \times 1{\cdot}70$	$d = (t \times 1{\cdot}25) + {\cdot}3125$
$d = t \times 1{\cdot}73$	$d = (t \times 1{\cdot}0625) + {\cdot}25$
$d = t \times 1{\cdot}75$	$d = (t \times {\cdot}6) + {\cdot}50$
$d = t \times 1{\cdot}77$	$d = (t \times {\cdot}5) + {\cdot}45$

The pitches used by different boiler-makers vary even more than the diameter of rivets used. For instance, for a single-riveted lap-joint of wrought-iron plates $\frac{5}{8}$ inch thick and iron-rivets, it will be seen from the following Table that pitches are used varying from ·75 inch diameter of rivet $\times$ 2·4 = 1·8 inch to ·75 $\times$ 3 = 2·25 inches.

The Pitch of the Rivets of Treble-Riveted Lap-Joints used in practice for steam-boilers, is on an average = diameter of rivet $\times$ 4 to 4·25 for wrought-iron plates and iron-rivets; and = diameter of rivet $\times$ 3 to 3·33 for mild-steel plates and steel-rivets.

The Pitch of the Rivets of Butt-Joints with Double Butt-Straps for steam-boilers of iron and steel plates usually employed in practice, may be ascertained by multiplying the diameter of the rivet in inches by one

TABLE 51.—VARIOUS PITCHES OF RIVETS USED IN PRACTICE FOR SINGLE-RIVETED LAP-JOINTS FOR STEAM-BOILERS.

Thickness of Plate in Inches.	WROUGHT-IRON PLATES AND IRON-RIVETS.	MILD-STEEL PLATES AND STEEL RIVETS.
	Pitch = Diameter of Rivet in Inches multiplied by the following numbers :—	Pitch = Diameter of Rivet in Inches multiplied by the following numbers :—
1/4	2·5, 2·75	2, 2·2, 2·3
5/16	2·5, 2·6, 2·75, 2·8	2, 2·23, 2·4
3/8	2·4, 2·5, 2·66, 2·88, 3·0	2, 2·25, 2·66
7/16	2·5, 2·66, 2·7, 2·75, 2·78	2, 2·25, 2·5
1/2	2·45, 2·47, 2·57, 2·66	2, 2·33, 2·66
9/16	2·4, 2·45, 2·5, 2·56, 2·66	2, 2·25, 2·66
5/8	2·36, 2·57, 2·63, 2·66	2, 2·2, 2·4
11/16	2·33, 2·35, 2·4, 2·45	2, 2·25, 2·58
3/4	2·25, 2·33, 2·5, 2·56	2, 2·3, 2·61
13/16	2·18, 2·22, 2·25, 2·3	2, 2·2, 2·5
7/8	2·18, 2·25, 2·5, 2·7	2, 2·25, 2·62
15/16	2·22, 2·25, 2·6, 3	2, 2·2, 2·57
1	2·2, 2·25, 2·66, 2·95	2, 2·25, 2·61
1 1/8	2·12, 2·23, 2·71, 2·84	2, 2·26, 2·64

TABLE 52.—VARIOUS PITCHES OF RIVETS USED IN PRACTICE FOR DOUBLE-RIVETED LAP-JOINTS OF STEAM-BOILERS.

Thickness of Plate in Inches.	WROUGHT-IRON PLATES AND IRON-RIVETS.	MILD-STEEL PLATES AND STEEL RIVETS.
	Pitch = Diameter of Rivet in Inches multiplied by the following numbers :—	Pitch = Diameter of Rivet in Inches multiplied by the following numbers :—
5/16	3·4; 3·6, 3·8, 4	2·66, 2·75, 2·86, 3·10, 3·33
3/8	3·33, 3·4, 3·5, 3·6, 3·91, 4	2·86, 3, 3·33, 3·4, 3·5, 4
7/16	3·25, 3·33, 3·5, 3·85, 4	2·58, 2·82, 3, 3·5, 3·7
1/2	3, 3·14, 3·33, 3·57, 3·8, 4	2·6, 2·75, 3, 3·25, 3·5
9/16	3·14, 3·2, 3·6, 3·71, 4	2·66, 3, 3·10, 3·5, 3·55
5/8	3·14, 3·43, 3·6	2·62, 3, 3·26, 3·45
11/16	3, 3·1, 3·2, 3·66	2·68, 3, 3·31, 3·48
3/4	3, 3·2, 3·25, 3·66	2·57, 3, 3·33, 3·49
7/8	2·9, 3·1, 3·22, 3·33	2·66, 3, 3·1, 3·25
15/16	2·8, 3·11, 3·33, 3·5	2·76, 3, 3·2, 3·3
1	2·8, 2·84, 3, 3·11, 3·56	2·59, 2·8, 3, 3·35
1 1/16	2·75, 2·85, 3, 3·14	2·55, 2·75, 2·9, 3
1 1/8	2·73, 2·8, 2·86, 3	2·57, 2·76, 3, 3·1
1 3/16	2·8, 2·95, 3·1	2·54, 2·86, 3
1 1/4	2·82, 2·93, 3.12	2·6, 2·88, 3

of the following multipliers according to the percentage of plate retained between the rivet-holes:—

Percentage of plate between the rivet-holes	78,	79,	80,	81,	82,	83,	84,	85.
Multiply the diameter of rivet by	3·545,	4·761,	5,	5·263,	5·555,	5·882,	6·25,	6·666.

For instance, the pitch of rivets for a butt-joint with double butt-straps to retain 80 per cent. between the rivet-holes with rivets 1 inch diameter is = 1 inch diameter of rivet × 5 = 6 inches.

Working-pressure of Steam-Boilers for a Given Strength of Riveted-Joint.—The working-pressure for the shells of cylindrical steam-boilers with riveted joints, expressed in terms of the strength of the joint, may be determined by the following *Rule*:—

Working-pressure of steam for steam-boilers =

$$\frac{\text{S} \times \text{percentage of strength of joint} \times t \times 2}{\text{internal diameter of boiler in inches} \times \text{factor of safety}}.$$

In which S = the tensile strength of the material = 21 tons, or 47000 lbs. per square inch for wrought-iron plates and rivets, and 28 tons, or in round numbers 63000 lbs. per square inch for mild steel plates and rivets.

t = the thickness of the plates in inches.

The smaller of the two percentages of the strength of the joint should be used. The factor of safety may be as given at page 173.

Example: Required the working-pressure suitable for a Cornish-boiler of 5 feet 6 inches internal diameter, of mild-steel plates $\frac{3}{8}$ inch thick, the strength of the riveted joints is 70 per cent. of that of the solid plate; the factor of safety is 5?

$$\text{Then } \frac{63000 \text{ lbs. per square inch} \times \cdot 7 \times \cdot 375 \text{ inch thick} \times 2}{66 \text{ inches internal diameter} \times 5, \text{ the factor of safety}} = 100 \text{lbs.}$$

per square inch, the working-pressure of steam suitable for this boiler.

Strength of Riveted Joints of Steam-Boilers.—The strength of a riveted joint depends principally upon the strength and ductility of the plates, the proportions of the pitch and diameter of the rivets to the thickness of the plates, the manner of perforating the plates for the rivets, the shearing-strength of the rivets, the percentage of plate between the rivet-holes, the form of the joint, and upon the stress being equally borne by all the rivets in the joint.

Experiments on the strength of riveted joints were made by Fairbairn, and the following ratios were deduced expressing the strength of the riveted joints of steam-boilers:—

Tensile strength of solid plate	100
Strength of double-riveted joints	70
Strength of single-riveted joints	56

These ratios of strength only apply to well-proportioned joints.

These estimates of the strength of joints have been confirmed by subsequent experiments, but the results of other and more recent experiments give higher ratios of strength.

In properly proportioned and well-made riveted-joints, having 60 per cent. of the plate remaining between the rivet-holes in single-riveted lap-joints, and 70 per cent. in double-riveted lap-joints, the strength of the joints as compared with the solid plate may be assumed to be as follows:—

Tensile strength of solid plate	100
Strength of double-riveted butt-joint	80
Strength of double-riveted lap-joint	75
Strength of single-riveted lap-joint	62

The seams in each case breaking joint properly.

Elastic Limit of the Riveted Joints of Steam-Boilers.—The riveted seams of a steam-boiler would cease to be steam-tight for some time before the internal pressure was equal to the elastic limit of the plates. If a boiler were stretched beyond the elastic limit of the material, the rivet-holes would become stretched and the joints of the plates would be disturbed, resulting in large leakage from the rivet-holes and seams.

The elastic limit of riveted joints of wrought-iron and mild-steel is as follows:—

	Elastic Limit in Tons per Square Inch.
Mild-steel boiler-plates of very best quality	16 to 17
Mild-steel boiler-plates of good ordinary quality	14 to 15
Wrought-iron boiler-plates of very best quality	12 to 13
Wrought-iron boiler-plates of good ordinary quality	10 to 11

These may be taken as the average values of the joints.

Tests of Riveted Joints.—The results of careful tests of riveted joints of wrought-iron and mild-steel by Mr. Codman are given in the following Table.

The proportions of the riveted joints in terms of the thickness of the plate, t, were as follows:—

		Wrought-iron joint.	Mild steel joint.
Diameter of rivets	=	$t \times 2$	$t \times 2$
Pitch of rivets in line	=	$t \times 9{\cdot}3$	$t \times 8{\cdot}7$
Distance apart of pitch-lines	=	$t \times 4$	$t \times 3$
Distance from edge of plate	=	$t \times 3$	$t \times 2{\cdot}7$

The dimensions being given in inches.

The iron-joint broke by fracture of the plates, and the steel-joint by shearing of the rivets.

TABLE 53.—RESULTS OF TESTS OF RIVETED-JOINTS OF WROUGHT-IRON AND MILD-STEEL.

Description.	Wrought-Iron Plate. Iron Rivets in Punched Holes, Hydraulic-Riveted.	Mild-Steel Plate. Steel Rivets in Drilled Holes, Machine-Riveted.
Width of plate in inches	6·96	6·51
Thickness of plate in inches	·38	·391
Gross sectional area of plates in square inches	2·645	2·548
Tensile strength of plates in lbs. per square inch	48350	57128
Stress on joint, total in lbs.	81800	98900
Stress per square inch of gross area of joint in lbs.	30926	38815
Net sectional area of plates in square inches	2·01	1·913
Stress per square inch of net area of plate in lbs.	40700	51700
Number of rivets	4	4
Diameter of rivets in inches	·75	·75
Area of rivets in square inches	1·767	1·767
Shearing-stress per square inch of rivet-area in lbs.	46300	55970
Ratio of joint to solid plate per cent.	64	70
Shearing-strength of rivets compared with the tensile strength of net section of plate, per cent.	...	90

TABLE 54.—AVERAGE RESULTS OF EXPERIMENTS ON SINGLE-RIVETED LAP-JOINTS OF IRON BROKEN BY TEARING AND SHEARING.

Authority.	Mode of preparing the Holes.	Mean apparent Tenacity of Joint in Tons per Square Inch.	Ratio of Mean Apparent Tenacity to the Tenacity of the original Plate, per Cent.	Mean Shearing Resistance of the Rivets in Tons, per Square Inch.	Ratio of the Tenacity of the Plate to the Shearing Resistances of the Rivets, per Cent.
Fairbairn	Punched	17·55	68·11	22·40	78·3
Hendry	Punched	17·96	—	—	—
Stoney	Punched	17·16	79·97	18·84	91·0
Stoney	Drilled	19·39	88·31	18·27	106·1
Fairbairn	Punched	—	—	20·41	—
Fairbairn	Drilled	—	—	19·47	—
Master Mechanics' Association	Punched	22·30	83·52	—	—
	Drilled	—	—	20·80	—
Greig & Eyth	Punched	16·80	75·50	—	—
Greig & Eyth	Drilled	19·75	88 70	18·43	107·1
Mean Result	Punched	18·35	76·77	20·55	84·6
Mean Result	Drilled	19·57	88·50	19·24	106·6

TABLE 55.—AVERAGE RESULTS OF EXPERIMENTS ON DOUBLE-RIVETED LAP-JOINTS AND DOUBLE-RIVETED BUTT-JOINTS OF IRON BROKEN BY TEARING.

Description.	Authority.	Apparent Tenacity of Joint in Tons per Square Inch.	Ratio of apparent Tenacity of the Joint to the Tenacity of the Original Plate, per Cent.
LAP-JOINTS.			
Single Shear, punched .	Fairbairn . .	23·50	91·2
,, ,, ,, .	Kirkaldy . . .	25·57	116·2
,, ,, ,, .	Easton & Anderson	16·35	87·4
,, ,, drilled .	Greig & Eyth . .	21·17	95·0
,, ,, punched .	Knight . . .	12·08	56·4
BUTT-JOINTS.			
Single Cover, punched .	Fairbairn . . .	24·07	93·4
,, ,, ,, .	Martell . . .	19·95	—
Double Cover ,, .	Fairbairn . . .	21·44	83·2
,, ,, ,, .	Kirkaldy . .	19·39	86·4
Single Cover, drilled .	Greig & Eyth . .	18·07	81·2
Double Cover ,, .	,, ,, . .	20·65	92·8
,, ,, punched .	Knight . . .	17·52	90·0

TABLE 56.—AVERAGE RESULTS OF EXPERIMENTS SHOWING THE RESISTANCE TO SHEARING OF IRON AND STEEL RIVETS IN STEEL-PLATES.

Description of Joint.	Authority.	Holes made by	Shearing Resistance of Rivets in Tons per Square Inch.
STEEL-PLATES & IRON-RIVETS.			
Lap, double-riveted . .	Martell and Knight .	Punch	19·44
Lap, treble-riveted . . .	Parker	Drill	17·27
Butt (one experiment) . .	Martell	Punch	23·39
STEEL PLATES & STEEL RIVETS.			
Lap, single-riveting . . .	Sharp and Greig and Eyth . .	Punch	25·62
,, ,, . .	,,	Drill	23·67
Lap, double-riveting . .	Sharp, Martell, and Kirkaldy . .	Punch	23·99
,, ,, . .	Sharp and Martell .	Drill	24·86
,, ,, . . .	Easton and Anderson .	Punch	19·85
Lap, treble-riveting . .	Martell	Drill	26·91
,, ,, (chain) .	Parker and Denny .	,,	24·65
,, ,, (zigzag) .	,,	,,	20·00
Lap, quadruple-riveting . .	,,	,,	19·20
Butt, single-riveting . .	Sharp and Greig and Eyth . .	...	22·72
,, double-riveting . .	,,	...	24·93

TABLE 57.—AVERAGE RESULTS OF EXPERIMENTS ON DOUBLE-RIVETED STEEL LAP-JOINTS BROKEN IN TEARING.

Plates.	Authority.	Apparent Tenacity of Joint.	Tenacity of Plate.	Ratio of Tenacity of Joint to that of Plate.
		Tons per square inch.	Tons per square inch.	Per Cent.
Unannealed . .	Martell . .	30·72	28·93	106·2
„ . .	Kirkaldy . . .	39·00	32·50	120·3
„ . .	Easton & Anderson	26·98	25·85	104·4
Annealed . .	Martell . . .	31·40	28·93	108·5
Drilled, unannealed	Greig & Eyth . .	26·89	25·83	104·1

TABLE 58.—AVERAGE RESULTS OF EXPERIMENTS ON STEEL PLATES, WITH TREBLE-RIVETED LAP-JOINTS BROKEN IN TEARING.

Riveting.	Authority.	Holes made by	Apparent Tenacity of Joint.	Tenacity of Plate.	Ratio of Tenacity of Joint to that of Plate.
					Per Cent.
Treble, iron rivets .	Parker & Denny	Punch	22·90	30·0	76·3
„ „ .	„ „	Drill	32·93	30·5	108·0
Treble, steel rivets .	„ „	„	30·99	28·8	107·5

TABLE 59.—AVERAGE RESULTS OF EXPERIMENTS ON DOUBLE-RIVETED STEEL BUTT-JOINTS BROKEN IN TEARING.

Description.	Authority.	Apparent Tenacity of Joint.	Tenacity of Plate.	Ratio of Tenacity of Joint to that of Plate.
UNANNEALED.		Tons per square inch.	Tons per square inch.	Per Cent.
Double-riveted. One cover	Sharp . .	41·44	36·22	114·4
„ Two covers	„ . .	41·02	36·22	113·2
„ „	Martell . .	25·65	28·93	88·7
„ „	Boyd . .	25·41	29·15	87·2
„ „	Greig & Eyth	27·91	25·83	108·0
„ „	Parker . .	24·17	27·5	87·8
ANNEALED.				
Double-riveted. One cover	Martell. .	22·90	28·93	79·2
„ Two covers	Kirkaldy . .	33·66	36·20	93·0

Experiments on Riveted Joints.*—The average results of a number of careful experiments on several forms of riveted joints, broken by tearing

* The Author is indebted for Tables 54—60 to the Report on Riveted Joints by the Committee of the Institution of Mechanical Engineers.

and shearing, are given in Tables 54—60: in which the ratio of the tension on the joint to the tearing section of the plate at the place of fracture is termed the "apparent tenacity" of the joint.

The apparent tenacity is rendered less than the original tenacity of the iron, by injury done in drilling or punching; by any irregularity of stress due to the way in which the rivets load or crush the plate; and by any irregularity of distribution of stress due to the bending of the joint as a whole under the action of the load.

TABLE 60.—RESULTS OF EXPERIMENTS ON DOUBLE-RIVETED LAP AND BUTT-JOINTS OF STEEL, HAND AND MACHINE RIVETED.

Particulars.	Double-riveted Lap-Joints.			Double-riveted Lap and Butt-Joints.		
	Hand-riveted.	Machine-riveted.		Machine-riveted, with Hydraulic Pressure of 100 Tons per Square Inch.		
		Hydraulic Pressure 30 to 35 Tons.	Hydraulic Pressure 60 to 80 Tons.	Lap-joint.	Butt-joint.	Butt-joint.
Thickness of steel plate, inches	·404	·399	·745	·739	·757	·991
Visible slip began at, tons	31·15	37·64	54·62	69·74	84·00	84·71
Total breaking load, tons	87·34	87·16	173·8	242·4	187·5	199·3
Tensile stress per square inch . . . tons	25·83	26·10	29·30	28·23	26·33	26·51
Tensile strength per square inch of original plate in tons	25·57	24·94	28·37	28·75	28·65	28·23
Shearing stress per square inch . . . tons	21·34	21·30	22·61	20·27	11·76	18·62
Bearing-pressure per sq. inch . . . tons	33·46	33·84	26·37	34·34	38·90	38·55
Load per inch of breadth when visible slip began, tons	2·57	3·05	4·44	4·31	6·60	8·32
Breaking load per inch of breadth of nominal thickness on original plate, in tons	9·45	9·61	21·28	21·56	21·73	28·41
Breaking load per inch of breadth of nominal thickness at joint, in tons .	6·98	7·06	14·13	15·00	14·74	19·72
Efficiency of joint, per cent.	73·9	73·5	66·4	69·6	67·80	69·40

Diagonal Seams.—The riveted seams of the plates composing a belt of the shell of a steam-boiler have in some cases been arranged in an inclined direction, as shown in Fig. 141, with the object of increasing their strength.

The circumferential seams being double as strong as the longitudinal seams of a cylindrical boiler-shell, to resist internal pressure, by inclining the latter seams they may be made to partake of the strength of the former, and the strength of the boiler-shell may be considerably increased.

The ratio of strength, R, of an inclined or diagonal seam, to that of a straight seam, or ordinary longitudinal seam of a boiler-shell, may be found by this *Rule*:—

$$R = \frac{2}{\sqrt{[(\text{cosine of angle of inclination of seam})^2 \times 3 + 1]}}.$$

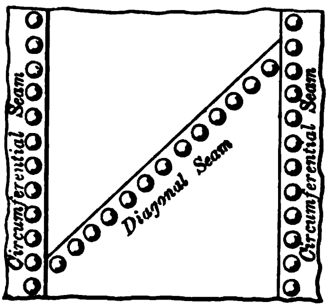


Fig. 141
Diagonal seam of boiler-shell.

The cosine of the angle of inclination may be found from a table of natural sines of angles.

Example: How much stronger is a riveted seam when inclined at an angle of 45°, than a similar straight seam, such as the longitudinal seam of a cylindrical boiler-shell?

Then, the cosine of angle 45° is ·707, and ·707 × ·707 = ·499 × 3 = 1·497 + 1 = 2·497, and $\sqrt{2{\cdot}497} = 1{\cdot}573$, then $2 \div 1{\cdot}573 = 1{\cdot}27$. Showing that a diagonal seam at this angle is 27 per cent. stronger than a similar seam placed longitudinally.

The objection to this form of seam is that it is very wasteful of material, a quantity of plate having to be cut to waste in forming the ends of the plates to the required angle.

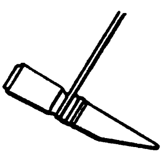


Fig. 142.

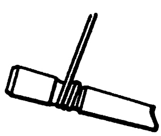


Fig. 144.

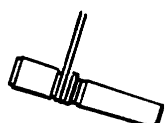


Fig. 145.

Fig. 142 & 144.—Caulking tools. Fig. 145.—Fullering tool.

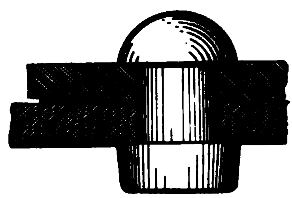


Fig. 143.—Example of caulking with a sharp-nosed tool.

Caulking the Riveted Joints of Steam-Boilers.—The joints or seams of the plates of boilers are frequently caulked to render them tight and free from leakage. But a riveted joint, of good proportions with well-fitted plates riveted in the best manner, should not require caulking, as the joints should be perfectly water-tight. Caulking is liable to injure the edges of the plates and produce flaws which may afterwards develop into fractures.

When caulking is performed with a thin-pointed tool as shown in Fig. 142, a thin piece of the plate is driven between the plates and they are not metal to metal, being separated from the edge of the lap to the line of rivets as shown in Fig. 143.

A sharp-nosed tool is also liable to cut the skin of the plates along the edge of the lap of the joint, and induce corrosive grooving. When caulking is adopted, it should be performed with a round-nosed tool as shown in Fig. 144.

When joints are not tight it is much better to fuller than caulk the plates. By fullering the edges of the plates with a square-nosed tool, as shown in Fig. 145, the plates are closed and brought into contact in slack places, and a metal-to-metal joint may be secured.

The surfaces forming the joint should be sponged with a weak solution of sal-ammoniac and water, to remove the oxide and obtain clean faces of the metal.

The edges of the plates should be planed to an angle of 75°, as shown in Fig. 146, to obtain a true edge and facilitate fullering.

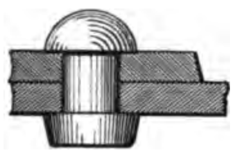


Fig. 146.—Riveted joint.

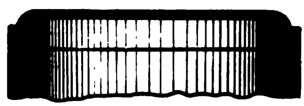


Fig. 147.—Cylindrical shell with flat-end.

Flat-Ends of Cylindrical Shells.—The unstayed flat-ends of a cylindrical shell are very weak compared with the strength of the shell.

The bursting-pressure of the unstayed flat end of a cylindrical shell, as shown in Fig. 147, may be found approximately by this *Rule*:—

Bursting-pressure of flat unstayed end in pounds per square inch =

$$\frac{\text{Thickness of plate in inches} \times \text{tensile strength of the metal in pounds}}{\text{Area of the end plate in square inches} \times \cdot 09}.$$

The diameter of the plate for this *Rule* is to be measured from centre to centre of the rivet-holes when the end is attached to the shell by an angle-hoop, or from the root of the flange when the end-plate is flanged.

Example: Required the bursting-pressure of the unstayed flat crown of a vertical boiler, flanged to join the shell. Diameter of end-plate measured from the root of the flange, 30 inches. Thickness of end-plate, ⅜ inch; tensile strength of the steel 60000 lbs. per square inch?

$$\text{Then } \frac{\cdot 375 \text{ inch thick} \times 60000 \text{ lbs.}}{30 \times 30 \text{ inches} \times \cdot 7854 \times \cdot 09} = 354 \text{ pounds per square inch,}$$

the bursting-pressure of the flat end of this shell.

The thickness of the flat unstayed end required to resist a given bursting-pressure may be found by this *Rule*:—

Thickness of flat unstayed end of a cylindrical shell =

$$\frac{\text{Area of the end-plate in square inches} \times \cdot 09 \times \text{bursting-pressure}}{\text{Tensile strength of the metal in pounds per square inch}}.$$

Example: Required the thickness of the end-plate described in the preceding example, to be equal to a bursting-pressure of 354 pounds per square inch?

$$\text{Then } \frac{30 \times 30 \text{ inches} \times \cdot 7854 \times \cdot 09 \times 354 \text{ lbs.}}{60000 \text{ lbs. per square inch}} = \cdot 375 \text{ inch,}$$

the thickness required for this unstayed flat end-plate.

The internal diameter of this shell is 33 inches, and the thickness and strength of the plates is the same as those of the end-plates.

The bursting-pressure of the shell is =

$$\frac{\cdot 375 \text{ inch thick} \times 2 \times 60000 \text{ lbs.}}{33 \text{ inches internal diameter}} = 1364 \text{ lbs. per square inch,}$$

or $1364 \div 354 = 3{\cdot}85$ times as great as that of the end-plate of this shell.

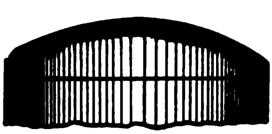


Fig. 148.—Cylindrical shell with dished-end.

Dished-Ends of Cylindrical Shells.—In order to make an unstayed end of a cylindrical shell equal in strength to that of the shell, it is necessary to dish the end to a radius equal to the diameter of the shell, as shown in Fig. 148.

Gusset-Stays for Flat Ends of Cylindrical Shells.—The flat ends of cylindrical boilers are stayed either by longitudinal bolt-stays, bar-stays, or gusset-stays. When the design of the boiler permits their use, gusset-stays are most efficient

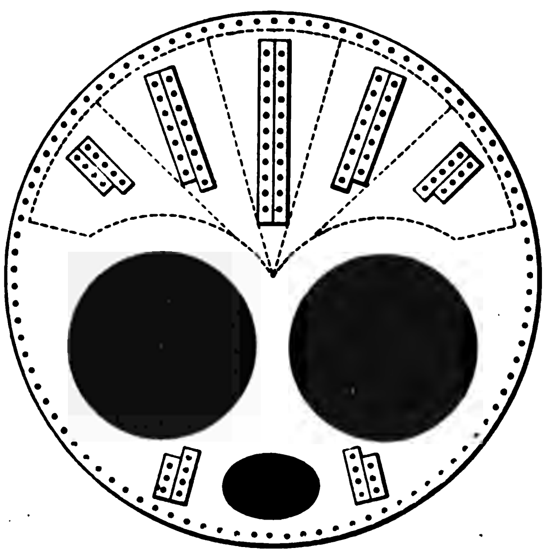


Fig. 149.—End plate of a Lancashire boiler.

for this purpose, and they do not interfere with access to the boiler for cleaning or examination.

Gusset-stays are arranged in planes radiating from the centre of the end-plate. Each gusset-stay sustains a pressure against a sector of the circular area supported, as shown by the dotted lines in Fig. 149. The rivet-section requisite for the ends of a gusset-stay is generally considerably stronger than the narrowest portion of the web of the stay, so that it is only necessary to compute the strength of the gusset-stay at the weakest part of its web, or at A in Fig. 150.

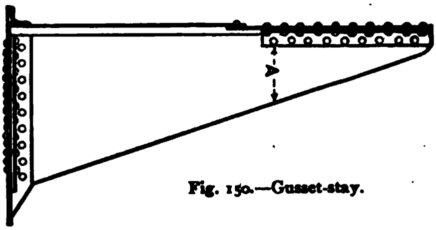


Fig. 150.—Gusset-stay.

The strength of gusset-stays may be calculated by the following *Rules*, in which the working tensile stress allowed on the stays may be the same as that given for stud-stays on page 235.

The working tensile stress on a gusset-stay may be found by this *Rule*—

$$\frac{\text{Area of surface supported in square inches} \times \text{working pressure of steam}}{\text{Depth of web at narrowest part in inches} \times \text{thickness of web in inches}}.$$

Example: One of the gusset-stays of a Lancashire boiler of steel-plates has a web 14 inches deep at the narrowest part, and ½ inch thick. It supports a surface of 525 square inches of the end-plate, and the working pressure of the steam is 120 lbs. per square inch. Required the tensile stress on the gusset-stay?

Then the working tensile stress on the gusset-stay in lbs. per square inch is =

$$\frac{525 \text{ square inches of supported surface} \times \left(\begin{array}{c}\text{120 lbs. per square inch}\\ \text{steam pressure}\end{array}\right)}{\text{14 inches depth of web at narrowest part} \times \text{·5 inch thickness of web}} =$$

9000 lbs. per square inch, the tensile stress on the steel-web of the gusset-stay.

The depth of web of the gusset-stay in inches may be found by this *Rule*—

$$\frac{\text{Area of surface supported in square inches} \times \text{working pressure of steam}}{\text{Working tensile stress on the stay} \times \text{thickness of web in inches}}.$$

For instance, the depth of the web of the gusset-stay described in the preceding example is =

$$\frac{525 \text{ square inches of supported surface} \times \left(\begin{array}{c}\text{120 lbs. per square inch}\\ \text{working pressure}\end{array}\right)}{\text{9000 lbs. per square inch working tensile stress} \times \text{·5 inch thickness of web}} =$$

14 inches, the depth of the web of the gusset-stay at the narowest part.

The thickness of the web of the gusset-stay in inches may be found by this *Rule*—

$$\frac{\text{Area of supported surface in square inches} \times \text{working pressure of steam}}{\text{Working tensile stress on the stay} \times \text{depth of web in inches}}.$$

Applying this *Rule* to the preceding example, it gives—

$$\frac{525 \text{ square inches of supported surface} \times \left(\begin{array}{c}120 \text{ lbs. per square inch}\\ \text{working pressure}\end{array}\right)}{9000 \text{ lbs. per square inch working tensile stress} \times 14 \text{ inches depth of web}}$$

= ·5 inch, the thickness of the web required for this gusset-stay.

The working pressure of steam in lbs. per square inch suitable for a gusset-stay may be found by this *Rule*—

$$\frac{\text{Working tensile stress allowed on the stay} \times \left(\begin{array}{c}\text{depth of stay in inches} \times\\ \text{thickness of stay in inches}\end{array}\right)}{\text{Area of surface in square inches supported by the stay}}.$$

Applying this *Rule* to the preceding example, it gives:—

$$\frac{9000 \text{ lbs. per square inch working tensile strength} \times \left(\begin{array}{c}14 \text{ inches deep} \times\\ \cdot 5 \text{ inch thick}\end{array}\right)}{525 \text{ square inches area of supported surface}}$$

= 120 lbs. per square inch, the working pressure of steam to be allowed on this stay.

The area of the sector of the circular area supported by a gusset-stay may be found by this *Rule*:—Multiply the length of the arc by its radius, and one-half of the product is the area of the sector.

For instance, the length of the arc of the sector shown on the centre of the end-plate in Fig. 149 is 25 inches, its radius is 42 inches, and its area is = (25 inches × 42 inches) ÷ 2 = 525 square inches.

Stays for the Flat-Surfaces of Water-Spaced Plates.—Stud-stays, screwed into plates forming water-spaced compartments of steam-boilers, shown in Figs. 151 and 152, each sustain a pressure measured by a rectangle contained between four stays, as shown by dotted lines in Fig. 151, the side of the rectangle being equal to the distance between the centres of the stays, or to their pitch.

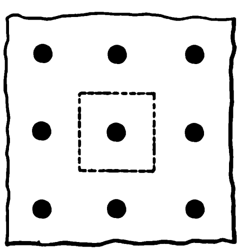


Fig. 151.

Fig. 152.

Figs. 151 & 152.—Stud-stays for flat surfaces.

Each stud-stay should be proportioned to the stress upon it, by the following *Rules*—

Let S = the area of surface in square inches supported by each stay.

P = the working-pressure of the steam in lbs. per square inch.

A = the sectional area of the stay at the bottom of the thread, or smallest part, in square inches.

T = the working tensile stress allowed on the stay, in lbs. per square inch.

Then, for equilibrium of stresses, S × P must be equal to A × T.

For instance, if the area supported by a stud-stay be 47 square inches, the working-pressure of the steam 150 lbs. per square inch, the sectional area of the stay 1 square inch, and the working tensile strength of the stay 7050 lbs. per square inch.

Then, 47 square inches × 150 lbs. per square inch = 7050 lbs., and 1 square inch area × 7050 lbs. working tensile strength = 7050 lbs., showing that the strength of the stay is equal to the strain upon it.

Fig. 153.—Stud-stay.

The sectional area in square inches of screwed stud-stays having a head at one end and a nut at the other, as shown in Fig. 153, or having each end of the stay riveted over to form a good head, may be found by the following formula, in which the notation is the same as given above,

$$A = \frac{S \times P}{T}.$$

The factor of safety for these stays should not be less than seven, and T should not be greater than $\frac{1}{7}$ of the ultimate tensile strength of the stay, or as follows:—

	Ultimate Tensile Strength in lbs. per square inch.				Working-stress, or value of T.
Copper-stays	= 33600	÷	7	=	4800 lbs. per square inch.
Wrought-iron stays	= 49000	÷	7	=	7000 „ „ „
Mild-steel stays	= 63000	÷	7	=	9000 „ „ „

Example: Required the sectional area of a stud-stay of mild-steel to support 49 square inches of area of surface, with a working-pressure of steam of 150 lbs. per square inch?

Then $\frac{49 \text{ square inches} \times 150 \text{ lbs. pressure per square inch}}{9000 \text{ lbs. per square inch}} = \cdot 81167$ square inch, sectional area, and $\sqrt{\frac{\cdot 81167}{\cdot 7854}} = 1\frac{1}{16}$ inch, the diameter of the stay at the bottom of the thread.

The pitch in inches of screwed stud-stays for flat surfaces, when provided with substantial heads, or fitted with nuts, may be found by this formula, in which the notation is the same as in the preceding *Rule*:—

$$\text{Pitch of stud-stays in inches} = \sqrt{\frac{A \times T}{P}}$$

Example: Required the pitch of the stud-stays described in the previous example.

Then $\frac{\cdot 81667 \text{ square inch} \times 9000 \text{ lbs. per square inch}}{150 \text{ lbs. per square inch pressure}} = 49$ square inches, the area of surface supported by each stud-stay, and $\sqrt{49} = 7$ inches, the pitch of the stays required.

The strain on the stay of a flat surface per square inch of section may be found by this formula:—

$$\text{Strain per square inch of section of each stay} = \frac{S \times P}{A}.$$

For instance, the strain on the screwed stud-stay given in the preceding

$$\text{example is} = \frac{49 \text{ square inches} \times 150 \text{ lbs. pressure per square inch}}{\cdot 81667 \text{ square inch area of stay}} = 9000,$$

the strain in lbs. per square inch of section of each stay.

The working-pressure of steam suitable for screwed stud-stays for flat surfaces when provided with substantial heads, or fitted with nuts, may be found by this formula:—

$$\text{Working-pressure of steam in lbs. per square inch} = \frac{A \times T}{S}.$$

For instance, the working-pressure suitable for the stayed surface described in the preceding example is =

$$\frac{\cdot 81667 \text{ square inch} \times 9000 \text{ lbs. per square inch}}{49 \text{ square inches of supported surface}} = 150 \text{ lbs. per square}$$

inch, the working-pressure of steam suitable for this stayed surface.

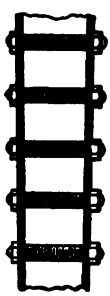


Fig. 154.
Stud-stays fitted with nuts.

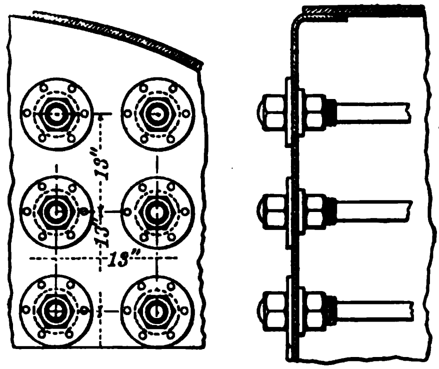
Stud-stays fitted with nuts as shown in Fig. 154 are more reliable, and give more effective support to a plate, than when formed with riveted heads.

Stud-stays between the sides of a fire-box and fire-box shell frequently break at a point close to the fire-box shell-plate. The difference of temperature between the parallel plates causes one plate to expand more than the other, and bends the stay. This continual bending causes the scale to be frequently thrown off the stay and oxidation is promoted, leading to grooving and ultimate fracture of the stay. This defect may be greatly remedied by making the flanges of the plates with large curves to accommodate the expansion.

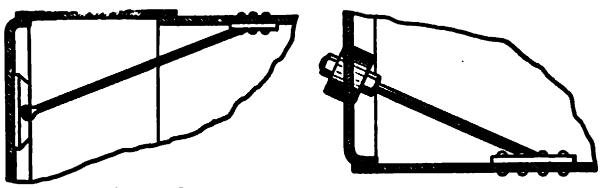
Copper stud-stays for locomotive fire-boxes formed of copper-plates ½ inch thick, are generally either ⅞ inch diameter, screwed into the plates with 14 threads per inch, or $\frac{15}{16}$ inch diameter with 12 threads per inch. The maximum pitch of these stays is 4 inches, from centre to centre.

Threads of Screwed Stud-Stays.—Screwed stud-stays may have 12 threads per inch up to 1⅛ inches diameter; 10 threads per inch from 1¼ inches to 1¾ inches diameter; and 8 threads per inch from 1⅞ inches to 3 inches diameter.

Longitudinal Stay-Bolts.—Longitudinal stay-bolts for the steam-space of boilers, shown in Figs. 155 and 156, and other stay-bolts for plates not exposed to the impact of heat and flame, may be proportioned by the same *Rules* as given above for stud-stays, by employing a different factor of safety giving a different value for T, as follows:—When the stays are fitted with nuts and washers, the latter being riveted to the plate and not less in diameter than three times that of the stay, and having a thickness equal to $\frac{2}{3}$ that of the plate they cover, a factor of 6 may be used with these rules, and T may be = 49000 lbs. ÷ 6 = 8166 lbs. per square inch for wrought-iron stay-bolts, and = 63000 ÷ 6 = 10500 lbs. per square inch for mild-steel stay-bolts. When the stays are fitted with nuts and washers, the latter being riveted to the plate and of a diameter equal to at least $\frac{2}{3}$ of the pitch of the stays, and having a thickness equal to that of the plate they cover, a factor of safety of 5 may be used, and T may be = 49000 lbs. ÷ 5 = 9800 lbs. per square inch for wrought-iron stay-bolts, and = 63000 lbs. ÷ 5 = 12600 lbs. per square inch for mild-steel stay-bolts.



Figs. 155 & 156.—Longitudinal stay-bolts.



Figs. 157 & 158.—Diagonal stays or palm stays.

Diagonal-Stays or Palm-Stays.—Diagonal bolt-stays, bar-stays, or palm-stays, shown in Figs. 157 and 158, are employed in some cases for staying the flat ends of the cylindrical shells of steam-boilers. The resultant tension is greater on a diagonal-stay than on one placed longitudinally. The tension on a diagonal-stay is equal to the tension which a stay placed perpendicular to a flat surface, supporting the same area of surface, would sustain, divided by the cosine of the angle which the diagonal-stay forms with a perpendicular to the supported surface, as shown in Fig. 159.

The tension in lbs. per square inch on a diagonal bolt-stay is =

$$\frac{\left(\begin{array}{c}\text{Pressure of steam in lbs. per square inch} \times \text{area}\\ \text{of supported surface in square inches}\end{array}\right)}{\text{Cosine } \alpha}.$$

For instance, if a diagonal bolt-stay at an angle of 60° supports a surface of 120 square inches of the end-plate of a cylindrical shell, with a steam-

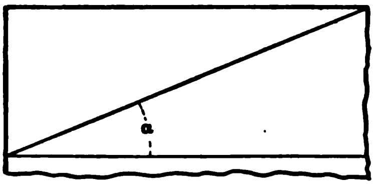


Fig. 159.—Diagonal stay.

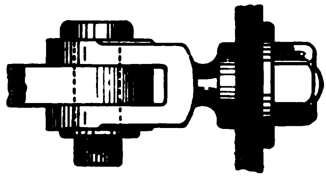


Fig. 160.—Fork-jointed stay-bar.

pressure of 120 lbs. per square inch. Then, a stay placed perpendicular to the end-plate would sustain a tensile stress = 120 square inches × 100 lbs. per square inch pressure = 12000 lbs. per square inch. The cosine of angle 60° is ·5, and 12000 ÷ ·5 = 24000 lbs. per square inch, the tensile stress on this diagonal stay.

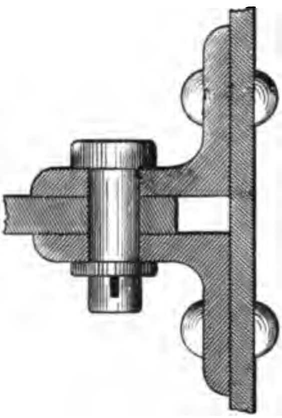
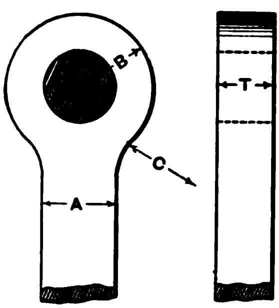


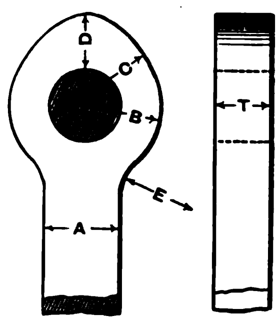
Fig. 161.—Stay-bar.

Proportions of Flat and Round Eye-Bar Stays.—When a stay is fixed between a fork-joint, as shown in Fig. 160, or between two plates, as shown in Fig. 161, the eye and pin should be proportioned to obtain uniformity of strength. When proportioned by the following *Rules*, the strength is approximately uniform in all parts for any thickness of bar less than, or not exceeding, its width.

Proportions of stay-bars with forged eyes, drilled holes, and turned pins, shown in Figs. 162 and 163.



Figs. 162 & 163.—Stay-bar with forged end



Figs. 164 & 165.—Stay-bar with stamped end.

Diameter of wrought-iron pin = width of flat bar, A, or = diameter of round bar.

Diameter of steel pin = width, or diameter, of bar × ·75.

Radius of eye, B, = diameter of pin × 1·2.

Radius of curve, C, = radius of eye, B.

Depth of eye, T, = thickness of flat bar or diameter of round bar.

Proportions of stay-bars with eyes stamped or cut out of bars, with drilled holes and turned pins, shown in Figs. 164 and 165.

Diameter of wrought-iron pin = width of flat bar, A, or = diameter of round bar.

Diameter of steel pin = width, or diameter, of bar × ·75

Radius of eye at the sides, B, = diameter of pin × 1·25.

Radius of curve, C, = diameter of pin × 1·5.

Radius of crown, or end, of eye, D, which is struck from the top of the pin, = diameter of pin × ·75.

Radius of curve, E, = diameter of pin × 1·25.

Depth of eye, T, = thickness of flat bar or diameter of round bar.

The rules apply to eye-bar stays of both wrought-iron and mild-steel.

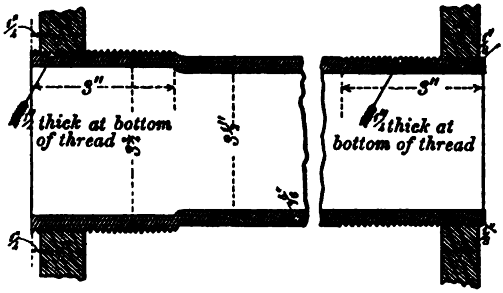


Fig. 166.—Stay-tube.

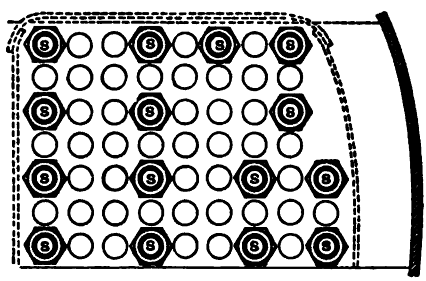


Fig. 167.—End view of stay-tubes.

Stay-Tubes for Staying the Tube-Plates of Multitubular Boilers.—Stay-tubes are employed in marine return-tube and other multitubular boilers to give support to the tube-plates. The stay-tubes are screwed into each tube-plate, and secured by nuts on one side, or on both sides, of the plate. A longitudinal section of a stay-tube is shown in Fig. 166; and an end-view of a tube-plate with a number of stay-tubes, marked S, is shown in Fig. 167. The sectional area of stay-tubes may be found by the following formula:—

Let A = the area of that portion of the tube-plate containing the tubes.

a = the area of the holes in the tube-plates for the tubes.

P = the working-pressure of the steam in lbs. per square inch.

T = the working tensile stress allowed on the tubes per square inch of section.

Total sectional area of stay-tubes in square inches =

$$\frac{\overline{A - a} \times P}{T}.$$

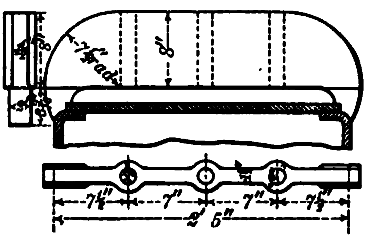
Example: In a return-tube boiler, working at 180 lbs. per square inch steam-pressure, one of the nests of tubes is 48 inches high and 44 inches wide, containing 73 tubes inclusive of stay-tubes. The thickness of the stay-tubes is $\frac{1}{4}$ inch at the bottom of the thread, and the working tensile stress on the tubes is to be 7000 lbs. per square inch. Required the total sectional area of the stay-tubes, and their number?

Then, the area of the tube-plates is $= 48 \times 44$ inches $= 2112$ square inches. The area of the end of the tubes is $= 3{\cdot}5$ inch $\times$ $3{\cdot}5$ inch $\times$ $\cdot 7854 = 9{\cdot}621$ area of one tube $\times$ 73 tubes $= 702$ square inches,

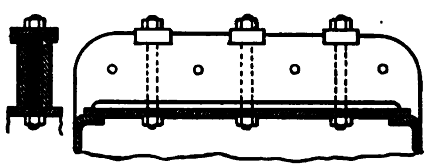
$$\text{and } \frac{2112 - 702 \text{ square inches} \times 150 \text{ lbs. per square inch}}{7000 \text{ pounds per square inch working tensile stress}}$$

$= 30{\cdot}21$ square inches, the total sectional area of the stay-tubes required. The sectional area of one stay-tube is $= (3{\cdot}5 \times 3{\cdot}5 \times 7854) - (3 \times 3 \times \cdot 7854) = 2{\cdot}552$ square inches. Therefore $30{\cdot}21$ square inches total sectional area $\div$ $2{\cdot}552$ square inches area of one tube $= 12$ stay-tubes are required for this nest of tubes; but to provide for the thinning of the tubes, due to wear and corrosion, at least double this number of stay-tubes would be provided in practice.

Screw-Threads of Stay-Tubes.—Stay-tubes of all sizes may have screws of nine threads per inch.



Figs. 168–170.—Solid girder-stays.



Figs. 171 & 172.—Split girder-stay.

Girder Roof-Stays of Fire-Boxes and Combustion-Chambers.—The flat crowns of fire-boxes of boilers of the locomotive type, and of combustion-chambers of return-tube boilers, and other boilers, when not stayed by bolt-stays to the outer shell, are strengthened by girder-stays fitted to the crown-plates. The girder-stays are either in one piece, as shown in Figs. 168—170, or split, that is, formed of two plates, either welded together at the ends, or riveted together with distance-pieces between them, as shown in Figs. 171 and 172. A water-space of from $1\frac{1}{2}$ to 2 inches deep is provided under the girders to permit the circulation of water.

Girder-stays are sometimes formed with bosses on the bottom surface, into which screws are inserted from the inside of the fire-box.

When girder-stays are secured by bolts which are not shouldered, ferrules are placed on the bolts in the water-space to permit the bolts being tightened without bending the crown-plate. The ends of the girders are fitted only to the edges of the crown-plate in some cases; and in others they bed both upon the edges of the crown-plate and on the round corners of the end-plates, as shown in Fig. 170, which is the firmer arrangement. The length of the girder-stay is equal to the length of the crown-plate in the former case, and to the outside length of the fire-box in the latter case. Girder-stays are sometimes attached by sling-stays to the roof of the shell in order to relieve the pressure on the top of the tube-plate.

Girder-stays for the roofs of fire-boxes and combustion-chambers may be proportioned by the following formula:—

Let C = 5400 for girder roof-stays of wrought-iron.
C = 6000 for girder roof-stays of mild-steel.
d = the depth of girder at the centre in inches.
T = the thickness of girder at the centre in inches.
N = the number of bolts supporting the girder, or bolting it to the crown-plate.
L = the length of the girder from end to end in inches.
P = the pitch of the supporting bolts in inches.
D = the distance the girders are spaced apart from centre to centre in inches.
S = the working-pressure of the steam in lbs. per square inch to be allowed on the girder-stays.

The depth of the girder-stay, d, in inches =

$$d = \sqrt[2]{\frac{S \times D \times L \times (L - P)}{C \times T \times \sqrt[2]{N}}}.$$

Example: The top of a combustion-chamber of a marine return-tube boiler is strengthed by girder-stays of mild-steel, each 30 inches long and 2 inches thick. Each girder-stay is secured by three supporting bolts of 8 inches pitch, and the girders are spaced $8\frac{1}{4}$ inches apart from centre to centre. What should be the depth of the girder for a working pressure of steam of 187 lbs. per square inch?

$$\text{Then } d = \frac{187 \text{ lbs.} \times 8{\cdot}25 \text{ inches} \times 30 \text{ inches} \times (30 \text{ inches} - 8 \text{ inches})}{6000 \times 2 \text{ inches thick} \times \sqrt[2]{3}}$$

$= 49$ and $\sqrt[2]{49} = 7$ inches, the depth of girder-stay required.

The thickness of the girder-stay in inches, T, may be found by this formula:—

$$T = \frac{(L - P) \times D \times L \times S}{d^2 \times C \times \sqrt[2]{N}}.$$

Applying this rule to the previous example,

$$T = \frac{(30 - 8) \times 8{\cdot}25 \text{ inches} \times 30 \text{ inches} \times 187 \text{ lbs.}}{7 \times 7 \text{ inches} \times 6000 \times \sqrt[2]{3}} = 2 \text{ inches},$$

the thickness required for this girder-stay.

The working-pressure of the steam in lbs. per square inch, S, to be allowed on the girder may be found by this formula:

$$S = \frac{C \times d^2 \times T \times \sqrt[3]{N}}{(L - P) \times D \times L}.$$

Applying this *Rule* to the previous formula,

$$S = \frac{6000 \times 7 \times 7 \times 2 \times \sqrt{3}}{(30 - 8) \times 8{\cdot}25 \times 30} = 187 \text{ lbs. per square inch,}$$

the working-pressure of steam suitable for these girder-stays.

The diameter of the supporting bolts for girder-stays may be found by the same *Rule* as given on page 235 for stud-stays. For instance, each bolt of the girder-stays in the previous example supports a surface = 8 inches pitch × 8·25 inches distance between the girders from centre to centre = 66 square inches, and

$$\frac{\left(\begin{array}{c}187 \text{ lbs. per square inch steam pressure} \times 66 \text{ square} \\ \text{inches of supported surface}\end{array}\right)}{9000 \text{ lbs. per square inch, the working tensile stress}}$$

= 1·37 square inches, the sectional area of the mild-steel bolt, and = $\sqrt{\frac{1{\cdot}37}{{\cdot}7854}} = 1\frac{5}{16}$ inches, the diameter at the bottom of the thread of each bolt supporting the girder-stay.

The working-pressure suitable for the supporting bolts of girder roof-stays, and the strain on them per square inch of section may be found by the *Rule* given for stud-stays on page 236.

Stay-Bolts for the Roofs of Fire-Boxes.—The roofs of the fire-boxes of locomotive boilers are in some cases stayed directly to the shell by stay-bolts. Boilers of modified locomotive-type have the roofs of the shells of the fire-boxes formed flat, and the roofs of the fire-boxes are stayed directly to the shell by stay-bolts of $\frac{7}{8}$ or 1 inch diameter. These stays are very efficient, and do not impede the circulation of the water. They may be proportioned by the same *Rules* as given for stud-stays on pages 234—236.

SECTION V.

DESCRIPTION AND PROPORTIONS OF CORNISH, LANCASHIRE, AND OTHER TYPES OF STATIONARY BOILERS; BOILER-SETTING; MULTITUBULAR-BOILERS; LOCOMOTIVE-BOILERS; PORTABLE-BOILERS; MARINE-BOILERS; VERTICAL-BOILERS; WATER-TUBE BOILERS, ETC.

The evaporative performance of these boilers is seldom more than from 6 to 8 lbs. of water from and at 212° Fahr. per lb. of coal.

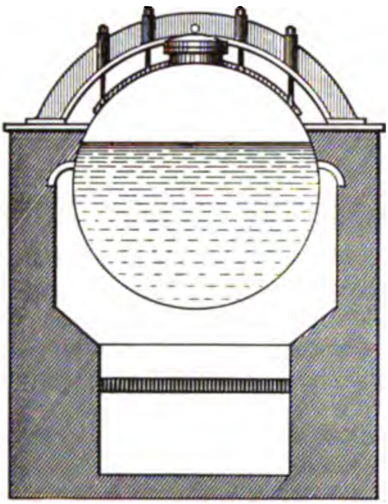


Fig. 174.—Setting of an egg-end boiler.

Fig. 175.—Section of Cornish boiler.

Cornish Boilers.—The Cornish boiler shown in cross section in Fig. 175, and in longitudinal section in Fig. 176, is an efficient form of boiler for small power. It is of simple and durable construction, requires few repairs, and will burn inferior qualities of fuel with economy. It consists of

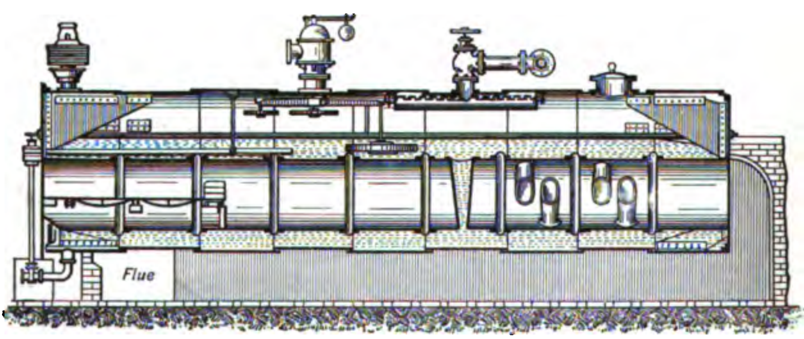


Fig. 176.—Longitudinal section of Cornish boiler.

a cylindrical shell with flat ends, having a furnace-tube placed inside, extending from end to end of the boiler. The furnace-tube is fitted with Galloway-cone circulating-tubes. The fire is placed on a grate fixed inside the front end of the furnace-tube. The end-plates are each formed in one piece; the hole for the furnace-tube is cut out in a lathe. The longitudinal seams of the shell are double riveted, and the circumferential seams single riveted. The longitudinal seam of each belt of plates of the furnace-tube is welded.

The diameter of the furnace-tube is generally = diameter of boiler × ·5 to ·6. The water-space between the bottom of the furnace-tube and the shell should not be less than 6 inches. The length of the boiler is generally equal to from 3 to 4 times the diameter of the shell.

This boiler works well, even with moderately impure feed-water, because impurities deposited as sludge by the water tend to settle below the intensely heated parts, or at the bottom of the boiler, whence it can be removed by blowing off, and what little settles on the furnace-tube may be removed when the boiler is cleaned.

The results of a number of evaporative tests of Cornish boilers are given in Table 80, page 353.

Heating-surface of Cornish Boilers.—The heating-surface of a Cornish boiler, fitted with the usual number of Galloway tubes in the furnace-tube, may be estimated approximately by the following *Rule*:—

Effective surface in square feet =

(diameter of boiler in feet)^2 × length of boiler in feet × ·55.

For instance, the effective heating-surface of a Cornish boiler of 5 feet diameter and 20 feet long, having a furnace-tube equal in diameter to one-half that of the boiler, and fitted with Galloway-tubes, is approximately = 5 × 5 feet diameter × 20 feet long × ·55 = 275 square feet.

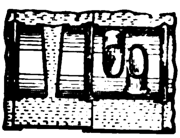


Fig. 177.—Galloway cone-tubes.

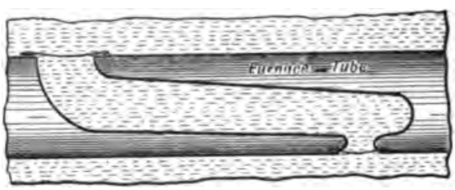


Fig. 178.—Water-leg in a furnace-tube.

The Circulation of the Water in Cornish Boilers is very defective, and there is considerable difference of temperature in different parts of the boiler. In getting up steam, the top of the boiler becomes much hotter than the bottom, and causes unequal expansion and injurious straining of the structure. This evil is mitigated by the employment of Galloway cone-tubes in the furnace-tube, as shown in Fig. 177, which improve the circulation and increase the temperature of the dead-water below the tube.

Many devices have been employed to remedy the sluggish circulation under the furnace-tube, and render the temperature more uniform throughout the boiler, but they have not proved to be of much advantage. A device for this purpose is shown in Fig. 178, but it has not proved so effective in practice as ordinary cone-circulating-tubes in the furnace-tube. It consists of a water-leg fixed in the furnace-tube a little beyond the fire-bridge, connecting the water-space above the furnace-tube with that below it. In another device a current of water from above to below the furnace-tube is induced by means of an injector.

Lancashire Boilers.—The Lancashire boiler is a favourite type of boiler for supplying steam for factory-engines. It is economical, durable, inexpensive in repairs, and having a large evaporative surface produces a good and steady supply of practically dry steam.

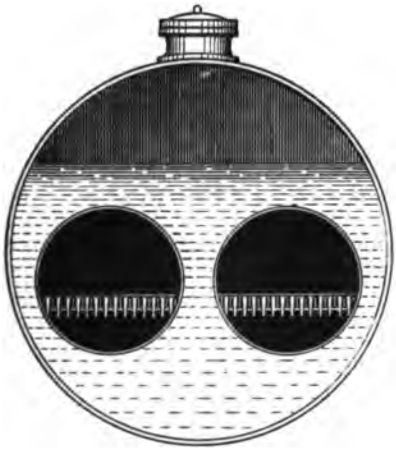


Fig. 179.—Cross-section of Lancashire boiler.

It consists of a cylindrical shell with flat ends, having two internal furnace-tubes extending from end to end of the shell. A cross-section of the boiler is shown in Fig. 179, and a longitudinal section in Fig. 180, The shell is composed of rings or belts of plates about 3 feet wide, arranged alternately in outer and inner belts, with single-riveted circumferential seams. Each belt is preferably formed of one plate, with longitudinal seams placed right and left alternately of the top centre-line of the boiler and at a suitable distance from it. When arranged in this manner the longitudinal seams are not concealed by the brickwork-setting. The longitudinal seams are lap-jointed and double-riveted for moderately high pressures, and butt-jointed with double butt-straps treble-riveted for high-pressures of steam.

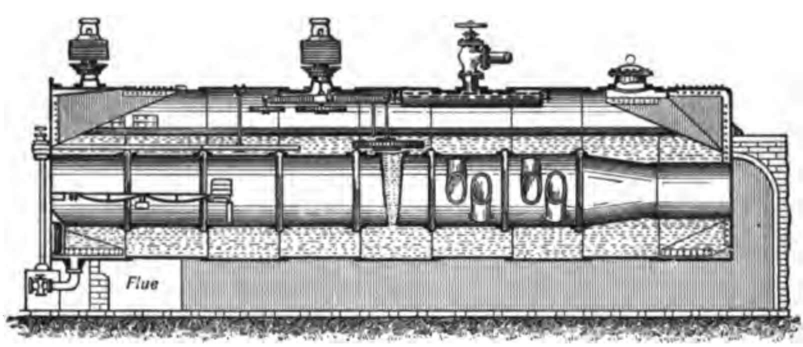


Fig. 180.—Longitudinal section of Lancashire boiler.

The end-plates are each formed in one piece. The back end-plate is either flanged to join the shell, as shown in Fig. 181, or attached by an inside angle-ring, as shown in Fig. 182, to prevent injury from impingement of flame. The front end-plate is attached to the shell by an external angle-ring, as shown in Fig. 183, for the purpose of increasing its elasticity. The end-plates are usually from $\frac{1}{16}$ inch to $\frac{1}{8}$ inch thicker than the shell-plates. They are stayed to the shell by seven gusset-stays, five being placed above the furnace-tubes and two below them. The diameter of both the furnace-tubes and the shell is measured internally, that of the shell being taken inside the inner belt of plates.

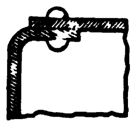


Fig. 181.—Back end-plate.

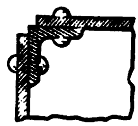


Fig. 182.—Back end-plate.

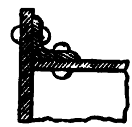


Fig. 183.—Front end-plate.

The diameter of the boiler is governed by the size of the furnace-tubes. The fire is placed on a grate fixed in the front end of the tubes. The size of each furnace-tube is limited by the width necessary for the maintenance of a good fire. The mimimum internal diameter of tube necessary for this purpose is 2 feet 9 inches. The internal diameter of each furnace-tube is equal to the internal diameter of the shell of the boiler multiplied by ·4. For instance, the diameter of each furnace-tube for a shell 7 feet 6 inches internal diameter is $7{\cdot}5 \times {\cdot}4 = 3$ feet. The space between the furnace-tubes should not be less than 5 inches, and that between the furnace-tube and the shell not less than 4 inches, in order to permit free circulation and facilitate cleaning.

The furnace-tubes are generally $\frac{3}{8}$ inch thick for a working-pressure of steam up to 70 lbs. per square inch; $\frac{13}{32}$ inch thick for 85 lbs. pressure per square inch; $\frac{7}{16}$ inch thick for 100 lbs. pressure per square inch; and $\frac{1}{2}$ inch to $\frac{9}{16}$ inch for higher pressures.

Each belt of plates forming a furnace-tube, consists of a single plate bent to the required diameter and welded at the longitudinal seam, to secure a truly circular tube. The circumferential seams are formed either with Adamson's flanged seams, or strengthening hoops of bridge-section. It is necessary to allow breathing room all round the outside of the ends of the furnace-tubes, to permit the end-plates to spring and accommodate the greater expansion of the tubes than that of the shell. This is obtained by leaving a space of 10 or $10\frac{1}{2}$ inches between the centres of the bottom rivets of the gusset-stays and the centres of the rivets securing the furnace-tubes to the end-plates. The furnace-tubes are reduced in diameter at the back end, to make room for the attachment of the back end-plate by an inside angle-ring. This arrangement permits a man to get down between the tubes to the bottom of the boiler. The furnace-tubes are generally fitted with Galloway tubes, which are either riveted or welded into their places.

The confined space between the furnace-tubes makes it difficult to clean Lancashire boilers. The cleaning would be greatly facilitated if the diameter of the furnace-tubes were reduced about one-sixth in diameter for two-thirds of the length from the back-end of the boiler. For instance, the furnace-tubes of a Lancashire boiler of 7 feet 6 inches in diameter, and 30 feet in length, might be 36 inches diameter for 10 feet in length from the front-end of the boiler, and $= 36 - 6 = 30$ inches diameter for the remaining 20 feet in length of the tubes.

Two longitudinal stay-bolts are employed to assist the end-plates, and relieve the strain on the gusset-stays. They are frequently $1\frac{7}{8}$ inches diameter, with swelled screwed-ends secured to each end-plate by inside and outside nuts and washers, and they are placed sufficiently high above the furnace-tubes to prevent restriction of the elasticity of the end-plates. The manhole in the shell is strengthened by a strong raised mouthpiece, and the mud-hole beneath the furnace-tubes is also fitted with a mouthpiece. Fitting-blocks of the same material as the boiler are provided for the mountings.

The length of the boiler is generally equal to about four times the diameter of the shell. It was found in some experiments that no economy was gained by making the length greater than this.

As it is generally difficult to provide space in London for ordinary Lancashire boilers of four diameters long, a shorter Lancashire boiler is frequently employed there, equal to from two and one-half to three diameters long. Lancashire boilers of this length do not present a sufficiently long run of heating-surface for the fuel-gases to become efficiently absorbed, and much heat passes to waste up the chimney. When short boilers are used they should be fired with fuel which develops the greater portion of its heat on the fire-grate, and little passes away in the products of combustion.

The results of evaporative tests of a number of Lancashire boilers are given in Table 80, page 353.

Heating-Surface of Lancashire Boilers.—The heating-surface of a Lancashire boiler, fitted with the usual number of Galloway tubes in the furnace-tubes, may be estimated approximately by this *Rule* :—

$$\text{Effective heating-surface of a Lancashire boiler in square feet} = \\ (\text{Diameter of boiler in feet})^2 \times \text{length of boiler in feet} \times \cdot 63.$$

For instance, the effective heating-surface of a Lancashire boiler of 7 feet diameter and 28 feet long, with five Galloway tubes in each furnace-tube, is approximately $= 7 \times 7$ feet diameter $\times$ 28 feet long $\times \cdot 63 =$ 865 square feet.

Furnace-Mouthpiece of Lancashire Boilers.—The mouthpieces of the furnace-tubes of Cornish and Lancashire boilers are in some cases made of cast-iron, but it is not a suitable material for this purpose. The castings are liable to fracture, and they generally cover up so much of the end-plate as to interfere with the inspection of the joint of the furnace-tube with the end-plate. The furnace mouthpiece is best formed of two plates of wrought-iron or mild-steel, with an air-space between them. The outer plate is fitted in the end of the tube, and the joint is covered with a brass bead, placed inside the ring of rivets which attach the tube to the end-plate, as shown in Fig. 184. It is fitted internally with an arch-piece surrounding the fire-door, supporting a lining of fire-clay for protecting the front of the mouthpiece.

The furnace-door is generally about 12 inches high, and from 15 to 20 inches wide, according to the size of the boiler. It is provided with a grid for admitting air above the fire, which is distributed through a perforated box baffle-plate attached to the door. The perforations are usually $\frac{3}{16}$ or $\frac{1}{4}$ inch in diameter, and have an aggregate area of from 2 to 3 square inches per square foot of fire-grate surface. Some kinds of coal, however, require a greater space than this for the admission of air above the fire, and the area of perforations necessary to secure economical combustion is in some cases from one-thirtieth to one-fortieth of the area of the fire-grate.

Fig. 184.—Mouthpiece of furnace.

Brick-work-Setting of Cornish and Lancashire Boilers.—It is important to have boilers properly seated on a dry foundation. Cornish and Lancashire boilers are set in brick-work, lined with fire-brick on the surfaces exposed to heat. The brick-work should be laid on a concrete foundation of at least six inches thick, and be so constructed as to admit of being readily taken down in parts where required for inspecting and repairing the boiler, and it should be arranged to prevent harborage of moisture. To prevent the lodgment of moisture in contact with the plates the boiler is best set on fire-brick blocks, elevated above the bottom of the side flues, and having narrow surfaces for the boiler to rest upon, as shown in Fig. 185. The blocks should be recessed at the ring-seams to permit examination of the seams. The recess should be filled with fire-clay, which can be removed when the boiler is inspected.

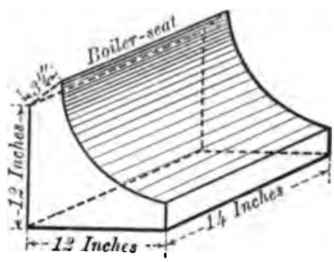


Fig. 185.—Boiler seating-block.

The flues should be large enough for a man to creep easily along from end to end when clearing or inspecting the boiler. They should be so constructed as to facilitate external examination of the shell, and to conceal as little of the surface of the plates of the boiler as possible. The side-flues should be from 9 to 10 inches wide at the top and be covered in with curved blocks of fire-bricks, as shown in Fig. 186, placed a little above the top of the furnace-tube. They should extend down to the level of the bottom of the boiler. The bottom flue should not be less than two feet deep at the vertical-centre line of the boiler, and its width should be equal to the radius of the shell. When the width of flue is thus proportioned, the angle that the bearing-surface of the heating-blocks makes with the horizon is 30 degrees for any diameter of boiler.

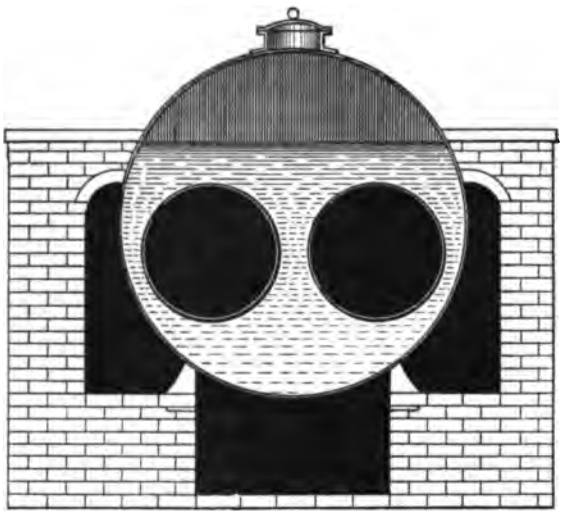


Fig. 186.—Section of Lancashire boiler and brickwork setting.

The best results are obtained when the flues are so arranged that the products of combustion pass from the furnace-tubes to the flue under the bottom of the boiler, near the front end of which the gases are divided into two currents, one of which passes along each side-flue, and the currents unite in the main flue leading to the chimney. In this arrangement the bottom of the boiler is maintained at a temperature more nearly approaching to that of the top, the dead-water underneath the furnace-tube is more effectively heated, and greater uniformity of temperature is obtained throughout the boiler, than when the products of combustion pass first through the side-flues and last under the bottom of the boiler.

The down-take flue at the back end of the boiler should be about 2 feet 6 inches wide lengthways of the boiler, and it should be covered in either with curved blocks of fire-brick or with an iron plate covered with brick-work. This flue should be divided into two compartments by a narrow wall between the furnace-tubes, so that the currents of the gases from the furnace-tubes may not unite until they reach the bottom flue, in order to prevent disturbance of the draught.

The thickness of the walls of the sides of the flues may be 14 inches, and the depth of brickwork above the side-flue may be 9 inches. Fire-clay should be used for jointing the bricks to the plates of the boiler. Mortar is not suitable for this purpose, because it readily absorbs moisture and induces corrosion. Care should be taken to prevent the formation of holes in the brickwork through which air may pass into the flues. The face of the brickwork at the front of the boiler should be set back 6 inches, so as to leave the angle-ring and its rivets open for ready inspection.

The flues of Cornish and Lancashire boilers are frequently cramped with the object of keeping the fuel-gases in contact with the plates of the boiler, but moderately wide flues are quite as efficient as narrow flues, and they permit ready access to the boiler. The hotter gases being lighter always rise to the top of the flues, whatever be their size or shape. Narrow flues do not permit proper cleaning of the heating-surfaces, which become covered more or less thickly with a permanent coating of soot, and results in their heat-transmissive efficiency becoming considerably reduced.

Lancashire Boilers shown in the Frontispiece. — The frontispiece shows four of a range of six Lancashire boilers each 7 feet 6 inches in diameter and 30 feet long, of mild-steel plates, a longitudinal section

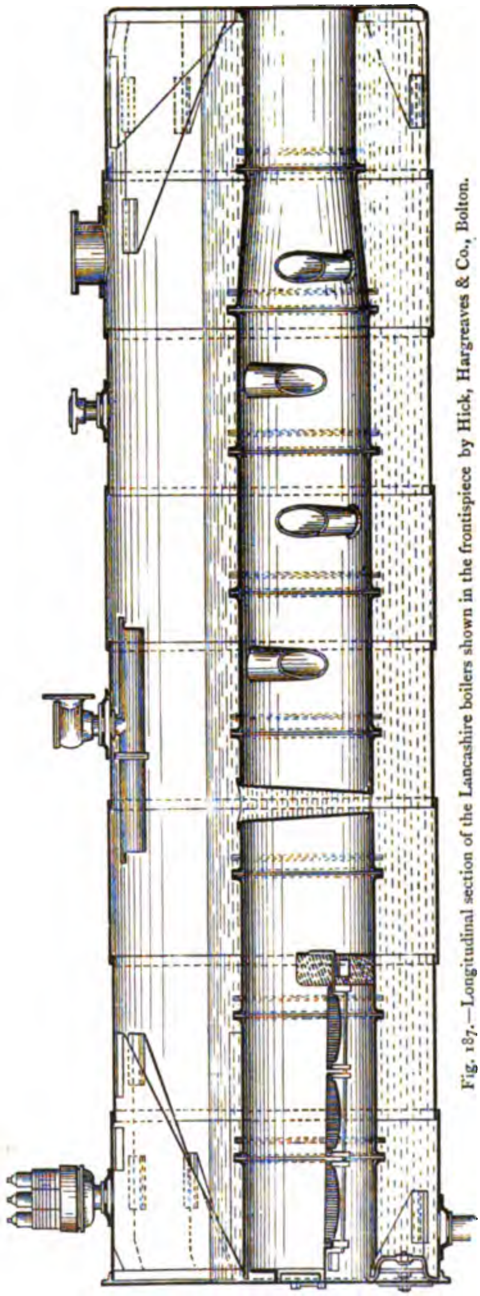


Fig. 187.—Longitudinal section of the Lancashire boilers shown in the frontispiece by Hick, Hargreaves & Co., Bolton.

of which is shown in Fig. 187. The furnace-tubes are 3 feet diameter, and are each fitted with five conical circulating tubes. They are $\frac{13}{32}$ inch in thickness, welded longitudinally, and have flanged transverse joints. The shell-plates are $\frac{7}{16}$ inch thick, each ring being formed of one plate, with longitudinal double-riveted butt-joints, having inside and outside butt-straps. The end-plates are each in one piece, $\frac{9}{16}$ inch thick, with openings for furnace-tubes bored out, and the edges of the plates are turned. All the rivet-holes were drilled in position after the plates were bent. The edges of the plates are planed. The seating-blocks for the mountings are of steel. The test-pressure of the boiler was 150 lbs. per square inch, and the working pressure is 80 lbs. per square inch.

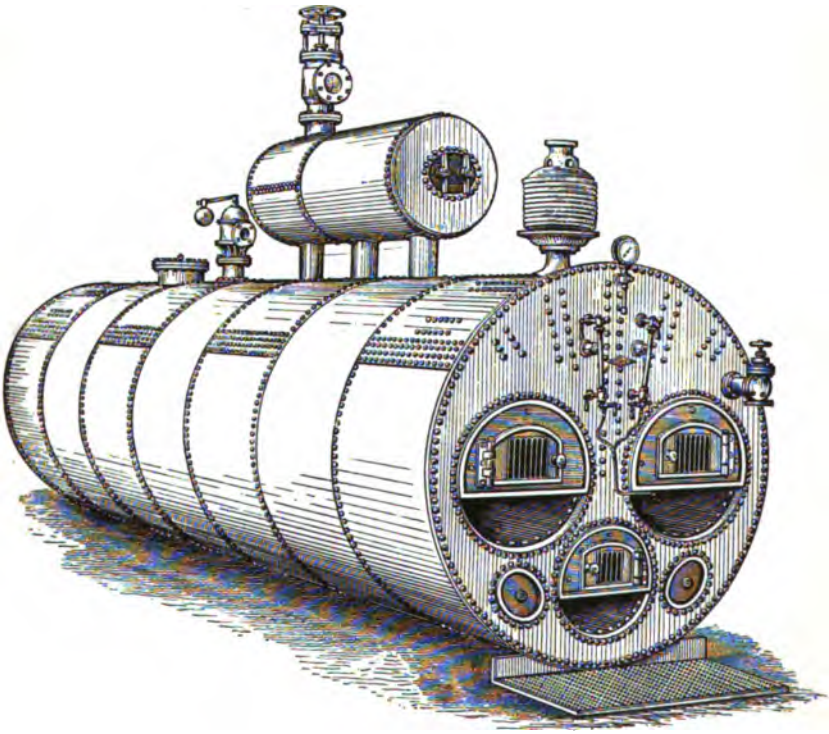


Fig. 188.—Three-flued boiler by Yates & Thom, Blackburn.

Three-Flued Boiler.—A modified Lancashire boiler is shown in Fig. 188. It is 8 feet diameter and 30 feet long, and has three furnace-tubes, two being 3 feet diameter, and one 2 feet 3 inches diameter. The advantages claimed for this boiler over an ordinary Lancashire boiler are that, it contains a larger heating-surface; a large volume of dead-water is displaced by the bottom furnace-tube; a more efficient circulation, and greater uniformity of temperature is obtained throughout the boiler, resulting in a considerable reduction in strains from unequal expansion and contraction. The bottom furnace-tube forms an efficient stay for the bottom portion of the end-plates of the boiler.

The total heating-surface of the boiler is 950 square feet, and the area of the fire-grate is 48 square feet. The boiler is set in a similar way to an ordinary Lancashire boiler, but a hood is fixed at the back-end of the bottom furnace-tube to prevent its draught being baffled by that from the furnace-tubes above it.

Average Evaporative Capacity of Lancashire and other Stationary-Boilers.—The average evaporative capacity and power of a few general sizes of stationary-boilers of different kinds are given in the following Table:—

TABLE 61.—AVERAGE EVAPORATIVE CAPACITY AND POWER OF CORNISH, LANCASHIRE, AND THREE-FLUED BOILERS.

Description of Boiler.	Size of Boiler. Diameter.	Size of Boiler. Length	Evaporation in lbs. of water per hour.	Indicated Horse-Power of good modern compound engine the boiler will supply steam for.
	Ft. In.	Feet.	Lbs.	I. H. P.
Cornish Boiler	4 6	15	1300	75
Cornish Boiler	5 0	20	2050	120
Cornish Boiler	5 6	21	2600	150
Lancashire Boiler . . .	7 0	28	5000	300
Lancashire Boiler . . .	7 6	30	6000	350
Lancashire Boiler . . .	8 0	30	6600	390
Lancashire Boiler . . .	8 6	30	7300	430
Yates & Thom's Three-Flued Boiler	8 0	30	8300	480
Yates & Thom's Three-Flued Boiler	8 3	30	8600	500
Yates & Thom's Three-Flued Boiler	8 6	30	9000	530

The evaporative capacities of these boilers are those generally obtained in practice with easy-firing; they may be increased by hard-firing. The last column of the Table gives the maximum power of engines the boilers are in a general way suitable for.

Weight of Cornish, Lancashire, and Three-Flued Boilers.—The weights of a few general sizes of Cornish, Lancashire, and Three-flued boilers of mild-steel, are given in Table 62.

Galloway-Boilers.—The Galloway-boiler, shown in Fig. 189, is an excellent and very economical steam-generator. It consists of a cylindrical shell with two circular furnace-tubes at the front-end, being in this respect similar to a Lancashire-boiler. These tubes unite beyond the fire-bridges in a special form of flue-tube, which has the upper and lower surfaces struck from one radius. This flue-tube contains a number of cone-tubes and side-pockets. All the tubes are alike, and have their flanges square to the centre-line of the tubes.

TABLE 62.—WEIGHT OF CORNISH, LANCASHIRE, AND THREE-FLUED BOILERS OF MILD-STEEL, WITH THE USUAL NUMBER OF CROSS-TUBES IN THE FURNACE-TUBES, FOR A WORKING-PRESSURE OF STEAM OF 100 LBS. PER SQUARE INCH.

Description of Boiler.	Diameter.		Length.	Thickness of Shell-Plates.	Approximate Weight.	
					Weight of Boiler.	Weight of Fittings.
	Feet.	Inch.	Feet.	Inch.	Tons.	Tons.
Cornish Boiler	4	6	15	$\frac{3}{8}$	4	1
Cornish Boiler	5	0	20	$\frac{3}{8}$	$5\frac{1}{2}$	$1\frac{1}{8}$
Cornish Boiler	5	6	21	$\frac{13}{32}$	$6\frac{1}{2}$	$1\frac{1}{2}$
Lancashire Boiler . . .	7	0	28	$\frac{1}{2}$	15	$2\frac{3}{4}$
Lancashire Boiler . .	7	6	30	$\frac{17}{32}$	17	3
Lancashire Boiler . . .	8	0	30	$\frac{9}{16}$	$18\frac{1}{2}$	$3\frac{1}{4}$
Lancashire Boiler . .	8	6	30	$\frac{5}{8}$	$21\frac{1}{2}$	$3\frac{1}{2}$
Yates & Thom's Three-Flued Boiler	8	0	30	$\frac{9}{16}$	21	$3\frac{3}{4}$
Yates & Thom's Three-Flued Boiler	8	3	30	$\frac{19}{32}$	$23\frac{1}{2}$	$3\frac{3}{4}$
Yates & Thom's Three-Flued Boiler	8	6	30	$\frac{5}{8}$	$24\frac{3}{4}$	4

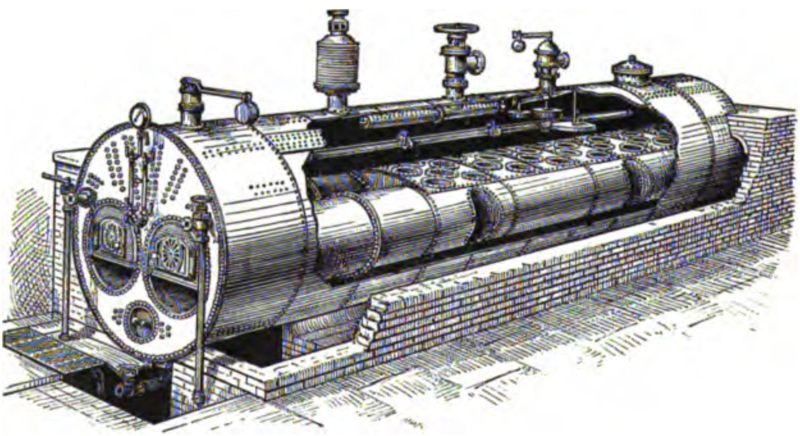


Fig. 189.—Galloway boiler by Galloways Limited, Manchester.

The cone-tubes and side-pockets effectually break up the current of the fuel-gases, and absorb a considerable quantity of heat. The meeting of the fuel-gases immediately behind the fire-bridges aids their combustion, and conduces to economy of fuel and the prevention of smoke.

The circulation in this boiler being very rapid and efficient, it is not necessary to run the products of combustion under the bottom of the boiler directly after leaving the flue-tube to obtain a more equable temperature of the water as in the case of Lancashire boilers. Therefore, the setting of the boiler is so arranged that the products of combustion on leaving the flue-tube at the back-end of the boiler divide and return along the side-flues to the front end, where they descend, re-unite, and pass under the bottom of the boiler to the main flue leading to the chimney.

The average evaporative capacity and power of several sizes of Galloway-boilers are given in the following Table:—

TABLE 63.—AVERAGE EVAPORATIVE CAPACITY OF GALLOWAY-BOILERS.

Size of Galloway-Boiler.	Evaporation in lbs. of water evaporated per hour.	Indicated Horse-Power of a good modern compound engine the boiler will supply steam for.
6 feet 6 inches diameter, and 22 feet long	4200	250
7 feet diameter, and 28 feet long	6000	350
8 feet diameter, and 30 feet long	8000	470

Weight of Galloway-Boilers.—The weights of three sizes of Galloway-boilers of mild-steel are given in the following Table:—

TABLE 64. WEIGHT OF GALLOWAY BOILERS OF MILD-STEEL FOR A WORKING-PRESSURE OF STEAM OF 100 LBS. PER SQUARE INCH.

Size of Boiler.			Thickness of Shell-Plates.	Approximate Weight.	
Diameter.		Length.		Weight of Boiler.	Weight of Fittings.
Feet.	Inch.	Feet.	Inch.	Tons.	Tons.
6	6	22	$\frac{7}{16}$	$10\frac{1}{2}$	$2\frac{1}{4}$
7	0	28	$\frac{1}{2}$	15	$2\frac{1}{2}$
8	0	30	$\frac{9}{16}$	20	3

The results of several evaporative tests of Galloway boilers are given in Table 80, page 353.

Modified Galloway-Boiler.—A modified type of Galloway boiler is shown in Fig. 190. The flue-tube is of the ordinary Galloway type, and is fixed in a circular shell. The front-end of the tube is formed with a fire-box similar to that of a locomotive boiler. The fire-box is nearly square, and sufficiently deep to admit of the fire-grate being placed low enough to permit a large amount of fuel to be heaped upon it, as is necessary for burning megass and other refuse-fuels.

The fire-box is stayed to the shell on all sides by screwed stays. The boiler is 6 feet 6 inches diameter, and 28 feet long, and contains 32 cone-tubes, 4 pockets, and 2 contracting pockets at the back-end. There is a manhole at the top, and a mudhole at the back, underneath the tubes, of the boiler. The fire-box is 6 feet 6 inches long, 6 feet wide, and 4 feet 9 inches high. There are 4 gusset-stays at each end, and the end-plate is flanged at the back-end, and at the front-end in the upper portion for attachment to the shell.

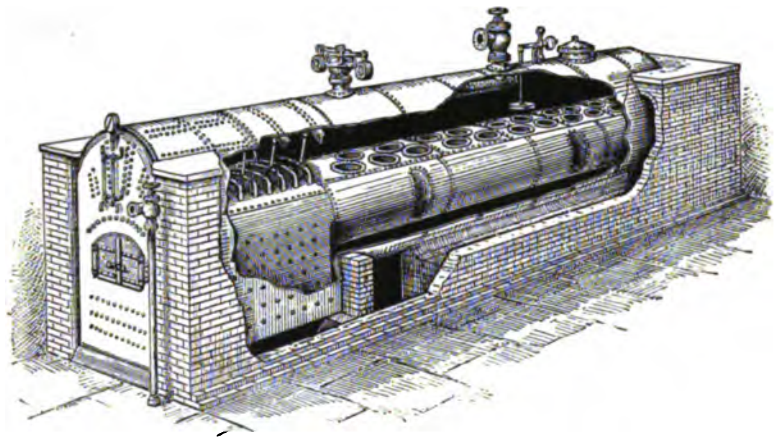


Fig. 190.—Modified Galloway-boiler by Galloways Limited, Manchester.

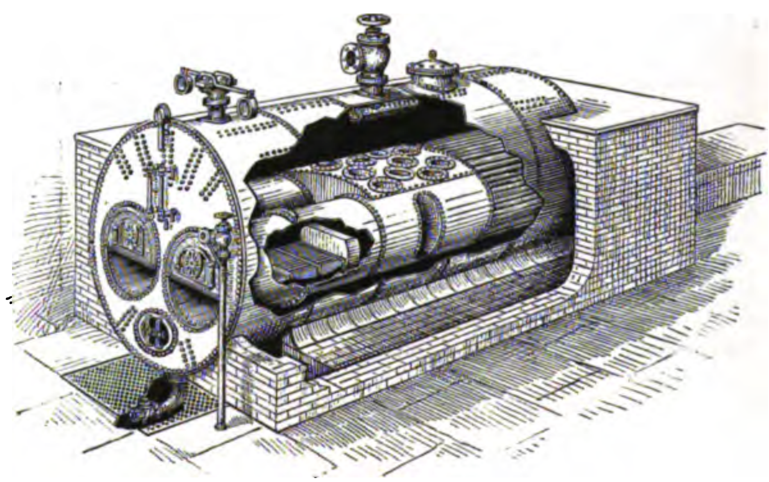


Fig. 191.—Compound Galloway-boiler by Galloways Limited, Manchester.

Compound Galloway-Boiler.—A compound Galloway boiler is shown in Fig. 191. It combines the Lancashire, the Galloway, and the multitubular boiler. It consists of three portions enclosed in a circular shell; the first portion consists of two plain furnace-tubes on the Lancashire principle. The products of combustion pass from the furnace-tubes into a chamber of the Galloway-type, containing cone-tubes, forming the second portion, and thence they pass through a number of smoke-tubes, forming the third portion.

The boiler is 7 feet 6 inches diameter, and 18 feet long; the two furnace-tubes are three feet diameter; the oval chamber contains 8 cone-tubes; there are 136 smoke-tubes of 3 inches external diameter; the end-plates are stayed by five gussets at each end, and 4 longitudinal stays. There is a manhole on the top of the shell, and a mudhole at the front-end, underneath the furnace-tubes. The shell of the boiler is constructed in the ordinary manner; the front end-plate is attached to the shell by an outside angle-hoop, and the back tube-plate is flanged to the shell.

This boiler contains great power in a small space. It can be set without the flame traversing the boiler-shell, an important point in some situations.

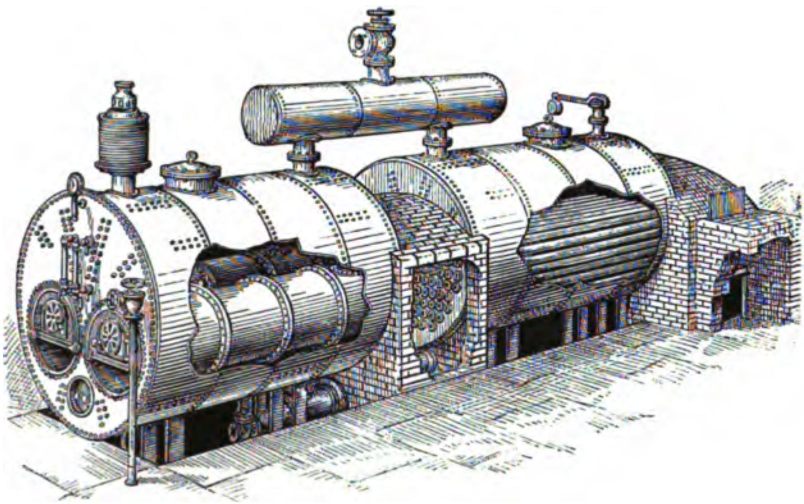


Fig. 192.—Compound Lancashire and multitubular boiler by Galloways, Limited, Manchester.

Compound Lancashire and Multitubular Boiler.—A compound Lancashire and multitubular boiler is shown in Fig. 192. This boiler consists of two portions of circular form. The front portion contains two ordinary furnace-tubes, and the second portion contains a number of smoke-tubes. There is a space 4 feet long between each portion of the boiler; it is enclosed by brickwork, and forms a capacious combustion-chamber. The portions of the boilers are connected by a pipe of large diameter placed underneath them, so as to maintain an equal water-level, and also for clearing away sediment; and the upper parts are connected by a steam-chest, so that the steam is taken equally away from the two portions of the boiler.

The first portion is 7 feet diameter, and 12 feet long, with 2 furnace-tubes of 2 feet 9 inches diameter, extending from end to end; the second portion is

7 feet diameter, and 12 feet long, and contains 98 smoke-tubes of $4\frac{1}{4}$ inches external diameter. The connecting-pipe at the bottom of the boiler is 10 inches diameter, and 9 feet long, attached by two 10 inch diameter stand-pipes to the bottom of the boiler. The steam-chest connecting the upper portions is 2 feet 6 inches diameter, and 12 feet long.

This boiler is as powerful as an ordinary one of equal diameter and length, and the weight is divided into two portions, a convenience in transport.

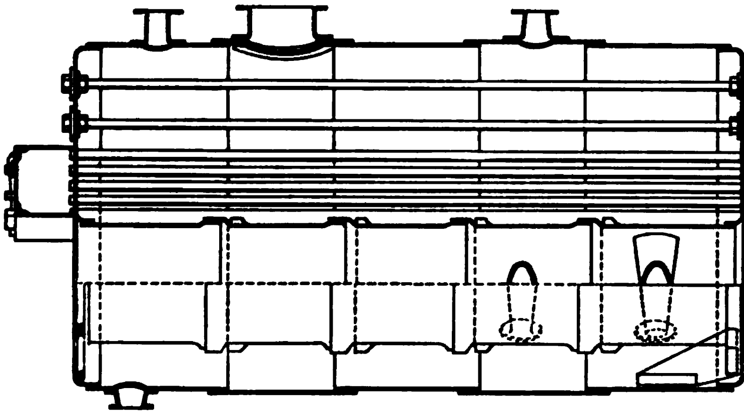


Fig. 193.—Stationary return-tube boiler by Davey, Paxman & Co., Colchester.

Multitubular Boilers.—Multitubular boilers of all kinds are excellent steam-generators when well-proportioned. They require to be fed with good water, otherwise the tubes become encrusted with mineral deposits from the water, and the evaporative efficiency of the boilers is thereby greatly reduced. When the feed-water is moderately pure the boiler may generally be maintained clean by blowing off a portion of the water two or three times a day, and by washing out the boiler occasionally with water under pressure from a hose. As the tubes are inaccessible for the purpose of inspection and cleaning they should be withdrawn, examined, cleaned, and replaced, once or twice a year, according to the scale-forming nature of the feed-water used.

In many multitubular boilers the run of the gases is too short to obtain efficient absorption of the fuel-gases by the heating-surfaces, and much heat escapes to waste up the chimney.

A stationary return-tube boiler is shown in Figs. 193 and 194. It is 7 feet 6 inches internal diameter, and 14 feet 6 inches long, with two furnace-tubes 2 feet 6 inches diameter, each fitted with two conical circulating-tubes at the back end. The circumferential seams of the furnace-tubes are formed with Paxman's strengthening and expansive joint. There are 74 smoke-tubes of 3 inches external diameter, and 14 feet 7 inches long. The products of combustion pass from the furnace-tubes into a brick combustion-chamber at the back-end of the boiler, and thence return through the smoke-tubes to a smoke-box at the front of the boiler. The total heating-surface of the boiler is 1219 square feet.

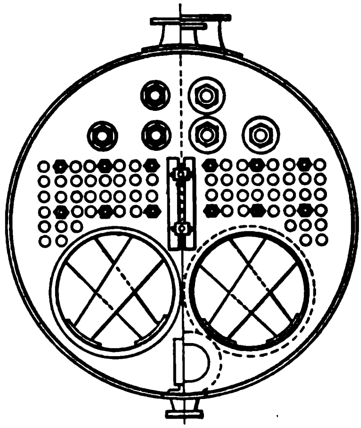


Fig. 194.—End-view of the boiler shown in Fig. 193.

A horizontal stationary cylindrical multitubular boiler is shown in Fig. 195. It is internally fired and has a large fire-box, surrounded by water-spaces, and fitted with a high bridge at the back of the fire, for the purpose of protecting the tube-plate from injurious action of flame, and of forming a combustion-chamber behind the fire to assist the combustion of the gases before they enter the tubes. The shell is 6 feet 6 inches diameter, and 16 feet long. There are 170 tubes of 3 inches external diameter and 7 feet 3 inches long. The heating-surface of the tubes is 967 square feet, and that of the fire-box 180 square feet, or a total heating-surface of 1147 square feet. The area of the fire-grate is 30 square feet. Air is admitted below the fire-grate through an ash-hole at the front of the boiler, and above the fire through perforations in the fire-door.

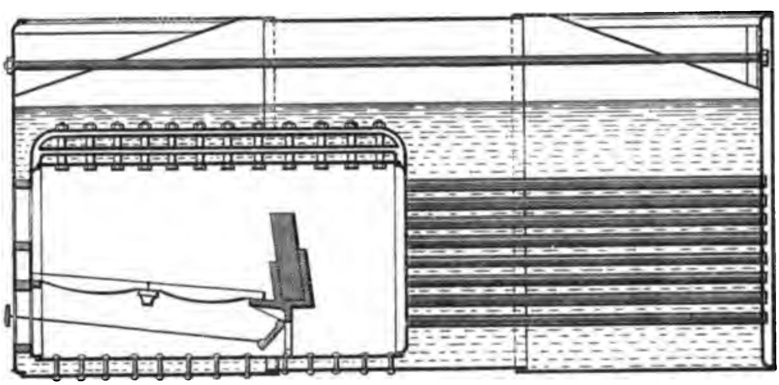


Fig. 195.—Horizontal cylindrical multitubular boiler.

Compound Lancashire and Multitubular Boiler.—A combined Lancashire and multitubular boiler is shown in Fig. 196. The furnace-tubes at the front-end of the boiler are arranged in the same manner as in a Lancashire boiler, and they terminate in a combustion-chamber connected by smoke-tubes to the back end of the boiler.

Stationary Boiler of Locomotive Type.—A stationary boiler of locomotive type with a long barrel is shown in Figs. 197 and 198. The fire-box is formed wider at the bottom than at the top. The fire-grate is 6 feet 6 inches by 4 feet 10 inches. The shell is 4 feet 2 inches diameter

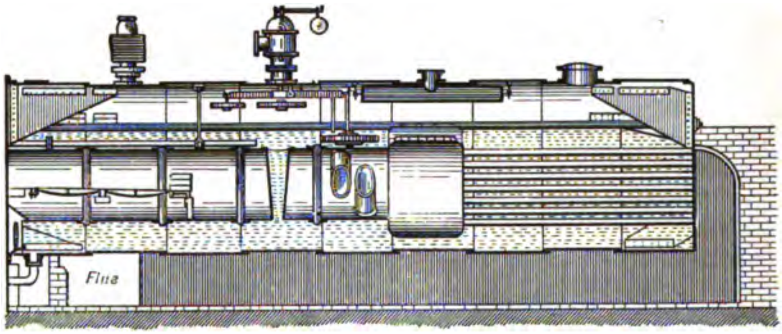


Fig. 196.—Compound Lancashire and multitubular boiler.

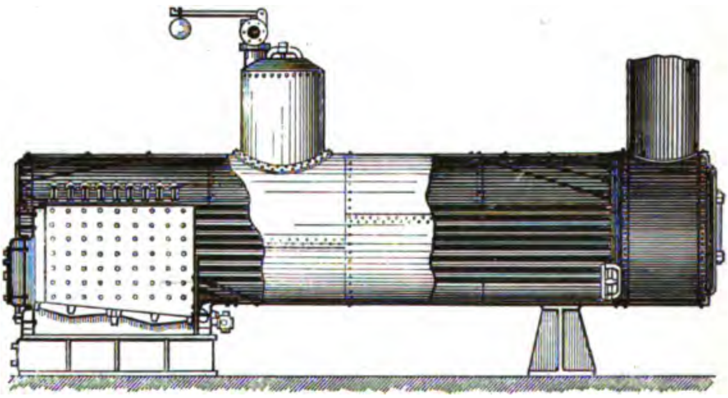


Fig. 197.—Stationary boiler of locomotive type.

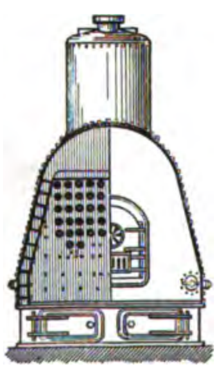
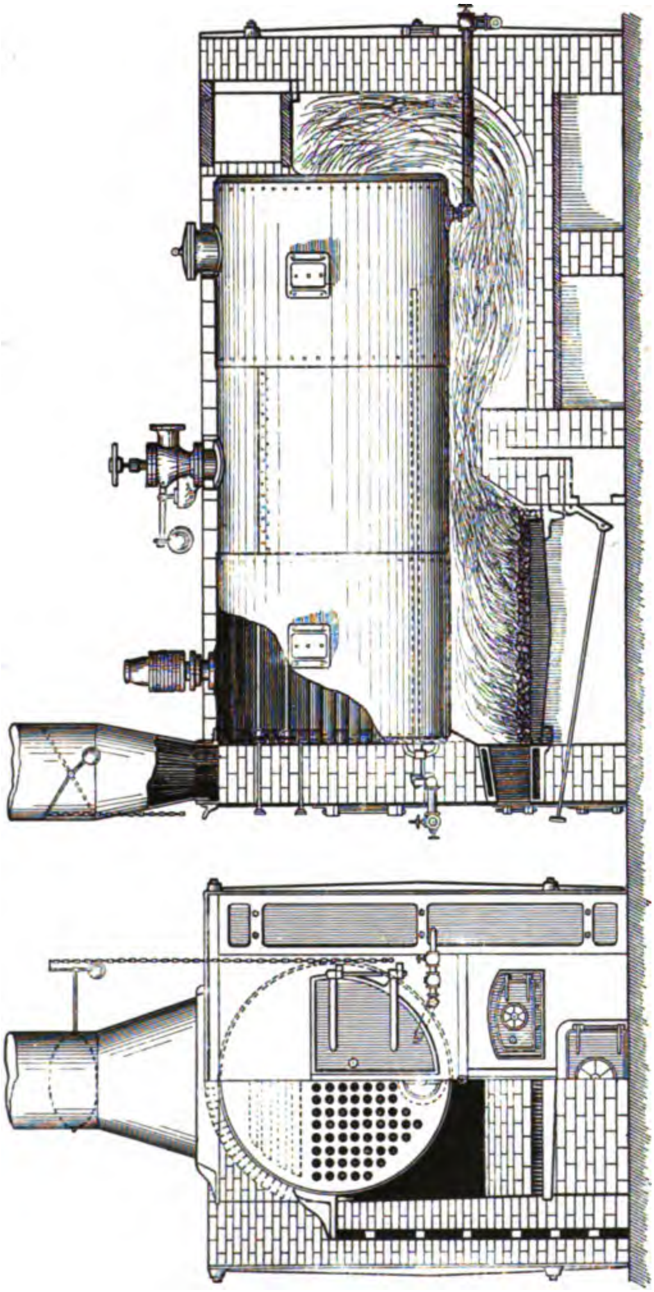


Fig. 198.—End-view of the boiler shewn in Fig. 197.

and 15 feet long. The total heating-surface is 844 square feet. The area of the fire-grate surface is 31·41 square feet.

Externally-Fired Multitubular Boilers.—An externally-fired stationary multitubular boiler is shown in Figs. 199 and 200. It consists of a cylindrical shell with flat ends, having a number of smoke-tubes extending from end to end of the boiler and communicating with a smoke-box surmounted by a chimney at the front end of the boiler. The tubes are generally from 3 to 5 inches diameter. The fire-grate is placed from 20 to 30 inches below the bottom of the boiler, according to the size of boiler and the nature of the coal. The top of the



Figs. 159 and 200.—Externally-fired stationary multitubular boiler.

fire-bridge is generally placed not less than 12 inches below the bottom of the boiler. A second bridge is sometimes placed near the back-end of the boiler of the same height as the fire-bridge, in order to form a combustion-chamber to assist the combustion of the fuel-gases. This bridge is sometimes perforated and carried up to the bottom of the boiler. A door is placed in the brick-work at the back of the boiler, at the floor-line, for removing soot and ashes.

The products of combustion pass along the bottom of the boiler, and return through the smoke-tubes to the chimney. The boiler is set in brick-work having air-spaces in the walls, to accommodate expansion from heat. The air-spaces also partly prevent radiation of heat. The fire-bridge is split to admit air behind the fire, to assist the combustion of the fuel-gases and the prevention of smoke.

This type of boiler is a good steam-generator, but it has the objection of all externally-fired boilers that sediment is liable to accumulate on the plates at the bottom of the boiler, causing overheating and burnt plates. It should only be employed when the feed-water is very pure and deposits little sediment, or when it is purified before entering the boiler.

The evaporative performance of these boilers is given at page 354.

Heating-Surface of Externally-Fired Cylindrical Multitubular Boilers.—From one-half to two-thirds the circumference of the shell of these boilers is exposed to the action of heat, according to the manner in which they are set.

The heating-surface of boilers of average proportions, arranged as shown in Figs. 199 and 200, may be found approximately by the following *Rule*:—

Total heating-surface in square feet of externally-fired cylindrical multitubular boilers =

$$(\text{Diameter in feet})^2 \times \text{Length of boiler in feet} \times 2{\cdot}25.$$

For instance, the heating-surface of a boiler of this type, 5 feet diameter and 13 feet long, is approximately = 5 × 5 feet × 13 feet × 2·25 = 732 square feet.

Weight of Externally-Fired Multitubular Boilers.—The weights of a few sizes of externally-fired cylindrical multitubular boilers as shown in Figs. 199 and 200, are given in Table 65.

Boiler with special Furnace for Burning Refuse-Fuels.—An externally-fired stationary multitubular boiler with a furnace specially arranged for the combustion of small coal of inferior quality, and refuse-fuels of various kinds, is shown in Fig. 201. The air for combustion is admitted at the back-end of the boiler. Part of the air passes through an opening under the fire-bridge to the fire-grate, and a portion is admitted behind the fire through a channel at the top of the bridge, and also through perforations in the bottom of the flue at the back of the bridge, as shown in the engraving.

TABLE 65.—WEIGHT OF EXTERNALLY-FIRED CYLINDRICAL MULTITUBULAR BOILERS OF MILD-STEEL PLATES, AS SHOWN IN FIGS. 199 AND 200, FOR A WORKING-PRESSURE OF 100 LBS. PER SQUARE INCH.

Diameter of Boiler.		Length of Boiler outside the Tube-Plates.	Number of Smoke-Tubes.	Diameter of Smoke-Tubes.	Thickness of Shell-Plates.	Approximate Weight.			
						Weight of Boiler.		Weight of Fittings.	
Feet.	Inch.	Feet.		Inch.	Inch.	Tons.	Cwt.	Tons.	Cwt.
3	0	8	30	3	$\frac{5}{16}$	1	5	1	5
3	6	9	38	3	$\frac{5}{16}$	1	15	1	6
4	0	10	52	3	$\frac{3}{8}$	2	5	1	8
4	3	11	56	3	$\frac{3}{8}$	2	15	1	11
4	6	12	32	4	$\frac{3}{8}$	3	4	1	14
5	0	13	42	4	$\frac{3}{8}$	4	2	2	0
5	6	14	54	4	$\frac{13}{32}$	5	8	2	6
6	0	15	72	4	$\frac{13}{32}$	6	10	2	12

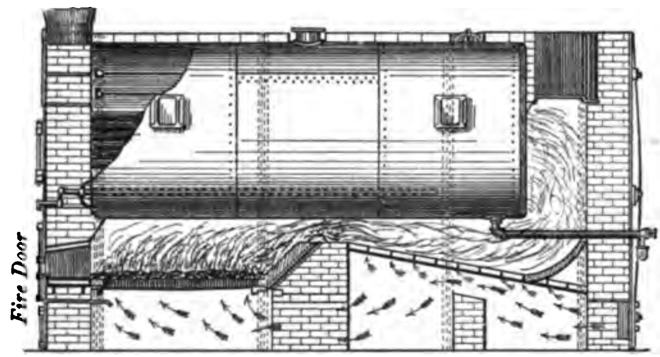


Fig. 201.—Externally-fired stationary boiler with Weitmyer's furnace.

Locomotive Boilers.—This design of boiler permits the employment of a shell of comparatively small diameter, containing a large amount of heating-surface. The flat surfaces of the fire-box can be rendered equal in strength to the shell by stays, and it is a strong and efficient form of boiler. The belts or rings of plates forming the barrel of a locomotive boiler are arranged telescopically when lap-jointed, or in rings of diminishing diameter, the first of which is lapped inside the fire-box shell or casing, the second inside the first, and the third inside the second. When the ring-seams are butt-jointed they are of equal diameter and are united by an outside covering-strip. The ring-seams are generally single-riveted. The longitudinal seams are butt-jointed with both inside and outside covering-strips double-riveted.

A locomotive boiler for a working-pressure of steam of 160 lbs. per square inch is shown in Fig. 202. The barrel is composed of mild-steel plates $\frac{9}{16}$ inch thick; the external diameter of the smallest plate is 4 feet 4 inches. The length of the barrel is 10 feet 4 inches. The fire-box casing is 6 feet long, and 4 feet wide externally at the bottom; the top and sides are in one plate of $\frac{9}{16}$ inch thick. The fire-box is composed of copper-plates $\frac{1}{2}$ inch thick, and it is 5 feet 3 inches long, 3 feet $6\frac{1}{2}$ inches wide at the top, and 3 feet $3\frac{5}{8}$ inches wide at the bottom; the height from the top of the fire-bars to the crown is 5 feet 6 inches at the tube-plate. The distance between the crown of the fire-box and that of the casing is 1 foot $7\frac{1}{2}$ inches.

The fire-box tube-plate is of copper $\frac{7}{8}$ inch thick where the tubes and stays pass through, and the remaining portion is reduced to $\frac{5}{8}$ inch thick. The fire-box is stayed to the fire-box casing on all sides with copper stays pitched 4 inches. The stays are 1 inch diameter, of 14 threads to the inch, screwed steam-tight into the plates of both the fire-box and its casing, and have their ends riveted over to form heads. The thread is turned off that portion of the stay which is in the water-space. The crown of the fire-box is supported by eight girder stays secured by bolts to the crown-plate, and slung from the fire-box casing.

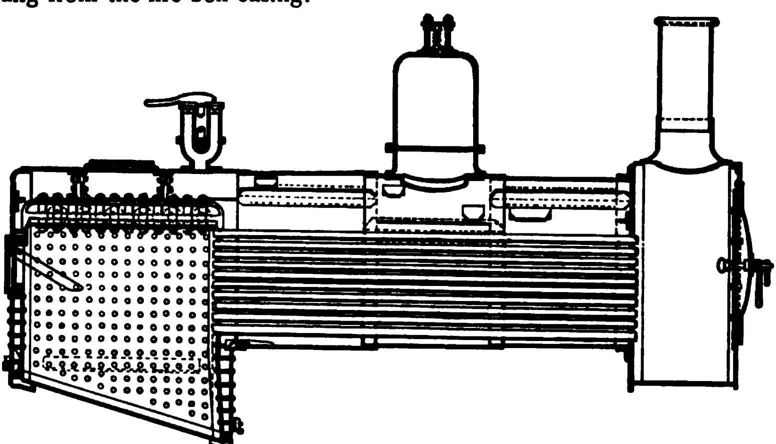


Fig. 202.—Locomotive boiler.

The smoke-box is 2 feet 10 inches long outside, 5 feet 2 inches wide, the side plates are $\frac{5}{16}$ inch thick and the front-plate is $\frac{1}{2}$ inch thick. The chimney is 1 foot 4 inches diameter, and 13 feet 3 inches high from the rails.

The boiler contains 220 tubes of copper, of $1\frac{5}{8}$ inch external diameter. The total heating-surface of the boiler is 1103 square feet. The area of the fire-grate surface is 17·4 square feet.

The results of a number of evaporative tests of locomotive boilers are given in Table 84, page 356.

Heating-Surface and Fire-Grate Surface of Locomotive Boilers.—The minimum area of the heating-surface and fire-grate surface of locomotive boilers burning good coal, requisite for economical evaporation, may be found by the following *Rules*:—

Total heating-surface in square feet of the boiler of a simple locomotive engine =

Area of one cylinder in square inches multiplied by 3·75.
Heating-surface of fire-box in square feet =
Total heating-surface in square feet divided by 10.
Area of fire-grate surface in square feet =
Total heating-surface in square feet divided by 60.
Area of tube-surface =
Area of total heating-surface — area of heating-surface of the fire-box.

The area of the air-space through the fire-door should not be less than one-seventy-fifth of the area of the fire-grate.

Example :—Required the area of heating-surface and fire-grate surface for the boiler of a locomotive having two cylinders of 19 inches diameter.

Then area of one cylinder = 19 × 19 inches × ·7854 = 283·529 square inches, and 283·529 × 3·75 = 1064 square feet, the total heating-surface required. The heating-surface of the fire-box is = 1064 ÷ 10 = 106·4 square feet. The heating-surface of the tubes is 1064 — 106·4 = 957·6 square feet. The area of the fire-grate surface is = 1064 ÷ 60 = 17·73 square feet.

When the heating-surface is calculated by the above *Rule*, the proportion of the heating-surface to the maximum indicated horse-power developed continuously by the locomotive, may be calculated approximately by assuming a piston-speed of 800 feet per minute, and a mean pressure of steam on the pistons equal to $\frac{1}{3}$ of the working pressure of steam. If the working-pressure be 160 lbs. per square inch, then, on this basis, the mean pressure is 160 ÷ 3 = say, 54 lbs. per square inch.

Taking the above size of cylinder, then

$$\frac{19 \times 19 \text{ inches} \times \cdot 7854 \times 2 \text{ cylinders} \times 54 \text{ lbs. pressure} \times 800 \text{ feet}}{33000} =$$

743, the indicated horse-power of this locomotive.

The heating-surface provided in this boiler is = 1064 square feet ÷ 743 = 1·432 square feet per indicated horse-power, when the engine is exerting its maximum continuous effort.

The above are the minimum proportions of heating-surface and fire-grate surface suitable for economical combustion. The following are the average proportions of locomotive boilers in practice :—

Total heating-surface in square feet of the boiler of a simple locomotive-engine =
Area of one cylinder in square inches multiplied by 4·75 to 5·75.
Heating-surface of fire-box in square feet =
Total heating-surface in square feet divided by 11 to 13.
Area of fire-grate surface in square feet =
Total heating-surface in square feet divided by 67 to 72.

The area of the tube-surface may be found by deducting the heating-surface of the fire-box from the total heating-surface of the boiler found by these Rules.

Weight of Locomotive Boilers.—The weights of several sizes of locomotive boilers, each complete with copper fire-box and tubes, but exclusive of the smoke-box and chimney, are given in the following Table, in which the dimensions are given internally.

TABLE 66.—WEIGHT OF LOCOMOTIVE BOILERS OF MILD-STEEL, EACH FITTED WITH A DOME, AND HAVING A FIRE-BOX OF COPPER.

Diameter of Barrel.		Length of Barrel.		Length of Fire-Box Shell.		Width of Fire-Box Shell.		Thickness of Shell-Plates.	Working-Pressure in lbs. per square inch.	Approximate Weight.		
										Weight of Boiler.		Weight of Fittings.
Ft.	Ins.	Ft.	Ins.	Ft.	Ins	Ft.	Ins.	Inch.		Tons.	Cwt.	Cwt.
2	0	6	0	1	9	2	0	$\frac{5}{16}$	140	1	2	4
2	6	7	6	3	4	2	6	$\frac{3}{8}$	”	2	4	$8\frac{1}{2}$
3	0	8	0	3	3	2	10	$\frac{3}{8}$	”	3	0	10
3	6	9	3	3	2	3	7	$\frac{3}{8}$	”	4	10	12
4	0	10	0	5	0	4	0	$\frac{1}{2}$	”	8	0	15
4	1	10	6	5	10	4	$0\frac{1}{2}$	$\frac{1}{2}$	”	8	10	17
4	3	10	3	6	0	4	1	$\frac{1}{2}$	”	9	0	18
4	6	9	2	5	5	4	0	$\frac{1}{2}$	”	9	0	16
4	9	11	0	6	8	3	5	$\frac{9}{16}$	”	10	16	20

The usual mountings are included in the weight of the fittings in this Table, but the weights of the ash-pan, hand-rail fixings, and clothing, are not included.

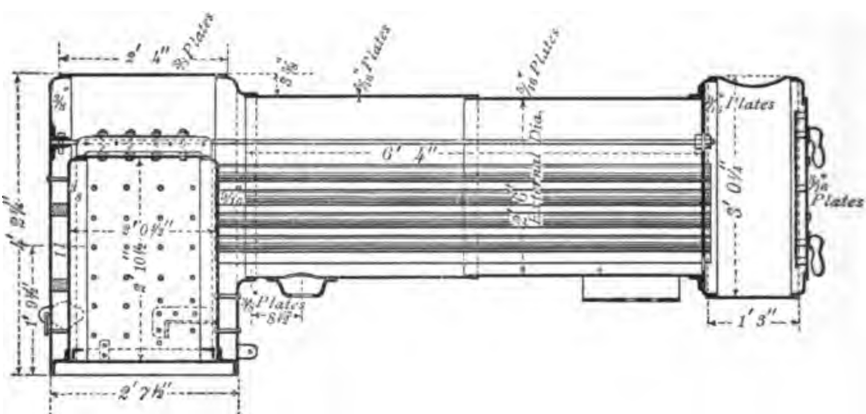


Fig. 203.—Portable-engine boiler by Ruston, Proctor & Co. Limited, Lincoln.

Portable-Engine Boilers.—The boilers of portable engines are of the locomotive type, and when well-proportioned are economical generators of steam. A boiler for a portable engine of 8 nominal horse-power is shown in Fig. 203. The barrel is 2 feet 6 inches diameter and 6 feet 4 inches long, of mild-steel plates $\frac{5}{16}$ inch thick. The fire-box shell is 3 feet $1\frac{3}{4}$

inches wide externally. The fire-box is 2 feet ½ inch long, 2 feet 7 inches wide, and 2 feet 10½ inches high. The water-space at the sides of the fire-box is 2⅝ inches wide. The fire-box plates are ⅜ inch thick, and the roof is supported by three girder-stays. There are 28 tubes of wrought-iron 6 feet 10¼ inches long, 2¼ inches external diameter, and ⅛ inch thick. The tube-plates are $\frac{9}{16}$ inch thick. The total heating-surface of the boiler is 167 square feet. The area of the fire-grate is 5·27 square feet. The chimney is 10 inches diameter.

Boilers of portable engines have generally from 17½ to 20 square feet of total heating-surface per nominal horse-power, and a ratio of heating-surface to fire-grate surface ranging from 23 to 30 in most cases. They are covered with felt, or other non-conducting material, and lagged with wood cased with sheet-iron. In some cases the non-conducting covering is omitted and they are simply lagged with wood cased with sheet-iron.

The results of a number of evaporative tests of portable-engine-boilers are given in Table 83, page 356.

Heating-Surface of Portable-Engine Boilers.—For boilers burning good coal, and supplying steam to simple engines having only one cylinder, the following are good proportions for the heating-surface and fire-grate surface, for effecting economical evaporation:—

Area of total heating-surface in square feet =
Area of cylinder in square inches × 3·2.
Heating-surface of fire-box in square feet =
Total heating-surface in square feet ÷ 6·5
Area of fire-grate surface in square feet for burning good coal =
Total heating-surface ÷ 50.

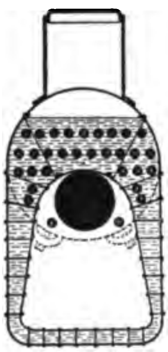
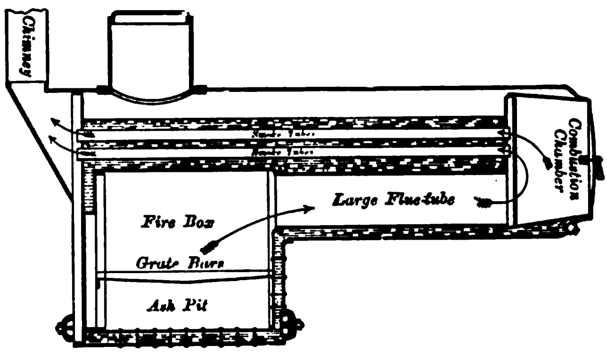
For instance, a portable engine with a cylinder of 10½ inches diameter, or of 10·5 × 10·5 × ·7854 = 86·59 square inches area, requires for economical evaporation a boiler, of the type shown in Fig. 203, with 86·59 × 3·2 = 277 square feet of total heating-surface. The fire-box should have 277 ÷ 6·5 = 42·6 square feet of heating-surface. And the area of the fire-grate for burning good coal should be = 277 ÷ 50 = 5·54 square feet.

Weight of Portable-Engine Boilers.—The weights of the usual sizes of portable boilers, of the locomotive type, as shown in Fig. 203, are given in Table 67.

Portable Return-Tube Boiler.—A portable return-tube boiler is shown in Figs. 204 and 205. The fire-box is arranged for burning refuse-fuel, but it can also be used for burning coal. The fire-box communicates with a flue-tube with the object of providing a long run of the flame, and to complete the combustion of the fuel-gases before they enter the smoke-tubes, which extend from the combustion-chamber to the smoke-box at the front of the boiler. The fire-box is surrounded by water, and so likewise is the combustion-chamber, excepting that portion which projects into the steam-space.

TABLE 67.—WEIGHT OF PORTABLE BOILERS OF LOCOMOTIVE TYPE, OF MILD-STEEL, FOR A WORKING-PRESSURE OF 100 LBS. PER SQUARE INCH.

Nominal Horse-Power.	Diameter of Barrel.		Length of Barrel.		Thickness of Shell-Plates.	Heating-Surface.	Approximate Weight.			
							Weight of Boiler.		Weight of Fittings.	
	Ft.	Ins.	Ft.	Ins.	Inch.	Square Feet.	Tons.	Cwts.	Tons.	Cwts.
4	2	2	4	6	$\frac{5}{16}$	84	1	7		13
6	2	6	5	8	$\frac{5}{16}$	119	1	11		15
8	2	8	6	1	$\frac{3}{8}$	165	1	18		18
10	2	9	6	6	$\frac{3}{8}$	195	2	4	1	0
12	3	0	6	9	$\frac{3}{8}$	235	2	12	1	2
16	3	4	7	0	$\frac{3}{8}$	325	3	14	1	7
20	3	7	7	6	$\frac{3}{8}$	388	3	18	1	15
25	3	9	8	0	$\frac{7}{16}$	514	4	12	1	18
30	4	0	8	6	$\frac{7}{16}$	590	5	16	2	2

Figs. 204 and 205.—Portable return-tube boiler.

Marine Return-Tube Boilers.—The marine return-tube boiler shown in Fig. 206, is an efficient steam-generator. It consists of a cylindrical shell with one or more furnace-tubes and combustion-chambers communicating with several rows of small tubes placed above each furnace-tube. The products of combustion pass from each furnace-tube into a combustion-chamber, and thence return through the smoke-tubes to a smoke-box placed at the front of the boiler above the furnace-tubes. The combustion-chamber is either confined to one furnace-tube, or is common to two, four, or six furnace-tubes.

A combustion-chamber common to two furnace-tubes conduces to uniformity of steam-supply, and if the furnaces are fired alternately it favours the combustion of the fuel-gases and the prevention of smoke. Separate combustion-chambers increase the heating-surface of a boiler, but the space available for combustion is frequently so restricted as to impede combustion of the fuel-gases. A separate combustion-chamber to each furnace-tube is, however, the best arrangement for accommodating expansion and obtaining free circulation of water to the tube-plate, and is preferable for forced combustion. A single combustion-chamber of large capacity with which all the furnace-tubes communicate, is conducive to a higher efficiency of combustion than is obtainable with separate combustion-chambers.

The shell of the boiler should be made with as few joints as possible, and the joints should be kept clear of the bottom of the boiler. Each belt of plates is preferably formed of one plate, with longitudinal seams placed right and left alternately of the top centre-line of the boiler, and at a suitable distance from it. For high steam-pressure the circumferential

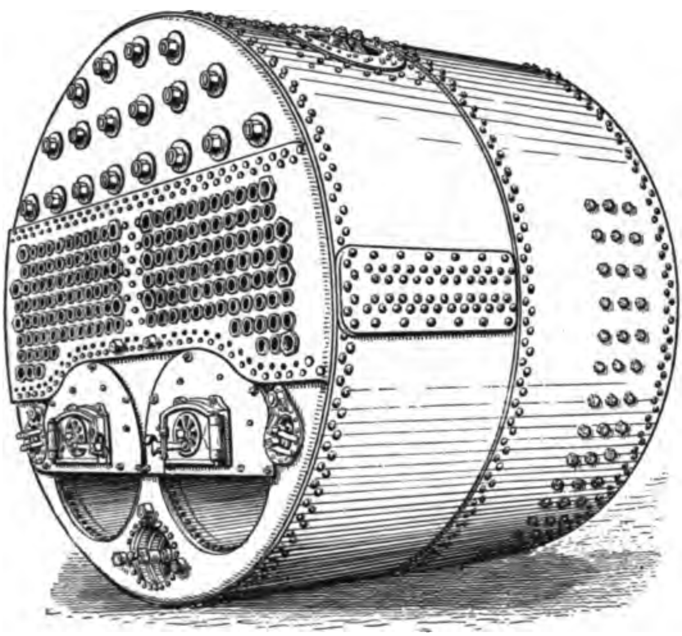


Fig. 206.—Marine return-tube cylindrical boiler.

seams are lap-jointed and generally double-riveted, and the longitudinal seams are butt-jointed and treble-riveted. The end-plates are flanged to join the shell; they are stayed with bolt-stays, having nuts and washers on both sides of the plate.

The top of the combustion-chamber is strengthened by girder-stays and the sides of the chamber are stayed with screwed stud-stays.

The water-spaces of these boilers are frequently so much cramped as to impede the flow of the convection currents. To facilitate circulation the furnace-tubes should not be placed closer together than 6 inches. The space between the top of the furnace-tube and the bottom of the bottom row of smoke-tubes should not be less than 12 inches. The space between the combustion-chambers should not be less than 8 inches, and that between the combustion-chamber and the shell not less than 6 inches. The water-space at the back of the combustion chamber of single-ended boilers should not be less than 6 inches at the bottom and 9 inches at the top. To facilitate cleaning, the space between the bottom of the furnace-tube and the shell should not be less than 8 inches. These are minimum proportions, which it is desirable to increase when convenient.

The results of a number of evaporative tests of marine return-tube boilers are given in Table 82, page 355.

Heating-Surface of Marine Return-Tube Boilers.—In calculating the heating-surface of these boilers, the surface of the front tube-plate is omitted because it is not effective heating-surface. The surface above the level of the fire-bars in the furnace-tube, the area of the back tube-plate, the area of the top and sides of the combustion-chamber, and the surface of the smoke-tubes, are reckoned as effective heating-surface, and the sum of these surfaces represents the total heating-surface of the boiler.

When the fire-bars slope considerably, it is more correct to take two-thirds the area of the surface of the furnace-tube than one-half of its total surface-area as the effective heating-surface.

The area of the surface of the smoke-tubes is equal to the circumference of one tube multiplied by the length between the tube-plates and by the number of tubes. The surface of the ends of the tubes covered by the tube-plates is approximately equal to the area of surface of the back tube-plate, therefore, it is sufficiently accurate for most purposes to omit the area of the back tube-plate, and to take the length of the smoke-tubes from outside to outside of the tube-plates in calculating the heating-surface.

Take, for instance, a double-ended marine return-tube boiler of 14 feet diameter, with 6 furnace-tubes 3 feet 6 inches external diameter, and 438 smoke-tubes of $3\frac{1}{2}$ inches external diameter and 7 feet long measured outside the tube-plates. The heating-surface of this boiler is computed as follows:—

Each smoke-tube has an area of heating-surface =

$$\frac{3\frac{1}{2} \text{ inches diameter} \times 3{\cdot}1416 \times 84 \text{ inches length of tube}}{144} = 6{\cdot}414$$

square feet.

Each furnace-tube of the boiler has an area of heating-surface = $3{\cdot}5$ feet diameter $\times\ 3{\cdot}1416 \times 7$ feet long $= 76{\cdot}969 \times \frac{2}{3} = 51{\cdot}312$ square feet.

	Square feet of Heating-Surface.
The surface of one smoke-tube is $= 6{\cdot}414 \times 438$, the number of tubes = a total tube-surface of . . . =	2809
The surface of one furnace-tube is $= 51{\cdot}312 \times 6$, the number of furnace-tubes = a total surface of . . =	308
The heating-surface of the top and sides of the combustion-chamber is . . . . =	183
	3300 sq. ft.

the total effective heating-surface of this boiler.

Average Heating-Surface of Marine Return-Tube Boilers.—The heating-surface of single-ended marine return-tube boilers necessary for economical evaporation may be found by this *Rule*.

Effective heating-surface in square feet =
(Diameter of boiler in feet)^2 × length of boiler in feet.

For instance, a return-tube boiler of 12 feet 6 inches diameter, and 10 feet 6 inches long, should have a total heating-surface of = 12·5 × 12·5 × 10·5 feet = 1640 square feet.

Weight of Marine Return-Tube Boilers.—The weight of a marine return-tube boiler of given external dimensions, suitable for a given pressure of steam, varies considerably in practice, according to the form of the details of its construction.

The weight of marine return-tube boilers of mild-steel, without fittings, may be calculated approximately by the following *Rule*, in which the constants represent the average minimum and maximum weights employed in practice.

Let D = the external diameter of the shell of the boiler in feet.
L = the external length of the shell of the boiler in feet.
H = the total heating-surface of the boiler in square feet.
P = the working-pressure of the steam in lbs. per square inch, or the pressure shown by the steam-gauge.
C = a constant = 19,200 for the minimum weight, and = 14,400 for the maximum weight, of single-ended boilers of mild-steel usually employed in practice. For double-ended marine return-tube boilers of mild-steel, for the average weight, C = 17,500.

Weight of a marine return-tube boiler in tons =

$$\frac{[(D^2 \times L) + H] \times P}{C}.$$

Example. Required the average minimum weight of a single-ended marine return-tube boiler of mild-steel plates; external diameter of the shell of the boiler 12 feet 6 inches; length of the shell of the boiler 10 feet 6 inches; total heating-surface of boiler 1,640 square feet; working-pressure of the steam 150 lbs. per square inch?

$$\text{Then } \frac{[(12·5 \times 12·5 \times 10·5 \text{ feet}) + 1640] \times 150}{19200} = \frac{492000}{19200} =$$

25·63 tons, the approximate average least weight of this boiler without fittings.

The actual weight of a few general sizes of marine return-tube boilres of mild-steel, from practice, are given in Table 68.

The fittings in this table include the smoke-box and casing, the funnel and damper, fire-bars, furnace-fittings, and all cocks, valves, and internal pipes.

TABLE 68.—WEIGHT OF MARINE RETURN-TUBE BOILERS OF MILD-STEEL, FOR A WORKING-PRESSURE OF STEAM OF 150 LBS. PER SQUARE INCH.

Diameter of Boiler.		Length, Single-Ended.		Length, Double-Ended.		Thickness of Shell-Plates.	Approximate Weight.	
							Weight of Boiler.	Weight of Fittings.
Feet.	Inches.	Feet.	Inches.	Feet.	Inches.	Inch.	Tons.	Tons.
11	4	9	3	—		$\frac{15}{16}$	21·25	12
11	9	10	0	—		$\frac{31}{32}$	22·50	13
12	6	10	6	—		$1\frac{1}{16}$	25·50	13·7
13	4	9	4	—		$1\frac{3}{32}$	27·30	14
13	6	9	6	—		$1\frac{5}{32}$	29·80	14·5
14	0	9	0	—		$1\frac{1}{4}$	31·85	15
13	9	9	9	—		$1\frac{1}{8}$	31·60	15
15	0	10	0	—		$1\frac{7}{32}$	37·80	16
12	2	—		14	0	$1\frac{1}{32}$	34·70	—
11	9	—		15	0	$\frac{15}{16}$	33·50	—
12	11	—		14	6	$1\frac{1}{16}$	37·00	—
13	3	—		15	9	$1\frac{1}{8}$	43·20	—
13	4	—		17	0	$1\frac{1}{8}$	50·00	—
15	0	—		17	0	$1\frac{1}{4}$	59·50	32
15	0	—		17	3	$1\frac{1}{4}$	61·90	29

Single-Ended Marine Return-Tube Boiler.—A single-ended marine return-tube boiler is shown in longitudinal section in Fig. 207. Two of these boilers supply steam for a set of quadruple expansion engines of 1,100 indicated horse-power. The working-pressure of the steam is 183 lbs. per square inch. The boiler is 13 feet 6 inches mean diameter, and 11 feet $\frac{1}{8}$ inch mean length. It has three ribbed furnace-tubes 3 feet 2 inches internal diameter. There are 228 wrought-iron tubes 7 feet 4 inches long inside the tube-plates, and $3\frac{1}{2}$ inches external diameter, 99 of these tubes are stay-tubes. The smoke-tubes are 8 B W G, or ·165 inch in thickness, and swelled $\frac{1}{16}$ inch at one end, as shown in Fig. 208. The stay-tubes are $\frac{1}{4}$ inch thick at the bottom of the thread; they are $3\frac{3}{4}$ inches diameter at the front tube-plate, and $3\frac{1}{2}$ inches diameter at the back tube-plate, and they are screwed into both tube-plates with a fine thread. All the tubes are expanded into the tube-plates.

The boiler-shell is of steel-plates $1\frac{5}{16}$ inches thick. It is formed of two rings with two plates in each ring, each plate weighing about 2·8 tons. The shell-plates are bent cold, and the rivet-holes are drilled in place. The arrangement of the riveted seams of the shell is shown in Figs. 209 and 210. The plates are joined by double butt-straps $1\frac{1}{16}$ inch thick. The outside row of rivets in the butt-straps are $8\frac{1}{2}$ inches pitch, and the rivets pass through the shell and one butt-strap; the two inside rows are $4\frac{1}{4}$ inches pitch, and the rivets pass through the shell and both butt-straps. All the rivets where practicable are riveted by hydraulic pressure, the pressure varying with the size of the rivet from 100 to 120 tons on each rivet. The

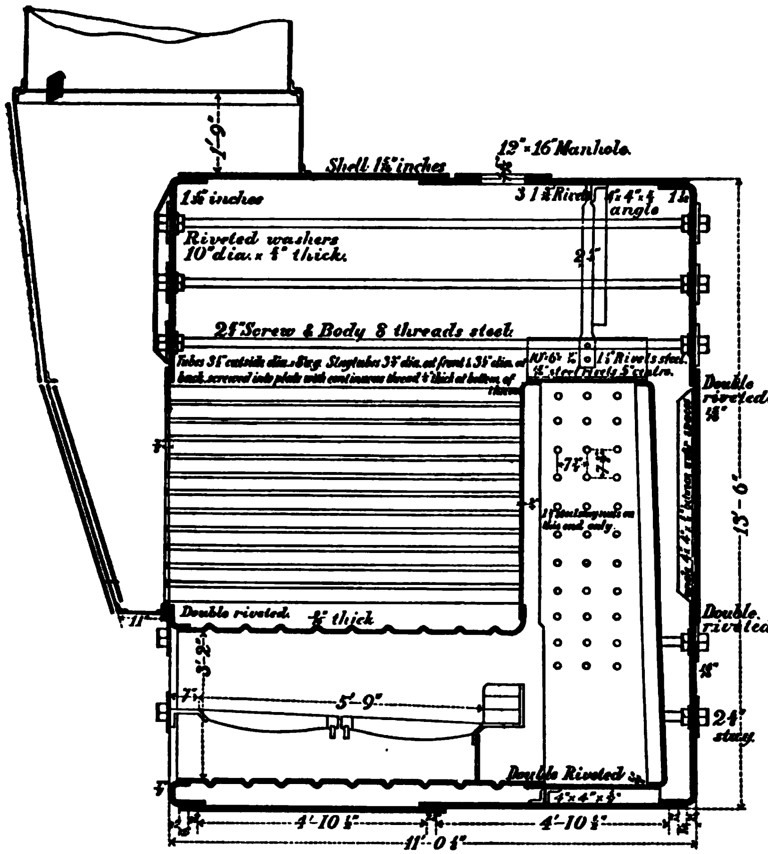


Fig. 207.—Longitudinal section of marine return-tube boiler by Rankin & Blackmore, Greenock.

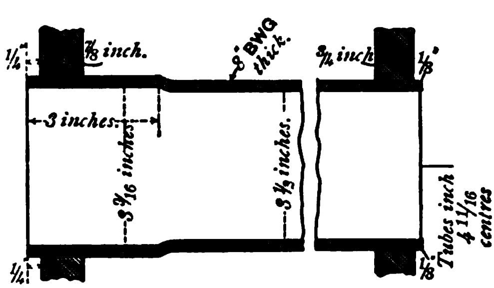
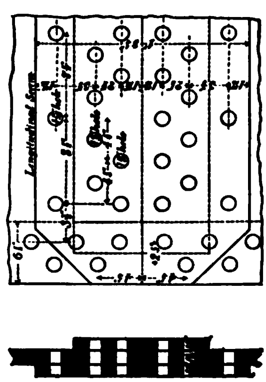


Fig. 208.—Smoke-tubes of return-tube boiler.



Figs. 209 and 210.—Riveted seams of the shell of return-tube boiler.

longitudinal stays in the steam-space are of steel $2\frac{5}{8}$ inches diameter, screwed at both ends 8 threads per inch, and have double nuts and riveted washers at each end.

The combustion-chamber is slung from the shell of the boiler by wrought-iron stays $2\frac{1}{4}$ inches diameter secured by angle-iron $4 \times 4 \times \frac{5}{8}$ inch to the shell, and to girder-stays $10 \times 6 \times \frac{9}{16}$ inch which are riveted to the top of the combustion-chamber. The stays of the combustion-chamber are of steel $1\frac{5}{8}$ inches diameter, pitched at the sides $7\frac{3}{8} \times 7\frac{3}{8}$ inches, and at the back $7\frac{1}{4} \times 7\frac{3}{8}$ inches. They are screwed into both plates with a fine thread and have nuts in the combustion-chamber and at the back of the plates.

The seams of the combustion-chamber are single-riveted, except those of the bottom-plate which are double-riveted. The seams of the back and front of the boiler are double-riveted. The edges of all the plates are planed. This boiler is shown in end view in Fig. 211, and in cross-section in Fig. 212.

This boiler was proportioned by Lloyd's rules, and the leading dimensions and particulars are as follows:—

Diameter of rivets = $1\frac{11}{32}$ inches, or 1·3438 inches.

Area of one rivet = 1·4181 square inches.

Mean diameter of boiler-shell 162 inches.

The upper end-plates are fitted with double-nuts and riveted washers, therefore the constant for their *Rule* is 160.

Pitch of stays for upper end-plates = 15·5 inches, and $15{\cdot}5 \times 15{\cdot}5 = 240{\cdot}25$ square inches of supported surface.

Area of stays in steam-space = 4·8 square inches.

Pitch of stays in combustion-chamber sides = 7·375 inches, and $7{\cdot}375 \times 7{\cdot}375 = 54{\cdot}39$ square inches of supported surface.

Pitch of stays in combustion-chamber back = $7{\cdot}375 \times 7{\cdot}25$ inches = 53·46 square inches of supported surface.

Outside diameter of furnace-tubes 39·125 inches.

Thickness of the Plates of the Boiler.

	Thickness in inches.		Thickness in inches.
Circumferential shell of steel .	$1\frac{5}{16}$	Back tube-plate of steel . .	$\frac{3}{4}$
Front and back upper plates .	$1\frac{1}{32}$	Combustion-chamber top. .	$\frac{9}{16}$
Back, mid	$\frac{15}{16}$	Combustion-chamber sides . .	$\frac{9}{16}$
Back, lower	$\frac{15}{16}$	Combustion-chamber back .	$\frac{9}{16}$
Front tube-plate	$\frac{7}{8}$	Combustion-chamber bottom .	$\frac{5}{8}$
Front lower plate . . .	$\frac{7}{8}$	Furnaces	$\frac{9}{16}$

The working-pressure for each part of the boiler was calculated by Lloyd's rules in the following manner:—

$$\text{Plate-section} = \frac{8{\cdot}5 - 1{\cdot}3438}{8{\cdot}5} \times 100 = 84{\cdot}2\ \%.$$

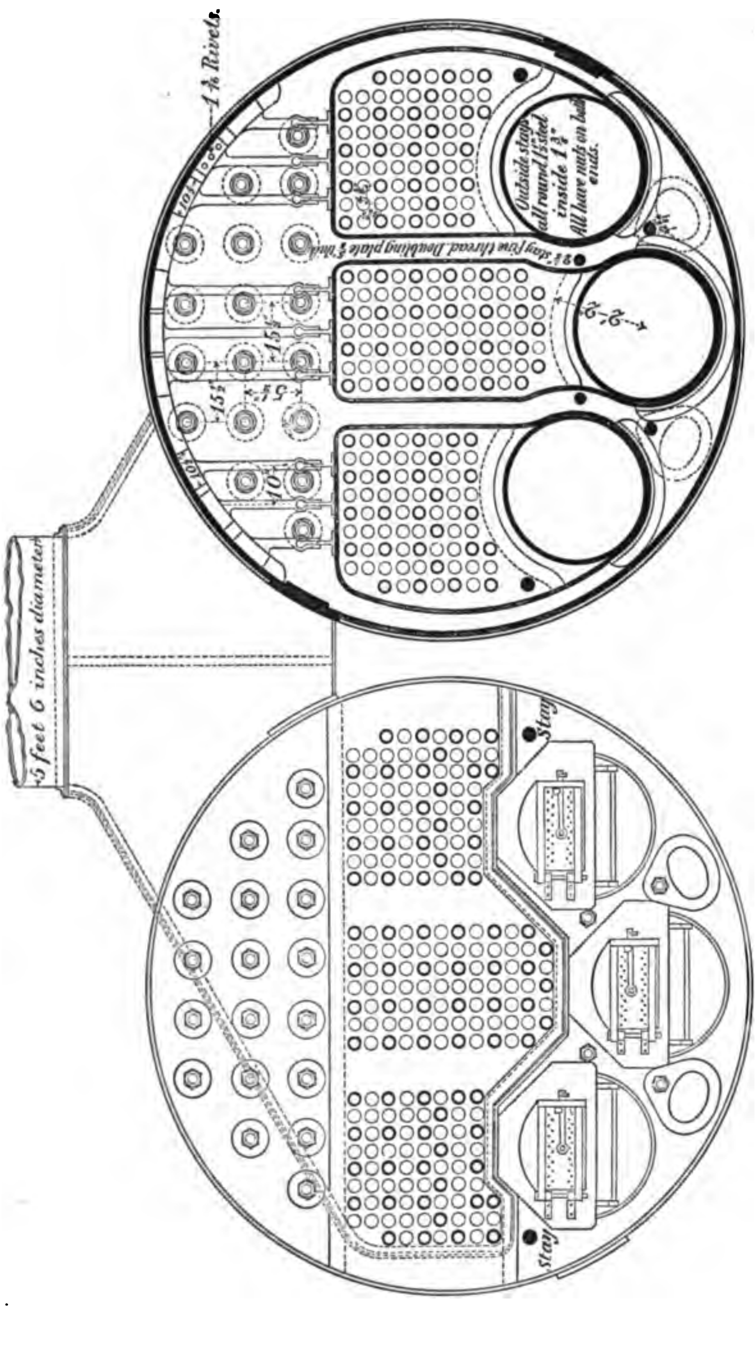


Fig. 211.

Fig. 212.

Figs. 211 and 212.—End view and cross-section of return-tube boiler by Rankin & Blackmore, Greenock.

$$\text{Rivet-section} = \frac{(4 \times 1\cdot4181 \times 1\cdot75) + 1\cdot4181}{8\cdot5 \times 1\cdot3125} \times 85 = 85\cdot85\ \%.$$

$$\text{Circumferential shell} = \frac{260 \times 1\cdot3125 \times 84\cdot2}{162} = 177\cdot4 \text{ lbs.}$$

$$\text{End-plates in steam-space} = \frac{160 \times (16\cdot5)^2}{240\cdot25} = 181\cdot3 \text{ lbs.}$$

$$\text{Stays in steam-space} = \frac{4\cdot8 \times 9000}{240\cdot25} = 179\cdot8 \text{ lbs.}$$

$$\text{Furnace-tubes} = \frac{1000 \times 7}{39\cdot125} = 178\cdot7 \text{ lbs.}$$

$$\text{Combustion-chamber sides} = \frac{120 \times 81}{54\cdot39} = 178\cdot7 \text{ lbs.}$$

$$\text{Combustion-chamber back} = \frac{120 \times 81}{53\cdot46} = 181\cdot8 \text{ lbs.}$$

$$\text{Combustion-chamber side-stays} = \frac{1\cdot227 \times 8000}{54\cdot39} = 180 \text{ lbs.}$$

$$\text{Combustion-chamber back-stays} = \frac{1\cdot227 \times 8000}{53\cdot46} = 183\cdot6 \text{ lbs.}$$

It will be seen that the scantlings of this boiler will pass "Lloyds" for 178 lbs. per square inch the calculated working-pressure, but they allow 5 lbs. extra pressure for pressures above 60 lbs. per square inch.

The working-pressure of the steam is therefore = 178 + 5 = 183 lbs. per square inch.

The heating-surface of this boiler is as follows:—

Heating-surface of the tubes	1531	square feet.
,, ,, furnace-tubes	111	,,
,, ,, centre combustion-chamber . .	85	,,
,, ,, side combustion-chamber . .	154	,,
Total heating-surface =	1881	square feet.

The fire-grate of each furnace-tube is 5 feet 9 inches long and 3 feet 2 inches wide, and the area of the fire-grate-surface is 5·75 feet × 3·166 feet × 3 = 54·61 square feet.

Double-Ended Marine Return-Tube Boiler.—A double-ended marine return-tube boiler is shown in Fig. 213. It is 13 feet mean diameter, and 16 feet 6 inches long, and has six furnace-tubes of 3 feet 4 inches external diameter. There are 740 tubes $3\frac{1}{2}$ inches external diameter and 6 feet 7 inches long. The total heating-surface is 2735 square feet. The area of the fire-grate surface is 84 square feet. The ratio of the total heating-surface to the fire-grate surface is 32·5.

Straight-Flued Marine Boiler.—A marine multitubular, or straight-through, boiler is shown in Figs. 214 and 215. It is 9 feet 2 inches diameter, and 18 feet 3 inches long. The tubes are in line with the furnace-tubes. The total heating-surface is 1542 square feet, and the area of the fire-grate is 50 square feet.

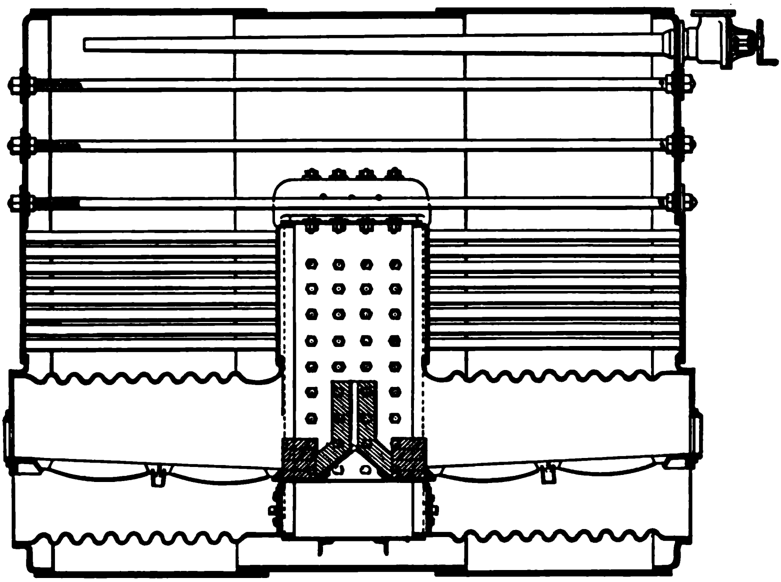


Fig. 213.—Double-ended marine return-tube boiler.

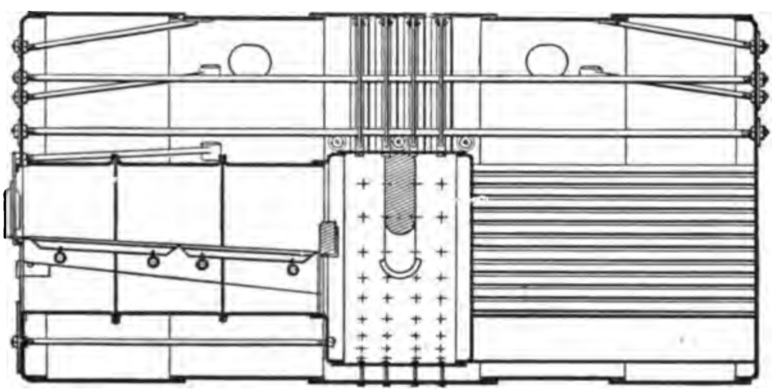


Fig. 214.—Straight-flued marine boiler by Palmer's Shipbuilding Co., Jarrow.

Marine Boilers of Modified Locomotive Type.—A marine boiler of modified locomotive type is shown in Figs. 216 and 217. The barrel is 4 feet 9 inches internal diameter, and 9 feet 3 inches long outside the tube-plates. It contains 195 solid drawn brass tubes of 2 inches external diameter, number 12 B W G thick at the fire-box end, and number 14 B W G thick at the smoke-box end, and 9 feet $3\frac{1}{2}$ inches long. The heating-surface of the tubes is 933·3 square feet, and of the fire-box 82·7

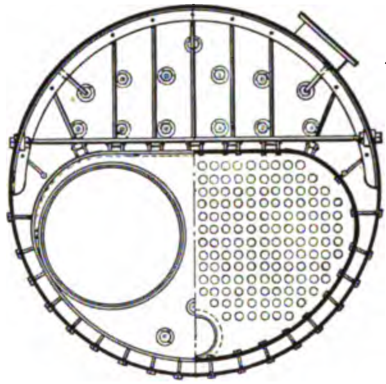


Fig. 215.—End view of the boiler shown in Fig. 214.

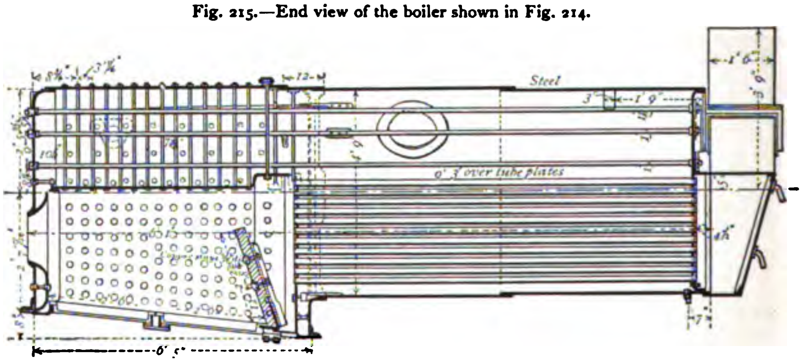


Fig. 216.—Boiler of torpedo-boat by Yarrow & Co., London.

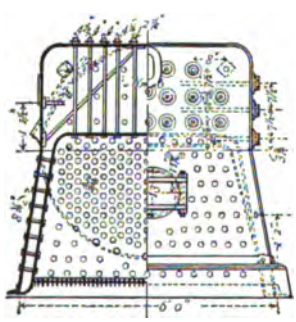
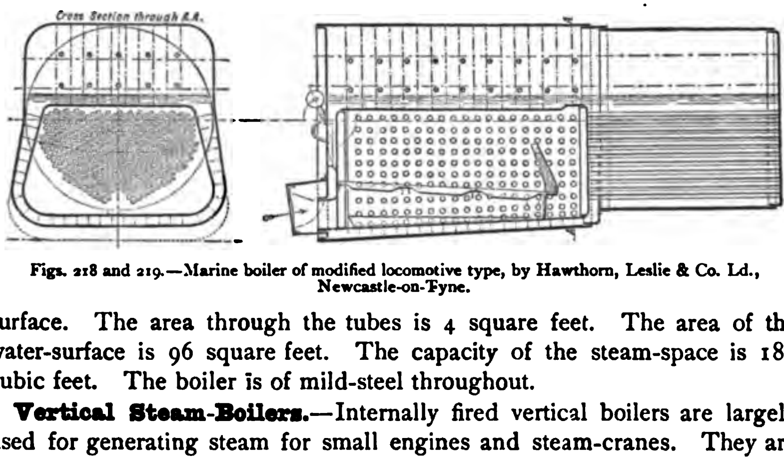


Fig. 217.—End-view of the boiler shown in Fig. 216.

square feet, the total heating-surface being 1016 square feet. The fire-box is of copper. The area of the fire-grate is 25 square feet. The area through the tubes is 487 square inches, and through the ferules 360 square inches. The funnel-area is 509 square inches.

Another marine boiler of modified locomotive type, with a long fire-box and short barrel, for a working-pressure of 155 lbs. per square inch, is shown in Figs. 218 and 219. The barrel is 6 feet $4\frac{1}{4}$ inches external diameter, and 6 feet $5\frac{1}{2}$ inches long inside the tube-plates. The shell-plates are

$\frac{17}{32}$ inch thick, and the tube-plates $\frac{11}{16}$ inch thick. The fire-box is 8 feet 7 inches long, 6 feet $5\frac{1}{2}$ inches wide at the bottom above the corner-curve, and 3 feet 2 inches deep; the thickness is $\frac{5}{8}$ inch at the crown, and $\frac{13}{32}$ at the bottom. The external width of the fire-box shell is 6 feet $5\frac{1}{4}$ inches at the top, and 7 feet 3 inches at the bottom. The roof of the fire-box is supported by direct stays to the shell, $1\frac{1}{8}$ inches diameter, pitched $5\frac{13}{16}$ inches lengthways, and 6 inches endways, the ends of the stays being jointed. There are 372 lap-welded steel smoke-tubes, $1\frac{7}{8}$ inches external diameter, 6 feet $7\frac{5}{8}$ inches long, and 12 S. W. G. thick, pitched $4\frac{1}{4}$ inches horizontally and $2\frac{1}{2}$ inches diagonally; and 20 stay-tubes, $1\frac{3}{4}$ inches external diameter and $\frac{3}{16}$ inch thick, screwed 12 threads per inch. The heating-surface of the tubes is 1,242 square feet, and the total heating-surface is 1,365 square feet. The area of the fire-grate surface is $45\frac{1}{2}$ square feet, or $\frac{1}{30}$ of the heating-



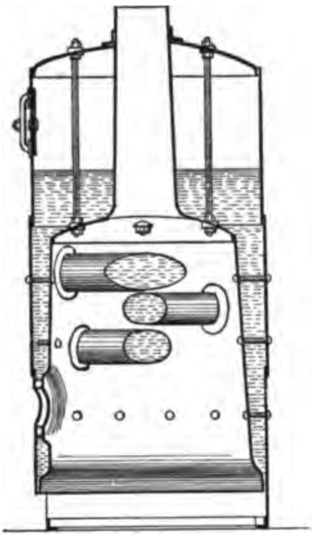
Figs. 218 and 219.—Marine boiler of modified locomotive type, by Hawthorn, Leslie & Co. Ld., Newcastle-on-Tyne.

surface. The area through the tubes is 4 square feet. The area of the water-surface is 96 square feet. The capacity of the steam-space is 183 cubic feet. The boiler is of mild-steel throughout.

Vertical Steam-Boilers.—Internally fired vertical boilers are largely used for generating steam for small engines and steam-cranes. They are handy, and suitable for this purpose, but they are not economical boilers. The simplest form of vertical boiler is the cross-tube boiler, having a conical fire-box and a single flue-tube, or uptake between the crown of the fire-box and that of the boiler-shell, and fitted with cross-tubes in the fire-box, as shown in longitudinal section in Fig. 220, and in cross-section in Fig. 221. The crown of the fire-box is dished, or formed convex, to increase its strength to resist collapse. The cross-tubes improve the circulation of the water and increase the heating-surface. They should always be inclined, instead of placed horizontally, in order to facilitate the escape of steam-bubbles from their surfaces.

The top of the shell of the boiler is frequently dished, to gain strength, but it is liable to be thereby rendered so rigid as to restrict the expansion of the fire-box and uptake, causing grooving either at the root of the uptake, or outside its flange at the crown of the fire-box. The top of the shell is best made flat, so that it may spring to accommodate the upward movement of the fire-box due to expansion, and it may be strengthened by stays so placed as not to interfere with the breathing-space necessary for the end-plate. The breathing-space may be 10 inches.

The water-space around the fire-box of this boiler is frequently so much cramped as to interfere with the flow of the convection-currents, resulting in the water being occasionally ejected from the heating-surfaces and the plates becoming burnt. It should in no case be less than 3 inches at the bottom, and 6 inches at the top, of the fire-box, in order to facilitate the flow of the ascending and descending currents of water from and to the heating-surfaces. As mud and sediment are liable to accumulate at the bottom of the water-space, the fire-bars should be placed sufficiently high above the bottom of the water-space to prevent danger from overheating. And it may be further protected by placing a narrow lining of fire-brick, or fire-clay, round the inside of the fire-box on the top of the fire-bars.



Figs. 220 and 221.
Vertical cross-tube boiler.

The shell of the boiler should extend sufficiently below the fire-box to elevate the bottom ring-seam of the fire-box above the influence of damp in the seating, and place it clear of wet ashes.

Proportions of Vertical Steam-Boilers.—The area of the fire-grate should be proportioned to the average rate of combustion and coal consumption. Small engines driven by these boilers are very extravagant, and seldom require less than 4 lbs. of coal per indicated horse-power developed. Taking the nominal horse-power at one-third the indicated horse-power, they require $4 \times 3 = 12$ lbs. of coal per nominal horse-power.

The rate of combustion of coal in these boilers is from 8 to 15 lbs. per square foot of fire-grate surface per hour. It is frequently 13 lbs. when there is a good draught, and on this basis the area of fire-grate required for this type of boiler may be found by this *Rule* :—

Area of fire-grate in square feet for a vertical cross-tube boiler =

$$\frac{\text{nominal horse-power of boiler} \times 12 \text{ lbs.}}{13 \text{ lbs. rate of combustion}}.$$

For instance, the area of fire-grate surface required for a vertical cross-tube boiler of 10 nominal horse-power is =

$$\frac{10 \text{ horse-power} \times 12 \text{ lbs.}}{13 \text{ lbs.}}$$

= 9·23 square feet.

The area of the fire-grate determines the internal diameter of the fire-box at a level with the fire-grate, and the diameter of this fire-box at that part is $= \sqrt[2]{\frac{9{\cdot}23}{{\cdot}7854}} =$, say, 3 feet 6 inches.

The height of the fire-box may be = the internal diameter of the fire-box measured on the top of the fire-grate, multiplied by 1·6; and the height of this fire-box is = 3·6 × 1·6 = 5·76 feet.

The water-space round the fire-box should not be less than 3 inches at the bottom, therefore the internal diameter of the boiler-shell should not be less than the external diameter of the fire-box, measured at the top of the fire-grate, plus 6 inches.

The height from the bottom of the boiler-shell to the top of its crown, is generally equal to from two to two and one-half times the internal diameter of the shell.

The area of the cross-section of the flue or uptake at the smallest part, should not be less than one-sixteenth the area of the fire-grate, but rather one-twelfth.

The diameter of the cross-tubes is generally equal to about one-fourth the diameter of the fire-box of small boilers, and one-sixth that of large boilers. The number of cross-tubes is from two to five, according to the size of boiler, but is most frequently three.

The fire-boxes of small vertical boilers are seldom supported by stays. But the fire-box of a donkey-boiler is so large in diameter as to form a weak structure, and it requires to be strengthened by staying the sides to the shell by screwed stud-stays, and the crown to the crown of the shell by vertical stay-bolts.

The results of a number of evaporative tests of vertical boilers are given in Table 82, page 355.

Heating-Surface of Vertical Cross-Tube Boilers.—The heating-surface suitable for vertical cross-tube boilers may be found by the following *Rule*:—

Effective heating-surface in square feet =
(diameter of boiler in feet) ^{2} × height of boiler in feet × ·8.

For instance, the heating-surface suitable for a vertical cross-tube boiler 3 feet 3 inches diameter, and 8 feet in height is = 3·25 × 3·25 × 8 feet × ·8 = 68 square feet.

Weight of Vertical Cross-Tube Boilers.—The weights of a few general sizes of vertical cross-tube boilers are given in Table 69.

Uptake-Protector for Vertical Boilers.—The uptake of a vertical cross-tube boiler is liable to become overheated and burnt above the water-level, unless shielded by a lining from the impact of flame. An uptake protected by a fire-clay lining is shown in Fig. 222.

TABLE 69.—WEIGHT OF VERTICAL CROSS-TUBE BOILERS OF MILD-STEEL, WITH THREE CROSS-TUBES IN THE FIRE-BOX, AS SHOWN IN FIG. 220, FOR A WORKING PRESSURE OF 100 LBS. PER SQUARE INCH.

Nominal Horse-Power.	Diameter of Shell.		Height of Shell.		Thickness of Shell-Plates.	Heating-Surface.	Approximate Weight.			
							Weight of Boiler.		Weight of Fittings.	
	Feet.	Inch.	Feet.	Inch.	Inch.	Square Feet.	Tons.	Cwts.	Tons.	Cwts.
2	2	6	5	0	$\frac{5}{16}$	21	0	12	0	7
3	2	6	6	0	$\frac{5}{16}$	31	0	13	0	11
4	2	9	6	6	$\frac{3}{8}$	42	0	18	0	13
6	3	3	7	6	$\frac{3}{8}$	62	1	5	1	1
8	3	6	8	6	$\frac{3}{8}$	83	1	12	1	6
10	4	0	9	6	$\frac{7}{16}$	105	2	1	1	11

When a boiler is not arranged for a lining of this kind, a wrought-iron tube may be fitted in the uptake, as shown in Fig. 223, to form an annular space, which should be filled with fire-clay.

Vertical Cross-Tube Boiler with Special Fire-box.—A vertical cross-tube boiler with a conical fire-box having an enlarged top is shown in Fig. 224. The products of combustion pass from the fire-box through a flue-

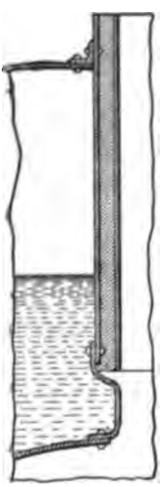


Fig. 222.
Uptake-protector

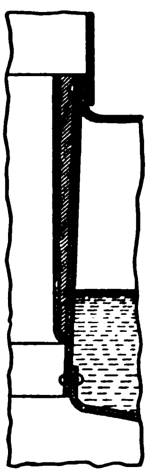


Fig. 223.
Uptake-protector.

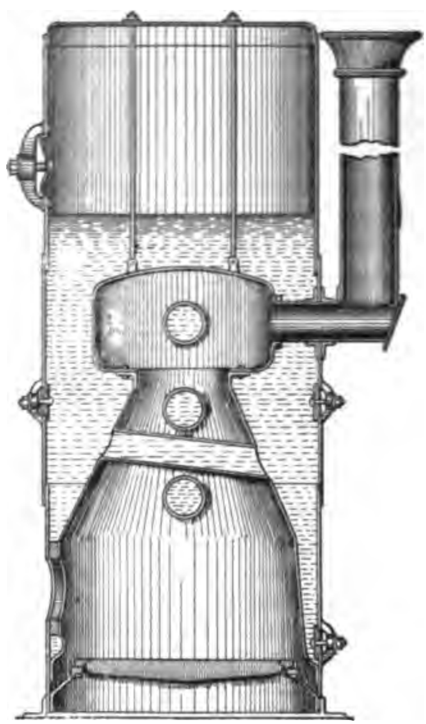


Fig. 224.—Vertical cross-tube boiler with special fire-box.

tube at the side of the shell. The crown of the fire-box is stayed to the top of the shell by stay-bolts.

Vertical Tubular Boilers.—Vertical tubular boilers, shown in Fig. 225, have a number of small tubes between the crown of the fire-box and that of the shell of the boiler. They are generally more economical and better steam producers than cross-tube boilers.

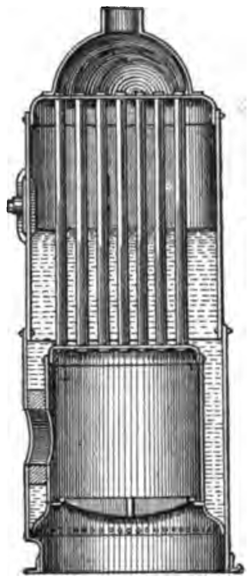


Fig. 225.—Vertical tubular boiler.

The area of the fire-grate may be proportioned in the same manner as that of cross-tube boilers on page 282, and the height and diameter of the shell of the boiler, and the diameter of the fire-box, may also be of the same proportions as those of cross-tube boilers, but the height of the fire-box does not require to be so great. The height of the fire-box of these boilers may be = the internal diameter of the fire-box multiplied by 1 to 1·3.

These boilers require a larger heating-surface than cross-tube boilers, to compensate for the loss of effective heating-surface of that portion of the tubes above the water-level.

The results of a number of evaporative tests of vertical tubular boilers are given in Table 82, page 355.

Heating-Surface of Vertical Tubular Boilers.—The heating-surface required for vertical tubular boilers may be found by this *Rule*:—

Effective heating-surface in square feet = (diameter of boiler in feet)^2 × height of boiler in feet × 1·2.

For instance, the heating-surface required for a vertical tubular boiler of 3 feet 6 inches diameter and 9 feet in height, is = 3·5 × 3·5 feet diameter × 9 feet high × 1·2 = 133 square feet.

Weight of Vertical Tubular Boilers.—The weights of several sizes of vertical tubular boilers are given in the following Table:—

TABLE 70.—WEIGHT OF VERTICAL TUBULAR BOILERS OF MILD-STEEL, FOR A WORKING-PRESSURE OF 100 LBS. PER SQUARE INCH.

Nominal Horse-Power.	Diameter of Shell.		Height of Shell.		Thickness of Shell-Plates.	Heating-Surface.	Approximate Weight.			
							Weight of Boiler.		Weight of Fittings.	
	Feet.	Inch.	Feet.	Inch.	Inch.	Square Feet.	Tons.	Cwts.	Tons.	Cwts.
4	2	9	6	6	$\frac{3}{8}$	60	0	18	0	13
6	3	3	7	6	$\frac{3}{8}$	91	1	6	1	1
8	3	6	8	6	$\frac{3}{8}$	122	1	12	1	6
10	4	0	9	6	$\frac{3}{8}$	153	2	0	1	11
12	4	0	11	0	$\frac{3}{8}$	185	2	7	1	17

Steam-Domes.—The object of a steam-dome is to assist the boiler to furnish dry steam, but it may have a contrary effect, by causing additional moisture, due to condensation of steam, from the large cooling-surface it presents to the atmosphere. The steam should be retained in the boiler-shell until required to be used, because it begins to lose heat the instant it passes out of the boiler, away from the influence of the furnace-heat. A dome adds very little to the steam-space, and should be dispensed with, as it cannot improve the quality of the steam, and it generally weakens the shell of the boiler.

When it is desirable to obtain a larger storage capacity for the steam than can be conveniently provided within the shell, a cylindrical steam-receiver placed horizontally, and supported on three stand-pipes connected to the shell, as shown in Fig. 226, is preferable to a dome.

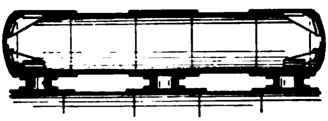


Fig. 226.—Cylindrical steam-receiver.

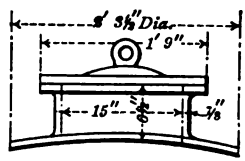
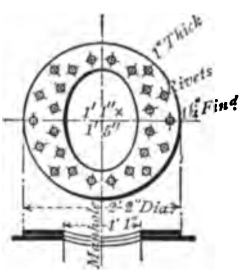


Fig. 227.—Raised manhole-mouthpiece.

Manholes.—Raised mouthpieces of manholes, shown in Fig. 227, should be of wrought-iron or mild-steel; cast-iron should not be used for this purpose. The mouthpiece should be 16 inches internal diameter, and not less than ¾ inch thick in the body and ⅞ inch thick on the flange. It should be attached to the boiler with a double row of rivets, passing through a strengthening hoop on the inside of the shell-plate. The cover is generally dished and secured by not less than 16 bolts of 1 inch diameter, the number and diameter of the bolts varying with the working-pressure of the steam. The bolts may be proportioned by the Rules on page 310.



Figs. 228 and 229.—Manhole strengthening ring.

Oval-shaped manholes are generally either 17 inches long and 13 inches wide, or 16 inches long and 12 inches wide, but in confined spaces they are sometimes 15 inches long and 11 inches wide. The larger axis should be placed in a circumferential direction of the shell. A strengthening ring, shown in Figs. 228 and 229, should be riveted round the hole in the plate to compensate for the metal cut away. To make this part of the shell as strong as the longitudinal seam of the shell, the area of the compensating ring should be equal to 70 per cent. of the metal cut away by the manhole.

For instance, in cutting an oval manhole 12 inches wide in a plate ½ inch thick, 12 × ·5 inch = 6 square inches have been removed from the shell in a longitudinal direction, and the cross-section of the ring at each end of the hole should be = (6 square inches × ·7) ÷ 2 = 2·1 square inches. If the ring be $\frac{3}{4}$ inch thick and riveted with two rows of rivets of $\frac{13}{16}$ inch diameter; then ·8125 inch × ·8125 × ·7854 = ·5184 square inch area of the rivet, and ·5184 × 2 = 1·0368 square inches area of two rivets, and 2·1 square inches + 1·0368 = 3·1368 ÷ ·75 inch thickness = 4·18 inch the width of the compensating ring.

Mudhole.—The mudhole at the bottom of the front end-plate of a cylindrical boiler should be strengthened by a raised mouth-piece of steel, placed inside the boiler, and riveted to the end-plate as shown in Fig. 230. The opening is oval-shaped 15 inches by 11 inches. The cover is secured by two cross-bars with bolts, which are enclosed by an outside cover secured by a bolt.

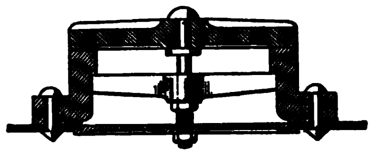


Fig. 230.—Mudhole mouthpiece.

Water-Tube Boilers.—Water-tube boilers are those in which steam is generated from water contained in thin tubes of small or moderately small diameter, by heat applied to the outside of the tubes from a fire generally placed underneath them. They are sometimes called safety-boilers, because they are supposed to render the destructive effects of an explosion as harmless as possible to the surrounding property. The violence of an explosion depends principally upon the weight of water and steam thereby set at liberty. The quantity of water contained in a water-tube boiler is very small compared with the contents of an ordinary cylindrical boiler, or fire-tube boiler, of the same power. The water is contained in a number of small spaces, and in case of an explosion the amount of destructive force is presumed to be reduced to a minimum, and the range of its action confined within narrow limits.

It is claimed for these boilers that, instead of the wide-spread destructive effects arising from the sudden liberation of the large amount of explosive force contained in the large body of water existing in a fire-tube boiler, the destructive effects are limited to the force due to the liberation of the small body of water which would escape through the orifices of the fractures of a single water-tube. The amount of force so liberated might not be sufficient to cause displacement of other portions of the boiler, or to seriously damage buildings.

The principle of all water-tube boilers is with few exceptions practically the same. They differ only in their arrangements, and in details of construction. They generally consist of a number of rows of water-tubes connected by connecting-boxes, placed either horizontally, in an inclined position, or vertically, above a fire-grate, and enclosed in a chamber formed by brick walls lined with fire-brick. The tubes are generally either $3\frac{1}{2}$ or 4 inches external diameter, of wrought-iron or mild-steel lap-welded. The tubes are most frequently inclined from the back to the front of the fire, a steam-chamber is placed above the boiler into which water ascends from the inclined tubes above the fire, through tubes at the front of the boiler, and returns to the inclined tubes through others placed at the back of the boiler, when it is again heated and again ascends to the steam-chamber. A mud-drum for collecting the impurities deposited from the water is placed beyond the action of the fire.

In this type of boiler a small quantity of water covers a large area of heating-surface, and a rapid circulation is necessary to carry off the heat absorbed by the tubes. It is essential that the tubes are regularly supplied with water, as any irregularity in the supply of feed-water is liable to cause a sudden and rapid generation of steam, which may accumulate in the tubes and cause priming.

A large capacity of both steam and water space is necessary for steady steaming. Ample passages should be provided for circulation, with free ascent and descent of the convection-currents, and a large area of surface at the water-line to secure steadiness of water-level. The feed-water should be pure and the heating-surface maintained clean and free from incrustation, to prevent over-heating. The firing should be regular and the boiler lightly worked, to avoid violent ebullition.

The tubes should be arranged to facilitate free escape of the steam. When the tubes are placed horizontally, they offer great resistance to the escape of steam-bubbles, which are compelled to travel along the tubes to the nearest vertical outlet before they can escape, and the tubes are consequently liable to be burnt. They frequently burn at the top, owing to the formation of a film of steam between the water and the heated metal. By inclining the tubes the escape of the steam-bubbles is facilitated, and circulation promoted. The tubes should be free to expand and contract.

It is difficult to obtain feed-water sufficiently pure for the satisfactory working of these boilers, and it is frequently necessary to employ feed-water purifiers in which the water is raised to a sufficiently high temperature, or to about 300° Fahr. to effect the deposit of its impurities before entering the boiler.

The principle of applying heat to a large body of water by subdividing it into small streams, or into a number of small columns contained in thin tubes around which the products of combustion freely circulate, is conducive to evaporative economy. It permits the heating-surfaces to be arranged in the best manner for absorbing the heat, by causing the tubes to intercept and break up the current of heated gases, and when the circulation of the water in the tubes is efficient, and they are placed so that the draught is not impeded, sufficient space being provided for the development of flame and combustion of the gases, then, the necessary arrangements for economical evaporation are theoretically complete.

But it is difficult in practice to develop this principle satisfactorily, because the products of combustion are not equably diffused among the tubes, and the tubes are not uniformly heated. Hence, water-tube boilers on an average are less economical, under ordinary working conditions, than fire-tube boilers.

Defects of Water-Tube Boilers. — Many water-tube boilers work satisfactorily, others are troublesome, and it may be useful to state some of the defects which several forms of these boilers have, or develop in practice. They are briefly as follows:—Complication of parts, and deficient accessibility. Inefficiency of heating-surface, resulting in extravagant consumption of fuel. Excessive wear and tear. Frequent repairs are necessary, especially to the brickwork of the furnace. A large quantity of heat is lost by passing through the brickwork-casing and dissipating in the air.

The tubes require frequent renewal. The tubes are frequently placed in such close proximity to the fire as to hinder the development of flame, and present so large a cooling-surface to the products of combustion as to lower their temperature excessively, resulting in very incomplete combustion of the gases, waste of fuel, and the production of a large quantity of dense black smoke. The tube-ends are frequently not readily accessible.

The boilers are troublesome to clean, a large number of joints having to be broken and re-made each time they are cleaned.

The tubes are liable to leak at the joints, and also to burst. They are liable to become encrusted with scale and mud, causing them to overheat and burn. The tubes frequently burn and burst from defective circulation. The expansion of the tubes is frequently restricted.

Mud and sediment which may be prevented by rapidity of circulation from settling on the heating-surfaces while the boiler is working, are liable to settle in the tubes and boxes when the fire is very low, or extinguished, and become baked on their surfaces when the fire is bright, or relighted.

Ebullition is frequently very violent, causing more or less ejection of the water, and such great fluctuations of the water-level as to render it scarcely discernible.

The boilers sometimes work very capriciously, the pressure of the steam appearing to rise and fall continually, producing sudden and frequently wide variations in the indications of the steam-gauge. The supply of steam is generally less regular than in other types of boilers, and the steam frequently contains a large amount of moisture. They frequently occupy an excessively large space.

Many of these defects are capable of being remedied, and do not exist in some forms of water-tube boilers.

The Advantages claimed for Water-Tube Boilers are that, when the circulation is efficient, a rapid current flows through the tubes, producing a tolerably uniform temperature in all parts of the boiler, and there are no serious strains from unequal expansion. The small diameter of the tubes permits the attainment of excessive strength over any desired ordinary steam-pressure even with thin heating-surfaces. As the boilers are made in sections of moderate size they are easily transported and can be conveyed through narrow openings of buildings, which would not admit a fire-tube boiler, and they may be fixed in confined spaces.

Various Kinds of Water-Tube Boilers.—There are numerous forms of water-tube boilers, a few of which are given in the following pages as representative examples of distinct systems of construction.

Thorneycroft's Water-Tube Boilers.—The Thorneycroft water-tube

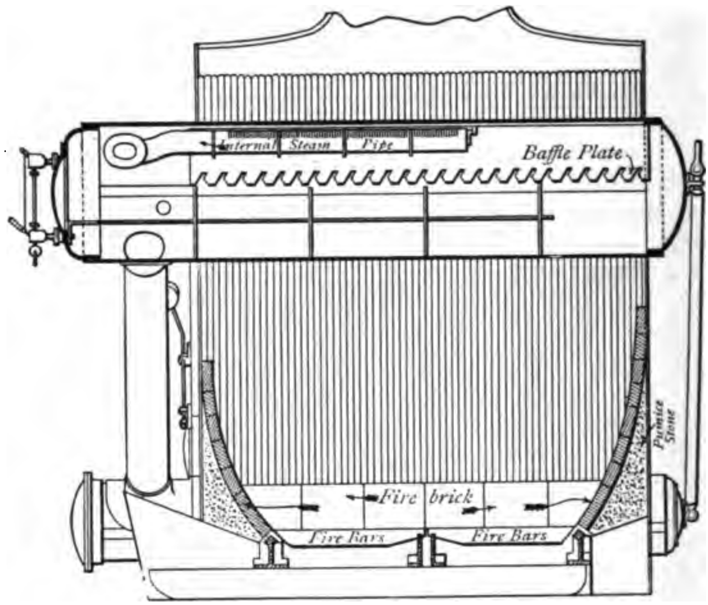


Fig. 231.—Longitudinal section of Thorneycroft's water-tube boiler.

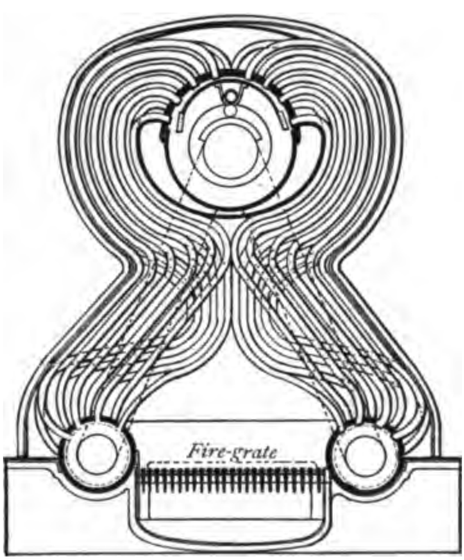


Fig. 232.—End view of the boiler shown in Fig. 231.

boiler is shown in longitudinal section in Fig. 231 and in cross section in Fig. 232. It consists of one top and two bottom cylindrical vessels connected by two pipes at the front, and stayed at the back-end by a triangular frame. These vessels are also connected by a number of small steam-generating tubes, placed so closely together that the products of combustion cannot pass between them. The fire-bars are placed between the bottom vessels. A curved baffle-plate with serrated edges is fixed in the top vessel, to protect the steam-pipe and effect the separation of the water and the steam.

The products of combustion pass through the spaces between the lower ends of the generating tubes, into the spaces between the tubes, and along them to the upper half of the top vessel, whence they pass to the funnel.

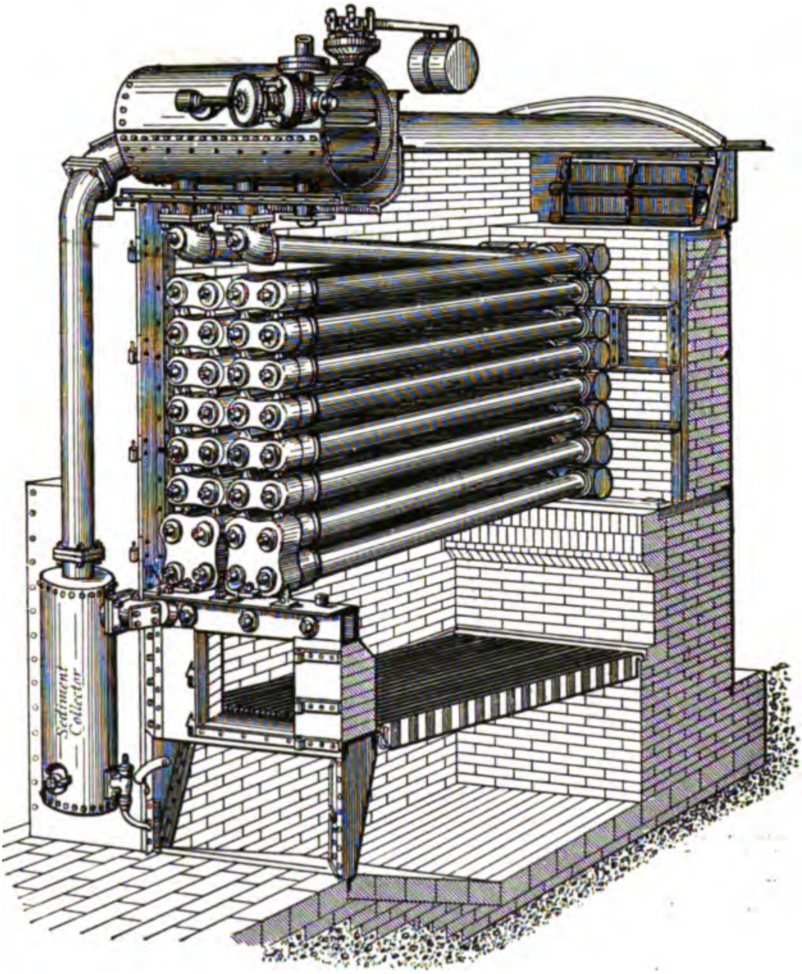


Fig. 233.—The Belleville water-tube boiler.

Belleville's Water-Tube Boiler.—In the Belleville water-tube boiler, shown in Fig. 233, a number of water-tubes are arranged horizontally over a furnace, the tubes are connected to a cylindrical steam-receiver at the top of the boiler. The feed-water is pumped into a purifying cylinder at the top of the boiler, where it is heated sufficiently to effect the precipitation cf calcareous matter which passes to a vertical sediment collector.

The purified water passes through a feed-pipe, placed above the furnace-door, up into the water-tubes, where it is generated to steam which collects in the steam-receiver.

Babcock and Wilcox's Water-Tube Boiler.—In this boiler a series of inclined tubes are placed over the furnace, in which the water is raised to a high temperature and rises through vertical connecting-boxes at the front-end into a horizontal steam and water drum, where the steam separates from the water. The remaining body of water returns through vertical tubes at the back-end of the boiler into the inclined water-tubes, where it is again heated and again passes into the steam and water drum. A continuous and rapid circulation is thus kept up, and a nearly uniform temperature maintained throughout the boiler. A mud-collector is fixed at the lowest point of the inclined tubes to receive the impurities from the water.

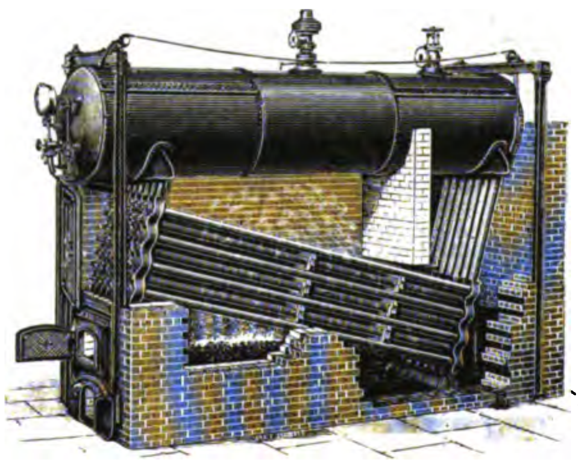


Fig. 234.—Babcock & Wilcox's water-tube boiler.

The boiler shown in Fig. 234, consists of nine sections of tubes, connected to a steam-drum of 48 inches diameter. The tubes are of wrought-iron lap-welded, 4 inches diameter and 18 feet long. They are connected to the drum by nipples and tubes expanded into the connecting-boxes of the drum, and to the top ends of the headers or collectors at the ends of the sections.

The nominal horse-power of the boiler is 159. The heating-surface is 1827 square feet, and the area of the fire-grate is $33\frac{1}{2}$ square feet. The water-contents at mean water-level are 1792 gallons. The steam-space at the normal working water-level is 150 cubic feet.

Root's Water-Tube Boiler.—This boiler consists of a number of tubes arranged in parallel layers in an inclined position over the furnace. The tubes are connected at both ends by caps which couple each tube to the

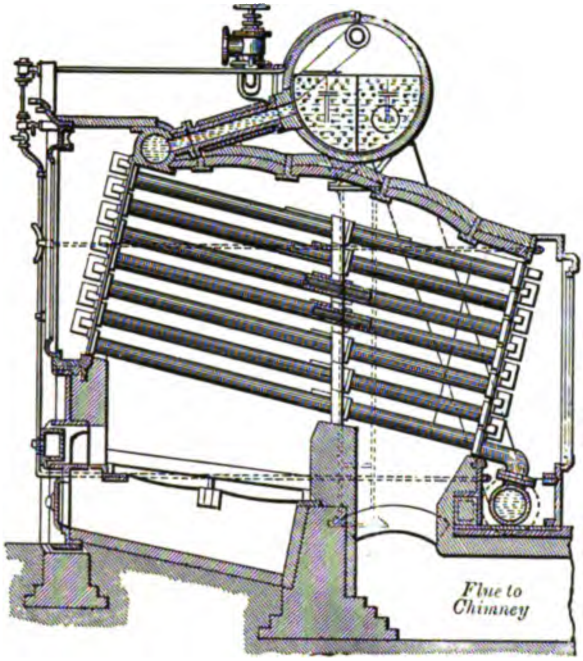


Fig. 235.—Root's water-tube boiler

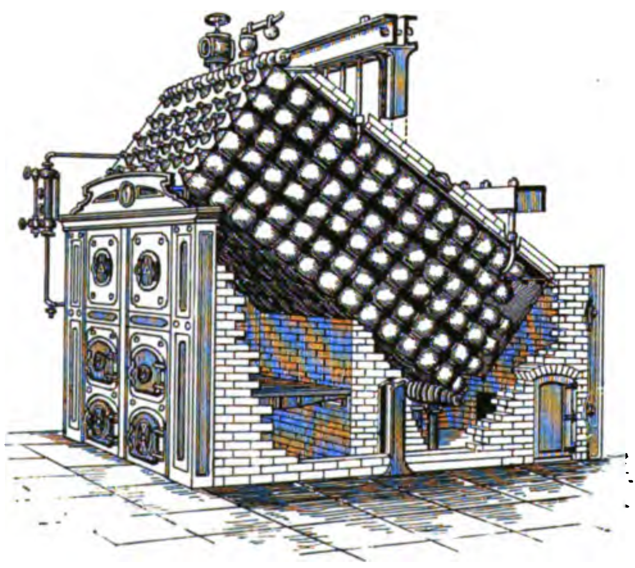


Fig. 236.—Harrison's water-tube boiler.

one below and the one above, so as to form a continuous communication between all the tubes in each successive row.

The boiler shown in transverse section in Fig. 235, is of 150 nominal horse-power. The area of the heating-surface is 1440 square feet, and the area of the fire-grate is 50 square feet. The steam-space at the normal water-level is 76½ cubic feet.

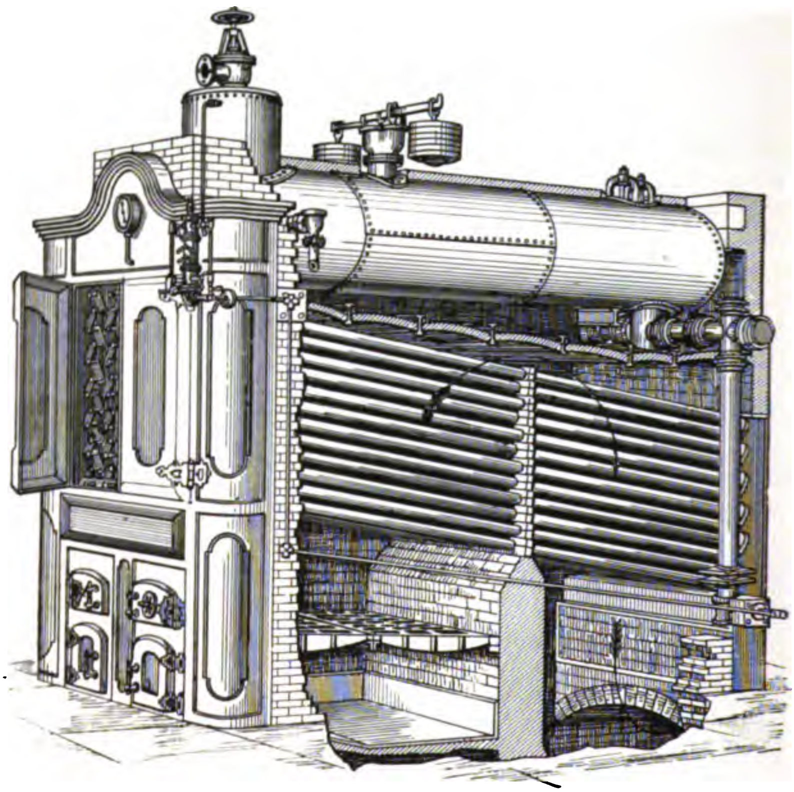


Fig. 237.—De Naeyers' water-tube boiler.

Harrison's Water-Tube Boiler.—This boiler consists of a number of small hollow balls, arranged in parallel inclined lines communicating with each other. The boiler shown in Fig. 236 is of 100 nominal horse-power. The heating surface is 949 square feet, and the area of the fire-grate is 35 square feet. The water-contents at mean water level are 488 gallons. The steam-space at the normal water-level is 30 cubic feet.

De Naeyers' Water-Tube Boiler.—This boiler consists of a number of tubes arranged in an inclined position over the furnace. The front-ends of the tubes communicate with a steam-drum at the top of the boiler. The back-end of the drum is connected by a pipe to the bottom row of the tube boxes. The feed is introduced at the front-end of the drum above the water-line. The water passes down from the steam-drum to the back set of tube-boxes, and the mixed steam and water discharged from the front end of the tubes ascends to the steam-drum.

The boiler shown in Fig. 237 is of 200 nominal horse-power. The heating surface is 1700 square feet and the area of the fire-grate is 40 square feet. The water-contents of the boiler at mean water-level are 1768 gallons. The steam-space at the normal working water-level is 90 cubic feet.

Heine's Water-Tube Boiler.—In this boiler the tubes are connected at each end to a flat hollow bracket attached to the steam-drum. The tubes are slightly inclined over a furnace, as in other water-tube boilers. The boiler shown in Fig. 238 has a total heating-surface of 900 square feet, and the area of the fire-grate surface is 20 square feet.

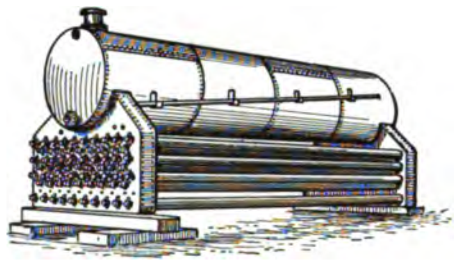
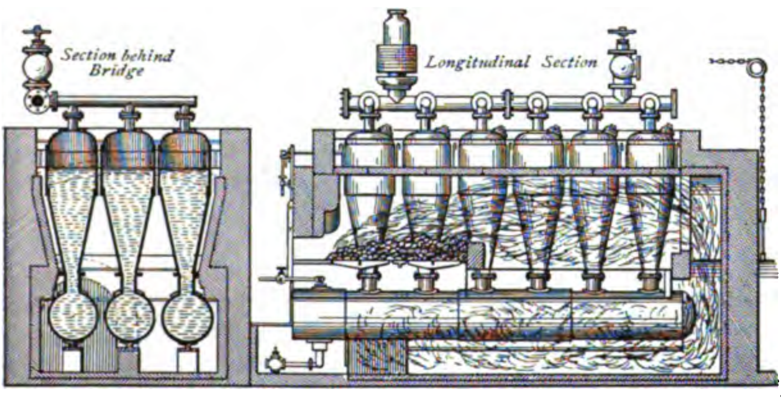


Fig. 238.—Heine's water-tube boiler.



Figs. 239 and 240.—Shepherd's water-tube boiler.

Shepherd's Water-Tube Boiler.—This boiler consists of three bottom cylindrical horizontal vessels each 2 feet diameter. On suitable blocks riveted to the top of these vessels, cone-tubes are placed. They are 6 feet 6 inches high and 2 feet diameter at the parallel belt, and they taper to 6 inches at the bottom. The fire-grate is placed in the flues formed by the line of cone-tubes. The boiler shown in Figs. 239 and 240, has a total heating-surface of 529 square feet, and the area of the fire-grate is 26·18 square feet. The water-contents of the boiler are 1830 gallons. The steam-space at working water-level is 87 cubic feet.

Perkins' Water-Tube Boiler.—This boiler is formed of successive horizontal rows of wrought-iron tubes, of 3 inches external diameter and $\frac{3}{8}$ inch thick, connected by vertical nipples. The tubes are contained in a wrought-iron double casing, having the space filled with vegetable black.

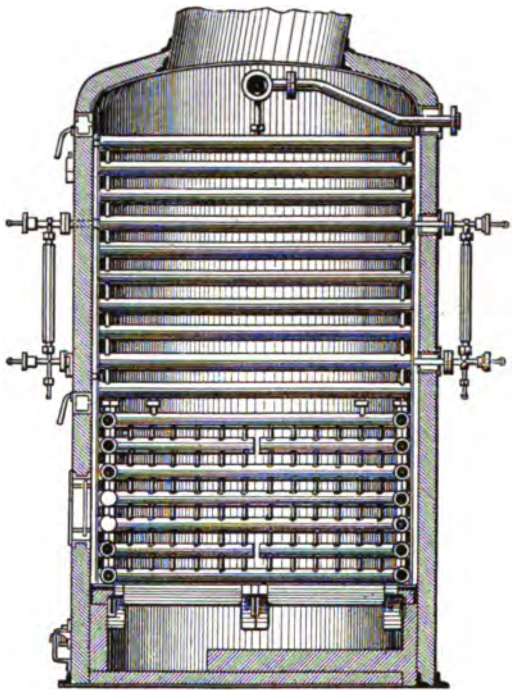


Fig. 241.—Perkins' water-tube boiler.

The boiler shown in sectional elevation in Fig. 241, has a total heating-surface of 301 square feet, and the area of the fire-grate is 15·3 square feet. The water-contents are 55 gallons. The steam-space is 10·2 cubic feet.

Herreshoff's Coil Water-Tube Boiler.—This boiler consists of a continuous coil of water-tube, of increasing diameter towards the lower end and arranged in the form of a hive. The feed-water is forced through the coil by a circulating-pump. This coil is placed above the fire-grate and it is surrounded by a second coil of water-tube. The feed-water is pumped into the bottom of the external coil and ascends to the top and returns down through the internal coil, from the bottom of which the mixture of steam and water is delivered into a separator at the side of the boiler, whence the steam is supplied to the engine, the water being pumped back again to the boiler.

The boiler shown in Fig. 242 has a total heating-surface of 485 square feet and the area of the fire-grate is 26 square feet. The bottom inside coil is 5 feet 9 inches diameter, the bottom tubes of which are $2\frac{1}{8}$ inches external diameter, and $\frac{3}{16}$ inches thick; all the tubes are of wrought-iron, lap-welded. The total contents of the boiler is 22 cubic feet, of which one-third is steam-space and the remainder water-space.

The heating-surface of this boiler is very efficient in the generation of steam. In one experiment, steam was raised from cold feed-water to 60

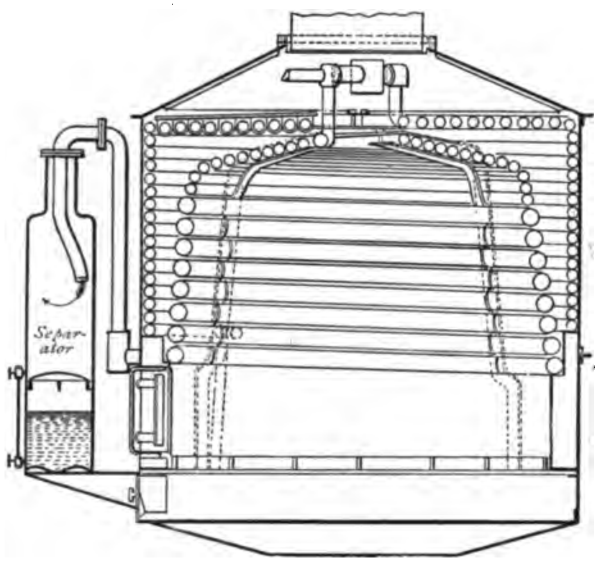


Fig. 242.—The Herreshoff coil-boiler.

lbs. per square inch pressure in about 6 minutes. But as very little water is contained in the boiler, very skilful firing is necessary to maintain a reasonably steady pressure. The water-level in the separator is subject to great and rapid variations.

It is difficult to maintain a uniform circulation, because the circulating-pump is liable to work capriciously owing to the stream of water not being solid. The boiler is of complex construction, and very difficult to repair. For these and other reasons, boilers of this kind with forced circulation are only used to a very limited extent.

Yarrow's Water-Tube Boiler.—The Yarrow water-tube boiler consists of a cylindrical upper part of 20 inches diameter and 6 feet long, and two lower prismatic water-pockets 6 feet in length. These three chambers are subdivided longitudinally, so as to permit free access to their interior. When the boiler is working, the upper cylinder is half full of water. This cylinder is connected to the water-pockets by a large number of straight galvanised steel-tubes, slanting downwards at an angle of 30 degrees, and forming in section a body something like an inverted V, of which the fire-grate forms the base. Owing to the chambers being capable of longitudinal sub-division, the tubes may be readily examined and renewed when required. The flames rise from the furnace and pass to the right and left, in between the slanting tubes, and round the top cylinder to the funnel.

The tubes are completely filled with water, and the system of circulation is maintained, owing to the water in the tubes subject to the greatest heat having an upward current, and that in those subjected to less heat having

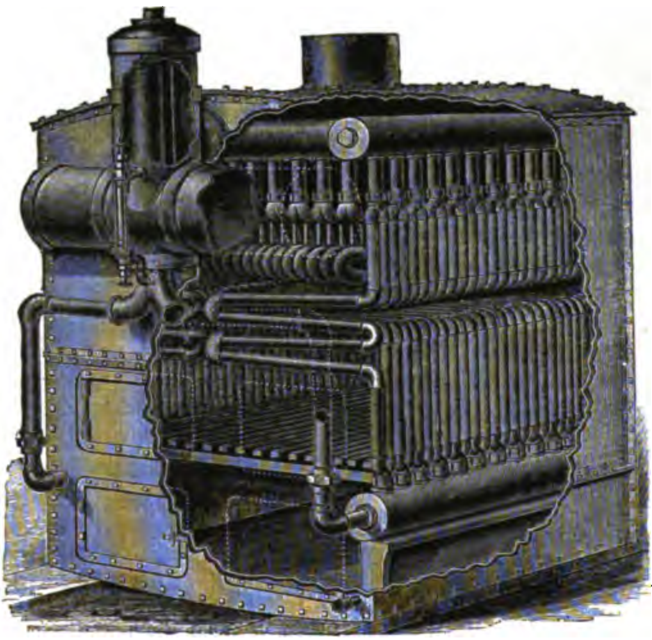


Fig. 243.—Almy's water-tube boiler.

a downward current. The water rises in the inner rows of tubes, and descends in the outer rows. Steam is produced in the outer rows; but the steam-bubbles are carried down by the descending currents. The tubes are 1 inch diameter and 4 feet long, of mild steel. The upper and lower chambers are castings in steel or tough cast-iron. The boiler is cased with sheet-iron, lined with fire-tiles.

Almy's Water-Tube Boiler.—This boiler is formed of wrought-iron or steel tubes of small diameter, arranged vertically and horizontally over a fire-grate placed near the bottom of the boiler. The boiler shown in Fig. 243 has a total heating-surface of 700 square feet; the area of the fire-grate is 30 square feet; and the water-contents are 200 gallons. The boiler weighs about $5\frac{1}{2}$ tons. The feed-water passes through a heater before entering the boiler. A separator is fixed on the front of the boiler, from which steam is drawn for the engine.

Proportions of Water-Tube Boilers.—The heating-surface of water-tube boilers is measured upon the internal diameter of the tube. In a general way, 1 square foot of heating-surface is required for the evaporation of $2\frac{1}{2}$ lbs. of water per hour, and 40 square feet of heating-surface are required for every 100 lbs. of water evaporated per hour. For instance, to evaporate 6000 lbs. of water per hour, a water-tube boiler is required with $6000 \div 100 = 60 \times 40 = 2400$ square feet of heating-surface. It is necessary to provide a large steam-receiver at the top of the boiler to obtain dry steam, especially for a boiler which only delivers the steam at one end of the receiver, because such unequable delivery causes violent disturbance of the water-level. The diameter in inches of the steam-receiver may be = heating-surface of the boiler in square feet ÷ 38 to 40. The receiver should be of mild-steel, with the longitudinal seams double or treble riveted according to the working-pressure.

The proportion of the heating-surfaces of several kinds of water-tube boilers are given at page 145; the quantity of water usually provided per square foot of heating-surface is given at page 146; the capacity of the steam-space is given at page 149; and the area of the fire-grate surface is given at pages 152 and 155.

These boilers are sometimes cased with sheet-iron lined with non-conducting material, or with fire-tiles, but they are most frequently set in a chamber of brickwork, having a cast-iron front. The number of bricks required varies, but when the walls are of ordinary thickness, it is, on an average, seven stock-bricks and two fire-bricks per square foot of the heating-surface of the boiler.

The evaporative performances of a number of water-tube boilers of various kinds are given in Table 85, page 357.

Weight of Water-Tube Boilers.—The weight of water-tube boilers of different types varies considerably. Taking the average of a number of water-tube boilers having tubes of 4 inches external diameter arranged in an inclined position over the furnace, it appears that they weigh from 20 to 22 lbs. per square foot of heating-surface, exclusive of the brickwork-casing. For instance, a water-tube boiler of this kind having 1200 square feet of heating-surface, weighs, approximately, 1200×22 lbs. $= 26400$ lbs. $\div 2240 = 11{\cdot}8$ tons, without the casing.

Allowing, in a general way, 10 square feet of heating-surface per indicated horse-power, this boiler is suitable for supplying steam for an economical engine of $1200 \div 10 = 120$ indicated horse-power.

Comparison of the Weight of Steam-Boilers of Various Types.—The weight of steam-boilers of different types, of average proportions, of mild-steel, without fittings, expressed in terms of the indicated horse-power developed by economical engines, averages, with ordinary draught, in a general way, as follows:—

	Pounds' weight of boiler per indicated horse-power of the engine.
Locomotive-boilers	29
Marine-boilers, of modified locomotive type, for triple-expansion engines	31
The Galloway boiler	96
Cornish-boilers for economical compound-engines	103
Lancashire-boilers for " "	108
Externally-fired cylindrical multitubular-boilers for compound-engines	110
Marine return-tube boilers for triple-expansion engines	114
Boilers of locomotive type with long fire-boxes adapted for stationary engines	140
Portable boilers, of locomotive type	175
Vertical tubular boilers	200
Vertical cross-tube boilers	220
Water-tube boilers of different kinds, exclusive of the brickwork-casing, average	230
Plain cylindrical or egg-ended boilers	300

The weight of steam-boilers of average proportions, of mild-steel, without fittings, expressed in terms of the total heating-surface, averages, with ordinary draught, in a general way, as follows:—

	Pounds' weight of boiler per square foot of total heating surface.
Externally-fired cylindrical multitubular boilers for compound-engines	12
Locomotive boilers	17
Marine boilers of modified locomotive type for triple-expansion engines	19
Portable boilers of locomotive type	26
Water-tube boilers of different kinds, exclusive of the brickwork-casing, average	28
Vertical tubular boilers	30
The Galloway boiler	35
Plain cylindrical or egg-ended boilers	36
Cornish-boilers for economical compound engines	37
Marine return-tube boilers for triple-expansion engines	38
Lancashire-boilers for economical compound-engines	41
Vertical cross-tube boilers	46
Boilers of locomotive type with long fire-boxes, adapted for stationary engines	57

It will be seen that the weights of boilers of different types vary considerably, and that the locomotive boiler contains the greatest power in the least weight.

As an example of the use of the above Table, suppose it be required to find the approximate weight of return-tube boilers suitable for triple-expansion engines of 1000 indicated horse-power. Then the weight of the boilers required for these engines is $= (114 \text{ lbs.} \times 1000) \div 2240 =$ nearly 51 tons, when empty, and without fittings.

SECTION VI.

SAFETY-VALVES; STEAM-PIPES; STOP-VALVES AND OTHER MOUNTINGS FOR BOILERS; INCRUSTATION AND CORROSION; FEED-WATER HEATERS; EVAPORATORS; TESTING BOILERS; EVAPORATIVE PERFORMANCES OF STEAM-BOILERS; STEAM-BOILER EXPLOSIONS, ETC.

Area of Safety-valve for a given area of Fire-grate.—The area of safety-valve in square inches required for a given working-pressure of steam, or pressure above the atmosphere, may be found by this *Rule*:—

$$\frac{\text{Area of fire-grate in square feet} \times 4}{\sqrt{\text{working-pressure of steam in lbs. per square inch}}}.$$

Example: Required the area of the safety-valve of a steam-boiler with a fire-grate surface of 33 square feet, for a working-pressure of steam of 100 lbs. per square inch.

$$\text{Then } \frac{33 \text{ square feet} \times 4}{\sqrt{100}} = 13{\cdot}2 \text{ square inches.}$$

Area of Safety-valve in square inches for a given Rate of Evaporation. The area of safety-valve for a given quantity of water evaporated per hour, may be found by this *Rule*:—

$$\frac{\text{Evaporative capacity of boiler in lbs. per hour}}{40 \times \sqrt{\text{working-pressure of steam in lbs. per square inch}}}.$$

Example: A steam-boiler evaporates $8\frac{1}{4}$ lbs. of water per lb. of coal, and burns 21 lbs. of coal per square foot of fire-grate surface per hour; the fire-grate surface is 33 square feet, and the working-pressure of the steam is 100 lbs. per square inch. Required the diameter of the safety-valve.

Then 21 lbs. of coal × 8·25 lbs. of water × 33 square feet of fire-grate = 5718 lbs. of water, the evaporative capacity of the boiler, and

$$\frac{5718 \text{ lbs.}}{40 \times \sqrt{100 \text{ lbs.}}} = 14{\cdot}29 \text{ square inches, area of safety-valve.}$$

$$\text{Then } \sqrt[2]{\frac{14{\cdot}29}{{\cdot}7854}} = 4\tfrac{1}{4} \text{ inches, the diameter of the safety-valve required.}$$

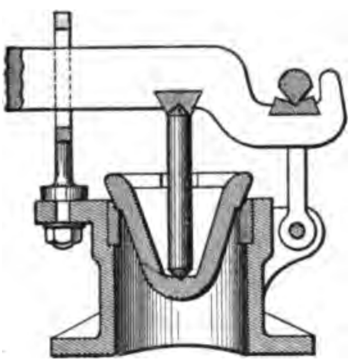


Fig. 244.—Lever safety-valve.

Two small safety-valves give more prompt relief to a boiler than one large one. The diameter of a safety-valve, loaded with a lever and weight, should not exceed $3\frac{1}{2}$ inches. When loaded with a direct-acting spring, the diameter of the valve should not exceed 6 inches. The width of the face of the valve should not exceed one-tenth inch.

Lever Safety-valves.—Lever safety-valves are frequently used for stationary boilers, but they have the objection that, the friction of the joints causes an extra resistance, and consequent increase of steam-pressure when the valve is rising. To reduce the friction to a minimum, they should be constructed with a fulcrum-link having a knife-edge bearing as shown in Fig. 244. The length of the link, from centre to centre, may be $4\frac{1}{2}$ inches for all sizes of safety-valves.

The load on the valve, and the pressure on the valve due to the weight of the lever, may be found by the following formulæ:—

Let a = the area of the safety-valve in square inches.
b = the distance in inches of the centre of gravity of the lever from the fulcrum: the point at which the lever balances on a knife-edge being termed the centre of gravity of the lever.
c = the distance in inches of the centre of the valve from the fulcrum; not to be less than the valve's diameter.
w = the weight of the lever in lbs.

Then, the load on the valve due to the lever =

$$\frac{b \times w}{c}.$$

The pressure in lbs. per square inch on the valve due to the weight of the lever =

$$\frac{w \times b}{a \times c}.$$

If the pressure of the steam in lbs. per square inch be represented by p, the weight of the valve and pin by v, and the distance between the weight hung on the lever and the fulcrum by l; then the weight in lbs. to be placed on the end of the lever is =

$$\frac{(p \times a \times c) - [v + (b \times w)]}{l}.$$

For instance, the load on a safety-valve, having a lever with the centre of gravity 12 inches from the fulcrum, weight of lever 10 lbs. and a distance of 4 inches from the centre of the valve to the fulcrum is = $(12 \times 10) \div 4 = 30$ lbs. If the area of the valve be $12\frac{1}{2}$ square inches, the pressure on the valve due to the weight of the lever is = $(10 \times 12) \div (12{\cdot}5 \times 4) = 2{\cdot}4$ lbs. If the pressure of the steam be 100 lbs. per square inch, the distance between the centre of the weight at the end of the lever and the fulcrum 42 inches; and the weight of the valve and pin = 5 lbs.

Then, the weight to be placed at the end of the lever is =

$$\frac{(100 \times 12{\cdot}5 \times 4) - [5 + (12 \times 10)]}{42 \text{ inches distance between weight and fulcrum}} =$$

116 lbs., the weight to be placed at the end of the lever of this safety-valve.

The ball-weight for the end of the lever should, for convenience of calculation, weigh an even number of pounds, and its weight should be stamped on the top of the fork of the ball.

When the lever is loaded by a spring-balance, in order that the load on the valve may be indicated correctly by the pointer on the scale of the balance, it is necessary that the quotient of the leverage of the spring by the leverage of the valve be equal to the area of the safety-valve in square inches.

Dead-Weight Safety-Valves.—Dead-weight safety-valves, shown in section in Fig. 245, are frequently employed for stationary-boilers. As the centre of gravity of the casing and weights with which the valve is loaded is below the valve, it keeps its position on its seat without the aid of guides. This form of valve is sensitive and efficient, and it is difficult to tamper with it by the addition of further weights than the valve is constructed to carry.

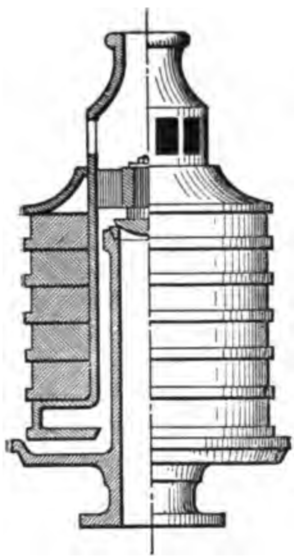


Fig. 245.—Dead-weight safety-valve.

Safety-Valve for High Steam and Low Water.—A safety-valve for high steam and low water, frequently used for Lancashire and other stationary boilers, is shown in Fig. 246. The central spindle projecting into the boiler is loaded by a weight. A float is attached to a lever, and when the water falls below the desired level, the float also falls and raises the valve and allows the steam to blow off, thus lowering the pressure and giving an alarm. The cage of the valve is loaded by an external lever and weight, and lifts along with the valve in the case of steam rising in pressure above the desired limit, and relieves the pressure in the boiler.

The valve-lever requires to be accurately balanced to secure efficient action of the valve. The valve and connections should be examined at each cleaning of the boiler, and maintained in good order.

Spring-Loaded Safety-Valves. — Spring-loaded safety-valves are suitable for all types of steam-boilers, rules for which, with other rules and data for safety-valves, are given in the author's works, "The Works' Manager's Handbook" and "The Practical Engineer's Handbook."

Steam-Pipes for Boilers.—Steam-pipes should take as short and direct a course as possible to the engine, and have as few bends as possible. To prevent reduction of pressure between the boiler and the engine, due to frictional resistance of the flow of steam, the velocity of steam through a steam-pipe of moderate length with several bends, should not exceed 85 feet per second when the engine is at its maximum speed, that is $= 85 \times 60 = 5100$ feet per minute. When a steam-pipe is very short and straight, the velocity of the steam may be 6600 feet per minute.

The area of steam-pipe may be found by this *Rule*:—

Area of cross-section of main steam-pipe in square inches =

$$\frac{\text{Area of cylinder in square inches} \times \text{speed of piston in feet per minute}}{5100, \text{ the velocity of the steam in feet per minute}}.$$

For instance, the size of main steam-pipe required for an engine with a cylinder of 15 inches diameter, length of stroke 3 feet, number of revolutions per minute 80, is =

$$\frac{15 \times 15 \text{ inches diameter} \times \cdot 7854 \times 3 \text{ feet stroke} \times 2 \times 80 \text{ revolutions}}{5100, \text{ the velocity of the steam}}$$

$= 16 \cdot 63$ square inches of area, and $\sqrt[2]{\frac{16 \cdot 63}{\cdot 7854}} = 4\frac{5}{8}$ inches, the diameter of steam-pipe required.

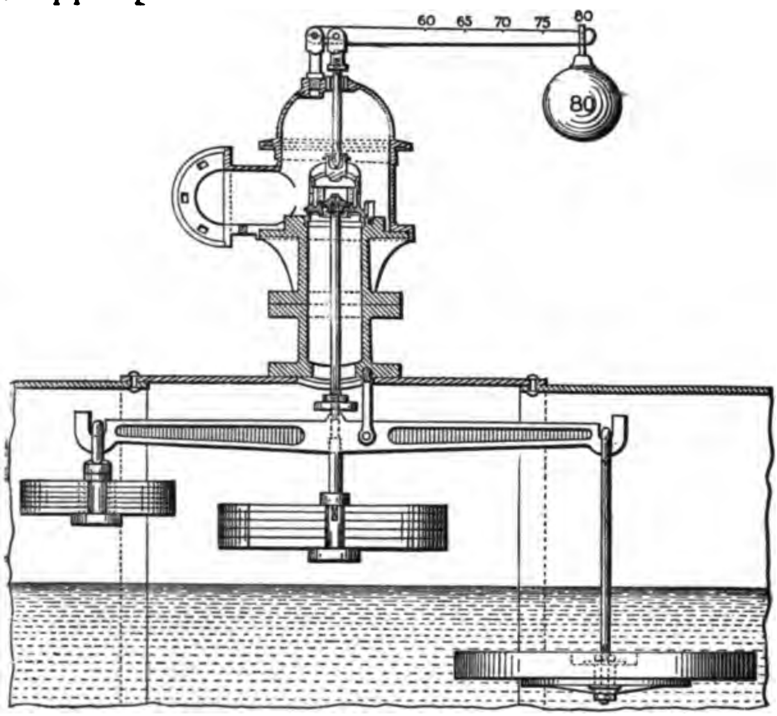


Fig. 246.—Safety-valve for high steam and low water.

The area of a main steam-pipe for a boiler of a given evaporative capacity may be found by the following *Rule*:—

Cross-sectional area of main steam-pipe in square inches =

$$\frac{\text{Lbs. of water evaporated per minute} \times \text{relative volume of the steam}}{\text{Velocity of steam in feet per minute} \times 62 \cdot 42} \times 144.$$

For example, the area of the main steam-pipe to which a range of six

Lancashire-boilers are connected, each boiler evaporating 7020 lbs. of water per hour to steam of 100 lbs. per square inch working-pressure, may be found as follows:—

The relative volume of steam of 100 lbs. per square inch working-pressure is, from Table 37, page 107, 238·2. The water evaporated by each boiler is $= 7020 \div 60 = 117$ lbs. per minute.

Then,

$$\frac{117 \text{ lbs. of water evaporated per minute} \times 238{\cdot}2}{5100 \text{ feet per minute velocity of steam} \times 62{\cdot}42} \times 144 =$$

12·6 square inches, the cross-sectional area of the steam-pipe required for one boiler, and $12{\cdot}6 \times 6 = 75{\cdot}6$ square inches of cross-sectional area of the main steam-pipe are required for the six boilers. Therefore, the main steam-pipe, to which the branch steam-pipe of each boiler is connected, should be $= \sqrt[2]{(75{\cdot}6 \div {\cdot}7854)} =$, say, 10 inches diameter.

Steam-pipes should be tested by hydraulic-pressure to at least double the working-pressure.

Thickness of Cast-Iron Steam-Pipes.—Steam-pipes should be of good tough metal, and they should have an excess of strength to provide for wasting due to corrosion and for accidental strains. However carefully the pipes are made there is generally some variation in thickness, due to the shifting of the core in casting, and they should be drilled to ascertain the least thickness of the metal.

The thickness of cast-iron steam-pipes may be found by this *Rule*:—

Thickness of cast-iron main steam-pipes in inches =

$$\frac{\left(\begin{array}{c}\text{Internal diameter of pipe in inches} \times \text{pressure} \\ \text{of steam in lbs. per square inch}\end{array}\right)}{4000} + {\cdot}5 \text{ inch.}$$

For instance, the thickness of a cast-iron main steam-pipe of 6 inches internal diameter for steam of 150 lbs. per square inch working-pressure, should be =

$$[(6 \text{ inches} \times 150 \text{ lbs.}) \div 4000] + {\cdot}5 \text{ inch} = {\cdot}725 \text{ inch.}$$

Table 71 has been calculated by this *Rule*.

Flanges of Cast-Iron Steam-Pipes.—The finished thickness of the flanges of these pipes may be equal to the thickness of the pipes in inches multiplied by 1·3.

The width of the flanges above the outside of the pipe, may be equal to the diameter of the bolts through them multiplied by 3 to $3\frac{1}{2}$.

Bolts of Steam-Pipes.—The bolts may be proportioned by the rules given at page 310. In a general way the diameter of the bolts may be equal to the thickness of the cast-iron pipe, measured to the nearest eighth of an inch above that thickness.

The number of bolts of this diameter may generally be four for pipes up to 3 inches diameter, five for pipes from $3\frac{1}{2}$ to 5 inches diameter, and above that size there should be one bolt for every inch in diameter of the pipe, that is, 10 bolts for a pipe of 10 inches diameter.

TABLE 71.—THICKNESS OF CAST-IRON MAIN STEAM-PIPES.

Internal Diameter of Pipe in Inches.	Working-pressure of Steam in the Boiler in Pounds per Square Inch.				
	100 lbs.	125 lbs.	150 lbs.	180 lbs.	200 lbs.
	Thickness of the Pipe in Inches.				
	Inch.	Inch.	Inch.	Inch.	Inch.
2	·5500	·5625	·5750	·5900	·6000
2½	·5625	·5781	·5937	·6125	·6250
3	·5750	·5937	·6125	·6350	·6500
3½	·5875	·6093	·6312	·6575	·6750
4	·6000	·6250	·6500	·6800	·7000
4½	·6125	·6406	·6687	·7025	·7250
5	·6250	·6562	·6875	·7250	·7500
6	·6500	·6875	·7250	·7700	·8000
7	·6750	·7187	·7625	·8150	·8500
8	·7000	·7500	·8000	·8600	·9000
9	·7250	·7812	·8375	·9050	·9500
10	·7500	·8125	·8750	·9500	1·0000
11	·7750	·8437	·9125	·9950	1·0500
12	·8000	·8750	·9500	1·0400	1·1000

Strain on the Bolts of the Flanges of Steam-Pipes.—The bolts of the flanges of steam-pipes are severely strained in tightening them sufficiently to secure steam-tight joints, and it is necessary to adopt a factor of safety high enough to provide for both the strain produced by tightening the nuts with a spanner, and the load due to the pressure of the steam. The load on the bolts of steam-pipes may generally be as follows:—

Iron bolts may be loaded to 2500 lbs. per square inch of section.
Mild steel bolts may be loaded to 3000 „ „ „

These loads are per square inch of the cross-section of the bolt at the bottom of the thread. The diameter and area of the cross-section at the bottom of the thread of bolts with Whitworth-thread are as follows:—

TABLE 72.—SIZE OF BOLTS AT THE BOTTOM OF THE SCREW-THREAD.

Diameter of Bolt.	Diameter at the bottom of the Thread.	Cross-sectional Area at the bottom of the Thread.	Diameter of Bolt.	Diameter at the bottom of the Thread.	Cross-sectional Area at the bottom of the Thread.
Inch.	Inch.	Square Inch.	Inch.	Inch.	Square Inch.
$\frac{5}{8}$	·509	·204	$1\frac{1}{8}$	·942	·697
$\frac{3}{4}$	·622	·304	$1\frac{1}{4}$	1·067	·894
$\frac{7}{8}$	·733	·422	$1\frac{3}{8}$	1·162	1·060
1	·840	·554	$1\frac{1}{2}$	1·286	1·30

Proportions of Bolts.—The total cross-sectional area in square inches at the bottom of the thread of the bolts for a steam-pipe, and the number and pitch of the bolts, may be found by the following *Rules* :—

Area of bolts =

$$\frac{\text{The total load on the bolts in lbs. per square inch}}{\text{Working-load on the bolts in lbs. per square inch}}.$$

Number of bolts =

$$\frac{\text{Total cross-sectional area of the bolts in square inches}}{\text{Cross-sectional area of one of the bolts in square inches}}.$$

Pitch of bolts =

$$\frac{\text{Circumference of the circle of the centres of the bolts}}{\text{Number of the bolts}}.$$

For instance, the sectional area and number of the bolts required for the blank-flange of the branch of a steam-pipe of 10 inches internal diameter, for a working-pressure of steam of 150 lbs. per square inch, are as follows:—

The surface of the blank-flange presents to the steam-pressure an area of 10 × 10 inches × ·7854 = 78·54 square inches. The flange resists a total pressure of 78·54 square inches × 150 lbs. per square inch = 11781 lbs. If iron-bolts be used, the total cross-sectional area of the bolts required is = 11781 lbs. ÷ 2500 lbs. per square inch, the working load on the bolts = 4·712 square inches. If Whitworth-bolts of 1 inch diameter be used, the number of bolts required is = 4·712 square inches ÷ ·554 square inch, the cross-sectional area of the bottom of the thread = 8·5, or 9 bolts are required for this flange; but 10 would generally be used in practice, for the purpose of securing a tight joint. If the diameter of the circle of the centres of the bolts in the flange be 15¼ inches, its circumference is = 15·25 × 3·1416 = 47·91 inches, and the pitch of the bolts is = 47·91 ÷ 10 =4·791 inches.

Copper Steam-Pipes.—Steam-pipes of copper should be of fine metal, containing not more than ½ per cent. of impurities. The pipes are best solid-drawn; a brazed-jointed pipe is of uncertain strength.

The thickness of copper steam-pipes may be found by this *Rule* :—

Thickness of copper main steam-pipe in inches—

$$\frac{\left(\begin{array}{c}\text{Internal diameter of pipe in inches} \times \text{pressure}\\ \text{of steam in lbs. per square inch}\end{array}\right)}{10000} + \cdot 125 \text{ inch.}$$

For instance, the thickness of a main steam-pipe of copper, 9 inches internal diameter, for steam of 180 lbs. per square inch working-pressure, should be =

$$\frac{9 \text{ inches diameter} \times 180 \text{ lbs. per square inch}}{10000} + \cdot 125 = \cdot 287 \text{ inch.}$$

The following Table has been calculated by this *Rule* :—

TABLE 73.—THICKNESS OF COPPER MAIN STEAM-PIPES.

Internal Diameter of Pipe in Inches.	Working-pressure of Steam in the Boiler in Pounds per Square Inch				
	100 lbs.	125 lbs.	150 lbs.	180 lbs.	200 lbs.
	Thickness of the Pipe in Inches.				
	Inch.	Inch.	Inch.	Inch.	Inch.
1½	·140	·143	·147	·152	·155
2	·145	·150	·155	·161	·165
2½	·150	·156	·162	·170	·175
3	·155	·162	·170	·179	·185
3½	·160	·168	·177	·188	·195
4	·165	·175	·185	·197	·205
4½	·170	·181	·192	·206	·215
5	·175	·187	·200	·215	·225
5½	·180	·193	·207	·224	·235
6	·185	·200	·215	·233	·245
6½	·190	·206	·222	·242	·255
7	·195	·212	·230	·251	·265
7½	·200	·218	·237	·260	·275
8	·205	·225	·245	·269	·285
8½	·210	·231	·252	·278	·295
9	·215	·237	·260	·287	·305
9½	·220	·243	·267	·296	·315
10	·225	·250	·275	·305	·325
10½	·230	·256	·282	·314	·335
11	·235	·262	·290	·323	·345
11½	·240	·268	·297	·332	·355
12	·245	·275	·305	·341	·365

Flanges of Copper Steam-Pipes.—The thickness of the flanges of these pipes may be equal to the thickness of the pipe multiplied by from 3 to 4.

The width of the flanges above the outside of the pipe may be equal to the diameter of the bolts through them multiplied by 3.

Effect of Heat on Copper.—The strength of copper is reduced considerably by heat. The tensile strength diminishes as the temperature increases above 32° Fahr., approximately as given in Table 74.

It will be seen that, when the temperature of copper is 360° Fahr., which is nearly the temperature of steam of 140 lbs. per square inch working-pressure, or pressure above the atmosphere, its strength is reduced 15 per cent.

Bursting-Pressure of Steam-Pipes.—The bursting-pressure in lbs. per square inch in a pipe, when it is considered as a ring of a length of one inch, is equal to the product of the diameter of the pipe in inches by the internal pressure in lbs. per square inch, and the resistance to the pressure is that due to the area of the metal on two opposite sides of the pipe.

TABLE 74.— SHOWING THE PER-CENTAGE OF STRENGTH LOST BY ELEVATION OF THE TEMPERATURE OF COPPER ABOVE 32° FAHR.

Temperature above 32° Fahr.	Diminution of Tensile Strength.	Temperature above 32° Fahr.	Diminution of Tensile Strength.
°	Per cent.	°	Per cent.
100	2	670	35
170	5	780	45
280	10	820	50
360	15	870	55
450	20	970	66
470	22	1000	68
520	25	1200	88

The bursting-pressure of steam-pipes may be found by this *Rule* :—

Bursting-pressure of steam-pipes in lbs. per square inch =

$$\frac{\left(\begin{array}{c}\text{Thickness of pipe in inches} \times 2 \times \text{tensile strength} \\ \text{of the metal in lbs. per square inch}\end{array}\right)}{\text{Internal diameter of pipe in inches}}.$$

The tensile strength of good metal for pipes averages as follows :—

	Tensile strength in lbs. per square inch.		Tensile strength in lbs. per square inch.
Cast-iron pipes . . .	15000	Wrought-iron tubes. . .	49000
Copper-pipes . . .	30000	Mild-steel tubes . . .	60000

The strength given for copper is that of metal of average quality when cold.

In calculating the bursting-strength of a copper-pipe, an allowance for the reduction of the strength due to elevation of temperature by the steam should be made according to Table 74. A reduction of 15 per cent. may, in a general way, be made for this purpose, and the tensile strength of a copper steam-pipe may be assumed to be = 30000 × ·85 = 25500 lbs. per square inch.

As an example of the previous *Rule* :—Required the bursting-pressure of a steam-pipe, 9 inches internal diameter, of copper ·26 inch thick ?

Then $\frac{\cdot 26 \text{ inch thick} \times 2 \times 25500 \text{ lbs.}}{9 \text{ inches internal diameter}} = 1473$ lbs. per square inch, the pressure required to burst this pipe.

Flow of Steam through Steam-Pipes.—The flow of steam from a boiler through a stop-valve, and through straight-pipes, may be found approximately by the following formula, which applies to all pressures above 12 pounds per square inch above the atmosphere.

Weight of steam in lbs. discharged per minute =

$\frac{\text{Absolute pressure of steam in lbs. per square inch}}{C} \times$ Area of pipe in square inches.

In which, C = 1·38 for steam-pipes up to 10 feet in length.

C = 1·39	„	„	40	„	„
C = 1·42	„	„	70	„	„
C = 1·46	„	„	100	„	„

Example: Required the weight of steam of 115 pounds per square inch absolute pressure, flowing from a boiler through a stop-valve with a steam-pipe 20 feet long.

Then, the area of the steam-pipe is = 4 × 4 × ·7854 = 13·36 square inches, and $\frac{115 \text{ lbs.}}{1·38} \times 13·36 = 1113·28$ pounds, the discharge of steam per minute, and 1113·28 × 60 = 66797 pounds of steam discharged by the pipe per hour.

When the steam enters a pipe with a round-edged orifice, and it has no stop-valve, the discharge is 15 per cent. greater than that obtained by this *Rule*.

The flow of steam is neither increased nor diminished by reducing the outside pressure below 58 per cent. of the absolute pressure in the boiler; that is, the same weight of steam will flow from a boiler under 100 lbs. per square inch pressure into steam of 58 lbs. per square inch absolute pressure as into the atmosphere.

Velocity of the Flow of Steam through Pipes. The velocity of steam through an orifice, such as that of a stop-valve, and through short pipes, may be found by the following *Rule*:—

Velocity of steam in feet per second =

$$32 \times \sqrt[2]{(\text{temperature of steam of absolute pressure} + 460)}.$$

Example: Required the velocity of steam of 100 lbs. per square inch absolute pressure, through a short pipe.

Then, steam of 100 lbs. per square inch absolute pressure is = 100 — 15 = 85 lbs. per square inch gauge-pressure, the temperature of which is from Table 37, page 107 = 327°·4 Fahr., and 327·4 + 460 = 787·4, the square root of which is = 28·06, and 28·06 × 32 = 898 feet per second, the velocity of the steam through this pipe.

Stop-Valves for Steam-Pipes.—The velocity of steam through a stop-valve should not be greater than that given for steam-pipes at page 306. A stop-valve is best fitted with an external nut on the spindle, fixed in a cross-bar on pillars, as shown in Fig. 247, so that the condition of the thread of the screw on the spindle may be seen. When the nut is fixed on the inside of the lid of the valve-box, the thread may become badly worn and partly stripped without being observed.

The cross-sectional area of a stop-valve may be found by this *Rule*:—

Area of stop-valve in square inches = the evaporative capacity of the boiler in lbs. of water per hour × ·0047.

For instance, the area of a stop-valve for a boiler evaporating 6000 lbs. of water per hour should be = 6000 lbs. × ·0047 = 28·2 square inches, and the diameter of the valve should be =

$$\sqrt[2]{\frac{28{\cdot}2 \text{ square inches}}{{\cdot}7854}} = 6 \text{ inches.}$$

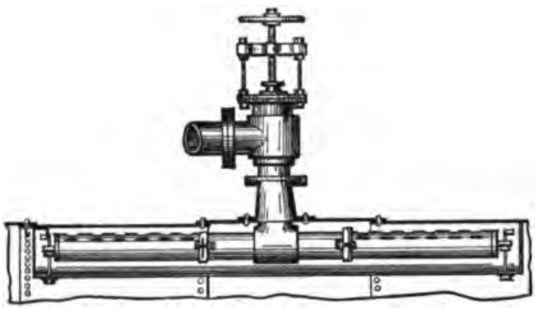


Fig. 247.—Stop-valve and steam-collecting pipe.

Steam - collecting - pipe.—Steam should be drawn from the boiler by a perforated collecting pipe, or anti-priming pipe, fixed below the stop-valve, close to the shell, as shown in the annexed engraving, into which the steam flows tranquilly, without creating a rapid current, which is liable to carry water with it. The total area of the perforations should not be more than one-third greater than the sectional area of the stop-valve in order to prevent priming, which excessive area tends to produce. The total area of the perforations divided by the area of one perforation will give the number of perforations in the collecting-pipe.

Steam-collecting-pipes are frequently made too short to be efficient. It is desirable that they should extend the entire length of the steam-space, or as great a portion of the length as convenient, in order to distribute the withdrawal of the steam over as great a length as possible.

Expansion of Steam-Pipes.—Means should be provided to permit free movement of steam-pipes as they expand and contract in length. This is best effected by the employment of bends of solid drawn copper-pipe, of sufficient length to permit them to spring and accommodate the expansion.

A stuffing-box is frequently placed in the line of pipe for this purpose, but as it is generally necessary to employ a guard-ring connected to the stuffing-box by bolts, as shown in Fig. 248, to prevent the pipe being drawn through the stuffing-box, the bolts do not expand in the same ratio as the pipe, and it is not an efficient arrangement. The stuffing-box is also liable to leak, and may cause more inconvenience than it was designed to obviate.

The expansion of steam-pipes is considerable. For instance, a cast-iron steam-pipe, 60 feet long, heated 300°, will expand, = 60 feet × 12 inches × ·0000063 × 300° = 1·3608 inches. A copper steam-pipe of the same length will expand = 60 × 12 × ·00001095 × 300 = 2·3652 inches. Therefore, it is essential to make provision for the expansion of steam-pipes, to prevent leakage at the joints and to obviate fracture.

Water-Hammer in Steam-Pipes.—When steam is turned into a pipe containing water, even in small quantities, a sudden condensation is produced, resulting in the water being driven violently by the steam against the sides of the pipe, causing a succession of blows or water-hammer, and a severe strain on the pipe. Steam-pipes have been burst by water-hammer when their bursting-pressure was more than ten times as great as the pressure of the steam in the pipes at the instant of fracture. Therefore, water-hammer may cause a strain or pressure of as much as or more than ten times as great as the working-pressure which the pipes were designed to sustain. Hence, it is important to avoid arrangements of pipes which permit lodgment of water, and to drain steam-pipes automatically, so that they may be maintained free from water.

A steam-pipe should not incline towards the boiler, for the purpose of draining the water from condensation back into the boiler, because the water will come in violent conflict with the steam, and may cause water-hammer.

The steam-pipe should incline towards the engine and drain into a separator.

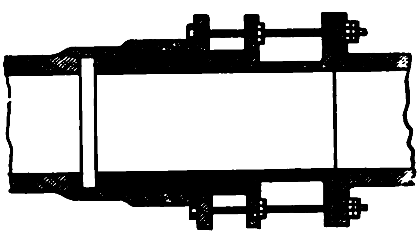


Fig. 248.—Expansion-joint.

Fig. 249.—Separator.

Separator for Steam-Pipes.—Steam generally contains more or less water held in suspension carried with it from the boiler, and a considerable quantity of water is carried with the steam in the case of priming. Much water is, therefore, frequently present in steam-pipes, which it is desirable to separate from the steam before it enters the cylinder of the engine.

A separator is an apparatus for depriving the steam of water. This is generally effected by causing the steam to take a sharp turn, so as to shoot off the water mixed with it into a catch-chamber. A simple and efficient separator of this description is shown in Fig. 249.

The separator should be fixed as close to the engine as convenient, and the steam-pipe should be inclined to it, so that the drain of water from the pipe may be intercepted by the separator. The water may be drained automatically from the separator by connecting it to a steam-trap.

Mountings for Steam-Boilers.—The mountings for steam-boilers should be of the best description, when of inferior quality they give trouble and soon wear out. The branches, or seating-blocks, for fixing the mountings, shown in Fig. 250, should be of mild-steel, and riveted to the boiler-plates. The face of the blocks should be turned, and scraped to a true surface, and the joint of the fittings made with boiled oil.

Steam-Gauges.—The movement of the index or pointer on the dial of a steam-gauge is derived either from the movement of an elastic corrugated plate caused by the pressure of steam against it, or from the movement of a bent flattened tube of metal in becoming straightened under pressure. The latter is the Bourdon-principle, and the majority of steam-gauges are made on this principle, as it is very simple and reliable.

If a metal-tube of flattened section be closed at one end and bent in the form of the letter

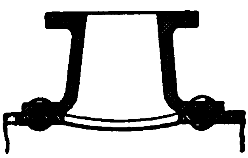


Fig. 250.—Seating-block.

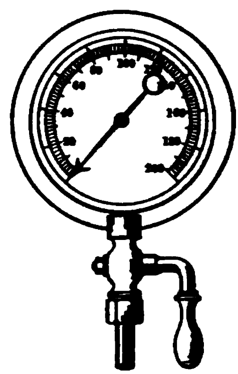


Fig. 251.—Bourdon steam-gauge.

C, or U, the application of internal pressure tends to change the shape of the tube to a circular section, which cannot be effected without partially unbending the tube.

In a steam-gauge, shown in Fig. 251, one end of a tube of this form is attached to the case of the gauge, and the other end is free to move under the internal pressure of the steam. The moving end of the tube is connected by a link to a lever, having at one end a toothed segment which gears into a pinion fixed on a spindle which carries the pointer. The tube is generally of brass for moderately high pressures, and of steel coated with tin for high pressures of steam. To prevent injury to the gauge by heat, it is fixed on a syphon-shaped bend, which becomes filled with water, condensed from the steam, of comparatively low temperature, which prevents steam entering the gauge.

A steam-gauge should have a stop-cock, so that it may be removed, tested, and replaced while the boiler is working. When there is a pipe between the syphon-tube and the boiler, a blow-off-cock should be fixed level with the bottom of the steam-gauge, so that sediment may be blown from the gauge-pipe without disturbing the water in the syphon-tube. A steam-gauge should not be fixed where a current of steam flows past the entrance to the gauge-pipe.

Steam-gauges should be removed from the boiler and tested at regular intervals, because they are liable to get out of order when in constant use and to give inaccurate indications of pressure. They should also be frequently tested by shutting off the steam, putting them into communication with the atmosphere, and allowing the pointer to run back to zero. The gauge-pipe should have a branch with a stop-cock, for the purpose of fixing a standard steam-gauge to test the accuracy of the one employed.

Water-Gauges.—Water-gauges should be of phosphor-bronze, or fine gun-metal; inferior metal is seriously deteriorated by heat, and becomes rotten or brittle.

The cocks are best made with solid bottoms and stuffing-box tops as shown in Fig. 252. The steam and water passages should not be less than $\frac{1}{2}$ inch, but rather greater. The glass tubes expand considerably, and they should be at least $\frac{1}{16}$ inch less in diameter than the glands of their stuffing-boxes.

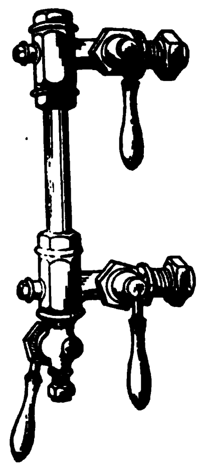


Fig. 252.—Water-gauge.

The bottom portion of the gauge is usually fixed so that when the water is in sight in the glass the level of the water is from 4 to 6 inches above the crown of the furnace-tubes of Cornish and Lancashire boilers, and from 2 to 5 inches above the top of the fire-box or combustion-chamber of other types of boilers. In marine return-tube boilers the bottom of the gauge-glass is generally placed from 3 to 4 inches above the top of the combustion-chamber, the length of the glass between the outside of the glands of its stuffing-boxes is frequently 11 inches, and the working-water level is at about the middle of the glass.

When water-gauges are attached to pipes, the internal diameter of the pipes should not be less than $1\frac{1}{2}$ inches, but rather greater; pipes of less diameter than this soon become choked with incrustation.

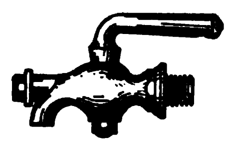


Fig. 253.—Test-cock.

A boiler should have two water-gauges, one being a check on the other, and a convenience in case of accident to one gauge. Water-gauges should be frequently blown through every day, and the steam and water passages and glasses maintained clean.

The taper of the plugs of water-gauges may be $\frac{1}{8}$ inch in diameter for each 1 inch in length.

Test-Cocks.—Test-cocks, shown in Fig. 253, are employed for testing the water-level of steam-boilers as a check on the water-gauges. They are generally three in number, the top test-cock is frequently placed 1 inch above the top of the glass of the water-gauge, and the bottom test-cock 1 inch below the bottom of the gauge-glass; the other test-cock is placed midway between the top and bottom test-cocks. In some cases the bottom test-cock is placed on a level with the top of the combustion-chamber, and the top test-cock is fixed at the high water-level.

The taper of the plugs of test-cocks may be 1 in 6.

Scum-Cock.—When a scum-cock, shown in Fig. 254, is employed it is to blow off scum from the surface of the water in a steam-boiler. A trough-shaped pipe is carried from the scum-cock inside the boiler, the top of which is placed a little below the working water-level of the boiler, for the purpose of collecting scum and refuse floating on the surface of the water.

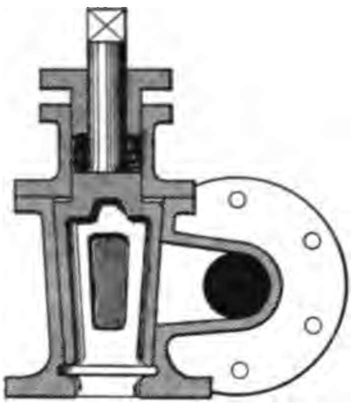


Fig. 254.—Scum-cock.

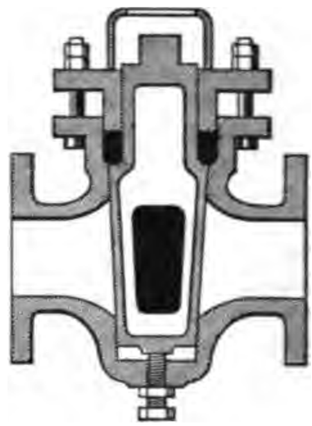


Fig. 255.—Blow-off-cock.

The cross-sectional area of a scum-cock in square inches may be = the evaporative capacity of the boiler in lbs. of water per hour × ·00053.

For instance, the sectional area of a scum-cock for a boiler evaporating 6000 lbs. of water per hour may be = 6000 lbs. × ·00053 = 3·18 square inches, and the diameter of the scum-cock should be =

$$\sqrt[2]{\frac{3{\cdot}18}{{\cdot}7854}} = 2 \text{ inches.}$$

The taper of the plugs of scum-cocks may be 1 in 8.

Blow-off-Cock.—Blow-off-cocks, shown in Fig. 255, are preferably of gun-metal, but they are sometimes of cast-iron with or without gun-metal linings for the plugs to work in, and with gun-metal plugs. The gun-metal may be composed of 88 parts of copper, 10 of tin, and 2 of zinc. The taper of the plug may be 1 in 6 for steam-pressures up to 90 lbs. per square inch; 1 in 8 up to 180 lbs. per square inch; for higher pressures the taper should be 1 in 10. The cock should have a solid bottom and a stuffing-box top. Metallic packing is the best for blow-off-cocks for boilers producing steam of very high pressure. A screw should be fitted to the bottom of the cock to ease the plug when it sticks fast.

The cross-sectional area of a blow-off-cock in square inches may be = the evaporative capacity of the boiler in lbs. of water per hour × ·00082.

For instance, the sectional area of a blow-off-cock for a boiler evaporating 6000 lbs. of water per hour may be = 6000 lbs. × ·00082 = 4·92 square inches, and the diameter of the blow-off-cock should be =

$$\sqrt[2]{\frac{4 \cdot 92}{\cdot 7854}} = 2\tfrac{1}{2} \text{ inches.}$$

A guard should be fixed over the plug, with a featherway to receive a feather on the box-key for turning the plug, so that the key cannot be withdrawn unless the cock be shut.

Blow-off-cocks are liable to stick fast. They should be opened at least once a day, and the plug should be removed, cleaned and replaced, and the packing of the stuffing-box adjusted at each cleaning of the boiler.

The elbow-pipe for attaching the blow-off-cock to the boiler should be of steel. The diameter of the end attached to the seating-block on the boiler should be double that of the end to which the blow-off-cock is fixed.

Mud-Plugs.—The mud-plugs of multitubular boilers should be of gun-metal, $1\frac{5}{8}$ or $1\frac{3}{4}$ inches diameter at the small end, and screwed 11 threads per inch. The taper of the plugs should be 1 in 6. A zinc-plug, having a collar projecting into the water-space, should be fitted in the hollow portion of the mud-plug, to prevent the corrosion of the screw-thread in the plate of the boiler.

Fusible-Plugs for the Crowns of Furnace-Tubes and Fire-Boxes of Steam-Boilers.—The function of a fusible-plug is to melt when the water accidentally becomes low, and present an opening for the escape of steam and water from the boiler into the furnace, to quench the fire and prevent the furnace-tube becoming dangerously hot, and thus avoid the bulging or collapse of the tube. Fusible-plugs, when of good and reliable construction, are useful safeguards against overheating of the furnace-plates in the event of shortness of water. They should, therefore, be fitted to all steam-boilers, and it is essential that they be frequently examined, and maintained in a clean and sound state.

Numerous forms of fusible-plugs are in use, many of which have the objection of not presenting a sufficient quantity of fusible metal to be acted on by the heat. Fusible-alloys have a tendency to become less fusible after being in use a short time in a boiler, and the smaller the quantity of metal the more rapidly is it affected by usage. The quantity of fusible metal employed is frequently so small that it suffers rapid deterioration, and soon becomes inoperative. The metal is also frequently confined in small or narrow spaces, which arrangement is liable to lead to fusion of only a portion of the metal, and the escape of steam in small jets, which prevents further fusion of the plug, and results in the fire being only partially quenched. In some cases, a portion of one of the metals forming the alloy fuses, and the remainder remain unmelted. That which fuses becomes replaced by carbonate of lime, and the alloy is rendered useless as a fusible metal.

A good form of fusible-plug is shown in Fig. 256. It consists of a gun-metal case filled with fusible metal, screwed into an outer casing of gun-metal. It contains a sufficient quantity of fusible metal to be efficiently acted upon by heat, and when fusion of the plug takes place, an opening is presented large enough for the escape of sufficient water to rapidly quench the fire. The plug is screwed into a hollow seating or socket screwed into the top of the furnace, so as to bring the bottom of the plug level with the water-side of the furnace-crown or a little above it as may be desired. The plug should be cleaned, and the scale scraped from the top of the fusible metal each time the boiler is cleaned.

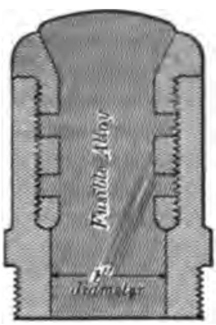


Fig. 256.—Fusible-plug.

Fig. 257.—Fusible-plug.

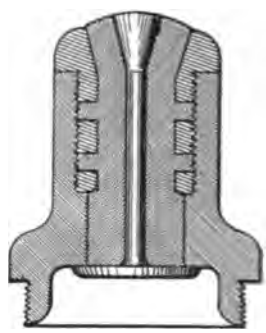


Fig. 258.—Fusible-plug.

A modification of this form of fusible-plug is shown in Fig. 257. It is provided with a cap of zinc to prevent the formation of scale on the surface of the fusible alloy, the action of zinc being very effective for this purpose in some kinds of feed-water.

Another form of this plug is shown in Fig. 258. It is provided with a cover at the bottom, to prevent the escape of any of the fusible metal until the whole is melted. The bottom of the plug is so arranged that it may be screwed direct into the furnace-crown without the intervention of a socket.

The best position for a fusible-plug, is on that part of the crown of the furnace which receives the most intense heat from the fire.

Fusible-Metal for Fusible-Plugs.—Tin of pure quality is the most reliable metal for fusible-plugs, because its nature is not impaired by use in a boiler, as is the case with fusible-alloys. It melts at 446° Fahr.

There are numerous alloys suitable for fusible-plugs; the composition and melting-points of a few of which are as follows:—

Fusible alloy composed of	13	parts of tin and	4	of lead melts at	360° Fahr.
,, ,, ,,	8	,, ,,	11	,, ,,	400°
,, ,, ,,	8	,, ,,	17	,, ,,	450°
,, ,, ,,	8	,, ,,	33	,, ,,	500°
,, ,, ,,	4	,, ,,	48	,, ,,	550°

The alloy should be employed which melts at a temperature corresponding to that of steam of the pressure to which the boiler is limited, or a little higher pressure than the working-pressure. The fusible-plug is then a protection against over-pressure, as well as against injury from shortness of water, so long as the nature of the alloy remains unchanged.

In casting these alloys, care should be taken to obtain sound castings, because if there be air-holes in the metal, it is liable to be blown out of its place when in use. The plugs should be frequently examined and renewed.

Lead is sometimes used for fusible-plugs. It melts at 620° Fahr.

Feed-Pumps for Steam-Boilers.—The capacity of the feed-pump of a steam-boiler should be equal to supplying at least double the quantity of water evaporated by the boiler per hour, in order to have a margin, and to enable the pump to keep down the supply of steam in case the engine is stopped for any reason unexpectedly when the fires are brisk.

The cross-sectional area of the ram of a feed-pump, having a length of stroke equal to $1\frac{1}{2}$ times the diameter of the ram, may be found by this rule:—

Cross-sectional area of ram of feed-pump in square inches =

The evaporative capacity of the boiler in pounds of water per hour × ·002.

For instance, a Lancashire boiler evaporating 6,000 pounds of water per hour, should have a feed-pump with a ram having a cross-sectional area = 6,000 lbs. × ·002 = 12 square inches. The ram should be =

$$\sqrt[2]{\frac{12 \text{ square inches}}{\cdot 7854}} = 4 \text{ inches diameter in round numbers, and the}$$

length of its stroke should be = 4 inches diameter × 1·5 = 6 inches.

The speed of the ram of a feed-pump may be 50 feet per minute, and should never exceed 100 feet per minute. Feed-pumps are frequently worked at too high a speed. The efficiency of a feed-pump is greater, and the wear and tear of the valves is less, at a low than at a high speed.

The friction of the ram of the pump may be reduced to a minimum by the employment of metallic-packing for its stuffing-box.

In pumping hot water, the pump should be so fixed that the water may flow into its barrel by gravity, so that the pump may only have to force and not lift the water.

Feed Back-Pressure Valves.—A feed back-pressure valve is best fitted with an external nut on the spindle; but it is generally arranged with an internal nut, as shown in Fig. 259. The valve should be sufficiently large in diameter to deliver the water with a lift not exceeding $\frac{1}{8}$ inch; a higher lift results in rapid destruction of the valve-seat, from the hammering action of the valve. The bottom of the delivery-branch to the boiler should be placed at a distance above the valve seat of at least equal to the diameter of the valve, in order to permit the pressure to act on the top of the valve, and ensure its closing properly and wearing evenly.

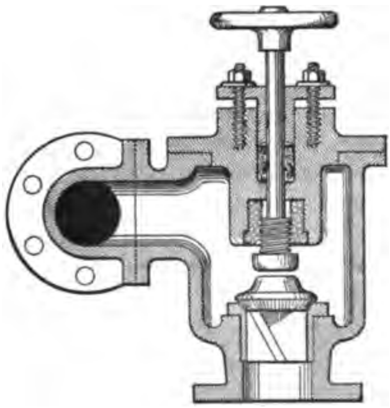


Fig. 259.—Feed back-pressure valve.

The cross-sectional area of a feed-valve in square inches may be = the evaporative capacity of the boiler in lbs. × ·00082.

For instance, the sectional area of a feed back-pressure valve for a boiler evaporating 6000 lbs. of water per hour may be = 6000 lbs. × ·00082 = 4·92 square inches, and the diameter of the feed-valve is =

$$\sqrt[2]{\frac{4 \cdot 92}{\cdot 7854}} = 2\tfrac{1}{2} \text{ inches.}$$

Feeding Steam-Boilers.—Feed-water forced into a boiler requires time to raise it to the same temperature as that of the boiler, and if it be rapidly injected in a considerable quantity it lowers the temperature of the water in the boiler excessively, and causes extravagant consumption of fuel.

The feed-water should be supplied in the best manner to maintain the temperature of the water in the boiler as uniform as possible.

The supply of feed-water should be regulated to the demand for steam so as to maintain a steady water-level, and it should be continuous. An intermittent feed-supply lowers the temperature of the water in the boiler excessively, and is prejudicial to economical evaporation and to a steady supply of steam.

The feed-water should enter the boiler above the level of the furnace-crowns so that in case of grit preventing the closing of the feed-valve, or of fracture of the valve-box, the water may not run out of the boiler and leave the furnace-crowns bare. When feed-water is delivered near the bottom of internally fired boilers it interferes with the circulation and increases the quantity of dead-water at that part.

The feed-water is generally delivered to Cornish and Lancashire boilers through a pipe suspended from the shell and placed about 2 inches above the crowns of the furnaces, and extending about one half the length of the boiler, the last one-third of the length of the pipe is perforated for dispersing the feed-water. In other types of boilers the feed-water is generally delivered about 7 inches below the working water-level.

Supplementary Supply of Feed-Water for Marine-Boilers.—In the processes of evaporating water to steam in a steam-boiler, condensing it after being exhausted from the engines, returning it to the boiler, and again evaporating it, there should theoretically be no loss, but in practice there is a considerable loss from the leakage of cocks, valves, pumps, cylinder-glands, safety-valves and various fittings. This loss is generally made up from fresh water either carried in the vessel in tanks, or produced by evaporating sea-water in an evaporator, the heating-surface of which is generally composed of rows, or coils, of copper pipe.

The loss by leakage varies considerably, but in engines with well-packed glands it may, in a general way, be assumed to average 1 lb. of water per indicated horse-power per hour. Therefore, the boilers of a set of engines developing 1,000 indicated horse power will require in working 24 hours, 1,000 × 1 lb. of water × 24 hours = 24000 lbs. of supplementary feed-water.

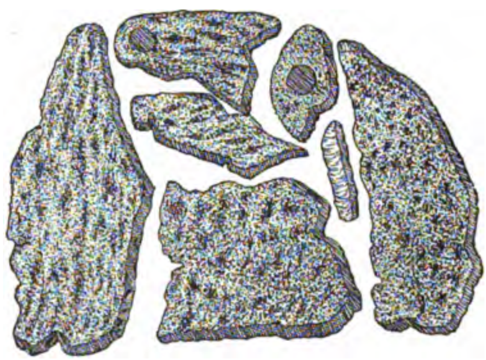


Fig. 260.—Fragments of boiler-scale.

Softening Feed-Water for Steam - Boilers.—Water in a natural state always contains more or less mineral impurities, which, on heating the water to a high temperature, are rendered insoluble, and coalesce into hard incrustation on the surfaces of heated plates of boilers. Fragments of thick scale removed from a steam-boiler are shown in Fig. 260.

The impurities held in solution in feed-water for steam-boilers, which produce incrustation, are principally sulphate of lime, and carbonates of lime and magnesia.

Sulphate of lime is common plaster of Paris. It may be dissolved in large quantities, being soluble to the extent of one part in from three to four hundred parts of water.

The carbonates of lime and magnesia are almost insoluble, except in the presence of carbonic acid. They can, therefore, be dissolved in water containing carbonic acid gas, giving rise to double carbonates or bi-carbonates of these compounds. This gas is driven off by boiling, and the lime and magnesia held in solution in the water are thrown down as insoluble deposits.

Several substances combine with carbonic acid gas and release the carbonate of lime, such as lime and caustic soda. . The soluble sulphate of lime may be converted into the insoluble carbonate of lime by treatment with carbonate of soda or common washing soda.

These softening materials require to be used in definite proportions, according to the chemical composition of the water. In softening water with caustic soda, it requires to be heated to a temperature of not less than 180° Fahr.; and to soften water with carbonate of soda, it requires to be heated to a temperature of not less than 212° Fahr.

Purification of Water by Heat.—The scale-forming substances held in solution are precipitated by heating and evaporating the water. A part of the mineral matter is deposited in the form of fine powder, which forms mud or sludge, and the remainder settles on the plates of the boiler as hard scale. The temperatures at which mineral substances are precipitated vary considerably. Carbonates of lime and magnesia precipitate at a less heat than sulphate of lime and other minerals.

The precipitating temperatures are as follows:—

Carbonates of lime precipitate at temperatures of from	180° to 260°	Fahr.
Sulphates of lime „ .. „ „	303° to 430°	„
Chlorides of magnesium „ .. „ ..	212° to 270°	„
„ sodium .. „ „	330° to 380°	„

When feed-water is not purified, these salts are released from the water when it is heated to the above temperatures in the boiler, and are deposited on the plates, and form incrustation. By heating the feed-water in a separate vessel to these temperatures, the deposit of the scale may be effected therein, and the introduction of scale-forming impurities into the boiler may be greatly prevented.

Treatment of Water in a Cold State.—When water is treated in a cold state with lime and soda, it requires considerable space and time, as the water must remain undisturbed for some time to effect deposit of the precipitate and permit clarification. There are numerous special processes and mechanical appliances for the treatment of water in large quantities, some of which are either too costly or cannot be conveniently applied, for the treatment of small quantities of water, such as are used for feed-water for steam-boilers.

Purification of Water by Chemicals.—The necessity of purifying feed-water led to the introduction of innumerable secret chemical compounds, generally sold under fancy names, for the prevention or removal of scale from steam-boilers. They generally contain lime or soda, or a mixture of both; but caustic soda forms the basis of most boiler compositions. The soda is frequently coloured to disguise its presence, and it is sometimes adulterated with and supplemented by useless and even injurious substances, some compositions having been found to contain chloride of barium, which is very injurious to metal. These boiler compositions owe their efficiency entirely to the soda or lime contained, whatever else they may contain. There is no universal specific for the purification of water. Different waters require different treatment, and a chemical examination of the water is a necessary preliminary to remedial measures.

Some waters may be softened by being treated simply with lime, others require treatment with caustic soda and lime-water, and others require a mixture of caustic soda and carbonate of soda, and water containing much magnesia requires treatment with a mixture of caustic soda, carbonate of soda, and tribasic phosphate of soda. The quantities of these softening materials necessary for efficient purification, are calculated in each case from the analytical data of the water to be treated; they cannot be generally defined.

No other materials are required for softening, or removing the impurities held in solution in the water than lime and soda, and all chemical compounds and secret compositions for the prevention or removal of scale from steam-boilers should be avoided. The impurities mechanically suspended in the water may be removed by filtration.

The boiler is not the proper place to effect the deposit of mud and scale from feed-water. The water should be deprived of its scale-forming impurities before it enters the steam-boiler. But when scale has formed in a boiler, it should be removed by introducing with the feed-water a solution of caustic soda, and frequently blowing off a portion of the water. The amount of caustic soda required when it is pure, is 1 ounce to every 5 gallons of very hard water, or half this quantity if the water be of medium hardness.

To get rid of earthy matter, or other impurities mechanically suspended in the water, and sludge, frequent blowing-off and cleaning of the boiler is necessary.

Prevention of Scale by Zinc.—Zinc has been used with success in some cases for preventing the formation of scale in steam-boilers using moderately hard feed-water, which yielded a deposit of sulphate of lime. It causes the lime-deposit to become pulverent and sedimentary, instead of forming into hard scale. From one and a half to two pounds of zinc are provided for every ten square feet of heating-surface of the boiler. Zinc is not equally effective for this purpose in all waters, and it does not prevent the formation of scale in water containing much organic matter.

When zinc is employed in water containing organic matter, an incrustration, composed of zinc oxide, organic matter, and sediment, is liable to form, which may have an injurious effect on the plates of the boiler, and may also cause over-heating. Therefore, when zinc is used in feed-water of a variable and dirty nature, the boiler should be frequently examined to ascertain the effect of the action of the zinc.

Prevention of Scale by Vegetable Substances.—Numerous vegetable substances have been used in steam-boilers for the prevention and removal of scale, the most effective of which are, eucalyptus, shumac, oak-bark, oak-chips, and logwood, but they are not suitable for all kinds of water. They contain soluble extractive colouring matter and more or less tannic acid. The soluble constituents are dissolved by the water, and basic tannate of lime is formed, which separates as a loose deposit and does not form scale.

Sugar has been employed with success in some waters for the removal of scale from steam-boilers.

Weight of Impurities in Water.—The weight of impurities held in solution and in mechanical suspension in moderately good water varies considerably, but it is frequently from 20 to 50 grains per gallon. A stationary steam-boiler, evaporating 5000 lbs. of water per hour, containing 28 grains of solid matter per gallon, would collect a deposit of 5000 lbs. ÷ 10 lbs. = 500 gallons of water evaporated × 28 grains = 14000 grains ÷ 7000 grains per lb. = 2 lbs. per hour, part of which would form sludge, and the remainder would be converted into hard scale. This boiler, in working a week of 54 hours, would accumulate 2 lbs. × 54 hours = 108 pounds, or nearly 1 cwt. of solid matter.

Prevention of Scale by Mineral-oil.—Mineral-oil rapidly dissolves scale, and is effective in preventing the formation of scale in steam-boilers using some kinds of feed-water. One pint of kerosine-oil, put into the boiler daily, is sufficient to keep a boiler of 40 nominal horse-power free from scale. When kerosine-oil is used for this purpose, it should be of best refined quality, having a vaporising point very much higher than the temperature of the steam in the boiler. The sediment and sludge should be removed by daily blowing-off a portion of the water from the boiler, which should be frequently cleaned, by washing out with water from a hose.

Injurious Effects of Incrustation.—Boiler-scale is a very bad conductor of heat, and resists the passage of heat through it, causing extravagant consumption of fuel and loss of evaporative efficiency. Incrustation frequently causes over-heating, burnt furnace-plates, and bulged furnace-crowns. It increases unequal expansion of the structure which has a weakening effect, and frequently causes leaky seams, and sometimes causes cracks in the plates near the rivet-holes. It causes excessive wear and tear, which may occasion numerous repairs, and it necessitates frequent cleaning of the boiler. It is also a frequent contributory cause of boiler explosions.

Loss of Fuel caused by Boiler-scale.—The loss of fuel caused by a coating of scale on the heating-surfaces of a steam-boiler varies considerably, because it depends upon the composition of the scale, and some kinds of scale are much more heat-resisting than others. In some cases the loss has been found to be 15 per cent. for every sixteenth of an inch of thickness of the scale, in other cases it has been very much less than this.

The loss of fuel caused by a coating of scale in a steam-boiler may, in a general way, be averaged as follows :—

Scale	$\frac{1}{64}$	inch thick	causes	a loss of	2	per cent.	of fuel.
"	$\frac{1}{32}$	"	"	"	4	"	"
"	$\frac{1}{16}$	"	"	"	9	"	"
"	$\frac{1}{8}$	"	"	"	18	"	"
"	$\frac{3}{16}$	"	"	"	27	"	"
"	$\frac{1}{4}$	"	"	"	38	"	"
"	$\frac{3}{8}$	"	"	"	48	"	"
"	$\frac{1}{2}$	"	"	"	60	"	"
"	$\frac{5}{8}$	"	"	"	74	"	"
"	$\frac{3}{4}$	"	"	"	90	"	"

The waste of fuel and loss of evaporative efficiency caused by thick scale is so great, that it is always expedient to incur the expense of purifying feed-water which contains much scale-forming matter.

The advantage obtained by freedom from scale in a steam-boiler is liable to be counterbalanced by corrosion. The effect of corrosion is more insidious and injurious than scale, and it is consequently advantageous to permit the formation of scale of the thickness of an egg-shell as a preventive against corrosion, when the feed-water is of a corrosive nature.

Corrosion and Pitting in Steam-boilers.—Gases absorbed by water, such as sulphuretted hydrogen, oxygen, and carbonic-acid, are very active in the corrosion of boiler-plates.

Even the purest water, when containing air, will cause corrosion. All water contains more or less air, which is liberated when the water reaches the boiling point, and it escapes into the steam-space of the boiler. When the liberated air does not find a free passage from the boiler, it collects in bubbles and causes irregular bead-like corrosion at the parts where it collects.

When much air is present in a boiler, it is not distributed in the steam-space, but being heavier than steam it remains below the steam and forms a layer between the water and the steam, and rapidly corrodes the plates in the vicinity of the water-line. The admission of air to a boiler is a frequent cause of corrosion, and care should be taken to prevent the suction-pipes of feed-pumps drawing air.

The slimy deposit of black or blackish-red colour found in steam-boilers, contains acids which rapidly corrode the plates. It is generally formed from grease carried with the feed-water from the hot-well of the engine into the boiler. Many lubricants decompose at 220° Fahr., and numerous cylinder-oils at 330° Fahr., and form compounds which combine with the mineral substances precipitated in the boiler and form a soft greasy deposit, containing from 60 to 85 per cent. of such mineral substances as iron-oxide, lime, silica, magnesia, iron, and zinc, and from 15 to 40 per cent. of oil.

Part of this greasy deposit rises to the top of the boiler in the form of froth or scum, and the remainder settles at the bottom of the boiler in the form of sludge. The evil effect of this deposit does not stop at the boiler. The deposit is frequently carried into the cylinder of the engine where it forms a black paste, which becomes baked on the ends of the cylinder and on the piston, and it also frequently causes wear and scoring of the cylinder-surfaces. In water containing organic matter free-acids may form and cause corrosion.

Pitting proceeds from the same cause as corrosion. It is the result of the chemical action causing the corrosion becoming intensified and concentrated at a particular part, due principally to the presence of a coating of rust and variations of temperature. Pitting can generally be remedied by scraping the affected parts and cleaning them with a solution of soda, and coating the surface with Portland cement.

Prevention of Corrosion in Steam-boilers.—Zinc suspended in the water is an effective protector from corrosion in boilers. The action of zinc in preventing corrosion is due to the fact that, when two metals of dissimilar character are immersed in a liquid capable of chemically acting on both of them, and are connected or in metallic contact, the metal which is most affected or acted on by the exciting medium becomes the positive or corroded element, and the other becomes the negative or inactive element, and thus escapes corrosion as long as the metals are in contact. Zinc being the most readily acted on it becomes the corroded element, and concentrates and absorbs corrosion which would attack the metal of the boiler if the zinc were not present and in metallic contact.

A very small quantity of zinc suspended in the water, or about two pounds weight of zinc per ton weight of the boiler, will prevent corrosion. But a larger quantity should be provided to allow for diminution of the range of its influence as it becomes wasted by corrosion. A good proportion is $2\frac{1}{2}$ ounces, or ·16 lb., of zinc, per square foot of heating-surface of the boiler. The zinc is applied in slabs or plates suspended in the water, or in sleeves placed on round stay-bolts, or in discs supported by studs.

When the water is good, or moderately pure, an enamel-like film or coating, efficiently protective against corrosion, may be given to the interior surfaces of a boiler when clean, by using zinc-plates suspended in the water and adding common soda continuously to the feed-water.

Another method of applying zinc is to place a ball of zinc, through which a bar of pure copper is passed, in a cage suspended in the water, and to connect the ball by wire to those parts of the boiler which suffer most from corrosion. But the current of electricity that is developed by the zinc and copper is rather feeble, and it is necessary in some cases to augment it by the employment of an auxiliary galvanic battery placed outside the boiler.

A zinc-plug should be inserted in the ends of wash-out plugs of locomotive boilers to prevent corrosion of the screw-threads in the plates, and the ends of water-gauges and test-cocks should have zinc-washers riveted on them for the same purpose.

When feed-water contains acids which cause corrosion, their effect may generally be neutralized by treatment with alkalies, for which purpose soda-ash is frequently employed.

Acids have little or no effect upon petroleum, and refined mineral-oil has been found effective in preventing corrosion in steam-boilers using acid feed-water. Only the best refined mineral-oil should be used for this purpose, and it should have a vaporising-point of at least 600° Fahr. The interior of the boiler becomes coated with a thin film of the oil which protects it against corrosion. The quantity required depends upon the nature of the feed-water, but in some cases a pint of this oil put into a 40 nominal horse-power steam-boiler has been sufficient to maintain it free from both corrosion and scale. A portion of the water should be blown-off daily from the boiler, which should be frequently cleaned and examined to ascertain the effect of the oil.

Heating Feed-water for Steam-boilers.—The feed-water delivered to a steam-boiler requires to be heated to the same temperature as that of the steam before ebullition and vaporization commences. When the feed-water is heated to the necessary ebullition-temperature in the boiler, it is at the expense of fuel which should be utilised in generating steam. Economical evaporation, therefore, depends upon the temperature of the feed-water entering the boiler, and it is desirable to raise the temperature of the feed-water to as near that of the steam as possible before it is delivered to the boiler. The heat necessary for heating the feed-water in a suitable water-heater, may be obtained from heat escaping to waste either in the exhaust-steam or in the products of combustion, and a considerable saving of fuel may be effected.

Saving Effected by Heating Feed-water by Exhaust-steam.—The heat required to raise the temperature of one pound of water from 0° to 212° Fahr. is 1178 units, and if the temperature of the feed-water be, say, 48° Fahr., the heat required to convert one pound of water to steam is $= 1178 - 48 = 1130$ units. If a feed-water heater, heated by exhaust-steam, be employed to raise the temperature of the feed-water to 200° Fahr., then, $1178 - 200 = 978$ units are only required to convert one pound of water to steam at 212°, instead of 1130 units; and a saving of $978 \div 1130 = \cdot 865$, or, say, 13 per cent. is effected by heating the feed-water.

The greatest saving that can be effected by an exhaust-steam heater which raises the temperature of the feed-water from 32° to 212° Fahr. is,

$$= \frac{1178 - 212}{1178 - 32} = \cdot 843, \text{ or, say, 15 per cent.}$$

But in practice the temperature of the feed-water heated by this form of heater seldom averages more than 180° Fahr.; and the saving effected by heating feed-water of 48° Fahr. is generally =

$$\frac{1178 - 180}{1178 - 48} = \cdot 892, \text{ or, in round numbers, 10 per cent.}$$

Quantity of Steam Condensed in Heating Feed-Water by Steam.—When feed-water is heated by discharging exhaust-steam into it, the volume of the water is considerably increased by the water produced by condensing the steam. For instance, to heat 100 lbs. of water from 32° to 212° Fahr. with exhaust-steam, requires $212° - 32° = 180° \times 100$ lbs. = 18000 units of heat. To obtain this quantity of heat it is necessary to condense $18000 \div 966° = 18{\cdot}63$ lbs. of the steam, and the quantity of water will become $100 + 18{\cdot}63 = 118{\cdot}63$ lbs.

The weight of steam condensed by mixing it with feed-water may be found by the following formula:—

Let W = the weight of water in the feed-tank in lbs.
T = the sensible heat of the steam, or the boiling-point of water and the condensing-point of steam under a given pressure.
To = the ultimate temperature produced by the mixture of water and steam in degrees Fahr.
t = the temperature of the water used to condense the steam with.
L = the latent heat of the steam.
w = the weight of steam condensed in lbs.

Then, the weight of steam condensed in heating the water is:—

$$w = \frac{W \times (To - t)}{L + (T - To)}.$$

Example: The exhaust-steam of a non-condensing engine is delivered into a feed-tank at a pressure of 2 lbs. per square inch above the pressure of the atmosphere. The tank contains 10 gallons of water at 62° Fahr. The water is raised in temperature by the exhaust-steam to 182° Fahr. Required the weight of steam condensed and added to the water?

Then, from Table 37, page 105, the boiling-point of water under a pressure of 2 lbs. per square inch above that of the atmosphere is = 218°·6 Fahr., and the latent heat of the steam is = 961·4. The weight of water to be heated is = 10 gallons × 10 lbs = 100 lbs, and

$$\frac{100 \times (182° - 62°)}{961{\cdot}4 + (218°{\cdot}6 - 182)} = \frac{12000}{998} = 12{\cdot}02 \text{ lbs}$$, the weight of steam condensed in heating the feed-water, and consequently the total weight of water in the tank is = $100 + 1{\cdot}2 = 112$ lbs.

Temperature of Feed-Water Heated by Exhaust-Steam.—The temperature of the water produced by the mixture of water and steam in heating water by discharging exhaust-steam into it, may be found by the following formula:—

Let L = the latent heat of the steam, from Table 37.
T = the sensible heat of the steam.
t = the temperature of the water used to condense with.

C = the weight of water in lbs. used to condense each lb. of steam with,

To = the ultimate temperature produced by the mixture of water and steam in degrees Fahr.

Then the temperature of the mixture of water and steam is :—

$$\text{To} = \frac{L + (T - t)}{C + 1} + t.$$

For instance, taking the data from the previous example, the weight of water used to condense each lb. of steam is = 100 lbs. of water ÷ 12·02 lbs. of exhaust steam = 8·319 lbs.

Then, $\frac{961{\cdot}4 + (218{\cdot}6 - 62)}{8{\cdot}319 + 1} + 62° = 182°$ Fahr. the temperature resulting from the mixture of steam and water.

The pressure of steam necessary to drive a non-condensing engine against its own friction and expel the exhaust-steam from the cylinder, is generally about 5 lbs. per square inch. The latent heat of steam of a pressure of 5 lbs. per square inch above the pressure of the atmosphere is = 955·3, and the temperature of the steam is = 227·2° Fahr., which data may be taken for general calculations.

When exhaust-steam is used in this manner for heating feed-water for steam-boilers, the cylinder of the engine should be lubricated only with good mineral oil, having a vaporising point very much higher than the temperature of the steam in the boiler, in order that the oil mixed with the feed-water may have no injurious effect on the boiler.

Saving Effected by Heating Feed-Water by Waste Fuel-Gases. —When a feed-water heater is employed, which is heated by the products of combustion, the water can be raised to a much higher temperature than when heated by exhaust steam.

The total heat required to convert water to steam varies with the temperature of the steam, but it is frequently convenient in estimating the saving effected by heating feed-water to take the temperature of evaporation in the boiler at 212° Fahr. On this basis, if a water-heater heated by the products of combustion raised the temperature of feed-water from 48° Fahr. to 330° Fahr.

a saving of $\frac{1178 - 330}{1178 - 48} = {\cdot}75$ or ·25 per cent. would be effected.

Therefore 75 lbs. of coal would evaporate the same quantity of water heated to 330° as 100 lbs. of coal with feed-water supplied at 48° Fahr.

Saving Effected by Heating Feed-Water to Various Temperatures.—The saving effected by heating feed-water from 50° Fahr. to various temperatures, and evaporating it in a steam-boiler at 212° Fahr., is in round numbers as follows :—

TABLE 75.—SAVING EFFECTED BY HEATING FEED-WATER WHEN EVAPORATED AT 212° FAHR.

Temperature of Feed-water entering the Boiler.	Saving when Evaporated at 212° Fahr.	Temperature of Feed-water entering the Boiler.	Saving when Evaporated at 212° Fahr.
Degrees Fahr.	Per cent.	Degrees Fahr.	Per cent.
106	5	270	20
130	7	280	21
150	9	290	22
160	10	310	23
170	11	320	24
180	12	330	25
190	13	340	26
200	14	350	27
210	15	360	28
230	16	370	29
240	17	380	30
250	18	390	31
260	19	400	32

In condensing engines, the feed-water is generally taken from the hot-well at about 100° Fahr., and the saving effected by raising the temperature of the water from 50° to 100° Fahr. is $= 1178 - 100 = 1078 \div (1178 - 50) = \cdot955$, or $4\frac{1}{2}$ per cent.

The previous method of calculating the saving effected by using hot feed-water as compared with cold water, is sufficiently accurate for most practical purposes. Where great accuracy is required, it is necessary to take the total heat of the steam corresponding to the required pressure, because the total heat varies slightly with the temperature at which the water is evaporated. The total heat of steam of the required pressure may be taken from Table 37, pages 105—111, but it is necessary to add 32 to the total heat there given, because the temperature of the feed-water is measured from zero of Fahrenheit scale.

For instance, if the pressure of steam in a boiler, as shown by the steam-gauge is 100 lbs. per square inch, and the feed-water is raised in temperature from 50° to 280° Fahr. in passing through a heater heated by the products of combustion.

Then, the total heat of steam of 100 lbs. per square inch pressure is from Table 37, $= 1184\cdot9$, and $1184\cdot9 + 32 = 1216\cdot9$ units of heat, reckoned from zero of Fahrenheit scale, and $\frac{1216\cdot9 - 280°}{1216\cdot9 - 50°} = \cdot80$, or the saving effected by using hot instead of cold feed-water is 20 per cent. in this case. This result is one per cent. less than that given by the previous method of calculation.

Feed-Water-Heaters Heated by Exhaust-Steam.—A coil water-heater is shown in Fig. 261. The feed-water is forced by a pump slowly through a coil-pipe, which is enclosed in a cylindrical shell through which the exhaust-steam from the engine passes. This form of heater is not very effective, because the water passes so rapidly through the coil that it cannot

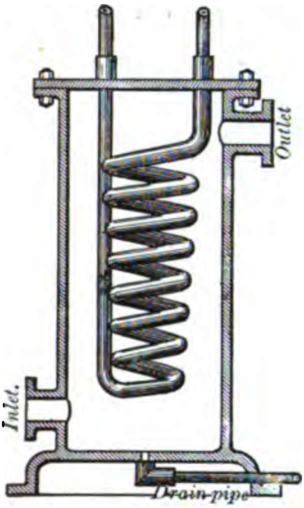


Fig. 261.—Feed-water-heater.

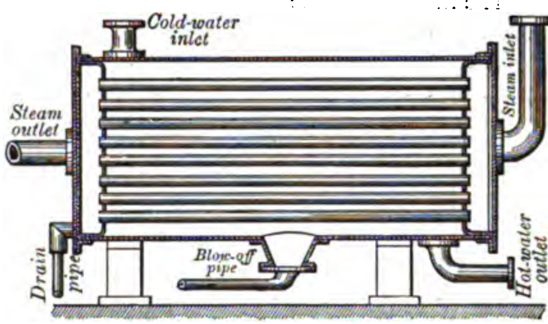


Fig. 262.—Horizontal feed-water-heater.

be raised to the highest possible temperature, and the pipes are liable to become choked with scale. A better arrangement is to force the water through the shell and pass steam through the coil.

A horizontal water-heater is shown in Fig. 262. The feed-water is forced through a cylindrical shell containing a number of small tubes, through which the exhaust-steam from the engine passes.

A vertical feed-water-heater is shown in Fig. 263. The feed-water is forced slowly through a cylindrical shell containing a number of small brass-tubes, through which the exhaust-steam from the engine passes. The large capacity of the heater affords time during the passage of the water through it for the impurities to be deposited to a greater

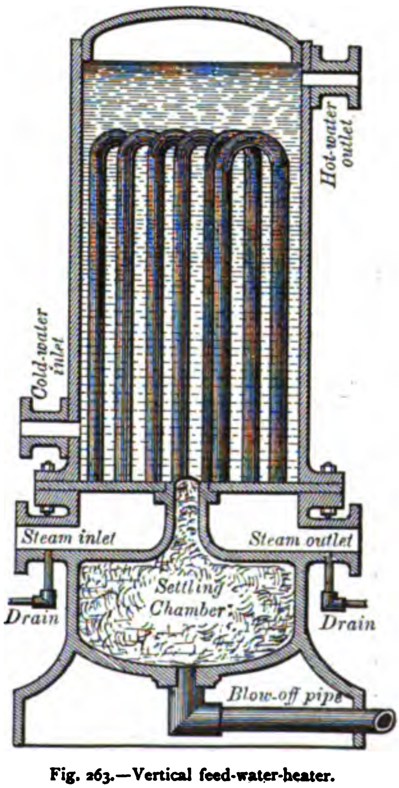


Fig. 263.—Vertical feed-water-heater.

or less degree, and the water may be raised to from 170° to 207° Fahr., according to the efficiency of the heater.

The proportions of this feed-water-heater may generally be as follows:—

The internal diameter of the cylindrical shell in inches should be at least = the diameter of the exhaust-pipe of the engine in inches multiplied by 4.

The height from the bottom to the top of the heater may be = the diameter of the shell multiplied by 3 to 4.

The total heating-surface in square feet may be = the indicated horse-power of the engine multiplied by 1·5 to 2.

In heating water by exhaust-steam in a well-arranged heater, one square foot of heating-surface is sufficient for each three gallons of water forced through the heater per hour, but many heaters contain from $1\frac{1}{2}$ to 2 times as great a proportion of heating-surface as this.

Heat Transmitted by Water-Heaters.—The heat transmitted by the heating-surface of the heater may be found by the following formulæ:—

Let W = the weight in lbs. of the water heated.
T = the temperature to which it is raised by the heater in degrees Fahr.
t = the temperature of the water to be heated.
M = the number of minutes occupied in heating the water.
H = the heating-surface of the heater in square feet.

The heat-units per hour per square foot of heating-surface is =

$$\frac{W \times (T - t) \times 60}{M \times H}.$$

For instance, if 63 gallons or 63 × 10 = 630 lbs. of water, at 53° Fahr. are raised to 188° Fahr. in ten minutes, by steam entering the heater at a pressure of 5 lbs. per square inch, or a temperature of 227·2° Fahr., and the heating-surface of heater is 20 square feet.

$$\text{The heat-units transmitted are} = \frac{630 \text{ lbs.} \times (188 - 53) \times 60 \text{ minutes}}{10 \text{ minutes} \times 20 \text{ square feet}} =$$

25515 units per square foot of heating-surface per hour.

The mean temperature of the water is = (53 + 188) ÷ 2 = 120·5° Fahr., and the difference of temperature between that and the steam is = 227·2 − 120·5 = 106·7, and the heat transmitted per hour per square foot per degree of difference of temperature is = 25515 ÷ 106·7 = 239 units.

Fuel-Economisers.—An economiser is a water-heater formed of pipes, placed in the main flue between the boiler and the chimney for the purpose of economising fuel by utilizing, in heating the feed-water, a portion of the heat of the products of combustion which would otherwise escape to waste up the chimney.

It is more economical to employ an economiser than to increase the length of a Lancashire boiler beyond that equal to four times the diameter

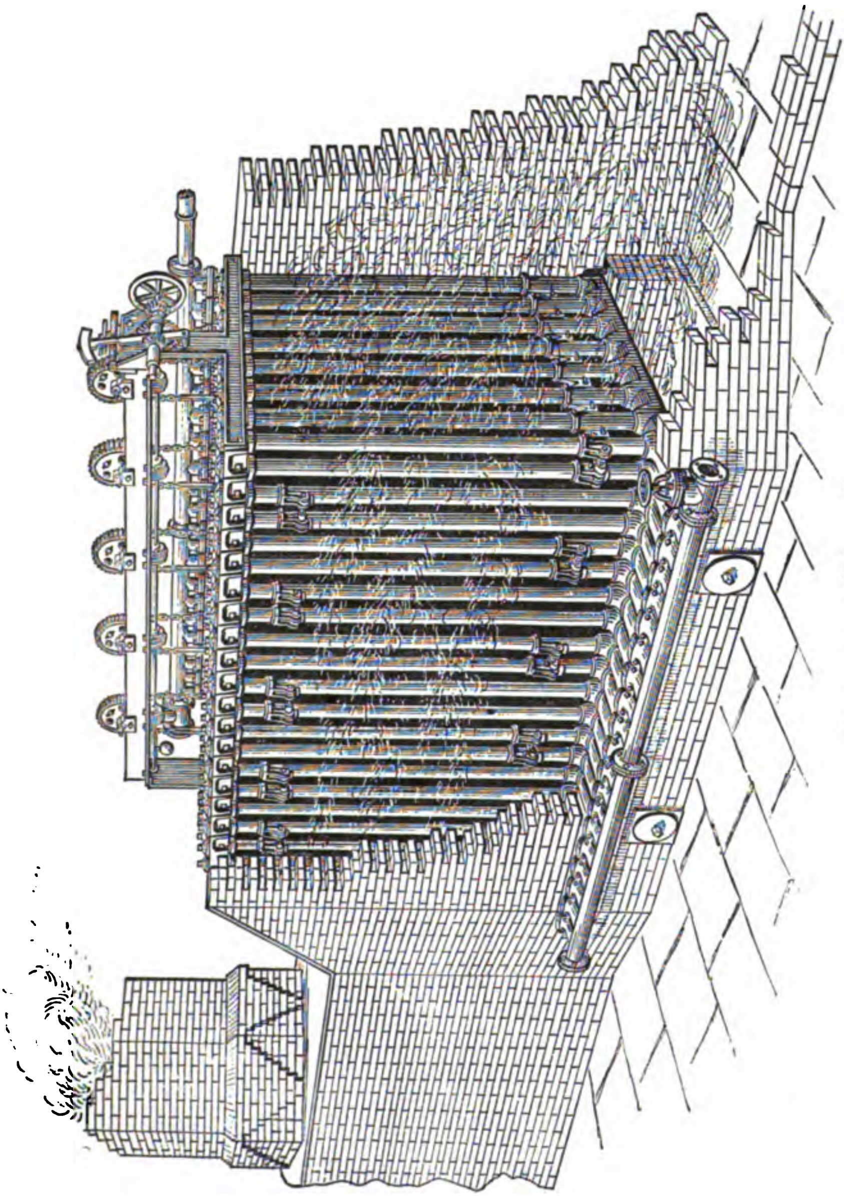


Fig. 264.—Green's fuel-economiser.

of the boiler, with the object of providing a greater area of absorbing-surface for the fuel-gases to traverse. Because, the pipes of the economiser being maintained free from soot by self-acting scrapers are much more efficient in absorbing heat than the back-end of the boiler, as its heating-surfaces are covered with a non-conducting coating of soot. Again, there is too small a range of temperature, or difference between the temperature of the boiler and that of the escaping fuel-gases, to permit the absorption of much heat by the plates of the back-end of the boiler. But in heating feed-water taken from the hot-well at a temperature of about 100° Fahr., there is a great difference between the temperature of the feed-water and that of the products of combustion, and a considerable quantity of heat is available for absorption by the heating-surfaces of the economiser.

The efficiency of the economiser depends greatly upon the efficiency of the apparatus employed for maintaining the outside of the pipes free from soot, and upon the inside of the pipes being kept clean and free from scale and mud.

The economiser, shown in Fig. 264, consists of a number of vertical pipes of 4 inches diameter and 9 feet long, connected at the top and bottom by cross-tubes, termed boxes. These boxes are connected by top and bottom branch-pipes, placed lengthways on opposite sides of the economiser, and on the outside of the brickwork with which it is encased. The feed-water is pumped into the economiser at the lower branch-pipe nearest the point of exit of the fuel-gases, and emerges from the upper branch-pipe nearest the point where the fuel-gases enter.

The velocity of the water through the economiser is from $\frac{1}{8}$ to 1 inch per minute according to its sectional area. It is frequently $\frac{1}{4}$ inch per minute.

Proportions of Economisers.—The number of pipes of the economiser, shown in Fig. 264, required for a Lancashire boiler is determined by the area of its fire-grate. The economiser should consist of $2\frac{1}{2}$ pipes for each square foot of fire-grate surface of the boiler. The water contents of the economiser per pipe is about 7 gallons, or $= 7 \times 10 =$ 70 lbs.

For instance:—A Lancashire boiler with a fire-grate surface of 32 square feet, should have an economiser with $32 \times 2{\cdot}5 = 80$ pipes; the water contents of which is about $80 \times 7 = 560$ gallons, or $560 \times 10 =$ 5600 lbs.

It is customary to consider the whole surface of the pipes as effective heating-surface, and that each pipe contains 9 square feet of heating-surface. Therefore the total heating-surface of an economiser of 80 pipes is $= 80 \times 9 = 720$ square feet.

The temperature of the feed-water on leaving the economiser is from 230° to 350° Fahr., according to the temperature of the products of combustion. A considerable saving of fuel is therefore effected by the employment of an economiser.

Test of an Economiser.—The results of a test of an economiser is given in the following table:—

TABLE 76.—RESULT OF A TEST OF STEAM-BOILERS WORKING WITH AND WITHOUT A GREEN'S ECONOMISER.

Particulars of Test.	Boilers working with the Economiser.	Boilers working without the Economiser.
Duration of Test . . . . . in Hours	$11\frac{1}{2}$	$11\frac{1}{2}$
Weight of coal consumed in pounds . . .	7856	10282
Working pressure of steam by the steam-gauge, in pounds per square inch	58	57
Temperature of the feed-water entering the Economiser in degrees Fahr.	88	...
Temperature of feed-water entering the Boilers, degrees Fahr.	225	85
Number of degrees the feed-water was heated by the Economiser	137	...
Temperature of the fuel-gases entering the Economiser in degrees Fahr.	618	...
Temperature of the fuel-gases entering the chimney in degrees Fahr.	365	645
Number of degrees the fuel-gases were cooled by the Economiser	253	...
Water evaporated per pound of coal from and at 212° Fahr.	10·613	8·235
Saving effected by using the Economiser, per cent.	28·9	...

Advantages of Feed-Water-Heating.—Numerous other advantages may be gained by the use of a feed-water heater besides economy of fuel. With an intermittent supply of cold feed-water, the water in the boiler is continually alternating in temperature, resulting in alternate expansion and contraction, which seriously strain the structure and frequently cause leaking joints. By heating the feed-water, the temperature throughout the boiler is rendered more uniform, and the effects of unequal expansion and contraction are greatly reduced, resulting in increased durability of the boiler. Air, grease, impurities, and scale-forming matter, are, to a greater or less extent, released from the water and collected in the heater. These substances would otherwise pass into the boiler and produce deleterious effects. A cold feed-supply produces so many evil effects that it is barbarous treatment of a steam-boiler to feed it with cold water.

Evaporators for Producing Fresh-water from Sea-water.—Evaporators for evaporating sea-water for auxiliary feed-water for marine-boilers are small steam-boilers, in which the heat necessary for evaporating the water is supplied by condensing steam in rows or coils of tubes forming the heating surface of the evaporator.

The area of the heating-surface of coils of copper-tube necessary for evaporating sea-water in an evaporator may be found approximately by the following *Rule*:—

Square foot of surface of copper-coil required for each gallon of sea-water evaporated per hour =

$$\frac{1\cdot 4}{\sqrt[3]{\text{Absolute pressure of steam in the coils}}}.$$

For instance, if the coils of copper-pipe of an evaporator be heated by steam of 100 lbs. per square inch absolute pressure, then, $1\cdot 4 \div \sqrt[3]{100} =$

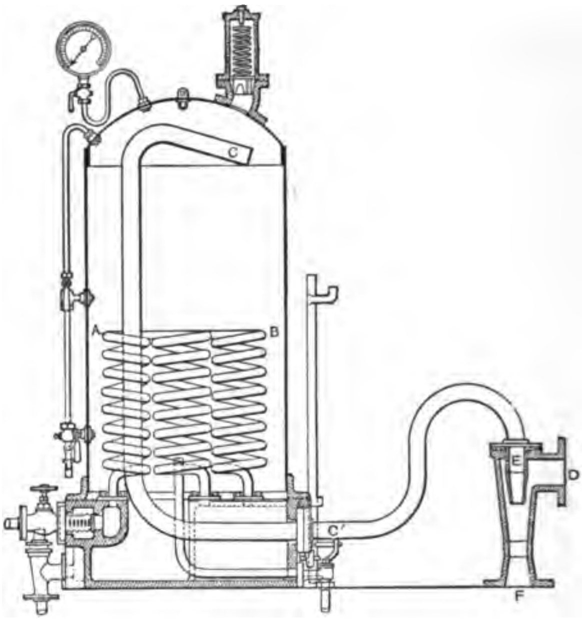


Fig. 265.—Morison's sea-water evaporator. Fig. 266.—Water-heater.

·14 square foot of coil is required for each gallon of sea-water to be evaporated per hour. This is the minimum area of heating-surface that should be provided. The feed-supply may be as given at page 323.

There are numerous forms of evaporators, two of which may be briefly described as follows :—

Morison's Evaporater.—This apparatus consists of an evaporator combined with a feed-water heater, as shown in Figs. 265 and 266. The sea-water is pumped into a cylindrical vessel, placed over a box-shaped casting, on which are fixed a number of heating-coils of solid-drawn copper-pipe. The heating-coils are arranged in pairs ; the steam from the boiler passes up through one set A, and down through the other set B, the condensation-water falling into a receiver formed in the bottom casting.

The steam generated in the evaporator is discharged into the water-heater, and is utilized in heating the feed-water. All the available heat is, therefore, utilized, except that lost by radiation from surface of the evaporator. The steam and the condensation-water pass from the evaporator through the pipe C to the heater. The feed-water enters the heater at D, and mingles with the steam entering at E, becomes heated, and flows out at F, and is pumped into the boilers. The evaporator is filled with a steam-gauge, water-gauge, and safety-valve.

The results of experiments with this evaporator when evaporating sea-water against an absolute pressure of 18 lbs. per square inch are given in the following Table :—

TABLE 77.—RESULTS OF EXPERIMENTS WITH MORISON'S EVAPORATOR.

Absolute pressure of steam in the coils in lbs. per square inch .	35	45	55	65	75
Area of coil-surface in square feet provided for each gallon of sea-water evaporated per hour .	·271	·210	·175	·154	·139
Sea-water evaporated in gallons per square foot of coil-surface per hour	3·65	4·76	5·71	6·53	7·19

The steam for the coils is generally taken from the steam-pipe of the steering-engine, so that fresh-water may be produced when in port, but it may be taken from the steam-chest of the intermediate cylinder of a triple-expansion engine.

Weir's Evaporator.—This evaporator, shown in Fig. 267, consists of a cylindrical shell, having heating-surfaces composed of a number of U-shaped tubes having contracted outlet-ends. Any steam which passes through the small holes in the contracted ends, is condensed in a tube, which returns through the evaporator and carries off the water of condensation to the hot-well. By this means there is a lower pressure at the outlet-ends, and a constant current is maintained through all the tubes, which prevents the accumulation of air and water. Steam for heating the evaporator is generally taken from the intermediate receiver of triple-expansion engines, and steam generated in the evaporator is discharged to the low-pressure receiver.

A number of experiments were made by Mr. Lang with a Weir's Evaporator, consisting of a cylindrical shell 3 feet diameter, and 4 feet 3 inches long. The heating-surface was composed of 12 solid-drawn copper tubes, $1\frac{1}{2}$ inches external diameter, and 10 B.W.G., or ·134 inch, thick, having a total heating-surface of 38 square feet, which was reduced to 21·95 square feet in some of the experiments. The results of some of the experiments are given in the following Table :—

TABLE 78.—RESULTS OF EXPERIMENTS WITH WEIR'S EVAPORATOR.

Heating-surface of tubes in square feet	38	21·95	21·95	21·95	21·95
Absolute pressure of steam in lbs. per square inch	50	80	105	135	165
Temperature of steam in the tubes in degrees Fahr.	281	312	331·3	350·1	366
Temperature of steam in the shell of the heater in degrees Fahr.	231·1	234·2	255·9	254·4	259·3
Weight of water from steam condensed in the tubes in lbs.	112	56	56	112	56
Time occupied, in minutes and seconds	3m. 8½s.	1m. 32¼s.	1m. 33s.	2m. 10s.	1m. 3½s.
Steam condensed in the tubes per hour in lbs.	2138·9	2185·36	2167·7	3101·5	3174·8
Latent heat of steam in the shell of heater	949·9	949·9	934	937·9	931·6
Units of heat given up by 1 lb. of steam	930·4	927·5	919·6	907·4	903·2
Units of heat transmitted per square foot of heating surface per hour	52343	92341	90816	128214	130637
Water evaporated per square foot of heating-surface per hour in lbs.	55·10	97·21	97·23	135·33	140·23
Units of heat transmitted per square foot of heating surface per hour per 1 degree Fahr. difference of temperature	1092	1187	1204	1286	1224
Water evaporated per square foot of heating-surface per hour per 1 degree Fahr. difference of temperature	1·146	1·249	1·289	1·357	1·314

The heat transmitted and the water evaporated per square foot of heating-surface per hour given in the above Table are calculated by the following formulæ:—

Let Q = the quantity of steam condensed in the tubes in lbs. per hour.
S = the total heating-surface of the evaporator in square feet.
H = the number of units of heat given up by each lb. of steam condensed in the tubes of the evaporator.
T = the temperature of the steam in the tubes of the evaporator in degrees Fahr.

t = the temperature of the steam in the shell of the evaporator in degrees Fahr.

L = the latent heat of the steam in the shell of the evaporator.

Then, the units of heat transmitted per square foot of heating-surface per hour are =

$$\frac{Q \times H}{S}.$$

The water evaporated per square foot of heating-surface per hour in lbs. is =

$$\frac{Q \times H}{S \times L}.$$

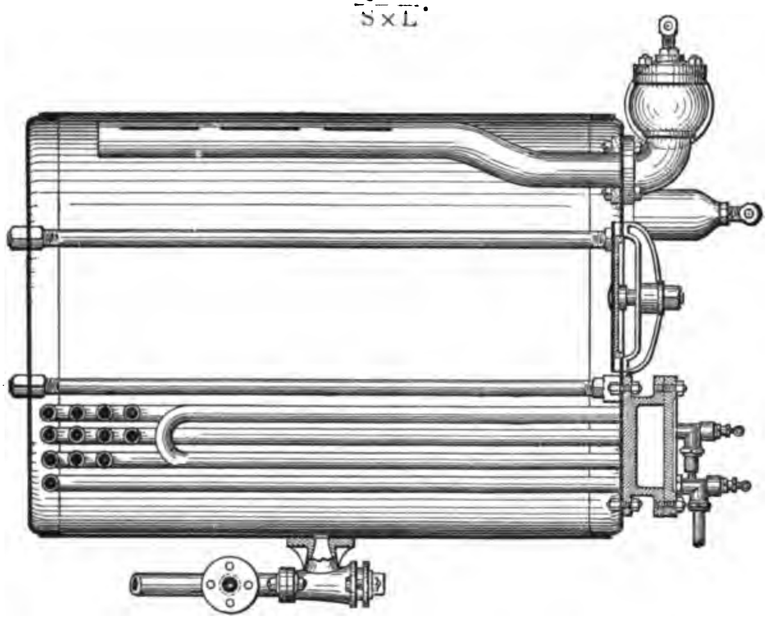


Fig. 267.—Weir's sea-water evaporator.

The units of heat transmitted per square foot of heating-surface per hour for 1 degree Fahr. difference of temperature are =

$$\frac{Q \times H}{S \times (T - t)}.$$

The water evaporated per square foot of heating-surface per hour in lbs. for 1 degree Fahr. difference of temperature is =

$$\frac{Q \times H}{S \times L \times (T - t)}.$$

In one of the experiments with this evaporator 1334 units of heat were transmitted per square foot of heating-surface per hour for 1 degree Fahr. difference of temperature between the tubes and the shell of the evaporator.

Blowing-off the Water from Steam-Boilers.—Boilers should not be blown-off while hot, because, rapid cooling causes sudden contraction, which may result in cracked or ruptured plates and stays. It is also detrimental in causing mud and scale to become baked on the heating-surfaces.

When it is necessary to cool a boiler quickly, the steam should be blown off, and the boiler filled with water, the blow-off cock should then be opened and cold water allowed to run in as fast as the hot water runs out, until the temperature of the boiler is reduced to about that of the atmosphere.

Boilers set in brick-work should not be emptied while the brick-work is hot.

Blowing-off should be frequently practised in boilers using feed-water producing deposits of either mud or scale.

Hydraulic-Pump for Testing Steam-Boilers.—A hydraulic-test-pump is shown in section in Fig. 268. It is of gun-metal, fitted on the top of a cast-iron tank on wheels; the tank is 26 inches long, $16\frac{1}{2}$ inches wide, and 15 inches deep. The barrel of the pump is $2\frac{1}{8}$ inches internal diameter, and $\frac{1}{2}$ inch thick. The suction-pipe is $\frac{3}{4}$ inch internal diameter, and the delivery-pipe is $\frac{1}{2}$ inch internal diameter. The ram of the pump is 2 inches diameter, and its stuffing-box is packed with leather-washers of the usual hydraulic form. The pump is fitted with a pressure-gauge, safety-valve with lever and weight, and with a suction-valve, delivery-valve, and release-valve. It is worked by a lever with socket-handle, attached to a cast-iron guide-standard.

Hydraulic Test for Steam-Boilers.—The hydraulic test is useful for ascertaining the tightness of seams, and for revealing indications of weakness, structural defects, and the bulging or movement under pressure of oval tubes and flat surfaces. The test-pressure should not be less than one-and-a-half times the working pressure, which is sufficient for the above purposes, and it should in no case exceed twice the working pressure, otherwise the materials may be overstrained and permanently injured. In testing a new boiler all the mountings should be fixed, and no blank-flanges should be used. The hydraulic pressure should remain on at least fifteen minutes without sign of leakage.

In applying the hydraulic test to old boilers, care should be taken to avoid the application of an unduly high pressure, as it may either create defects where none previously existed, or so much overstrain a weak part as to cause permanent injury, and also aggravate existing defects and contribute to the ultimate failure of the boiler. Careful observations should be made of the behaviour of the furnace-tube and flat-surfaces under the test-pressure, to prevent overstraining and starting of a depression or bulge in the plates.

Cold-water pressure is a very severe test for a boiler, and may cause leaks from joints which would be perfectly tight under an equal pressure of steam. Hot-water pressure is preferable to cold, as it gives proper expan-

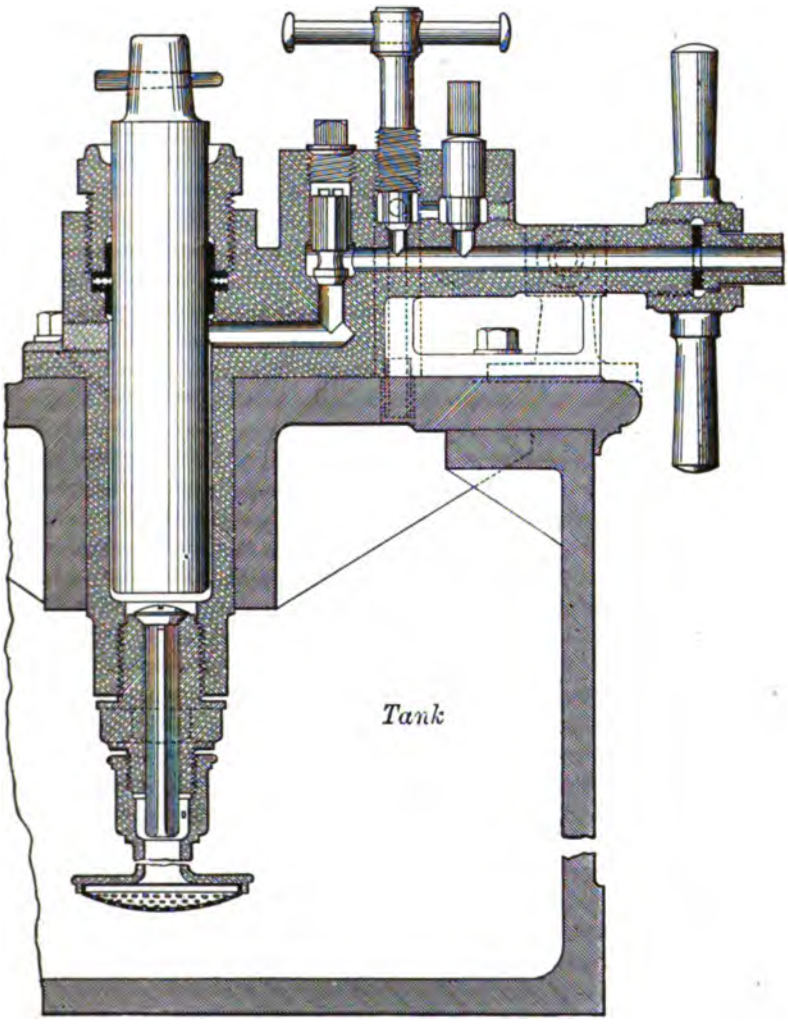


Fig. 268.—Hydraulic-pump for testing boilers.

sion to all the parts, and the boiler is strained approximately in the same manner as under ordinary working conditions, excepting strains from unequal expansion. Hot water creeps through very minute crevices and searches a joint thoroughly, it also exposes a leak more readily than steam

A test-pressure of this kind reaches every part and exposes defects which, through position or inadvertence, might escape observation.

But a successful hydraulic test does not afford conclusive evidence of the strength or safety of a boiler. It must be preceded by a consideration of the particular construction of the boiler, and a careful examination of its condition both internally and externally, before a fair estimation can be formed of its strength and safety.

Inspecting and Testing Steam-Boilers.—As it is impossible to determine by inspection the quality of the plates of a steam-boiler, an examination is confined to the ascertainment of the state or general condition of a boiler, and the excellence of its design. The boiler should be carefully examined externally for wasting from corrosion and leaks, and their location recorded. All internal seams should be carefully examined and the effects of corrosion or pitting noted. All the stays should be examined and tested by light blows of a hammer to reveal by sound if they are cracked or broken. After thorough examination externally and internally, the boiler should be subjected to a careful test by hot water at a temperature of from 180 to 208 degrees Fahr., and at a pressure which exceeds the working-pressure by 50 per cent.

The pressure should be applied after the boiler has had time to become sufficiently warm to produce uniform expansion of the plates. While this pressure is on, all exposed surfaces of the boiler should be tested with light blows of a hammer, and all flat stayed surfaces proved with a straight edge, and any change of form which would indicate weakness in design or material, and also defects of workmanship, should be marked and recorded.

Fire-box stud-stays, or other stud-stays, may have a hole $\frac{1}{8}$ inch diameter, and deep enough to bring it a little way into the water space, drilled in them from the outside, any stays which leak under this test should be replaced by new ones. The stays generally break on the inside of the fire-box-shell, consequently it is not necessary to drill the stays from the inside of the fire-box. Stays drilled in this manner should be left afterwards with the holes open, and not plugged.

When the test-pressure is removed, permanent set, or alteration of shape produced, should be noted, and search made for indications of weakness developed, or of undue strain having been thrown on individual members of the structure during the test.

A boiler should be inspected and tested in this manner periodically, say, once a year for the first three years of its life, and every six months thereafter. Efficient periodical inspections and tests are necessary for the safe working of steam-boilers. By this means defects and deterioration may be observed which would otherwise escape notice, and might lead to accident or cause explosion.

When a boiler is set in brick-work, proper inspection cannot be made unless that portion of the brickwork in contact with the plates be removed.

It is important that these parts be seen at every examination, as they are liable to secrete corrosion. The covers of the manhole and mudhole, and the fire-bars and fire-bridge should also be removed, and the damper should be tightly closed to stop the draught, so that the inspector may work comfortably by candle-light. It is essential to a proper examination that every part of the boiler shall be clean.

Method of Testing the Evaporative Performance of a Steam-Boiler.—The object of testing a steam-boiler should be to determine its

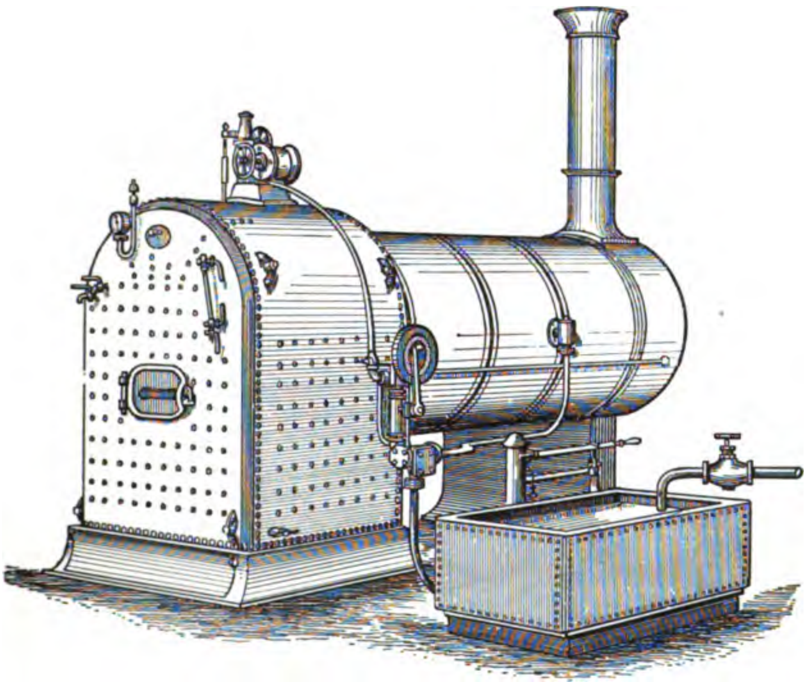


Fig. 269.—Method of determining the water-consumption in the evaporative test of a steam boiler.

evaporative efficiency under ordinary working conditions, and not under conditions differing widely from those obtained in practical working.

In conducting an evaporative test, it is necessary to have facilities for accurately determining the consumption of coal and water by the boiler. The consumption of coal may be determined by stoking the fire from weighed heaps of coal. It is convenient to provide the coal-supply for the test in sacks, each containing exactly 100 lbs. weight.

The consumption of water may be ascertained by either measuring or weighing the feed-water, the latter being the most reliable method. The measurement of the feed-water may be readily effected by a calibrated feed-tank.

To determine the water-consumption by weighing the feed-water, a tank should be placed upon a weighing machine, as shown in the arrangement for testing a boiler in Fig. 269. The suction-pipe of the feed-pump is connected to the tank by a short piece of flexible, or india-rubber, tube. The tank may, for convenience of calculation, be filled with an even number of pounds of water, say 600, and when the pump starts forcing feed-water into the boiler, it should continue to work until exactly 500 pounds of water have been taken from the tank, when the cock on the suction-pipe should be closed, and the tank refilled with exactly the same quantity of water ready for the next feeding of the boiler. This method of feeding should be continued during the test.

For large boilers two or more feed-tanks on weighing machines are necessary, the use of which permits the water to be supplied continuously to the boiler, which is an advantage.

In order that the quantity of water pumped into the boiler may accurately represent the quantity of water evaporated during the test, it is necessary that the water-level in the boiler and the pressure of the steam be precisely the same at the end as at the beginning of the test.

A chemical analysis of the coal should be made, and its calorific value determined by calculation and by experiment with a calorimeter.

Preparatory to starting the evaporative test of a steam-boiler, the steam should be raised to the required working-pressure. The fire should then be quickly removed from the grate, the damper closed, the ash-pit cleaned and the height of the water-level and pressure of the steam noted. Then commence the test by re-lighting the fire with known weights of coal and wood. Stoke the fire from weighed heaps of coal, charge into the furnace all cinders that fall into the ash-pit, and adopt the same manner of firing as employed under the ordinary working conditions of the boiler. Regulate the draught by the damper so as to maintain a nearly uniform pressure of steam, without permitting waste of steam by the safety-valve blowing off. The boiler should be worked continuously without stopping at meal-times. The temperatures of the atmosphere, the feed-water, and the products of combustion, should be frequently taken, and the mean of each during the test should be determined at the end of the test. At the close of the test the fire should be removed from the grate, and the unburnt fuel weighed and deducted from the quantity of coal fired during the test. The height of the water-level and the pressure of the steam should be the same at the close as at the beginning of the test.

When a steam-boiler is tested under ordinary working conditions, and it is not convenient to stop the engine while the fire is withdrawn and re-lighted, it is necessary to obtain the same condition of fire at the end as at the beginning of the test. This may be obtained by allowing the fire to burn down until the pressure of the steam falls a few pounds below the required working-pressure. Then commence the test by stoking the fire with weighed coal. At the end of the test, the fire should be allowed to burn down until the working-pressure of the steam falls to the same point as at the commencement of the test.

Another method of treating the fire, is to allow it to burn down until ready for stoking, clear out the ashes and clinkers, and note the condition and level of the fire. At the end of the test the fire should be permitted to burn down to the same condition as at the beginning of the test.

Calculations Relating to Boiler-Tests.—The evaporation is frequently estimated per pound of constituent carbon of the coal. In some cases a sample of the coal is dried in an oven, and the evaporation is estimated per pound of dry coal, but this is not necessary when it is desired to ascertain the evaporation under ordinary working conditions. The method of computing the various data usually tabulated in testing a steam-boiler may be illustrated by the following *Example* :—

In the trial of a boiler of locomotive type of a portable-engine, with one cylinder, of 8 nominal horse-power, 3249 lbs. of water were evaporated during a run of 6 hours, with a coal-consumption of 342 lbs., the engine developing 19 indicated horse-power. Heating-surface of boiler 170 square feet; area of fire-grate surface $5\frac{1}{4}$ square feet; working-pressure of steam by the steam-gauge 76 lbs. per square inch; temperature of the feed-water 60° Fahr.; calorific power of the coal 14800 units per lb. of coal.

Then, the coal-consumption per hour is = 342 lbs. ÷ 6 hours = 57 lbs. per hour; or = 57 lbs. ÷ 5·25 square feet = 10·857 lbs. of coal consumed per square foot of fire-grate surface per hour; and = 57 lbs. ÷ 170 square feet = ·335 lb. of coal consumed per square foot of heating-surface per hour.

The clinker and ashes weighed 22 lbs., or (22 lbs. × 100) ÷ 342 lbs. of coal = 6·43 per cent. of the quantity of coal consumed.

The water-consumption was 3249 lbs., or = 3249 ÷ 10 = 324·9 gallons.

The water evaporated was = 3249 ÷ 6 hours = 541·5 lbs. per hour, or 541·5 ÷ 10 = 54·15 gallons per hour: or = 541·5 ÷ 5·25 square feet = 103·14 lbs. per square foot of fire-grate surface per hour, and = 103·14 ÷ 10 = 10·314 gallons per square foot of fire-grate surface per hour. The water evaporated per square foot of heating-surface was = 541·5 ÷ 170 square feet = 3·185 lbs. per hour, and = 3·185 ÷ 10 = ·3185 gallon per hour.

The water evaporated from feed-water at a temperature of 60° Fahr. was = 3249 lbs. of water ÷ 342 lbs. of coal = 9·5 lbs. for each lb. of coal consumed.

The total heat of steam of a pressure of 76 lbs. per square inch is, from Table 37, page 106, = 1179·9 units. The factor of evaporation is = (1179·9 + 32°) − 60°, the temperature of the feed-water, = 1151·9 ÷ 966 = 1·192, and 9·5 lbs. of water × 1·192 = 11·324 lbs., the equivalent weight of water evaporated from and at 212° Fahr.

The heat evolved by combustion was = 342 lbs. of coal × 11·324 lbs. of water × 966 units per lb. = 3741133 units. The efficiency of a boiler is:—

$$\text{Efficiency} = \frac{\text{heat utilized}}{\text{heat applied}}.$$

The calorific power of the coal is = 14800 units per lb., therefore the efficiency of this boiler is =

$$\frac{3741133 \text{ units of heat}}{342 \text{ lbs. of coal} \times 14800 \text{ units}} = \cdot 74,$$ or 74 per cent., that is, presuming all the water to have been evaporated and none was carried from the boiler with the steam from priming.

This engine developed during the test 19 indicated horse-power, therefore the consumption of coal was = 57 lbs. of coal consumed per hour ÷ 19 = 3 lbs. of coal per indicated horse-power per hour. The actual horse-power of this boiler, if estimated by the rule for the power of boilers for simple engines on page 163, is = 541·5 lbs. of water evaporated per hour × 1·192, the factor of evaporation = 645·46 lbs. of water ÷ 34·5 lbs. of water per horse-power = 18·7 horse-power, or nearly the same as that developed in this test.

The temperature of the atmosphere was 64° Fahr.

The mean temperature of the products of combustion in the smoke-box was 392° Fahr.

The weight of air used for combustion, calculated from the analysis of the fuel-gases, was 23·65 lbs. per lb. of coal = 23·65 + 1 = 24·65 lbs. of gases per lb. of coal. The heat carried away by the gases is = 392° − 64 = 328° × 24·65 lbs. × ·238 specific heat = 1924 units; or = (1924 × 100) ÷ 14800 units, the calorific value of the coal, = 13 per cent. of the total heat developed by combustion.

These data are collected and arranged in the following Table:—

TABLE 79.—RESULTS OF A TRIAL OF THE BOILER OF A PORTABLE ENGINE OF EIGHT NOMINAL HORSE-POWER.

Heating-surface of boiler	in square feet	170
Area of fire-grate surface	in square feet	$5\frac{1}{4}$
Air-spaces of fire-grate	in square feet	1·25
Ratio of fire-grate surface to heating surface		$\frac{1}{32 \cdot 38}$
Duration of trial	hours	6
Designation of coal		Welsh
Calorific value of 1 lb. of coal	in thermal units	14800
Weight of coal consumed during the test	in lbs.	342
Weight of coal consumed per hour	in lbs.	57
Weight of coal consumed per square foot of fire-grate per hour	in lbs.	10·857

TABLE 79 *continued*.—RESULTS OF A TRIAL OF THE BOILER OF A PORTABLE ENGINE OF EIGHT NOMINAL HORSE-POWER.

Weight of coal consumed per square foot of heating-surface per hour . . . . in lbs.	·335
Weight of clinker and ash . . . . . . in lbs.	22
Weight of clinker and ash in parts of coal consumed, per cent.	6·43
Pressure of steam, average during the test, in lbs. per square inch	76
Temperature of the atmosphere . . in degrees Fahr.	64°
Mean temperature in the smoke-box . in degrees Fahr.	392°
Temperature of feed-water . . . in degrees Fahr.	60°
Weight of water fed into the boiler during the test . in lbs.	3249
Quantity of water fed into the boiler during the test, in gals.	324·9
Water evaporated per hour . . . . . . in lbs.	541·5
Water evaporated per hour . . . . . in gallons	54·15
Water evaporated per square foot of fire-grate surface per hour . . . . in lbs.	103·14
Water evaporated per square foot of fire-grate surface per hour . . . . in gallons	10·314
Water evaporated per square foot of heating-surface per hour, in lbs.	3·185
Water evaporated per square foot of heating-surface per hour, in gallons.	·3185
Water evaporated from feed-water at 60° . per lb. of coal	9·5
Equivalent evaporation from and at 212° Fahr., per lb. of coal	11·324
Factor of evaporation	1·192
Efficiency of boiler . . . . per cent.	74
Power developed by the engine during the test, indicated horse-power	19
Power of the boiler, estimated by calculation, indicated horse-power	18·7
Coal consumed per hour per indicated horse-power, developed by the engine . . . . . . in lbs.	3·0
Heat carried away by the products of combustion, per cent.	13

BALANCE SHEET SHOWING THE DISTRIBUTION OF THE HEAT.

Heat Developed per Pound of Coal.	Units.	Heat Expended per Pound of Coal.	Units.
Calorific value of 1 lb. of coal	14800	Heat expended in evaporating the water in the boiler = 14800 × ·74 . . =	10952
		Heat carried away by the products of combustion = 14800 × ·13 . =	1924
		Heat lost by imperfect combustion, radiation, and other sources = 13 per cent. = 14800 × ·13 =	1924
	14800	Total units .	14800

Dryness of Steam in Boiler-Tests.—The quantity of water carried with the steam varies considerably in different kinds of boilers.

When steam is generated in boilers which generally produce steam containing so little moisture as to be in a practically dry state, it is seldom necessary to test the dryness of the steam. But where great accuracy is required, or when steam is generated in types of boilers liable to produce moist steam, the quality of the steam should be tested by a calorimeter.

Fig. 270 —Calorimeter for testing the dryness of steam.

Calorimeter for Determining the Quantity of Moisture in Steam.—Steam taken from steam-boilers is never perfectly dry, but always contains more or less moisture, carried with it in the form of spray from the boiler. If the weight of water necessary for the condensation of a given weight of steam, and the range of temperature through which it is raised in the operation, be known, the quantity of the moisture in the steam may be readily calculated. The apparatus used for the purpose is termed a calorimeter.

A simple and efficient form of calorimeter, consists of a barrel placed upon a weighing machine as shown in Fig. 270. The barrel has a cover through which a rod is passed, having a disc at the end, for the purpose of stirring the mixture of water and steam to obtain a uniform temperature. Steam from the main steam-pipe is admitted through a pipe fitted with a rose at the bottom-end, placed a little above the bottom of the barrel, to permit tranquil discharge of the steam into the water.

The steam-pipe is provided with a branch, or waste-pipe, for the purpose of discharging steam to waste for a few minutes, in order to heat the pipe, and obtain steam free from water of condensation before steam is admitted to the calorimeter. When the barrel and pipes are fixed in position, they are thickly clothed with felt to prevent loss of heat by radiation.

The steam-pipe of the calorimeter should project a little into the main steam-pipe, in order to avoid drawing off water of condensation and water carried with the steam along the surface of the pipe.

In using the calorimeter, a certain weight of cold water, of say not less than 80 lbs., is poured into the barrel and the temperature of the water is noted.

Steam is then discharged through the waste-pipe until the pipe is properly heated, and the steam appears to be free from water of condensation.

The cock on the waste-pipe is then closed, steam is admitted to the calorimeter, and when 4 lbs. of steam are added to the water, the steam is turned off. The water is mixed with the stirring-rod and its final temperature noted.

With the data thus obtained, the quality of the steam may be determined by the following formulæ:—

Let T = the final temperature of the mixture of steam and water in degrees Fahr.
t = the initial temperature of the cooling water.
W = the weight of cooling water in lbs.
w = the weight of steam in lbs. mixed with the water.
L = the latent heat of steam of the given pressure.
H = the sensible heat of the steam, or the boiling point of water and the condensing-point of steam under a given pressure.
D = the weight of dry steam contained in the number of lbs. of steam as supplied from the boiler and discharged into the calorimeter.

$$\text{Then } D = \frac{[(T - t) \times W] - [(H - T) \times w]}{L}.$$

The percentage, p, of dry steam contained in 1 lb. of steam from the boiler is =

$$p = \frac{D \times 100}{N},$$

in which N = the number of lbs. of steam mixed with the water in the calorimeter.

Example: In a calorimeter containing 80 lbs. of water at a temperature of 56° Fahr., 4 lbs. of steam of a pressure of 76 lbs. per square inch by the steam-gauge were discharged, which resulted in raising the temperature of the water to 110° Fahr. Required the percentage of dry steam contained in the steam as supplied by the boiler, and also the percentage of moisture or water carried from the boiler with the steam?

Then, the boiling point of water under steam of 76 lbs. per square inch pressure is, from Table 37, page 106, = 320·6° Fahr., and the latent heat of the steam is = 888·9 units.

The weight of dry steam contained in the steam as supplied by the boiler is, =

$$\frac{[(110^\circ - 56^\circ) \times 80 \text{ lbs.}] - [(320^\circ{\cdot}6 - 110^\circ) \times 4 \text{ lbs.}]}{888{\cdot}9} = \frac{4320 - 842{\cdot}4}{888{\cdot}9} =$$

3·91 lbs. of dry steam, the steam therefore contains 4 lbs. — 3·91 lbs. = ·09 lb. of water.

The weight of dry steam contained in 1 lb. of the steam as supplied by the boiler is =

$$\frac{3{\cdot}91 \text{ lbs.} \times 100}{4 \text{ lbs.}} = 97{\cdot}75 \text{ per cent.}$$

The percentage of moisture carried with the steam from the boiler in the form of spray is $= 100 - 97{\cdot}75 = 2{\cdot}25$ per cent.

In practice the quantity of moisture in steam generally averages from 1 to 4 per cent. It is however frequently more, and is generally considerable in boilers in which steam is generated in confined spaces and free circulation is not obtained. The quantity of water carried with the steam from the boiler, as determined by a calorimeter, should be deducted from the quantity of water apparently evaporated in the test of a boiler.

For instance, if the percentage of moisture in steam be 4 per cent., and 20,000 lbs. of water were apparently evaporated by the boiler, then $20{,}000 \times {\cdot}04 = 800$ lbs. of water were carried from the boiler unevaporated, and the quantity of water actually evaporated is only, $20{,}000 - 800 = 192{,}000$ lbs.

The quantity of water carried with the steam was in some boiler-tests as follows:—

	lbs. per square inch.		per cent.
Lancashire boiler, working pressure .	78	Water in the steam .	1·56
Cornish boiler ,, ,,	70	,, ,,	2·38
Multitubular boiler ,, ,,	92	,, ,,	2·87
Water-tube boiler ,, ,,	83	,, ,,	3·46

The boilers were worked at less than their full capacity.

Evaporation per square foot of Heating-Surface of Boilers of different Types.—The evaporative power of different types of boilers, measured in lbs. of water evaporated per square foot of heating-surface per hour, varies considerably. The weight of water that may be evaporated from and at 212° Fahr. per hour per square foot of total heating-surface of boilers of different types, of average proportions, has been found by careful tests to be as follows:—

	lbs.		lbs.
Locomotive boilers . . .	13	Lancashire boilers . . .	6
Portable-engine boilers of locomotive type	10	Cornish boilers	5
Marine return-tube boilers .	9	Water-tube boilers of various types, maximum average of .	4

This is the maximum evaporation obtainable in a general way with each type of boiler when fired with good coal and having ordinary draught, that is, natural draught in all except the locomotive and portable boilers which have steam-blast in the chimney.

Evaporative Tests of Steam Boilers of Various Types.—The evaporative power of a boiler varies with the quality of the coal, the strength of the draught, and the efficiency of the combustion. The results of a number of carefully conducted evaporative tests of modern steam-boilers of different types are given in the following Tables, as representative examples of their efficiency. The evaporative performances of these boilers show the results which may generally be obtained in practice with boilers having tolerably clean heating-surfaces, when fired with good coal. The tests were made with ordinary hand-firing, and with draught of the strength usually obtained with each type of boiler under ordinary working conditions.

TABLE 80.—RESULTS OF A NUMBER OF EVAPORATIVE TESTS OF GALLOWAY, LANCASHIRE, AND CORNISH BOILERS.

Size of Boiler. Diameter in Ft.	Size of Boiler. Diameter in Ins.	Size of Boiler. Length in Feet.	Area of Fire-grate in Square Feet.	Total Heating Surface in Square Feet.	Description of Coal used.	Coal Consumed per square foot of Fire-grate Surface per Hour in Pounds.	Water Evaporated per Pound of Coal from and at 212° Fahr. in Pounds.
Galloway Boilers, shown in Fig. 189.							
7	0	26	33	748	Welsh	3·27	12·83
7	0	28	36	853	Steam-coal	18·07	12·15
7	0	28	39	973	„	7·27	11·72
7	0	27	32	780	„	19·65	10·75
7	0	26	33	748	„	6·40	10·24
Lancashire Boilers with Galloway Tubes, shown in Fig. 180.							
7	1	30	33	936	Welsh	15·9	12·02
7	1	32	29	982	„	16·6	11·60
7	6	28	24	920	„	20·2	11·25
7	0	28	27	870	Steam-coal	18·1	11·13
7	0	30	30	935	Lancashire	18·3	10·85
7	1	30	33	944	„	20·4	10·73
7	6	30	31	941	Rough slack	18·7	10·38
7	0	24	25	740	Welsh	18·6	10·32
7	0	27	25	842	Newcastle	19·8	10·27
7	1	30	33	937	„	17·2	10·18
7	0	30	23	934	„	22·3	10·00
7	1	30	31	938	Durham	18·3	9·81
7	0	30	33	898	Newcastle rough slack	17·4	9·70
7	0	30	32	894	„	18·5	9·61
7	0	28	33	886	„	16·9	9·57
7	0	26	30	810	Lancashire	17·8	9·50
7	0	18	22	583	„	23·6	9·36
7	6	28	30	916	Lancashire rough slack	18·8	9·10
7	6	28	33	923	„	17·1	8·75
7	6	28	27	910	Newcastle slack	20·2	8·40
7	0	18	22	584	„	20·9	8·32
7	6	28	33	916	Yorkshire slack	16·7	8·25
Cornish Boilers with Galloway Tubes, shown in Fig. 176.							
5	6	22	16	443	Welsh	19·5	11·56
6	0	24	18	515	„	18·7	10·75
5	0	25	18	448	Steam-coal	16·7	10·68
5	9	23	15	485	Rough slack	19·2	9·80
5	0	20	13	368	„	20·7	8·46
4	3	14	10	220	„	22·1	8·31
4	6	16	11	266	Small slack	20·4	7·75

TABLE 81.—RESULTS OF A NUMBER OF EVAPORATIVE TESTS OF MULTITUBULAR AND OTHER STATIONARY BOILERS.

Size of Boiler. Diameter in Ft. Ins.	Size of Boiler. Length in Feet.	Area of Fire-grate in Square Feet.	Total Heating Surface in Square Feet.	Description of Coal used.	Coal Consumed per square foot of Fire-grate Surface per Hour in Pounds.	Water Evaporated per Pound of Coal from and at 212° Fahr. in Pounds.
Horizontal Internally Fired Multitubular Boilers, shown in Fig. 195.						
6 7	16	28	1155	Steam-coal	16·8	11·76
5 0	12	15	483	,,	21·9	11·04
5 6	14	16	689	,,	19·7	11·32
5 9	15	18	740	,,	18·2	10·21
6 3	18	22	1192	,,	17·9	9·42
6 0	16	20	914	Rough slack	17·1	9·30
5 1	12	18	450	,,	16·3	8·40
5 1	12	16	450	,,	19·8	8·13
Horizontal Externally Fired Multitubular Boilers, shown in Fig. 200.						
5 0	15	30	948	Steam-coal	16·4	11·30
4 6	13	27	652	,,	17·6	11·06
5 6	16	28	1200	,,	18·9	10·76
4 2	12	26	523	,,	18·4	10·12
4 2	12	26	523	,,	19·1	9·88
6 0	16	33	1432	,,	16·1	9·61
4 0	12	20	476	,,	18·6	8·81
4 0	12	25	476	Rough slack	17·3	8·58
4 6	16	21	664	Anthracite	12·0	7·53
3 6	10	22	305	Rough slack	20·4	7·10
3 0	9	20	200	,,	19·4	6·50
Plain Cylindrical or Egg-ended Boilers, shown in Fig. 173.						
4 6	30	36	280	Small Slack	17·4	8·56
4 10	36	40	360	,,	16·2	8·04
4 0	26	32	240	,,	18·3	7·28
3 9	20	27	151	,,	15·6	7·11
3 0	12	22	73	,,	14·9	6·85
3 7	18	25	130	,,	13·1	6·52
Diameter of Barrel. Inches.	Length of Tubes. Feet.	*Stationary Boilers of Locomotive-Type, shown in Figs.* 197 *and* 198.				
60	16	40	1550	Steam-coal	20·0	11·55
54	15	35	1020	,,	18·3	10·30
54	15	29	1020	,,	19·6	9·61
50	15	32	844	Slack	17·8	7·90

TABLE 82.—RESULTS OF A NUMBER OF EVAPORATIVE TESTS OF VERTICAL BOILERS AND MARINE RETURN-TUBE BOILERS.

Size of Boiler. Diameter in Ft.	Ins.	Length in Ft.	Ins.	Area of Fire-grate in Square Feet.	Total Heating Surface in Square Feet.	Description of Coal used.	Coal Consumed per Square Foot of Fire-grate Surface per Hour in Pounds.	Water Evaporated per Pound of Coal from and at 212° Fahr. in Pounds.
Vertical Tubular Boilers, shown in Fig. 225.								
3	9	9	0	8·6	155	Welsh	12·8	10·21
3	6	8	6	7·5	110	,,	12·2	9·70
3	6	8	6	7·5	110	Newcastle	13·1	9·58
2	9	6	0	4·0	65	Steam-coal	13·7	8·75
2	6	5	0	3·1	32	,,	16·3	8·20
2	7	5	6	3·2	50	Slack	15·8	8·02
3	0	7	0	5·2	81	,,	12·5	7·85
Vertical Cross-tube Boilers, shown in Figs. 220 and 221.								
3	6	9	0	7·8	85	Steam-coal	12·4	8·52
3	6	9	0	7·8	85	,,	11·8	8·13
4	6	9	0	13·0	116	,,	13·0	7·96
4	6	9	0	13·0	116	,,	9·6	7·72
2	9	6	0	4·5	40	,,	15·8	7·53
3	0	7	0	5·0	45	Slack	16·1	6·95
3	3	7	6	6·6	68	,,	13·2	6·40
3	6	8	6	8·0	76	,,	12·3	5·57
Marine Return-Tube Boilers, shown in Fig. 206.								
8	0	8	3	18	510	Welsh	19·3	12·23
9	0	9	6	25	764	,,	18·4	12·00
10	0	9	6	30	948	,,	17·1	11·49
10	0	9	6	25	948	Steam-coal	19·5	11·27
11	3	9	9	33	1194	,,	17·3	11·18
11	0	10	0	36	1146	Welsh	16·5	11·10
13	0	10	6	57	1685	,,	15·6	11·00
12	6	10	8	48	1535	,,	17·9	10·86
10	6	10	0	30	1008	Steam-coal	18·7	10·28
12	9	10	6	50	1570	Newcastle	16·4	9·85
9	6	9	0	30	818	Welsh	19·8	9·58
12	6	10	9	48	1480	,,	17·1	9·41
10	0	9	6	34	960	Newcastle	17·8	9·25
9	0	9	9	24	772	,,	15·9	8·72
9	0	9	9	24	772	,,	18·8	8·56
5	0	6	0	7	151	Steam-coal	19·0	8·60
4	6	5	6	6	125	,,	18·3	8·34
4	6	5	6	6	125	,,	16·7	8·03

TABLE 83.—RESULTS OF A NUMBER OF EVAPORATIVE TESTS OF PORTABLE-ENGINE BOILERS OF ORDINARY LOCOMOTIVE TYPE, OR SIMILAR TO THAT SHOWN IN FIG. 203.

Size of Boiler. Diameter of Barrel. Ft. Ins.		Size of Boiler. Length of Barrel. Ft. Ins.		Area of Fire-grate in Square Feet.	Total Heating Surface in Square Feet.	Description of Coal used.	Coal Consumed per Square Foot of Fire-grate Surface per Hour in Pounds.	Water Evaporated per Pound of Coal from and at 212° Fahr. in Pounds.
*	—	—		4·32	226·7	Welsh	8·73	12·99
*	—	—		2·63	211·5	”	11·21	12·96
*	—	—		3·39	218·1	”	13·57	12·59
*	—	—		3·39	218·1	”	13·44	12·27
*	—	—		2·63	192·2	”	12·98	12·26
*	—	—		4·60	238·1	”	9·57	11·21
*	—	—		4·18	167·7	”	20·90	10·14
3	0	6	9	5·1	266	”	12·64	12·72
2	6	6	4	4·2	170	”	14·70	11·66
2	6	6	2	4·3	158	Steam-coal	20·04	11·55
4	0	9	1	10·4	486	Welsh	17·94	11·27
2	9	6	6	4·5	220	Steam-coal	11·90	11·20
2	7	6	2	4·1	162	Welsh	15·46	11·10
2	5	5	9	4·0	155	Steam-coal	16·21	9·50
2	2	4	8	3·5	117	”	13·34	9·12

* The tests marked * in the above Table were made by the Royal Agricultural Society.

TABLE 84.—RESULTS OF A NUMBER OF EVAPORATIVE TESTS OF LOCOMOTIVE BOILERS, OR SIMILAR TO THAT SHOWN IN FIG. 202.

Description of Engine.	Area of Fire-grate Surface in Square Feet.	Total Heating Surface in Square Feet.	Description of Coal used.	Coal Consumed per Square Foot of Fire-grate Surface per Hour in Pounds.	Water Evaporated per Pound of Coal from and at 212° Fahr. in Pounds.
Passenger engine .	21·0	1500	Steam-coal	68	12·00
” . . .	17·5	1080	”	74	10·90
” . . .	19·0	1210	”	87	10·52
Tank engine . .	16·0	906	”	85	10·48
Passenger engine .	18·0	1160	”	72	10·32
” . .	17·2	1276	”	57	10·28
” . . .	15·5	1138	”	90	10.19
Tank engine . .	13·0	600	”	76	10·12
Passenger engine .	19·0	1226	”	64	10·04
”	16·8	1310	”	66	9·73
” . . .	15·7	1141	”	86	9·22
”	17·5	1245	”	71	9·02
” . . .	15·1	1004	”	73	8·76
Tank engine . .	15·0	810	”	76	8·51
” . .	15·0	906	”	75	8·30

TABLE 85.—RESULTS OF EVAPORATIVE TESTS OF VARIOUS KINDS OF WATER-TUBE STEAM-BOILERS.

Description of Steam-Boilers.	Area of Fire-grate in Square Feet.	Total Heating Surface in Square Feet.	Description of Coal used.	Coal Consumed per Square Foot of Fire-grate Surface per Hour in Pounds.	Water Evaporated per Pound of Coal from and at 212° Fahr. in Pounds.
Thorneycroft's water-tube boiler	22·7	1410	Welsh	—	13·40
Babcock & Wilcox's water-tube boiler	25	1403	Scotch coal	13·4	12·38
Babcock & Wilcox's do. do.	25	1403	„	12·8	12·05
Babcock & Wilcox's do. do.	31	1380	Anthracite	12·8	11·22
Babcock & Wilcox's do. do.	23	1193	Steam-coal	13·7	10·70
Babcock & Wilcox's do. do.	31·1	1426	Coke	6·8	10·05
Babcock & Wilcox's do. do.	25	1610	Welsh	23	10·00
Babcock & Wilcox's do. do.	15	860	Slack	24·6	7·20
Belleville water-tube boiler	39·7	1264	Welsh	—	11·71
Belleville do. do.	39·7	1264	Briquettes	—	10·83
De Naeyer water-tube boiler	40	1700	Bituminous	14·25	11·70
Root's water-tube boiler	27	875	Anthracite	11·7	10·60
Root's do. do.	36	910	„	10·1	10·20
Root's do. do.	50	1440	„	10·0	9·50
Root's do. do.	16·0	845	Slack	28·0	8·34
Buttner's do. do.	12·6	1942	Steam-coal	31·0	10·50
Shepherd's do. do.	26·18	529	„	13·9	10·48
Shepherd's do. do.	26·18	529	„	12·9	9·98
Towne's do. do.			Coal	4·3	10·46
Harrison's do. do.	35·13	949	Anthracite	9·3	9·32
Perkin's do. do.	15·3	300	Semi-Anthracite	12·0	9·27
Herreshof coil water-tube boiler	26·0	485	Anthracite	13·0	9·10
Herreshof do. do.	26	485	„	18·0	8·10
Heine's water-tube boiler	19·4	869	Steam-coal	18·8	9·00
Steinmuller's do. do.	32·3	724	„	13·8	8·80
Watther's do. do.	15·7	996	„	17·5	8·80
Neuman's do. do.	13·7	494	„	14·5	8·70
Water-tube boiler with vertical tubes	21·2	928	„	22·7	8·33
Water-tube boiler with pendent tubes	19·6	958	„	21·9	8·10
Water-tube boiler with vertically inclined tubes	28	1600	„	20·7	7·34
do. do.	17·3	870	Slack	34	7·02
Ward's water-tube boiler			Coal	15·5	8·28
Water-tube boiler on Root's principle	33·2	1276	Steam-coal	19.9	8·17

Average Coal-Consumption in Evaporation.—Taking the average of a number of tests of steam-boilers, it appears that, the quantity of coal consumed in evaporating 1000 lbs. of water to steam, from and at 212° Fahr., under ordinary working conditions is frequently as follows :—

	Lbs. of coal used in evaporating 1000 lbs. of water.
Locomotive boilers	100
Portable-boilers of locomotive type .	104
Marine return-tube boilers . . .	108
Lancashire boilers	112
Cornish Boilers	125
Water-tube boilers	140
Vertical cross-tube boilers . . .	165

These quantities are for coal of the ordinary quality and kind common to each type of boiler.

Boilers for Supplying Steam for Heating Buildings.—The size of boiler required for supplying steam for heating a building of a given size, varies with the kind of building and the type of boiler used.

The size of boiler requisite for supplying steam for the efficient heating of buildings may, in a general way, be found by dividing the cubic contents of the building by the number corresponding to its description given in the following Table :—

TABLE 86.—NUMBER OF CUBIC FEET OF SPACE IN BUILDINGS OF VARIOUS KINDS THAT MAY BE HEATED WITH AN EFFICIENT ARRANGEMENT OF STEAM-PIPES BY ONE SQUARE FOOT OF HEATING-SURFACE OF A GOOD STEAM-BOILER.

Description of Building.			Cubic Contents of Building that may be Heated by Steam generated by One Square Foot of Heating-Surface of the Boiler.
			Cubic Feet.
Large public rooms,	of stone or brick	. .	600
Warehouses and sale-rooms,	do.	do. . .	550
Factories and work-rooms,	do.	do. . .	500
Shops and waiting-rooms,	do.	do. . .	450
Offices and dwellings,	do.	do. . .	400
Large draughty sheds,	do.	do. . .	350
Buildings of wood or galvanised sheet-iron .		. .	300
Buildings constructed largely of glass		. . .	250
Conservatories		.	125
Laundry drying-rooms		.	100

As an example of the use of this Table, suppose it is required to determine the total heating-surface of a boiler to generate steam for heating, by an efficient arrangement of steam-pipes, a factory consisting of two rooms of 80 feet long, 40 feet wide, and 14 feet high.

Then, the cubic contents of space to be heated is = 80 × 40 × 14 feet × 2 rooms = 89600 cubic feet; and a boiler will be required having a total heating-surface of 89600 ÷ 500 = 179·2 square feet.

Radiating Surface of Steam-Pipes for Heating Buildings.—The radiating surface required to heat buildings of various kinds, by steam-pipes placed on or near the floors, is given in the following Table:—

TABLE 87.—NUMBER OF CUBIC FEET OF SPACE IN BUILDINGS OF VARIOUS KINDS THAT MAY BE HEATED BY ONE SQUARE FOOT OF RADIATING SURFACE OF STEAM-PIPES PLACED ON OR NEAR THE FLOOR.

Description of Building.			Cubic Contents of Building that may be Heated by One Square Foot of Radiating Surface of Steam Pipes.
			Cubic Feet.
Large public rooms,	of stone	or brick	150
Warehouses and sale-rooms,	do.	do.	100
Factories and work-rooms,	do.	do.	90
Shops and waiting-rooms,	do.	do.	80
Offices and halls,	do.	do.	70
Large draughty sheds,	do.	do.	60
Buildings of wood or galvanised sheet-iron			50
Buildings constructed largely of glass			40
Conservatories			20
Laundry drying-rooms			10

The number of square feet of radiating surface of steam-pipes required to heat a building, may be found by dividing its cubic contents by the number in the above Table, corresponding to the kind of building.

For instance, to heat a work-room of 60 feet long, 35 feet wide, and 13 feet high, or having 60 × 35 × 13 = 27300 cubic feet of contents, requires steam-pipes having a radiating surface = 27300 ÷ 90 = 304 square feet, when the pipes are placed on or near the floor of the room.

Heating Buildings by Overhead Steam-Pipes.—In the overhead system of heating buildings by steam, horizontal rows of pipes are placed round the room near to the ceiling. The pipes are suspended in racks, fixed about 3 feet from the wall, and at the same distance from the ceiling. They are generally 1¼ inches diameter, and 1 foot in length of pipe of this diameter will heat 85 cubic feet of space. If the cubic contents of the space to be heated be divided by 85, the quotient is the length of overhead steam-pipe of 1¼ inch diameter required to heat it.

For instance, to heat a work-room having 34,000 cubic feet of contents, with over-head steam-pipes, requires a length of steam-pipe = 34000 ÷ 85 = 400 feet, when the pipes are 1¼ inches diameter, and they are placed about 3 feet from the ceiling, and at the same distance from the walls of the room.

Steam-Boiler Explosions.—The explosion of a steam boiler is not an accident, and is always preventable, because it proceeds from a cause which might have been foreseen, the defect could have been remedied, and the explosion prevented.

The causes from which explosions proceed are very numerous and various, but are principally as follows:—Weakness, and defects in the design, construction, or workmanship. Improper treatment, carelessness, neglect, or ignorance on the part of the boiler-attendant. Wasting from wear, tear, and corrosion. Overpressure, worn-out condition, and overheating from shortness of water. Defective condition of safety-valves and other mountings. Exposure to conditions which cause the development of defects in, or general deterioration of, the structure.

Power Liberated by the Explosion of a Steam-Boiler.—In the explosion of a steam-boiler an enormous amount of energy stored in the water and steam is suddenly liberated. The destructive effect of the explosion of a steam-boiler is produced by the sudden expansion of the liberated mass of water and steam from the pressure existing at the instant of the explosion down to that of the atmosphere.

The energy stored in the steam and liberated by the explosion of a boiler, may be found by multiplying the units of latent heat given up by the steam, in expanding from the pressure existing at the instant of the explosion down to atmospheric pressure, by 772, and by the weight of steam in pounds in the steam-space of the boiler. The weight of steam in a boiler is so small that the energy stored in the water veiy greatly exceeds that stored in the steam.

The energy stored in the water and liberated by the explosion of a boiler, may be found by the following formula:—

Let T = the temperature of the steam above that of the atmosphere at the instant of the explosion.

Let W = the weight of the water in the boiler in pounds.

The energy in foot-pounds stored in the water of a boiler and liberated by explosion is =

$$\frac{(T - 212^\circ)^2 \times 772}{1135 + T} \times W$$

As an example of these rules, calculation may be made of the power liberated by the explosion of a Lancashire boiler of 7 feet 6 inches diameter, and 30 feet long, weighing 34300 lbs., and containing 60 lbs. of steam and 42100 lbs. of water, the pressure of the steam at the instant of the explosion being 100 lbs. per square inch by the steam-gauge.

Then from Table 37, page 105, the latent heat of steam of atmospheric pressure is 966·1 units, and that of steam of 100 lbs. per square inch pressure is 876·5 units, and each lb. of steam contains 966·1 − 876·5 = 89·6 units of heat above that contained after expansion to atmospheric pressure. The energy stored in the steam and liberated by the explosion is = 89·6 units × 772 × 60 lbs. of steam = 4150272 foot-pounds.

The temperature of steam of 100 lbs. per square inch pressure is, from Table 37, $= 337^\circ{\cdot}7$ Fahr., and the energy stored in the water and liberated by the explosion is =

$$\frac{(337^\circ{\cdot}7 - 212^\circ)^2 \times 772}{1135 + 337{\cdot}7} = \frac{12197978{\cdot}28}{1472{\cdot}7} = 8283 \times 42100 \text{ lbs. of water}$$

= 348714300 foot-pounds + 4150272 foot-pounds contained in the steam

$= 352864572$ foot-pounds of total energy, and $= \dfrac{352864572}{33000} = 10693$ indicated horse-power released by the explosion of the boiler.

The power required to project a weight of one lb. one mile high is 5000 foot-pounds, and the power liberated by the explosion is sufficient to project the boiler to a height of,

$$\frac{352864572 \text{ foot-pounds liberated by the explosion}}{34300 \text{ lbs. weight of the boiler,} \times 5000 \text{ foot-pounds}} = 2{\cdot}05 \text{ miles.}$$

Boiler-Explosions from Internal Corrosion.—Boilers have been prematurely worn out by using feed-water of a corrosive nature, due to acids either present in the water, or proceeding from organic matter or grease introduced into the boiler with the feed-water. In some cases the whole of the interior surface of the boiler below the water level has been seriously corroded from this cause, as shown in Fig. 271.

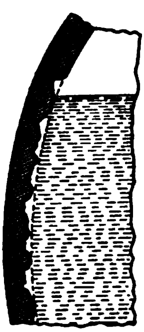


Fig. 271.—Corroded boiler-shell.

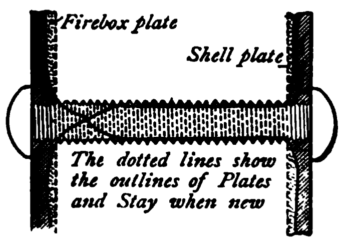


Fig. 272.—Corroded plates and stay.

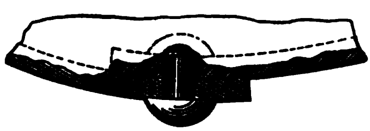


Fig. 273.—Boiler-shell wasted by internal corrosion.

Internal corrosion from acids in the feed-water was the cause of the explosion of a boiler in which the fire-box and stays were much wasted away, some of the stays being eaten completely through as shown in Fig. 272.

Weakness due to internal corrosion was the cause of a boiler explosion in which the plates were wasted away, as shown in Fig. 273. The boiler was worked without examination of any kind until the structure gave way.

The explosion of the Cornish boiler shown in Fig. 274,* was caused by internal corrosion, which wasted the furnace-tube so much that it was unable to bear the working pressure. It collapsed and ruptured the crown of the furnace.

Fig. 274.—Explosion of a Cornish boiler from internal corrosion.

Fig. 275.—Explosion of a vertical boiler from internal corrosion.

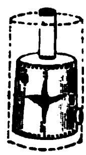


Fig. 276.—Explosion of a vertical boiler from internal corrosion.

Fig. 277.—Explosion of a vertical boiler from internal corrosion.

Fig. 278.—Explosion of a vertical boiler from corrosion of the uptake.

In the explosion of the boiler shown in Fig. 275, the plates of the fire-box were so much thinned by corrosion as to cause it to collapse.

Internal corrosion was the cause of the explosion of the vertical boiler shown in Fig. 276. The fire-box was so much reduced in strength by corrosion that it collapsed, and the boiler was projected through the roof of the building.

The explosion of the vertical tubular boiler shown in Fig. 277, was caused by internal corrosion. The upper ends of the tubes were reduced by corrosion to $\frac{1}{16}$ inch in thickness, and the shell-crown-plate to $\frac{1}{4}$ inch in thickness. The crown-plate was forced over the ends of the tubes and bulged upwards, the crown of the fire-box bulged downwards.

Numerous explosions of vertical cross-tube boilers have occurred from corrosive wasting of that portion of the uptake which passes through the steam-space.

This was the cause of the explosion of the boiler shown in Fig. 278. The uptake was so much thinned by corrosion that it ruptured in two places near the water-line.

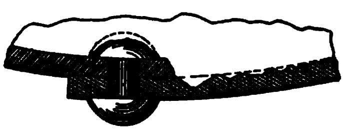


Fig. 279.—Boiler-shell weakened by corrosive grooving.

Fig. 280.—Explosion of a locomotive boiler from corrosive grooving.

Boiler-Explosions from Corrosive Grooving.—Grooving or furrowing is due to the action of corrosion on a surface or point at which expansion or alteration of shape is localized. Corrosive grooving has been a prolific source of boiler explosions. In locomotive boilers formed with

* The author is indebted for a number of engravings of boiler-explosions to Mr. E. B. Marten, of Stourbridge.

longitudinal lap-joints, a longitudinal buckling action is caused by alternations of pressure, owing to the plates not being truly circular, which results in the formation of a groove along the edge of the longitudinal seam of the barrel as shown in Fig. 279. The remedy for this evil is to form the plates a true circle by employing a butt-joint.

Corrosive grooving was the cause of the explosion of the locomotive boiler shown in Fig. 280. It ruptured at a deeply furrowed longitudinal seam, and the rent spread round the boiler. A test-hole drilled near to the line of furrow had failed to reveal it.

Corrosive grooving was the cause of the explosion of the locomotive boiler shown in Fig. 281. It ruptured at the left side central longitudinal seam where weakened by grooving.

In Cornish and Lancashire boilers, corrosive grooving may take place in the end-plates in the vicinity of the angle-hoops, as shown in Fig. 282.

Corrosive grooving was the cause of the explosion of the cylindrical boiler shown in Fig. 283. The back end-plate was so much weakened by grooving in the angle-hoop, caused by the movement of the flat end-plate under varying pressure, that the end was blown out, and the shell was thrown from its seat.

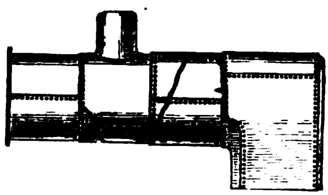


Fig. 281.—Explosion of a locomotive boiler from corrosive grooving.

Fig. 283.—Explosion of a cylindrical boiler from corrosive grooving.

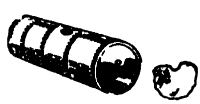


Fig. 284.—Explosion of a boiler from corrosive grooving.

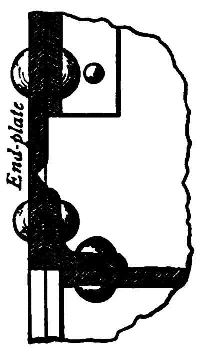


Fig. 282.—End-plate of Lancashire boiler weakened by corrosive grooving.

Corrosive grooving was the cause of the explosion of the boiler shown in Fig. 284. The back end-plate was grooved at the root of the flange and reduced to $\frac{1}{16}$ inch in thickness, and the lower portion of the plate was blown out.

In vertical boilers, grooving frequently takes place at the root of the flange, or outside the flange on the crown of the fire-box, at the bottom of the uptake. It is also liable to take place near the flange, or angle-hoop, on the outside of the bottom of the fire-box.

In the explosion of the boiler shown in Fig. 285, the flange at the base of the uptake was so much weakened by corrosive grooving that it ruptured, and the top of the fire-box bulged down.

Grooving has frequently been induced by improper caulking, which resulted in the skin of the plate being cut through, and an indent made along the outside of the lap of the joint.

Boiler-Explosions from External Corrosion.—External corrosion has been the cause of numerous boiler explosions. It may proceed from the boiler being placed in a damp situation, or from leakage and lodgment of water in contact with the plates.

In the explosion of the boiler shown in Fig. 286, the plates were so much wasted by external corrosion that the fire-box was unable to bear the usual working-pressure, and a piece was blown out of one of the plates.

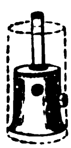


Fig. 285.—Explosion of a vertical boiler from corrosive grooving.

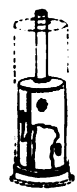


Fig. 286.—Explosion of a vertical boiler from external corrosion.

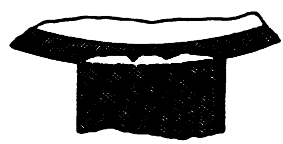


Fig. 287.—Explosion of a Cornish boiler from external corrosion.

In an explosion of a Cornish boiler, the bottom of the shell, where it rested on a centre-wall, or mid-feather, had wasted away by external corrosion for a considerable portion of its length, the plates being in some places less than $\frac{3}{16}$ inch thick, as shown in Fig. 287. The corrosion arose from moisture draining into the brick-seatings from the surface of the ground, which was on a higher level. Boilers should not be set in this way, as the mid-feather harbours moisture from leaking seams and other sources, and corrodes the plates.

Fig. 288.—Explosion of a Cornish boiler from external corrosion.

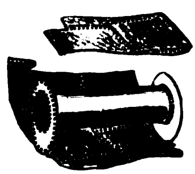


Fig. 289.—Explosion of a Cornish boiler from external corrosion.

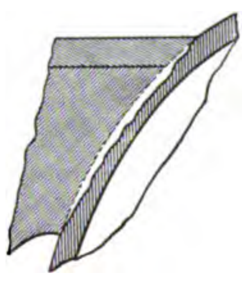


Fig. 290.—Corroded shell-plate of Lancashire boiler.

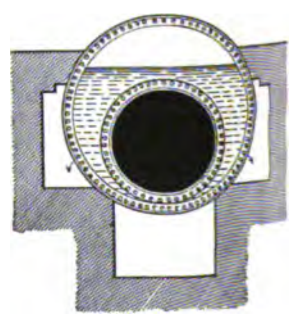


Fig. 291.—Explosion of a Cornish boiler from external corrosion.

External corrosion was the cause of the explosion of the Cornish boiler shown in Fig. 288. A piece was blown out of the bottom where thinned by external corrosion, and the issuing contents turned the boiler upside down.

In the explosion of the Cornish boiler shown in Fig. 289, the front shell-plates near the left side-wall were so much thinned by external corrosion as to be unable to bear the usual working-pressure, and they ruptured, some of the plates were blown out, and the boiler was turned upside down.

In the explosion of a boiler, the plates were seriously corroded under the brickwork of the top of the side flues, as shown in Fig. 290, which was excessively thick, and harboured moisture from a leaking seam.

In the explosion of a boiler, it had been improperly set on broad seatings as shown in Fig. 291; a layer of damp soot had collected in each side flue, and seriously corroded the plates. The seatings of boilers should be raised above the bottom of the side flues to prevent injury to the plate by accumulations of this kind, the seatings should be narrow, and if there are longitudinal seams near the bottom of the boiler they should be kept clear of the seatings.

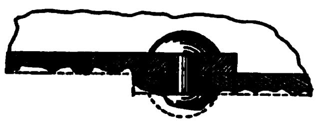


Fig. 292.—Corroded shell-plate of boiler from contact with damp brick-work.

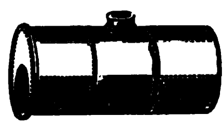


Fig. 293.—Explosion of Cornish boiler from external corrosion.

Damp brickwork in contact with the boiler has been a fruitful source of external corrosion. In the explosion of a boiler, the brickwork in contact with the plates had not been removed at the inspections of the boiler, and the plates had become much thinned, as shown in Fig. 292, by corrosion, which had been going on for some time undetected.

In the explosion of the Cornish boiler shown in Fig. 293, the right side of the back belt of plates was so much thinned by external corrosion where it rested on the brickwork that it ruptured, the rent extended round the boiler, and the plate was peeled off.

The outside of the cross-tubes of vertical boilers is frequently wasted by corrosion from rain passing down the chimney when the boilers are not at work, and from condensation-water dropping from the exhaust-pipe when steam-blast is used in the chimney.

The front end-plates of Cornish and Lancashire boilers are frequently corroded from contact with damp ashes. The hot ashes should be removed some distance from the boiler before they are slacked, otherwise the front wall is liable to become damp and produce corrosion.

Boiler-Explosions from Overheating.—The parts of a boiler liable to become overheated are the surfaces adjacent to the fire. Overheating can only take place when a plate is not in contact with water. It may occur if steam be in contact with the plate instead of water; and from any obstruction to the transmission of heat, either through the plate, or from the plate to the water. Overheating may arise from defective circulation, shortness of water, and from excessive incrustation or deposit on the heating-surfaces preventing the transmission of heat from them. When the crowns of furnace-tubes are overheated and become red-hot, they are so much weakened that the tubes are liable to collapse and rupture at the ordinary working-pressure of the steam.

The furnace-tube of the boiler shown in Fig. 294, collapsed from shortness of water.

In the explosion of the boiler shown in Fig. 295, the crown of the furnace-tube collapsed and ruptured from shortness of water.

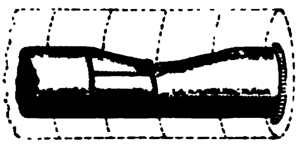


Fig. 294.—Explosion of a boiler from a collapsed furnace-tube.

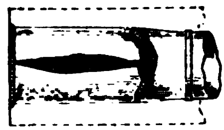


Fig. 295.—Explosion of a boiler from collapse of the furnace-tube.

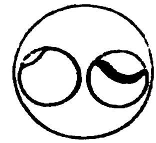


Fig. 296.—Explosion of a Lancashire boiler from the collapse of both of the furnace-tubes.

In the explosion of the Lancashire boiler shown in Fig. 296, both furnace-tubes collapsed and ruptured from shortness of water.

An accumulation of sediment on the top of a furnace-tube caused it to bulge down, as shown in Fig. 297.

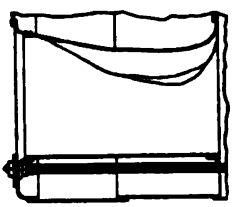


Fig. 297.—Collapse of the furnace-tube of a boiler from accumulation of sediment.

Grease or oil on the surface of a plate is a serious obstacle to the transmission of heat, and may cause overheating. Those oils which are most readily decomposed, or organic oils, such as lard-oil, rapeseed, linseed, and other vegetable oils, are most liable to cause overheating, and mineral oils are the least dangerous.

In non-condensing engines the exhaust-steam is frequently discharged into feed-tanks, and carries with it grease from the cylinders of the engines. In condensing-engines the water from condensed steam is used over and over again, and a considerable quantity of oil or grease from the cylinders becomes mixed with the feed-water. The grease mixes with sediment in the boiler and forms a greasy deposit, which, when it settles on the heating-surfaces, maintains the water out of contact with the plates, and causes overheating and bulged furnace-crowns.

Muddy feed-water is liable to cause overheating. A new multitubular boiler using dirty feed-water from a pond, had only been at work about two months when the tubes became burnt and destroyed from overheating caused by incrustation of mud.

Overheating from shortness of water has been caused in many cases by leakage through blow-off cocks inadvertently left partly open. To prevent accidents of this kind, the blow-off cock should be fitted with a spanner-guard, which will not permit the spanner to be withdrawn until the cock is closed. Boilers have frequently become short of water from leakage through fractured blow-off pipes, that is, the pipes connecting the blow-off cocks to the boilers.

In the explosion of the boiler shown in Fig. 298, the blow-off cock was left open too long in partially blowing-off, and the furnace-tubes collapsed from shortness of water.

Shortness of water and overheating caused the furnace-tube, shown in Fig. 299, to bulge down and rupture immediately over the fire, resulting in the explosion of the boiler.

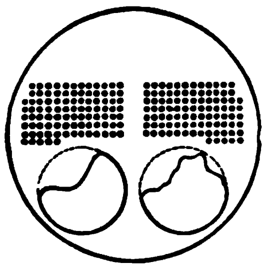


Fig. 298.—Explosion of marine return-tube boiler from collapse of the furnace-tubes.

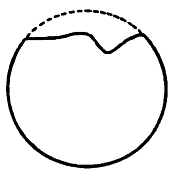


Fig. 299.—Collapsed furnace-tube.

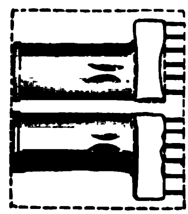


Fig. 300.—Explosion of a marine return-tube boiler from collapsed plates.

Fig. 301. Bulged top of combustion-chamber.

In the explosion of the marine boiler shown in Fig. 300, the plates became red-hot from shortness of water, and were unable to bear the working-pressure. The crowns of the furnaces slightly collapsed, the plates of the combustion-chamber bulged, and the end-plates of the chamber were forced over some of the stays.

The crowns of the combustion-chambers of marine return-tube boilers have frequently bulged and collapsed from over-heating.

Shortness of water caused the top of a combustion-chamber to become over-heated and bulge down, as shown in Fig. 301.

Boilers having heat applied to the shell-plates have frequently become overheated from having the side-flues carried above the water-level of the boiler.

Shortness of water has occurred from boiler-attendants having been misled by false indications of water-gauges, caused by the passages having become choked with sediment or mud.

Explosions of stationary boilers have occurred from shortness of water, owing to the attendants having been misled by float-gauges in a defective condition. These gauges require accurate adjustment, and should be frequently examined, and kept in good order and in a sensitive condition.

Overheating has frequently been caused by defective circulation due to cramped water-spaces.

Overheating has frequently taken place in that portion of the uptake of vertical cross-tube boilers which is above the water-line, or passes through the steam-space. It occurs most frequently in cases where the uptakes are not protected by fire-clay, or other lining, from destructive impingement of flame.

Deposits of scale on the heating-surfaces of steam-boilers is a frequent cause of overheating.

Overheating caused by deposit of scale was the cause of the explosion of the Lancashire boiler shown in Fig. 302. One of the furnace-tubes collapsed sideways and ruptured the angle-hoop at each end. The plates were so much weakened by overheating that they were unable to bear the usual working-pressure.

The vertical-boiler shown in Fig. 303, had the lower cross-tube bulged and cracked from deposit of scale.

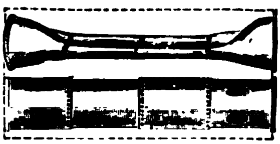


Fig. 302.—Explosion of a Lancashire boiler from collapsed furnace-tube.

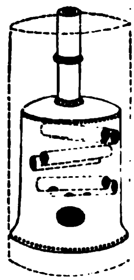


Fig. 303.—Explosion of a vertical boiler from fractured cross-tube.

Fig. 304.—Explosion of a boiler from collapsed furnace-tube.

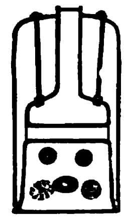


Fig. 305. — Explosion of a vertical boiler from collapsed fire-box.

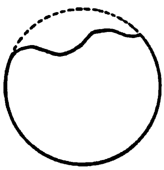


Fig. 306.—Bulged furnace-tube.

Overheating has been frequently caused in boilers fed with sea-water from deposit of salt. This was the case in the boiler shown in Fig. 304; the furnace-crown collapsed and allowed the contents of the boiler to escape, the plates having become overheated from accumulation of salt.

In the boiler shown in Fig. 305, the fire-box partly collapsed near the fire-hole, the plates were overheated from deposit of salt.

Explosions of vertical boilers have frequently been caused by overheating, bulging and rupture, of the plates of the fire-box at a little above the level of the fire-bars, from accumulation of salt at the bottom of the water-space of the fire-box.

The tubes of multitubular boilers frequently become overheated and burnt from deposits of scale. In some cases on removal of the tubes, some of them have been found completely embedded in scale and petrified into a solid mass.

Overheating has been frequently caused by the injudicious use of compositions for the prevention of scale. Chemical compositions should never be put into a steam-boiler. Overheating has in some cases resulted from the composition yielding a greasy deposit which mixed with the sediment, adhered to the heating-surfaces, and prevented contact of water. In other cases the compositions have driven the scale off in large pieces, which accumulated on the heating-surfaces and caused overheating. This was the cause of the furnace-tube bulging shown in Fig. 306; a composition for removing scale was put into the boiler, and caused the scale to become detached in large flakes, which settled in a heap along the top of the furnace-tube, and became baked on it during the night when the steam was down, and caused overheating.

When a furnace-tube is in danger of becoming overheated, it is generally expedient to withdraw the fire. When it is overheated, it is dangerous to disturb or withdraw the fire, and it is usual to smother the fire with wet ashes, but the attendant runs the risk of being scalded in the event of rupture of the furnace-crown.

Effect of Cold Feed-Water on Overheated Plates.—When cold feed-water is turned on to overheated plates, there is no danger of the explosion of the boiler from the sudden generation of steam, because metal has so little capacity for heat that it cannot retain sufficient to generate much steam. This has been proved by experiments.

In one experiment, steam of a pressure of 150 lbs. per square inch was raised in a boiler, the water was blown off, and the plates were permitted to become nearly red-hot. Water was then pumped into the boiler, but there was no sudden generation of steam. The water only cooled the plates, and the boiler did not explode.

In an experiment with another boiler, the water was blown off until the crown of the furnace was exposed and became red-hot, water was then pumped into the boiler, which resulted in cooling the plates, and causing the joints to leak badly, but no explosion took place.

The sudden cooling of overheated plates is liable to cause injury by the severe strain produced by sudden contraction after excessive expansion.

Boiler Explosions from Over-pressure.—Numerous boiler explosions have been caused by over-pressure, or by working the boiler at a pressure higher than it was fit to bear. Over-pressure has been frequently caused by safety-valves sticking fast on their seats, and becoming inoperative. In some cases safety-valves have been overloaded or wedged down for the purpose of temporarily obtaining higher pressure, and the overpressure has resulted in explosion.

Safety-valves loaded with spring-balances have been frequently found overloaded by being screwed down until the valves were fastened on their seats. To prevent the valve being loaded beyond the blowing-off pressure, a ferrule should be fitted on the screw of the balance.

Over-pressure was the cause of the explosion of the boiler shown in Fig. 307. The safety-valve was screwed down and rendered inoperative. The over-pressure caused the rupture of the plates in several places.

The explosion of the boiler shown in Fig. 308, was caused by over-pressure. One of the safety-valves was jammed, and the other was supposed to have been screwed down on its seat. The overpressure caused sixteen screwed stays to be drawn out of the left side of the fire-box, and both the fire-box and its shell were bulged.

Over-pressure was the cause of the explosion of the marine boiler shown in Fig. 309. The safety-valve was wedged and rendered inoperative, and the over-pressure ruptured part of the bottom of the furnace-tube and forced it upwards.

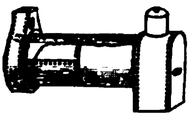


Fig. 307.—Explosion of the boiler of a traction engine from over-pressure.

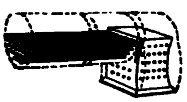


Fig. 308.—Explosion of the boiler of a traction-engine from over-pressure.

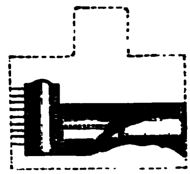


Fig. 309.—Explosion of a marine-boiler from over-pressure.

Lever safety-valves can be easily overloaded by the addition of a small weight. They have been found after explosions to have been overloaded with pieces of iron, bricks, and fire-bars. A wedge is sometimes driven between the top of the lever and its fork-guide, for the purpose of gagging the valve to obtain a higher pressure. A boiler should have at least one safety-valve which cannot be tampered with or gagged.

Safety-valves are frequently of deficient area, and incapable of preventing increase of pressure on a sudden stoppage of the flow of steam from the boilers. The lift of safety-valves is frequently too small to permit free escape of the steam and prevent accumulation of pressure.

Over-pressure has been caused by defective safety-valves, which could not permit the steam to escape as fast as was necessary to relieve the boiler when the limit of safety was reached.

Safety-valves have sometimes become overloaded by an accumulation of water in the escape-pipes, which has frozen in winter and rendered the valves inoperative. When waste-pipes are employed, they should be automatically drained of water from condensation. Waste-pipes should only be used for safety-valves when unavoidable. When the valves are open-topped, it can easily be seen when they are in a leaky condition, and when steam is blowing-off to waste.

Over-pressure has been caused in some cases by an error of judgment in fixing the working-pressure at too high a point for the safe working of old boilers, their strength having been estimated from the results of hydraulic tests. These tests are very misleading and unreliable. A boiler is strained gradually and uniformly by hydraulic pressure, but it is subject to intermittent and varying strains under steam-pressure, and also to racking strains from unequal expansion and contraction.

Old boilers having plates varying considerably in thickness frequently withstand a hydraulic test satisfactorily, and although water-tight are in an unsafe condition. For instance, an old boiler withstood satisfactorily a hydraulic pressure of 120 lbs. per square inch, and was declared safe and sound for a working-pressure of 60 lbs. per square inch. After the boiler was emptied, it was inspected and tested by sounding the plates with a hammer, when a thin place was discovered in the bottom of the shell, through which the hammer was easily driven, the plate at that point being little more than $\frac{1}{16}$ inch thick. The boiler was also found to be so much thinned by corrosion in two other places as to be unsafe for any working-pressure, and it was in a dangerously weak condition.

Boilers have frequently exploded a short time after being tested by hydraulic pressure, and at a considerably less pressure of steam than the pressure of water they satisfactorily withstood. In one case a boiler ruptured with a pressure of steam nearly one-half less than the water-pressure it had safely stood a short time previously.

In the explosion of the Lancashire boiler shown in Fig. 310, the right hand furnace-tube was weak, and was so much injured by overstraining during a hydraulic test that it collapsed sideways for its whole length.

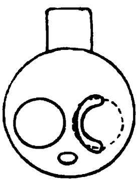


Fig. 310.—Explosion of a boiler from injury from over-straining by hydraulic pressure.

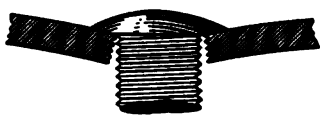


Fig. 311.—Bulged stayed-plate.

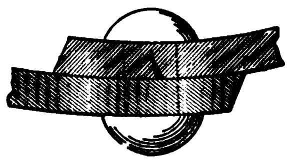


Fig. 312.—Fractured seam.

Over-pressure has in many cases caused the plates of water-spaced compartments to become so seriously bulged as to stretch the plates, and enlarge the rivet-holes so much as to permit the stud-stays to be drawn through the plate without stripping the thread of the screw, as shown in Fig. 311.

Over-pressure has in some cases arisen from reliance on inaccurate steam-gauges. Numerous roughly-made and inaccurate steam-gauges are made for the purpose of supplying cheap instruments, and steam-gauges are sometimes fitted to steam-boilers which are not only unreliable but dangerously misleading, because they are liable to indicate less than the true pressure. In some cases even apparently well-made steam-gauges have been found inaccurate. All steam-gauges should be carefully examined and frequently tested by a mercurial column.

Boiler Explosions from Hidden Flaws.—Hidden flaws have been the cause of numerous boiler explosions. Flaws along the line of rivet-holes, or between the rivet-holes, have been frequently caused by punching. When plates are of a hard and brittle nature, incipient flaws are liable to be started in them during the process of making the boiler, which may afterwards develop into a rupture from racking strains thrown on a boiler when at work. In an explosion of a boiler, the flaw from which the explosion arose occurred at the overlap of the longitudinal seam of the plates, and started from the centre of the joint, as shown in Fig. 312, so that it was entirely hidden from view.

Boiler Explosions from Fractured Plates.—Explosions have frequently arisen from fractured plates, due to a variety of causes.

Fractures have been started in many cases by the rough treatment the ends of plates occasionally receive in imparting the required set to the joint. When plates are bent after the rivet-holes are made, the lessened resistance of the plates at the rivet-holes permits the plate to become set instead of bending uniformly to the curve, and plates have been injured by flogging the bent edges back to the required curve.

Fractures have been frequently started by the injudicious use of the drift, for the purpose of drawing the rivet-holes fair.

Fractures have been started by caulking with a tool of improper shape.

Fractures have been frequently produced at the edges of unstrengthened manholes and mudholes by strains from screwing up the covers. The edge of these holes should always be strengthened by a ring riveted to the plate.

Fractured seams are frequently caused by unequal expansion and contraction, and in some cases by fretting of the metal from strains produced by uneasy seating, or uneven setting of the boiler.

Fractures have been frequently caused by staying plates so rigidly as to interfere with their expansion, and by strains produced on one part by the excessive expansion of another part of the structure.

Fractures have frequently been caused by sudden contraction from the impingement of a current of cold air on hot plates.

Fractures have been produced by sudden contraction due to the delivery of cold feed-water on hot plates.

Fractures have frequently been caused by sudden contraction and straining from running cold water into a boiler immediately after blowing-off, while both it and the brickwork-setting were hot, for the purpose of cooling it quickly.

Fractured Steam-Pipes.—Steam-pipes have frequently fractured for want of elasticity. They should never be fixed so rigidly as to be incapable of yielding to accommodate expansion.

Explosions of Steam-Pipes.—Explosions of stop-valves and steam-pipes have been frequently caused by wasting from corrosion, and by water-hammer due to accumulation of water from condensation of steam.

Cast-iron steam-pipes have frequently burst from defects, and uneven thickness, caused by the shifting of the core in casting.

Copper steam-pipes have frequently burst from defectively brazed seams.

Copper is liable to become reduced in strength and rendered more or less hard and brittle by long use, but it may be rejuvenated by annealing.

Burst Tubes.—The tubes of multitubular boilers have frequently burst from wasting due to the scouring effect of cinders and ashes, and from the deteriorating action of the gases from coke. The tubes of these boilers should be frequently withdrawn and inspected.

In the boiler of a tramway-locomotive shown in Fig. 313, one tube burst at the place where the part is left out in the engraving, being worn very thin by the scouring-action of coke-dust.

Worn-out Steam-Boilers.—Explosions have frequently resulted from working steam-boilers when in a worn-out condition. In the explosion of the Cornish boiler shown in Fig. 314, the shell was fractured near the bottom line, where in contact with brickwork. The rents continued round the boiler, and three rings of plates were blown out. The shell was completely worn out.

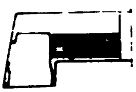


Fig. 313.—Burst smoke-tube.

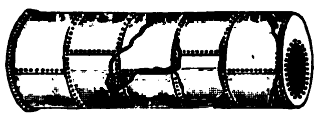


Fig. 314.—Explosion of a worn-out boiler.

In the explosion of a stationary steam-boiler, the shell fractured on one side where resting on the brickwork-seating, and it was torn into five pieces. The boiler had been in use for more than thirty years, and was completely worn out. It had not been inspected since it was made.

In the explosion of a worn-out cylindrical steam-boiler, the shell was so much reduced in strength that it ruptured at the bottom from the ordinary working-pressure. The rents spread over the whole shell, which was torn into three pieces with such violence as to wreck the premises.

In the explosion of a worn-out stationary steam-boiler, the plates were so much reduced in thickness as to be unable to bear the ordinary working-pressure. The shell ruptured at one side and the rent extended round the boiler. Both ends were blown out and the middle plates of the shell remained on the seating.

In the explosion of a worn-out portable steam-boiler of locomotive type, the fire-box was much reduced in thickness, and the heads of the screwed stud-stays were so much wasted away as to permit the plates to yield to the pressure, and they were forced over the ends of the stays.

In the explosion of a worn-out Lancashire boiler, the furnace-tubes were much reduced in thickness and were out of a true circle, being 1½ inches larger in diameter horizontally than vertically, and one of them collapsed at the ordinary working-pressure.

In the explosion of a worn-out vertical steam-boiler, the stays were reduced to nearly half their original diameter, and several of them were fractured. The fire-box was much reduced in thickness, and it collapsed and caused the destruction of the boiler.

In the explosion of a worn-out vertical steam-boiler, all the plates were much thinned by corrosion caused by bad feed-water, and the fire-box was so much weakened that it was unable to bear the usual working-pressure and it collapsed.

In the explosion of a worn-out vertical steam-boiler, the fire-box collapsed from weakness due to extensive wasting of the plates on the fire-side, caused by sulphurous fumes from the fuel.

In the explosion of a worn-out marine return-tube boiler, the shell was so much reduced in thickness by both external and internal corrosion, that it was unable to bear the usual working-pressure, and it ruptured and was torn completely from the tubes.

In the explosion of a worn-out stationary steam-boiler, some new plates which had been recently added to the shell tore away from the old plates, and the boiler was blown to pieces. The greater expansion of the new plates, which were much thicker than the old shell-plates, severely strained the old plates, and induced such weakness in the structure, that it was unable to bear the usual working-pressure.

Careful examinations of all these worn-out boilers would have revealed their dangerous condition, and the explosions might have been prevented.

Numerous explosions have occurred of partly worn or second-hand boilers. These boilers should never be purchased unless declared good and sound by competent experts.

Defective Fittings of Steam-Boilers.—The fittings of steam-boilers are frequently found in such a defective condition as to endanger the safety of the boilers.

Blow-off cocks frequently leak, and allow a considerable quantity of water to escape from the boiler. They are frequently so much wasted by corrosion as to be liable to be either broken or twisted off on opening and closing them. The blow-off elbow-pipe connecting the cock to the boiler is frequently so much wasted by corrosion as to be in a dangerous condition. The waste-pipes of the blow-off cock are frequently seriously corroded. The corrosion of blow-off cocks and pipes is frequently caused by accumulation of damp ash.

Water-gauges are frequently of such inferior metal that it is seriously impaired by high temperature and rendered rotten, and the gauges are liable to be easily broken, or twisted off, when moving the plugs in testing the cocks. The passages of water-gauges and test-cocks are frequently found more or less choked with scale, and the plugs of the cocks are frequently in a defective and leaky condition.

Feed-valves are frequently in a defective condition, the passages and feed-pipes are frequently more or less choked with scale, and the feed-pipes are frequently much thinned by corrosion.

Safety-valves have frequently been found rusted fast to their seats, and the levers rusted fast in the joints.

Fusible-plugs are frequently found either thickly encrusted with scale, or in a worn-out condition; or rendered inoperative by the adherence of a thick coating of hard deposit on the fire-side.

Steam-fittings should be frequently examined and adjusted, and kept clean and in good order.

Leaky Joints of Steam-Boilers. — Leaky seams may be caused by defects in the proportions or workmanship of the joints, deposits of scale, defective circulation, and unequal expansion and contraction. They are frequently caused by strains produced by difference in temperature in different parts of the boiler, by sudden contraction due to the admission of cold air or cold water to the hot plates, and also to blowing off the boiler when at a high temperature.

Leakage at the seams of the furnace-tubes has been frequently caused by unequal expansion due to raising steam too rapidly, and to lighting the fires in, and firing, one of two or more furnace-tubes instead of both or all of them in getting up steam.

Leakage is frequently caused by the omission of breathing space, or by staying the parts so rigidly as to interfere with expansion, and by strains produced on one part by the expansion of another part of the structure, and by the greater expansion of one part than another.

Leaks at seams of combustion-chambers of marine return-tube boilers may be caused by strains from the longitudinal expansion of the furnace-tube for want of elasticity in the tube-plates.

Leaks at the ends of tubes may be caused by a coating of grease or scale on the tube-plate, by imperfectly-expanded tubes, by the impingement of intense heat on the tube-plate, and by overheating due to insufficiency of access of water to the tube-ends and tube-plates. It is frequently caused by distortion of the holes in the tube-plate for want of accommodation of the expansion of the tubes, and by excessive pressure on the tube-plate of the fire-box or combustion-chamber produced by the roof-stays.

Leakage at the stays is frequently caused by unequal expansion, and excessive rigidity of the structure.

Leaky seams due to defective workmanship may generally be remedied by either caulking the plates, or inserting new rivets, but when the leakage proceeds from resisted expansion, the staying should be altered to provide the necessary breathing spaces. Leaks, from whatever cause they proceed, should receive immediate attention, as they generally indicate either bad treatment of the boiler, or defects in the construction or workmanship, which may produce more serious results.

Prevention of Steam-Boiler Explosions. — The employment of trustworthy and experienced attendants, the maintenance of boilers in good condition, and working them under efficient periodical supervision of reliable and competent independent inspectors, are the best means of preventing explosions.

Explosion of Fuel-Gases in the Flues of Steam-Boilers.—Accidents have frequently occurred from the accumulation of gas in the flues of boilers when the dampers are closed, resulting in either blowing the fire-doors open and scattering the fire over the stokehole, or in igniting the soot and setting fire to the chimney. If the dampers are tightly closed when the furnaces are heavily charged with coal, an accumulation of gas is liable to form, which may ignite and explode when air is admitted to the gas on raising the dampers.

Fuel-Economisers have been frequently shattered by explosions of fuel-gas inside their brick-chambers and outside the economisers.

Life of Steam-Boilers.—The life of a steam-boiler depends principally upon the excellence of its design, materials, and workmanship, the purity of its feed-water, the nature of the fuel, and also upon its treatment and the care bestowed upon it in service. Lancashire boilers frequently last at least a quarter of a century, during which time few, if any, repairs are required. The life of a locomotive-boiler is, on an average, ten years, during which time frequent repairs are necessary, especially to the fire-box, which only lasts from 3 to 6 years on an average. Marine boilers of locomotive type generally last ten years. Some marine return-tube boilers have been worn out in twelve years, others have lasted longer. It is difficult to state the life of a water-tube boiler, because as the tubes wear out they are replaced. Some water-tube boilers have been worn out in thirteen years, others have lasted longer.

Well-made steam-boilers, of good materials, properly cared-for, kept in good repair and used under favourable conditions, have been recorded to have satisfactorily worked continuously for the following periods:—

	Years of Service.		Years of Service.
Egg-ended boilers	40	Vertical cross-tube boilers	21
Cornish-boilers	35	Boilers of portable-engines, locomotive type	20
Lancashire-boilers	33	Return-tube boilers	18
Locomotive-boilers	24	Water-tube boilers	15
Multitubular-boilers, various	22		

These types of boilers, when worked under less favourable conditions, have been frequently worn out in from one-third to one-half these numbers of years. It cannot, therefore, be stated how long a steam-boiler should last. The principal causes of the premature decay of steam-boilers are bad feed-water, neglect, and bad treatment.

Special Types of Boilers.—There are so many patent boilers that they cannot be described in this work. Some are good steam-generators, others have no features of practical value.

For many of these boilers a very high evaporative efficiency is frequently claimed, but as the normal evaporative power of a boiler may during an ordinary test be easily increased from 30 to 50 per cent. by expert firing, it is prudent to be dubious of the performance, and to scrutinise the results of tests of boilers which, under ordinary working conditions, show an evaporation of more than 10 pounds of water per pound of coal, from cold, or moderately cold, feed water.

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Test made by ....

at ....

PRINCIPAL DIMENSIONS OF THE BOILER..

....

....

SUMMARY OF PARTICULARS OF THE EVAPORATIVE TEST.

Date of the Test.................................. *Duration of the Test, Hours*	....
HEATING-SURFACE AND FIRE-GRATE SURFACE OF THE BOILER.	
Total Heating-surface of the Boiler . . . . *square feet*	....
Area of the Fire-grate surface . . . . " "	....
Ratio of the Fire-grate surface to the Heating-surface	....
ADMISSION OF AIR, TEMPERATURE, AND DRAUGHT.	
Air-space between the Fire-bars . . . . . . *square feet*	....
Air-space in the Fire-doors . . . . " "	....
Temperature of the external Air . . . . . *degrees Fahr.*	....
Temperature of the Air in the Boiler-room . . . . " "	....
Temperature of the Fuel-Gases leaving the Boiler . " "	....
Chimney-draught . . . . *inches of water*	....
STEAM.	
Average pressure of Steam by the Steam-gauge *lbs. per square inch*	....
Temperature of Steam corresponding to the average pressure *degrees Fahr.*	....
ANALYSIS OF FUEL-GASES.	
Carbonic Acid, volume . . . . *weight per cent.*	....
Carbonic Oxide, " . . . . " "	....
Oxygen, " . . . . " "	....
Nitrogen, " . . . . " "	....
AIR USED FOR COMBUSTION.	
Weight of Air used per lb. of Coal . . . . . *lbs.*	....
Weight of Air used per lb. of dry Coal . . . . . . "	....
Ratio of the quantity of Air used to that theoretically required for Combustion	....

ANALYSIS OF COAL.

Carbon........................	*Hydrogen*	*Oxygen*........................
Nitrogen	*Sulphur*........................	*Ash* ....

CONSUMPTION OF COAL.

Total weight of Coal used during the Test	*lbs.*	
Weight of Coal fired per Hour	"	
Quantity of Moisture in the Coal	*per cent.*	
Weight of dry Coal fired per Hour	*lbs.*	
Weight of Coal consumed per square foot of Fire-grate surface per Hour	"	
Weight of Coal consumed per square foot of total Heating-surface of the Boiler per Hour	"	

CLINKER, ASH, AND FIRING.

Total Weight of Clinker and Ash produced	*lbs.*	
Percentage of Clinker and Ash in the total weight of Coal used,	*per cent*	
Thickness of the Fire, average	*inches*	
Weight of Coal put on the Fire at each Firing . . .	*lbs.*	
Number of times each Fire was stoked per Hour .	.	

FEED-WATER SUPPLY, AND EVAPORATION.

Total weight of Water fed to the Boiler during the Test . .	*lbs.*	
Weight of Water fed to the Boiler per Hour	"	
Temperature of the Feed-water	*degrees Fahr.*	
Water evaporated per lb. of Coal from the Temperature of the Feed-water per Hour	*lbs.*	
Water evaporated per square foot of Fire-grate surface per Hour	"	
Water evaporated per square foot of Heating-surface of the Boiler per Hour	"	
Factor of Evaporation	.	
Equivalent Evaporation from and at 212° Fahr. per lb. of Coal,	*lbs.*	

EFFICIENCY OF THE BOILER.

Efficiency of the Boiler	*per cent.*	

BALANCE-SHEET SHOWING THE DISTRIBUTION OF THE HEAT.

HEAT EVOLVED PER LB. OF COAL.	Units.	HEAT EXPENDED PER LB. OF COAL.	Units.
Calorific value of 1 lb. of the coal	....	*Heat expended in evaporating the water* . .	....
		Heat carried away by the products of combustion	....
		Heat lost by Radiation and Imperfect Combustion .	....

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