Chapter XXXVI.—BOILERS FOR STATIONARY STEAM ENGINES.

THE boiler for a steam engine requires the most careful usage and inspection, in the first case because a good boiler may be destroyed very rapidly by careless usage, and in the second case because the durability of a boiler depends to a great extent upon matters that are beyond ordinary control, and that in many cases do not make themselves known except in their results, which can only be discovered by careful and intelligent inspection. All that the working engineer is called upon to do is, to use the boiler properly, keep it clean, and examine it at such intervals as the nature of the conditions under which it is used may render necessary.

The periods at which a boiler should be cleaned and inspected depend upon the quality of the water, whether the feed water is purified or not, and to a certain extent upon the design of the boiler; hence these periods are variable under different circumstances.

The horse power of a boiler is estimated in various ways, and there is no uniform practice in this respect. Some makers estimate a boiler to have a horse power for every fifteen square feet of heating surface it possesses, while others allow but 12 square feet.

The heating surface of a boiler of any kind is the surface that is exposed to the action of the fire on one side, and has water on the other; hence the surface of the steam space is not reckoned as heating surface, even though it may be exposed to the action of the heat. The effectiveness of the heating surface of a boiler obviously, however, depends upon the efficiency of the fire, and this depends upon the amount of draught, hence the estimation of horse power from the amount of its heating surface, while affording to a certain extent a standard of measurement or comparison while the boiler is not in use, has no definite value when the boiler is erected and at work.

Thus whatever amount of steam a boiler may produce under a poor or moderate draught, it will obviously produce more under an increased draught; hence the efficiency of the same boiler depends to a certain extent upon the draught, or in other words upon the quantity of fuel that can be consumed upon its fire bars.

The amount of water required in steam boilers varies from 16 lbs. to 40 lbs., per horse power per hour, and it has been proposed to compute the horse power of boilers from the water evaporation, taking as a standard 30 lbs. of feed water at a temperature of 70 degrees, evaporated into steam at a temperature of 212 degrees, at which temperature the steam is assumed to equal the pressure of the atmosphere.

*"The strength of the shell of a cylindrical boiler to resist a pressure within it, is inversely proportional to its diameter and directly, to the thickness of the plate of which it is formed.

"For instance, take three cylindrical boilers each made of ½ inch plate, the first one 2 feet 6 inches in diameter; the second twice that, or 5 feet in diameter; and the third twice that again, or 10 feet in diameter; and if the 2 foot 6 inch boiler is fit for a safe working pressure of 180 lbs. per square inch, then the 5 foot boiler will be fit for exactly one-half that amount, or 90 lbs. per square inch; and the ten foot boiler will be fit for half the working pressure of the five foot boiler, hence we have:

Diameter of boiler shell.	Thickness of plate.	Relative working pressure.
2 feet 6 inches.	½ inch.	180 lbs. per square inch.
5 "	" "	90 " " " "
10 "	" "	45 " " " "

"The reverse applies to the thickness of the plate. For instance, if we take two cylindrical boiler shells, each 5 feet in diameter, the first one made of plate ½ inch thick, and the second twice that, or 1 inch thick, and if the first is equal to a safe working pressure of 90 lbs. per square inch, then the second is equal to a safe working pressure of twice as much, or 180 lbs. per square inch, providing, of course, that the riveted seams are of equal strength in each case, and that both boilers are allowed the same margin for safety; hence we have:

Diameter of boiler.	Thickness of shell.	Safe working pressure.
5 feet.	½ inch.	90 lbs. per square inch.
5 "	1 "	180 " " " "

"These principles (namely, that the strength of a boiler is, all other things or elements being equal, inversely proportional to its diameter, and directly proportional to its thickness) afford us a groundwork upon which we may lay down rules for determining by calculation the strength of the solid part* of any boiler shell, and the bases of these calculations are as follows:

"If the shell plate of a cylindrical boiler is ½ inch thick, there is

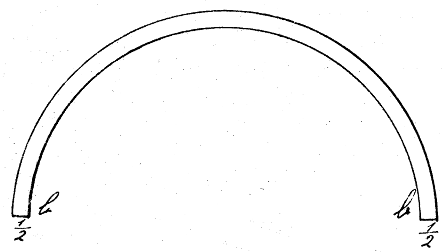


Fig. 3237.

one inch section of metal to be broken before the boiler can be divided into two pieces, that is to say there is ½ inch on each side

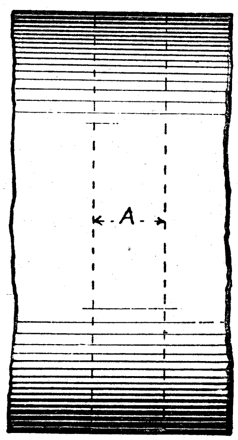


Fig. 3238.

of the shell, as shown in Fig. 3237, and the two together will make 1 inch. If we take a ring an inch broad, as, say, at A in Fig. 3238, we shall obviously have a section of 1 square inch of metal to break before the ring can be broken into two pieces.

* From "*Steam Boilers.*"

* In the case of the riveted joints or seams other considerations come in, as will be shown hereafter.

"The next consideration is, what is the average strength of a plate of boiler iron? Now suppose we have a strip of boiler iron 2 inches wide and ½ inch thick, or, what is the same thing, a bar of boiler iron 1 inch square, and that we lay it horizontally and pull its ends apart until it breaks, how many lbs. will it bear before breaking? Now for our present purpose we may assume this to be 47,040 lbs., and if this number of lbs. be divided by the diameter of the boiler in inches, it will give the bursting pressure in lbs. for any square inch in the ring, or any other square inch in the cylindrical shell of the boiler.

"The reason for dividing by the diameter of the boiler is as follows:

"Of course the steam pressure presses equally on all parts of the interior surface of the shell, and may be taken as radiating from the centre of the boiler, as in Fig. 3239, which represents an end

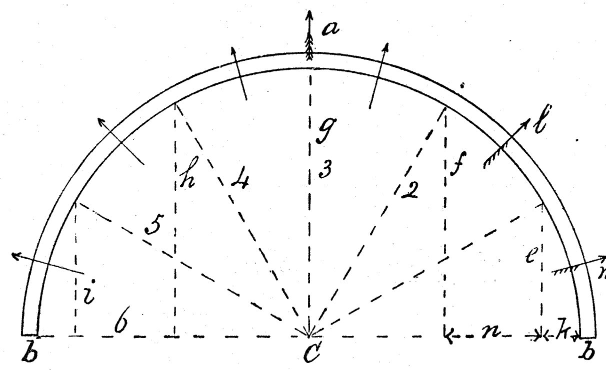


Fig. 3239.

view of a strip an inch wide, of one half of a boiler. Now leaving the riveted seam out of the question, and supposing the shell to be truly cylindrical, and the metal to be of equal quality throughout, it will take just as much pressure to burst the shell apart in one direction as it will in another, hence we may suppose that the boiler is to be burst in the direction of arrow *a*, and it is the section of metal at *b b* that is resisting rupture in that direction.

"Now suppose we divide the surface against which the steam presses into six divisions, by lines radiating from the centre C, and to find the amount of area acting on each division to burst the shell in the direction of arrow *a*, we drop perpendicular lines, as line *e*, from the lines of division to the line *b b*, and the length of the line divided off (by the perpendicular) on the diameter represents the effectiveness of the area of that division to burst the boiler in the direction of arrow *a*; thus for that part of the boiler surface situate in the first division, or from *b* to line *e*, the area acting to burst the boiler in the direction of *a* is represented by the length of the line *k*, while the general direction of the pressure on this part of the shell is represented by arrow *m*.

"Similarly, for that part of the shell situate between vertical line *e* and vertical line *f*, the general direction of the steam pressure is denoted by the arrow *l*, while the proportion of this part that is acting to sever the boiler in the direction of *a* is represented by the distance *n*, or from the line *e* to line *f* measured on the line *b b*.

"By carrying out this process we shall perceive that, although the pressure acts upon the whole circumference, yet its effectiveness in bursting the boiler in any one direction is equal to the boiler diameter. Thus in Fig. 3240, the pressure acting in the direction of the arrows *a* (and to burst the boiler apart at *b b*) is represented by the diametral line *b b*, while the pressure actually exerted upon the whole boiler shell is represented by the circumference of the boiler.

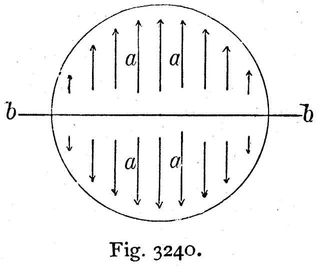


Fig. 3240.

"To proceed, then, it will now be clear that the ultimate strength of the boiler material, multiplied by twice the thickness of the boiler shell plate in inches or decimal parts of an inch, and this sum divided by the internal diameter of the boiler, in inches, gives the pressure (in lbs. per square inch) at which the boiler shell will burst."

We have here only considered the strength of the solid plate of the shell, and may now consider the strength of the riveted joints, because, as the boiler cannot be any stronger as a whole than its weakest part is, and as the riveted joints are the weakest parts of a cylindrical boiler,* therefore the strength of the riveted joint determines the strength of the boiler.

†"The strains to which a riveted joint is subjected are as follows: That acting to shear the rivet across its diameter is called the *shearing* strain. But the same strain acts to tear the plate apart; hence, when spoken of with reference to the action on the plate, it is called the *tearing* strain.

"The same strain also acts to crush and rupture the plate between the rivet hole and the edge of the plate, and in this connection it is called the *crushing* strain.

"Thus, Fig. 3241 represents a single riveted lap joint, in which

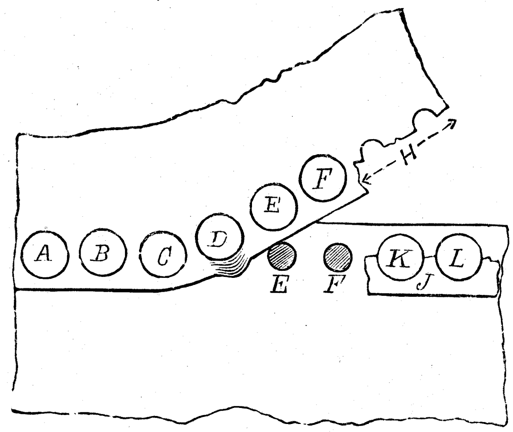


Fig. 3241.

the joint at rivets A, B, and C is intact, the metal outside of D has crushed, the rivets E, F have sheared, and the plate has torn at H, leaving a piece J on the rivets K L.

"It is obvious that, since it is the same strain that has caused these different kinds of rupture, the joint has, at each location, simply given way where it was the weakest.

"If a riveted joint was to give way by tearing only, the indica-

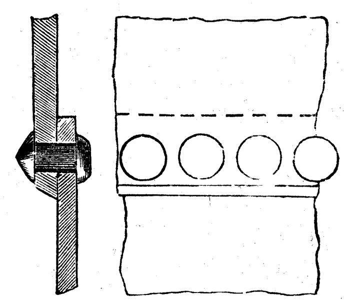


Fig. 3242.

tion would be that the proportion of strength was greatest in the rivets, which might occur from the plate being of inferior metal to the rivets, or from the rivets being too closely spaced. If the rivets were to shear and the plate remain intact, it would indicate insufficient strength in the rivets, which might occur from faulty material in the rivets, from smallness of rivet diameter, or from the rivets being too widely spaced.

"The object then, in designing a riveted joint is to have its re-

* It may be here noted that the riveted joint of a flat plate is stronger than the flat surface of the plate, because at the joint the plate is doubled, or one plate overlaps the other.

† From "*Steam Boilers.*"

sistance to tearing and shearing proportionately equal, whatever form of joint be employed."

The English Board of Trade recommends that the rivet section should always be in excess of the plate section, whereas, in ordinary American practice, for stationary engine boilers, the plate and rivet percentages are made equal.

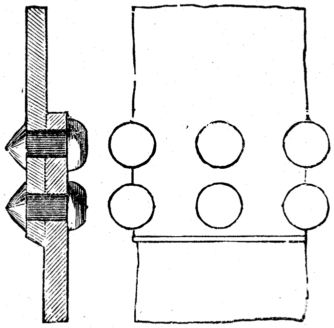


Fig. 3243.

The forms of riveted joints employed in boiler work are as follows:

Fig. 3242 represents a single riveted lap joint. Fig. 3243 represents a double riveted lap joint, chain riveted; and Fig. 3244, a double riveted lap joint, with the rivets arranged zigzag.

Fig. 3245 represents a single and Fig. 3246 a double riveted

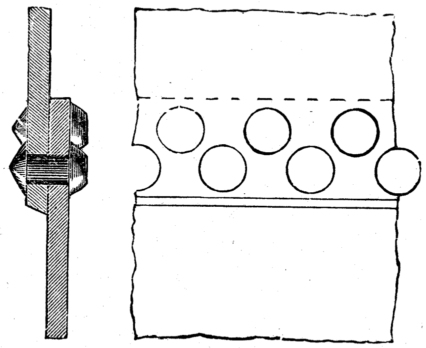


Fig. 3244.

butt joint, so called because the ends of the boiler plate abut together. The plates on each side of joint are called butt straps.

The advantages of the butt joint are, first, that the boiler shell is kept more truly cylindrical, and the joint is not liable to bend as it does in the lap joints, in the attempt of the boiler (when under pressure) to assume the form of a true circle, and second that the

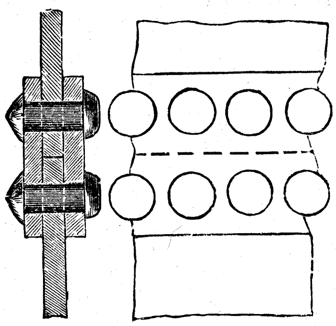


Fig. 3245.

rivets are placed in double shear. That is to say, if in a lap joint the rivet was to shear between the plates, the joint would come apart, whereas, in a butt joint, the rivet must shear on each side of the plate, and therefore in two places.

Fig. 3247 represents a form of joint much used in locomotive practice in the United States. It is a lap joint, with a covering plate on the inside of the joint; rivets E and F are in single and rivets D in double shear.

*"When we have to deal with comparatively thin boiler plates,

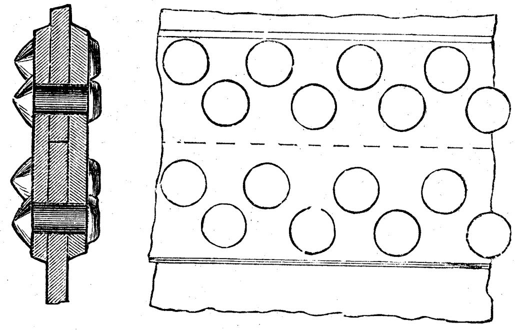


Fig. 3246.

there is no difficulty in obtaining a sufficiently high percentage of strength in the joints, by using the ordinary double riveted joint, but when we have to deal with thick plates, as in the case of large marine boilers, as 1 inch or upwards, a more costly form of joint

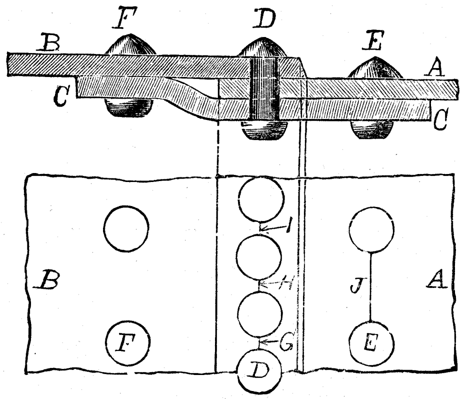


Fig. 3247.

must be employed, in order to obtain the required percentage of strength at the joint; hence the ordinary double riveted joint is replaced by various other forms as follows:

"First, a triple zigzag riveted lap joint, such as shown in Fig.

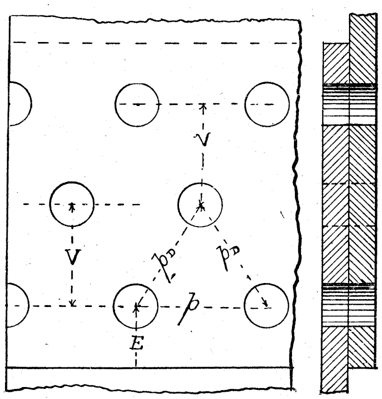


Fig. 3248.

3248, or a chain riveted joint as in Fig. 3249, in both of which the third row of rivets enables the rivet pitch to be increased, thus increasing the plate percentage, while the third row of rivets also increases the rivet percentage.

"Second, by employing butt joints with butt straps, either double or treble riveted.

*From "*Steam Boilers.*"

"A double riveted butt joint with double straps is shown in Fig. 3250, and a treble with double straps in Figs. 3251 and 3252.

"Third. By various arrangements of the rivets in conjunction

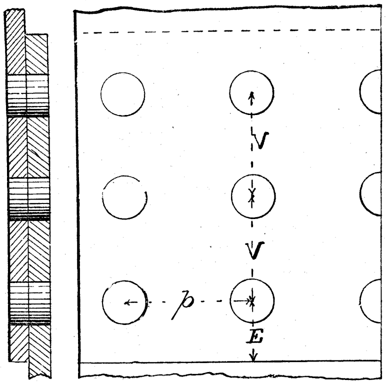


Fig. 3249.

with butt joints and double straps, with which it is not necessary, at this point, to deal.

"One of the great advantages obtained by the use of the double strap is that of bringing the rivet into double shear (or in other words, the rivet must shear on each side of the plate, or in two places, instead of between the plates only, before the joint can give way by shearing), and thus obtaining an increased calculated

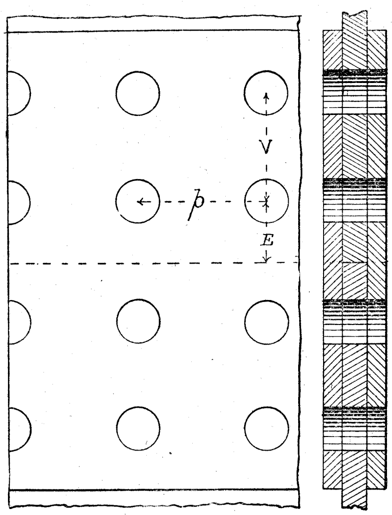


Fig. 3250.

strength of $1\frac{3}{4}$ times the ordinary or single shear, the rule being to find the rivet strength in the ordinary way (as before explained), and then multiply the result by 1.75.

"The Board of Trade rules for spacing the rivets of these joints are as follows:

"Dimension E is the distance from the edge of the plate to the centre of the rivet hole. Dimension V is the distance between the rows of rivets, dimension p is the pitch of the rivets, which is always measured from centre to centre of the rivets, and dimension pd is the diagonal pitch of the rivets.

"The rule for finding dimension E, whether the plates and rivets are either of steel or iron, is as follows:

"Multiply the diameter of the rivet by 3 and divide by 2, the formula being as follows:

$$\frac{3 \times d}{2} = E.$$

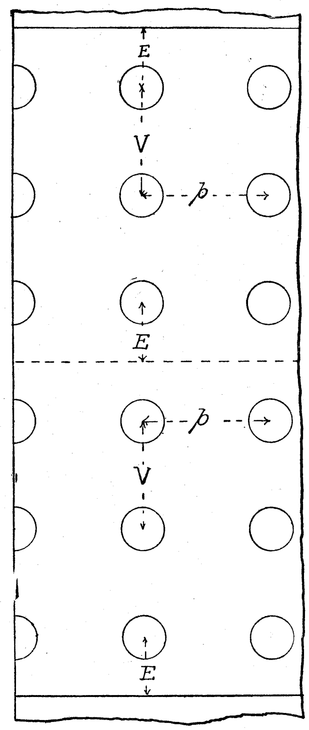
"To find the distance V between the rows of rivets in chain riveted joints. This distance must not be less than twice the rivet diameter, and a more desirable rule is four times the rivet diameter plus 1 divided by 2, thus:

$$\frac{4d + 1}{2} = V.$$

"To find the distance between the rows of zigzag riveted joints:

$$\frac{\sqrt{(11p + 4d) \times (p + 4d)}}{10} = V,$$

that is, multiply 11 times the pitch plus 4 times the rivet diameter, by the pitch plus 4 times the rivet diameter, then extract the square root and divide by 10.



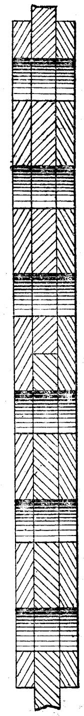


Fig. 3251. Fig. 3252.

To find diagonal pitch pd, multiply the pitch p by 6, then add 4 and divide by 10, thus:

$$\frac{6p + 4}{10} = pd."$$

Fig. 3253 represents a form of high percentage joint, used upon marine boilers of 10 to 14 feet diameter, and carrying from 100

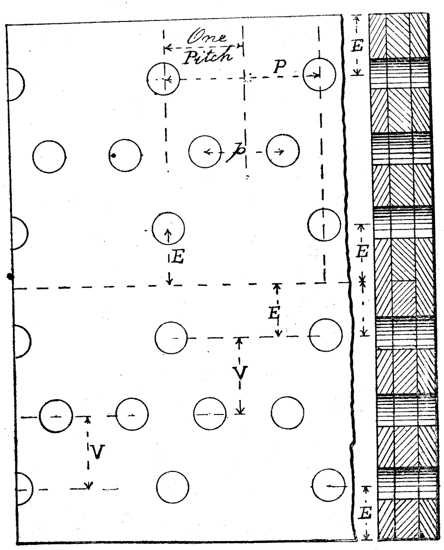


Fig. 3253.

to 190 lbs. pressure of steam. The rivets are what are termed unevenly pitched, or, that is to say, on each side of the joint, there are three rows of rivets, of which the inner and outer rows are wider pitched than the middle row.

* "The advantage gained by this spacing is that the shear of the outer row of rivets is added to the plate section at the narrow pitch, that is to say, if the plate section broke through the line of rivet holes at the narrow pitch, it has yet to shear the outer row of rivets before the plate can separate."

Fig. 3254 represents a second example of joint with rivets unevenly pitched, this form finding much favor in recent practice. The four inner rows of rivets are spaced at narrow pitch and the two outer rows are wide pitched.

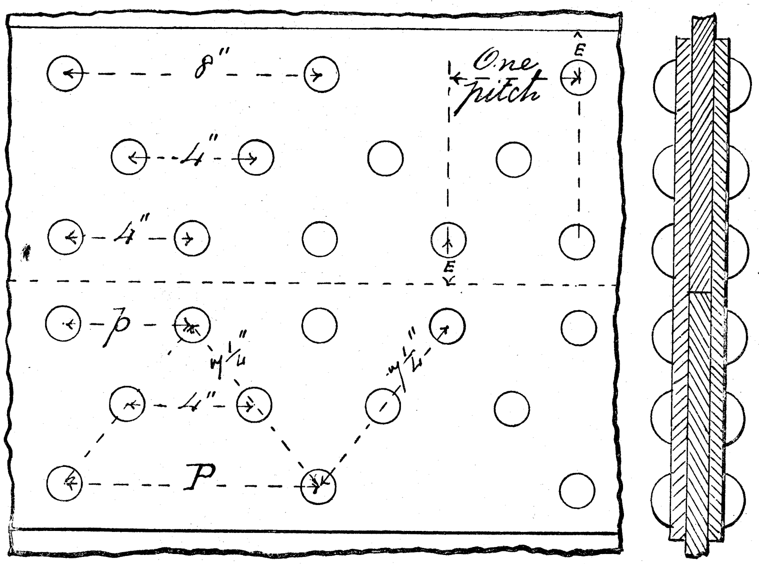


Fig 3254.

† "The strength percentage of this joint is calculated from three points of view, as follows:

"First. The plate section at the wide pitched rivets.

"Second. The rivet section in one pitch.

"Third. The plate section at the narrow pitch plus half the double shear of the outer or wide pitched rivet."

The steam pressures generally employed in the boilers of stationary engines range from about 60 to 100 lbs. per square inch, and as a result of these comparatively low pressures less perfect forms of construction are employed than would be permissible if higher pressures were used.

The strength of the shell plate of boilers of small diameter is always largely in excess of the requirements, and as a result the strength of the joints may bear a very low percentage to that of the solid plate, and yet give a sufficient factor of safety for the working pressure.

Take, for example, a boiler shell of 36 inches internal diameter with a shell plate $\frac{1}{4}$ inch thick, and allowing the strength of the material to be 48,000 lbs. per inch of section, and with a factor of safety of 4, the working pressure will be 166 lbs. per square inch, thus:

Strength of the material. Plate thickness × by 2.

$$\frac{48000 \times (.25 \times 2)}{36} = 666\tfrac{2}{3} \text{ lbs.} = \text{bursting pressure.}$$

Diameter of boiler.

By dividing this 666 by the factor of safety 4 we get $166\frac{2}{3}$ lbs. as the working pressure of the shell plate independent of the riveted joint. Usually, however, such a boiler would not be used for a pressure above about 60 lbs. per inch, and this leaves a wide margin for the reduction of strength caused by the riveted joints.

Suppose, for example, that a single riveted lap joint is used, and the strength of this joint is but 50 per cent. of that of the solid plate, and we have as follows:

Strength of material. % strength of riveted joint. Twice the plate thickness.

$$\frac{48000 \times .50 \times (.25 \times 2)}{36 \times 4} = 83\tfrac{1}{3} \text{ lbs.} = \text{W.P.}$$

Internal diam. of boiler. Factor of safety.

Here then we find that the working pressure of the solid plate is double that of the riveted joint, and that the working pressure of the boiler is 83 lbs. per square inch, notwithstanding that the strength of the riveted joints is but 50 per cent. of that of the solid plate. Such a boiler would not, however, be used for a pressure of over 60 lbs. per square inch.

If the above-named boiler was double riveted so as to bring the percentage of joint strength up to say 70 per cent. of that of the solid plate, its working pressure would be 116 lbs. per square inch, thus:

Strength of material. % strength of riveted joint. Twice plate thickness.

$$\frac{48000 \times .70 + (.25 \times 2)}{36 \times 4} = 116\tfrac{2}{3} \text{ lbs.} = \text{W.P.}$$

Internal diam. of boiler. Factor of safety.

But in practice such a boiler would not be used for pressures above about 75 lbs. per square inch, hence the shell plate thickness is still largely in excess of the requirements, and it may be remarked that plates less than $\frac{1}{4}$ inch thick are not used on account of the difficulty of caulking them and keeping them steam tight.

On account therefore of the excessive strength of the shell plates in boilers of small diameter, butt straps are rarely used in stationary boilers, while punching the rivet holes and other inferior modes of construction are employed.

We may now consider the circumferential seams of the boilers for stationary engines, such boilers sometimes being of great length in proportion to the diameter.

In proportion as the length of a boiler (in proportion to its diameter) is increased, the construction of the circumferential or transverse seams, as they are sometimes called, becomes of more importance.

The strength of the circumferential seams is so much greater than that of the longitudinal seams that it is often taken for granted that they are sufficiently strong if made with a lap joint and single riveted, but that such is not always the case will be shown presently.

In Fig. 3255 is represented a boiler composed of three strakes (*i. e.*, three rings or sections), and it is clear that as the thickness of

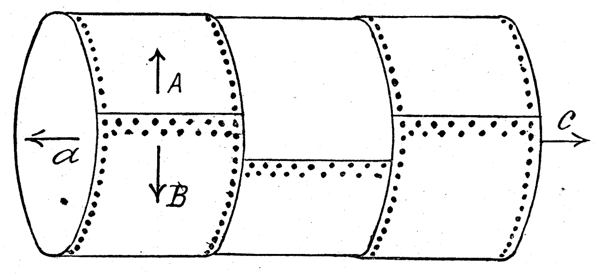


Fig. 3255.

the shell is doubled at the circumferential seams where the ends of the middle strake pass within the end strakes, therefore the strength of the lapped joint of the shell to resist rupture in a transverse direction, as denoted by the arrows A, B, is actually increased by reason of the lap of the riveted joint. But suppose this boiler to be supported at the ends only, and the weight of the shell and of the water within it will be in a direction to cause the middle of the boiler to sag down, and therefore places a shearing strain on the rivets of the circumferential seams.

* From "*Steam Boilers.*"
† From "*Steam Boilers.*"

Moreover, the temperature of the outside of the boiler cannot be made or maintained uniform, because the fire passing beneath the bottom of the boiler first will keep it hotter, causing it to expand more, and this expansion acts to shear the rivets of the circumferential seams. In proportion as the heat of the fire varies in intensity, the amount of the expansion will vary, and the consequence is that the circumferential seams may get leaky or the joint may work, especially in boilers that are long in proportion to their diameters. It is clear, therefore, that for the very best construction at least a double riveted circumferential joint should be employed.

Leaving these considerations out of the question, however, we may find the amount of stress on the circumferential seams by multiplying the area of the end of the boiler by the working pressure, and dividing by the cross-sectional area of all the rivets in one circumferential seam.

Suppose, for example, that the diameter of the boiler is 36 inches, the working pressure 60 lbs. per square inch, and that there are in each circumferential seam 50 rivets, each $\frac{3}{4}$ inch in diameter, and we proceed as follows:

The area of a circle 36 inches in diameter = 1017.87 square inches.
The area of a rivet $\frac{3}{4}$ inches in diameter = .4417 square inch.

Then

$$\frac{\underset{\text{Area of boiler end.}}{1017.87} \times \underset{\text{Working pressure.}}{60}}{\underset{\text{Number of rivets.}}{50} \times \underset{\text{Area of each rivet.}}{.4417}} = 2765 \text{ lbs. per cross-sectional square inch of rivet.}$$

By multiplying the area of the boiler end by the working pressure, we get the total steam pressure acting to shear the rivets, and by multiplying the number of rivets by the area of one rivet, we get the total area resisting the steam pressure, and then by dividing the one quantity into the other, we get the shearing stress per square inch of rivet section.

In the case of longitudinal seams, we have as follows, the pitch being say $2\frac{1}{8}$ and the rivets $\frac{3}{4}$.

$$\frac{\underset{\text{Diameter of boiler in inches.}}{36} \times \underset{\text{Steam pressure.}}{60} \times \underset{\text{Pitch.}}{2.125}}{\underset{\text{Rivets in one pitch.}}{2} \times \underset{\text{Area of rivet.}}{.4417}} = 5196 \text{ lbs. per square inch of rivet area.}$$

It is seen, therefore, that the stress placed by the steam pressure on the transverse seam is about one-half of that it places on the longitudinal seam. But, as before remarked, the transverse seam is subject to racking strains, from which the longitudinal seams are exempt; thus, for example, the expansion of the boiler diameter, whether uniform or not, does not strain the longitudinal seam, whereas it may severely strain the transverse seam.

The English Board of Trade rules, in assigning values to the various constructions and qualities of workmanship, assign a certain value, in the form of an addition to the factor of safety, which takes into account the difference in the stress upon the transverse and longitudinal seams, the quantities in each case having been determined both from experiment and from experience. A comparison of the different values may be made as follows:

The rules take a boiler shell made of the best material, with all the rivet holes drilled after the strakes are rolled into shape and put together, with all the seams (both longitudinal and transverse) fitted with double butt straps each at least five-eighths of the thickness of the shell plates they cover, and with all the seams at least double riveted, with rivets having an allowance of not more than 75 per cent. over the single shear, and provided that the boilers have been open to the inspection of their surveyors during the whole period of construction, and say that such a boiler shell shall be allowed a factor of safety (divisor of seam strength) of 5.

But for every departure from this, which they deem the best mode of construction, a penalty in the shape of an addition to the factor of safety is made. These additions to the factors of safety with reference to the longitudinal as compared to the transverse seams, are given in the following table:

Nature of the deviation in the construction or workmanship.	Addition to the factor 5 if the deviation is in the longitudinal seam.	Addition to the factor 5 if the deviation is in the transverse seam.
The holes not fair and good	.75	.2
Holes drilled out of place after bending	.15	.1
Holes drilled before bending	.3	.15
Holes punched after bending	.3	.15
Holes punched before bending	.5	.2
Joints lapped and double riveted instead of having double butt straps	.2	.1
Joints double riveted but have single butt straps	.3	.1
Joints single riveted and have a single butt strap	1.0	.2
Joints lapped and single riveted	1.0	.2

An addition of .25 is also made to the factor of safety, when the strakes are not entirely under or over. In Fig. 3256 for example,

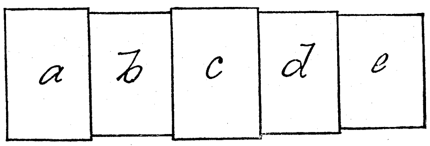


Fig. 3256.

strake b is within or under strake a at one end and strake c at the other end, hence b is entirely under; strake c is over b and d, and therefore entirely over; while strake d is under c, and over e, and therefore not entirely under nor entirely over.

When the rivet holes are punched they do not match properly, and unless the holes are punched somewhat smaller than the required size and reamed out afterwards, some rivets receive more stress than others, and may consequently shear in detail. It is customary, however, to punch the holes for ordinary stationary boilers, and it is with seams having punched holes therefore that we have at present to deal.

In the United States the rivet diameter and plate percentages are, in the boilers of stationary engines, usually made equal, and the reasons advanced both for and against this are as follows:

First, in favor of a greater plate percentage than rivet section, it is advanced that the plate gets thinner by wear, whereas the rivet does not, hence the wear reduces the plate section; that the plate is weakened by the punching process, and requires a greater percentage to make up its strength as compared to the rivet; that the rivets are usually of better material than the plates.

In favor of a greater rivet section than plate section, it is advanced that the shearing strength of iron is but about four-fifths of the tensile strength, and that with equal plate and rivet sections the rivet is therefore the weakest; that with punched holes the rivets may be sheared in detail, and that the rivets may be sheared gradually by the working of the joint from varying expansion and contraction.

From these premises the assumption is drawn that the weakening of the plate from being punched and from corrosion about offsets the excess of the tensile over the shearing strength, and that it is best therefore to employ such a pitch that the area of the rivet and of the metal left between the rivet holes shall be equal.

In order to do this the diameter of the rivet must be determined, and the following are the proportions given by the various authorities named:

TABLE OF THE DIAMETERS OF RIVETS FOR VARIOUS THICKNESSES OF PLATES WITH SINGLE RIVETED LAP JOINT.

Thickness of Plate.	Diameter of Rivets.					
	Lloyd's Rules.	Liverpool Rules.	English Dockyards.	Fairbairn.	Unwin.	Wilson.
in.	in.	in.	in.	in.	in.	in.
$\frac{5}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{5}{8}$
$\frac{3}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{11}{16}$
$\frac{7}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{21}{32}$	$\frac{13}{16}$	$\frac{3}{4}$
$\frac{1}{2}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{3}{4}$
$\frac{9}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{27}{32}$	$\frac{7}{8}$	$\frac{7}{8}$
$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{15}{16}$	$\frac{15}{16}$	$\frac{7}{8}$
$\frac{11}{16}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$1\frac{1}{32}$	1	$\frac{7}{8}$
$\frac{3}{4}$	$\frac{7}{8}$	$\frac{15}{16}$	1	$1\frac{1}{8}$	$1\frac{1}{16}$	1
$\frac{13}{16}$	$\frac{7}{8}$	1	1	$1\frac{7}{32}$	$1\frac{3}{32}$	1
$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{8}$	...	$1\frac{1}{8}$	1
$\frac{15}{16}$	1	$1\frac{3}{16}$	$1\frac{1}{8}$	...	$1\frac{3}{16}$	$1\frac{1}{8}$
1	1	$1\frac{1}{4}$	$1\frac{1}{8}$	..	$1\frac{1}{4}$	$1\frac{1}{8}$

From the above it is seen that with thin plates the diameter of rivet employed is about twice the thickness of the plate, whereas as the thickness of plate increases the proportion of rivet diameter decreases, and the reasons for this are, first, that with rivets twice the thickness of thick plates and pitched so as to equalize the rivet and plate sections the pitch would be too great to permit of the seams being caulked steam tight.

The diameter of the rivet having been determined, the rivet area and area of plate left between the rivet holes may be made equal by determining the pitch by the following rule :

Rule.—To the area of the rivet divided by the plate thickness add the diameter of the rivet, and the sum so obtained is the pitch. The correctness of this rule may be shown as follows :

Suppose the rivet diameter to be $\frac{7}{8}$ inch = decimal equivalent .875, and its area will be .6013 square inch. Suppose the thickness of the plate to be $\frac{9}{16}$ = decimal equivalent .5625, then by the rule :

```
                       Rivet area.
Plate thickness = .5625).6013(1.0689
                         5625
                          38800
                          33750
                            50500
                            45000
                             55000
                             50625
```

To this 1.0689 we are to add the rivet diameter, thus :

```
1.0689
 .8750 = rivet diameter.
1.9439 = pitch of the rivets.
```

We have thus found the required pitch to 1.9439 inches, and as the joint is single riveted there are two half rivets or one whole one to one pitch, and if we subtract the diameter of the rivet from the pitch we shall get the width of the metal or plate left between the rivets, thus :

```
1.9439 = pitch of rivets.
 .8750 = diameter of rivet.
1.0689 = distance in inches between the rivets.
```

If now we multiply this distance between the rivets by the thickness of the plate, we shall get the area of the plate that is left between the rivet holes, thus :

```
                1.0689 = width of plate between rivets.
                 .5625 = thickness of plate.
                 53445
                21378
Area of plate  64134
   between    53445
  rivets =   .60125625
```

Here then we find the area of plate left between the rivet holes to be 6.01 square inches, and as the area of the rivet is 6.01 square inches, the two are shown to be equal.

We may now place the various rivet diameters and the pitches that will make the rivet area and plate area in a single riveted joint equal in a table as follows :

TABLE OF RIVET DIAMETERS AND PITCHES FOR SINGLE RIVETED LAP JOINTS.

Thickness of Plate.	Diameter of Rivet.	Pitch.
$\frac{1}{4}$	$\frac{1}{2}$	$1\frac{1}{4}$
$\frac{5}{16}$	$\frac{5}{8}$	$1\frac{5}{8}$
$\frac{3}{8}$	$\frac{11}{16}$	$1\frac{11}{16}$
$\frac{7}{16}$	$\frac{3}{4}$	$1\frac{3}{4}$
$\frac{1}{2}$	$\frac{3}{4}$	$1\frac{5}{8}$
$\frac{9}{16}$	$\frac{7}{8}$	2
$\frac{5}{8}$	$\frac{7}{8}$	$1\frac{7}{8}$
$\frac{11}{16}$	$\frac{7}{8}$	$1\frac{3}{4}$
$\frac{3}{4}$	1	..
$\frac{13}{16}$	1	2
$\frac{7}{8}$	1	$1\frac{1}{8}$
$\frac{15}{16}$	$1\frac{1}{8}$	$2\frac{1}{8}$
1	$1\frac{1}{8}$	$2\frac{1}{8}$
$1\frac{1}{16}$	$1\frac{1}{8}$	$2\frac{1}{8}$
$1\frac{1}{8}$	$1\frac{3}{6}$	$2\frac{1}{4}$
$1\frac{1}{4}$	$1\frac{3}{16}$	$2\frac{1}{8}$

The rivets in double riveted lap joints, and in butt strap joints having a single cover, are spaced alike, because in both cases there are two rivets in one pitch, and the rivets are in single shear.

As there are two rivets in one pitch (instead of only one as in a single riveted joint), therefore the percentage of rivet section is doubled, and the plate section must therefore be doubled if the plate and rivet sections are to be made equal, and the rule for finding the required pitch is as follows :

Rule.—To the amount of rivet area in one pitch, divided by the thickness of the plate, add the diameter of the rivet.

Example.—Let the plate thickness be as in the last example $\frac{9}{16}$, decimal equivalent = .5625, and the rivet diameter be $\frac{7}{8}$ inch = decimal equivalent .875, the area of one rivet being .6013 square inch, and the pitch is calculated as follows :

```
                           .6013 = area of one rivet.
                               2 = the rivets in one pitch.
Plate thickness = .5625)1.2026(2.1377
                        1.1250
                          7760                2.137
                          5625                 .875 = rivet diameter.
                          21350               3.012 = pitch.
                          16875
                           43750
                           39375
                            43750
                            39375
                             4375
```

We find, therefore, that the pitch is 3.012, or 3 inches (which is near enough for practical purposes), and we may now make it clear that this is correct.

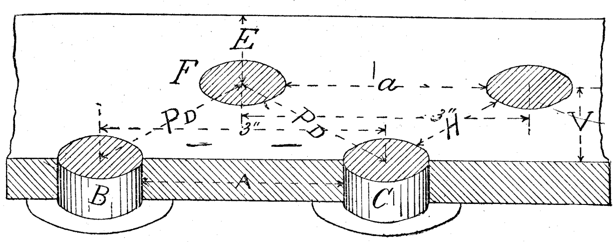


Fig. 3257.

In Fig. 3257 the joint is shown drawn one-half full size, and the length A of plate left between the rivet holes measures (as nearly as it is necessary to measure it) $2\frac{5}{32}$ inches, or 2.156, and if we multiply this by the thickness of the plate = .5625 inch, we get 1.2 square inches as the area of the plate left between the rivet holes.

Now there are two rivets in a pitch (as one-half of B, one-half of C, and the whole of F), and as the area of each rivet is .6, therefore the area of the two will be 1.2, and the plate section and rivet section are shown to be equal.

The area at *a* is obviously the same as that at A, because the pitches of both rows of rivets are equal, this being an ordinary zigzag riveted joint.

We may now consider the diagonal pitch of the rivets, using the rule below.

$$\frac{\text{The pitch} \times 6, + 4 \text{ times the rivet diameter}}{\text{divided by } 10} = \text{the diagonal pitch } p_D.$$

In this example the pitch has been found to be 3 inches, hence we have

```
      .875 = diameter of rivet.
         4 = constant.
     3.500

         3 = pitch of the rivets.
         6 = constant.
        18
       3.5 = rivet diameter multiplied by 4.
  10)21.5(2.15 = the diagonal pitch.
     20
      15
      10
       50
```

The diagonal pitch, that is, the distance p_D, Fig. 3257, is therefore found to be 2.15, or $2\frac{1}{8}$ inch full.

The amount of metal left between the rivets, measured on the diagonal pitch, is twice the dimension H multiplied by the thickness of the plate, and as this (with the diagonal pitch determined as above) always exceeds the pitch A or *a*, therefore if the plate fails, it will be along the line *a*, and not through the diagonal pitch.

We may now consider the total amount that the plates overlap in a double riveted lap joint zigzag riveted, this amount being twice the distance E, added to the distance V between the rows of rivets.

Placed in formula, the rule appears as follows, *d* representing the rivet diameter, and *p* the pitch.

$$\frac{\sqrt{(11p + 4d)(p + 4d)}}{10} = \text{distance V between the rows of rivets.}$$

As this rule involves the extraction of the square root of the sum of quantities above the line, and as in determining the diagonal pitch, we have already determined the distance V, it is unnecessary to our purpose to carry out this latter calculation, as it is easier to find the diagonal pitch, and then, after drawing the joint, the distance between the rows of rivets can be measured if it is required, as it might be in finding the length of plate required to roll into a strake for a boiler of a given diameter and having a double riveted lap joint.

We may now consider chain riveted joints in comparison with zigzag riveted joints, which is especially necessary, because it has been assumed by some that the second row of rivets in a chain riveted joint added nothing to the strength of the joint.

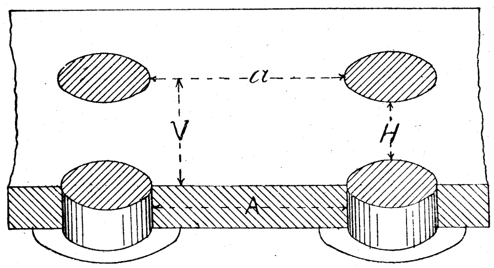


Fig. 3258.

Fig. 3258 represents a chain riveted joint, having the same thickness of plate, rivet diameter and pitch as the zigzag riveted joint in Fig. 3257, and it will be seen that the plate sections at A and at *a* are the same in the two figures, and as there are four half rivets, which are equal to two rivets, in one pitch, therefore the strength of the two joints is equal.

Each joint can be as efficiently caulked as the other, as the rivet spacing is the same and the edge of the plate is the same distance from the rivets in both cases.

The pitch of the rivets is obtained by the same rule as for zigzag riveted joints, and all we have now to consider is the distance apart of the two rows of rivets or distance V in the Fig. 3258, and

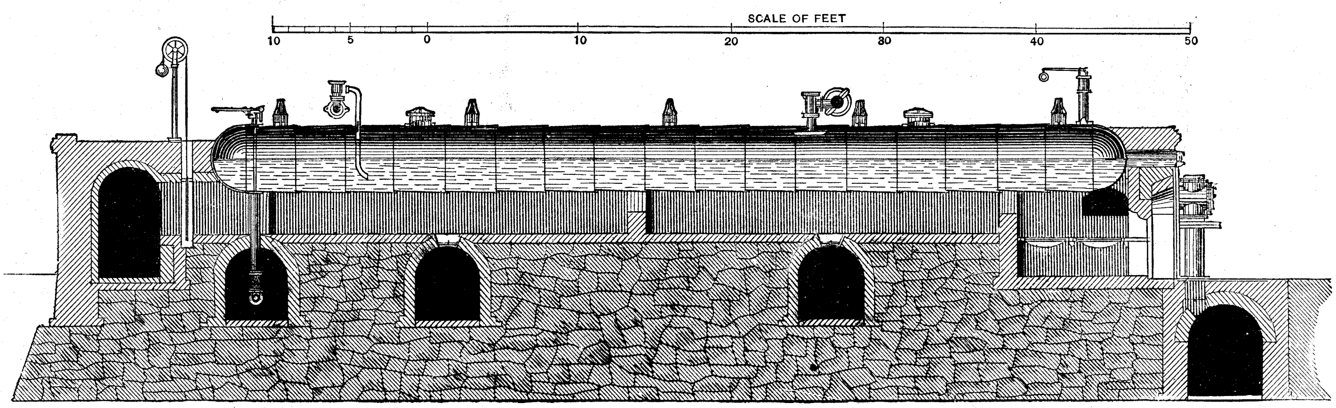


Fig. 3259.

The distance E, Fig. 3257, is usually made one and a half times the diameter of the rivet, this being found to give sufficient strength to prevent the edge of the plate from tearing out and to prevent the rivet from shearing the plate out to the edge, rupture not being found to occur in either of these directions.

The rule for finding the distance V, when the diagonal pitch has been determined by the rules already explained, is as follows:

Rule.—To the pitch multiplied by 11, add 4 times the rivet diameter, then multiply by the pitch, plus 4 times the rivet diameter. Then extract the square root and divide by 10.

for this there are two rules, the first being that it shall not be less than twice the diameter of the rivet, which would leave a dimension at H in the figure equal to the diameter of the rivet. The second rule is that a better proportion than the above is to multiply the diameter of the rivet by 3. This makes the dimension at H equal to twice the rivet diameter.

When the joints have double buttstraps, the rivets may be spaced as wide as the necessity for tight caulking will admit, because, on account of the rivets being in double shear, the rivet percentage exceeds the plate percentage.

The allowance for the rivets being in double shear is 75 per cent., or in other words, a rivet in double shear is allowed 1.75 times the area of the same size rivet in single shear.

STATIONARY ENGINE BOILERS.

The simplest form of horizontal boiler is the plain cylinder boiler, an example of which is given in Fig. 3259, and which is largely used in iron works and coal mines.

A water tube is one in which the water is inside and the fire outside, as distinguished from a fire tube, in which the fire passes through the tube and the water is outside. A water tube is stronger than a fire tube, because the former is subject to bursting pressure and the latter to collapsing pressure.

Vertical boilers are internally fired, and in the ordinary forms have no return tubes or flues, examples of those used for small stationary engines being given as follows.

Fig. 3261 represents an ordinary form with vertical tubes. The

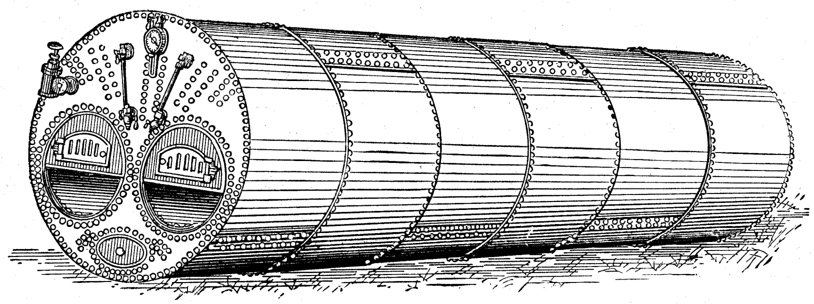


Fig. 3260.

Boilers of this class are easily cleaned, because the whole interior can be readily got at to clean.

As the bottom of this boiler gets thinned from wear, the boiler is turned upside down, thus prolonging its life.

Fig. 3260 represents an internally fired flue boiler, known as the

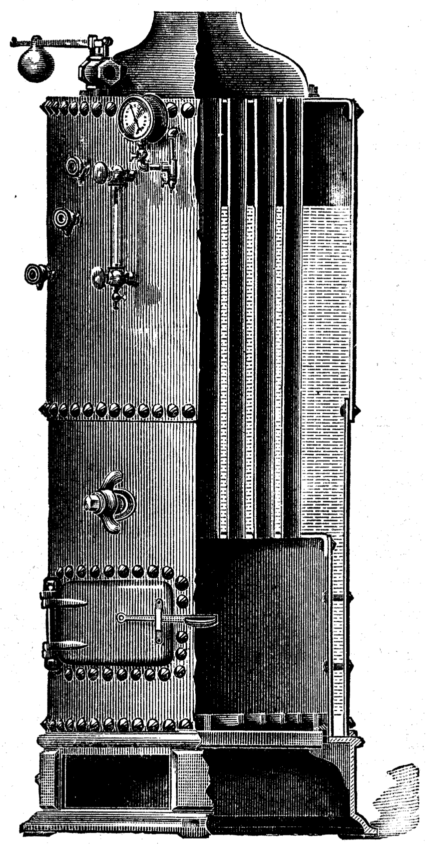


Fig. 3261.

Cornish or Lancashire boiler. The furnace is at one end of the flues, the fire passing through them to the chimney. There is here obviously more heating surface than in the plain cylinder boiler, but somewhat less facility for cleaning.

The Galloway boiler is of this class, but has vertical water tubes placed at intervals in the flues. These water tubes are wider at the top than at the bottom. They serve to break up the body of heat that passes through the flues, and increase the heating surface while extracting more of the heat and promoting the circulation of the water in the boiler.

upper ends of the tubes here pass through the steam space—a condition that under the moderate pressures and firing that this class of boiler is subjected to is of less importance than it is in boilers having higher chimneys and therefore a more rapid draught, and using higher pressures of steam. Furthermore, the small diameters and lengths or heights in which these boilers are made give them ample strength with shells and tubes of less

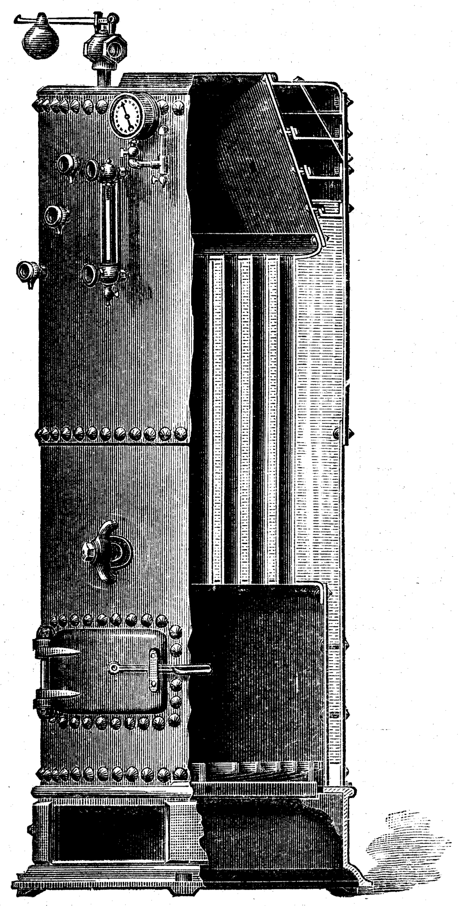


Fig. 3262.

thickness, while the condition of tube ends with steam on one side and fire on the other is permissible without the injurious effects that ensue under rapid combustion and high pressures.

The crown sheet of the fire boxes or furnaces of this class of boiler is very effective heating surface, first, because of the great depth (and therefore weight) of water resting upon it insuring constant contact between the water and the plate, while there is no danger of the crown sheet burning from shortness of water.

A similar boiler, but with the upper ends of the tubes below the water level, is shown in Fig. 3262.

From the small diameters of these boilers, the flat surfaces are not stayed except to the extent that the holding power of the tubes serves that end.

A return flue vertical boiler is shown in Figs. 3263 and 3264. The whole of the surfaces having contact with the fire also have

In boilers of this class, a majority contain water tubes, which, when properly arranged, promote rapid evaporation and circulation.

A boiler with *Field* tubes is shown in Fig. 3265. It consists of an outer shell and a cylindrical fire box, from the crown sheet of which a number of Field tubes are suspended in the fire box or combustion chamber.

Fig. 3266 is a sectional view of a Field tube, the construction being as follows:

The outer tube, which is expanded into the tube plate, is enclosed

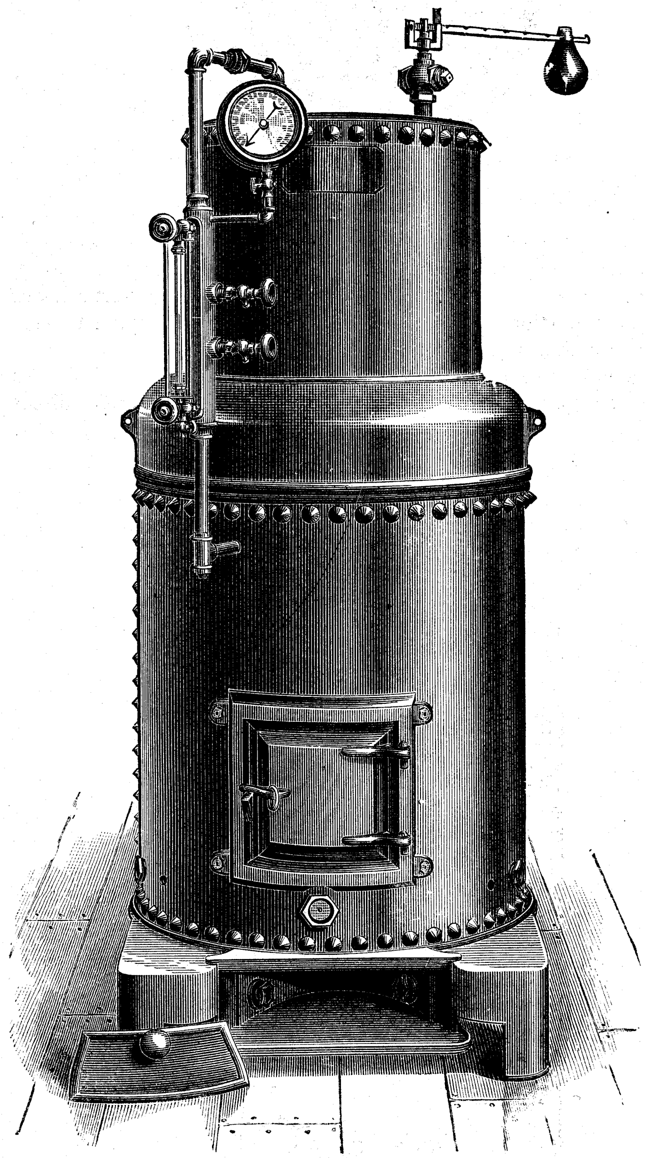


Fig. 3263.

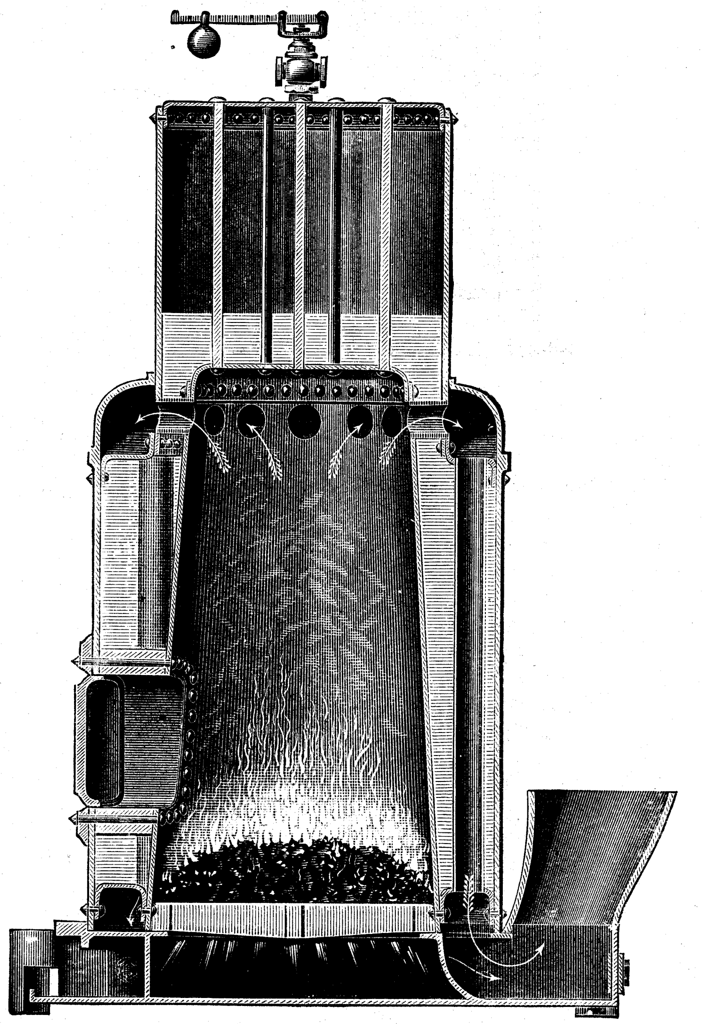


Fig. 3264.

contact with the water, and the height of the crown sheet removes it from the intense heat of the fire. It is stayed to the top of the boiler. The fire box or combustion chamber being taper increases the effectiveness of its sides as heating surface, since the heat in its vertical passage impinges against it.

The products of combustion pass from the top of the combustion chamber through short horizontal flues, which enter an annular space surrounding the lower section of the boiler, and from this space vertical flues pass to a corresponding space at the bottom of the boiler

The passage of the steam generated at the sides of the combustion chamber is facilitated by the taper of the chamber, which gives increased room for the steam as it gathers in ascending.

Vertical boilers for high pressures, as from 60 to 120 lbs. per inch, are represented in the figures from 3265 to 3269.

at its lower end, and has at its upper end in the water space of the boiler a perforated mouth piece, from which is suspended an inner tube that extends nearly to the bottom of the outer tube.

As the outer tube is bathed in the fire, steam is generated very rapidly, and a thorough and rapid circulation is kept up, the water passing down the inner and up the outer tubes, as denoted by the arrows.

The outer tube is spread out at the upper end to a slight cone, so that it cannot be forced out of the tube sheet by the pressure, and as it hangs free, there is no liability for it to loosen or get leaky from expansion and contraction.

From the great amount of heating surface obtained with these tubes, the fire box may be kept at a minimum diameter for the duty, while still leaving a wide space for the water leg, which facilitates the circulation.

The damper, which is suspended in the uptake, spreads the fire sideways.

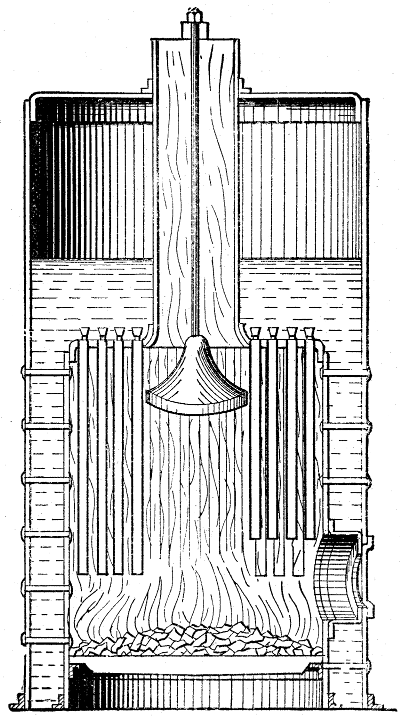


Fig. 3265.

Fig. 3267 represents the arrangement of Field tubes in a boiler.

A boiler of this form may for a given capacity be made lighter

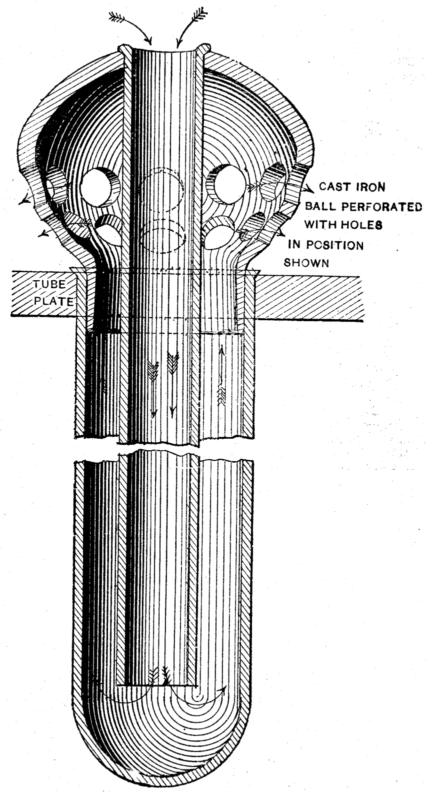


Fig. 3266.

and smaller than in any other of the ordinary forms, while the rapid circulation acts to keep the tubes clean.

The inner tubes may be thin, because they are under pressure both inside and out, while the outer tubes may be thin, because they are under a bursting strain, whereas a fire tube is under collapsing pressure.

A design of high rate boilers, in which the uptake does not

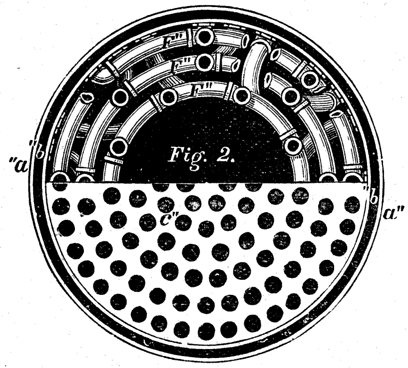


Fig. 3267.

come into contact with the water, and water tubes are employed, is shown in Fig. 3268. In the fire box is an inclined tube which promotes the circulation, and is very effective heating surface, and in the combustion chamber are a number of vertical water tubes.

Two manholes give access for cleaning purposes.

The efficiency of the heating surface in this class of boiler is increased from the fact that, as the heat does not pass direct through the boiler, it impinges against the surface. In Fig.

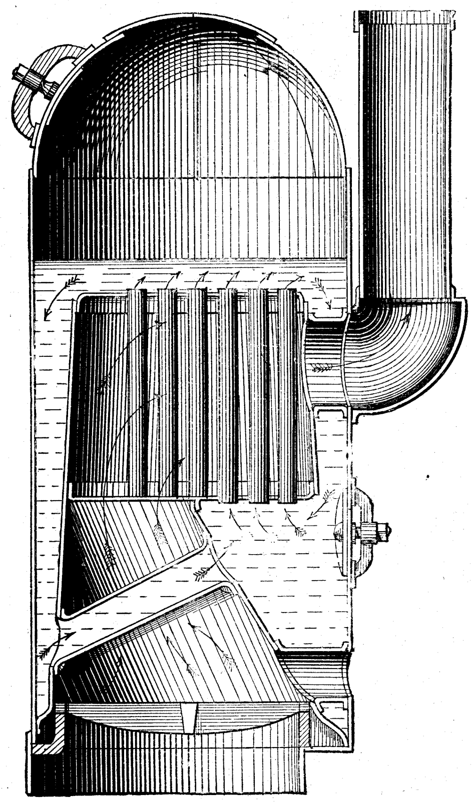


Fig. 3268.

3269, for example, the exit from the spherical fire box is on one side of the boiler, and the uptake on the other, the heat passing from the fire box into a combustion chamber, and thence through the horizontal fire tubes to the uptake.

The crown sheet is here stayed by gusset stays, but if made spherical, as in Fig. 3270, the stays may be omitted.

Figs. 3271, 3272, and 3273 illustrate a 60-inch horizontal return tubular boiler constructed by the Hartford Steam Boiler Inspection and Insurance Company. This class of boiler has found much favor in the United States. It is an externally fired, return tube boiler, the fire passing beneath the boiler and returning through

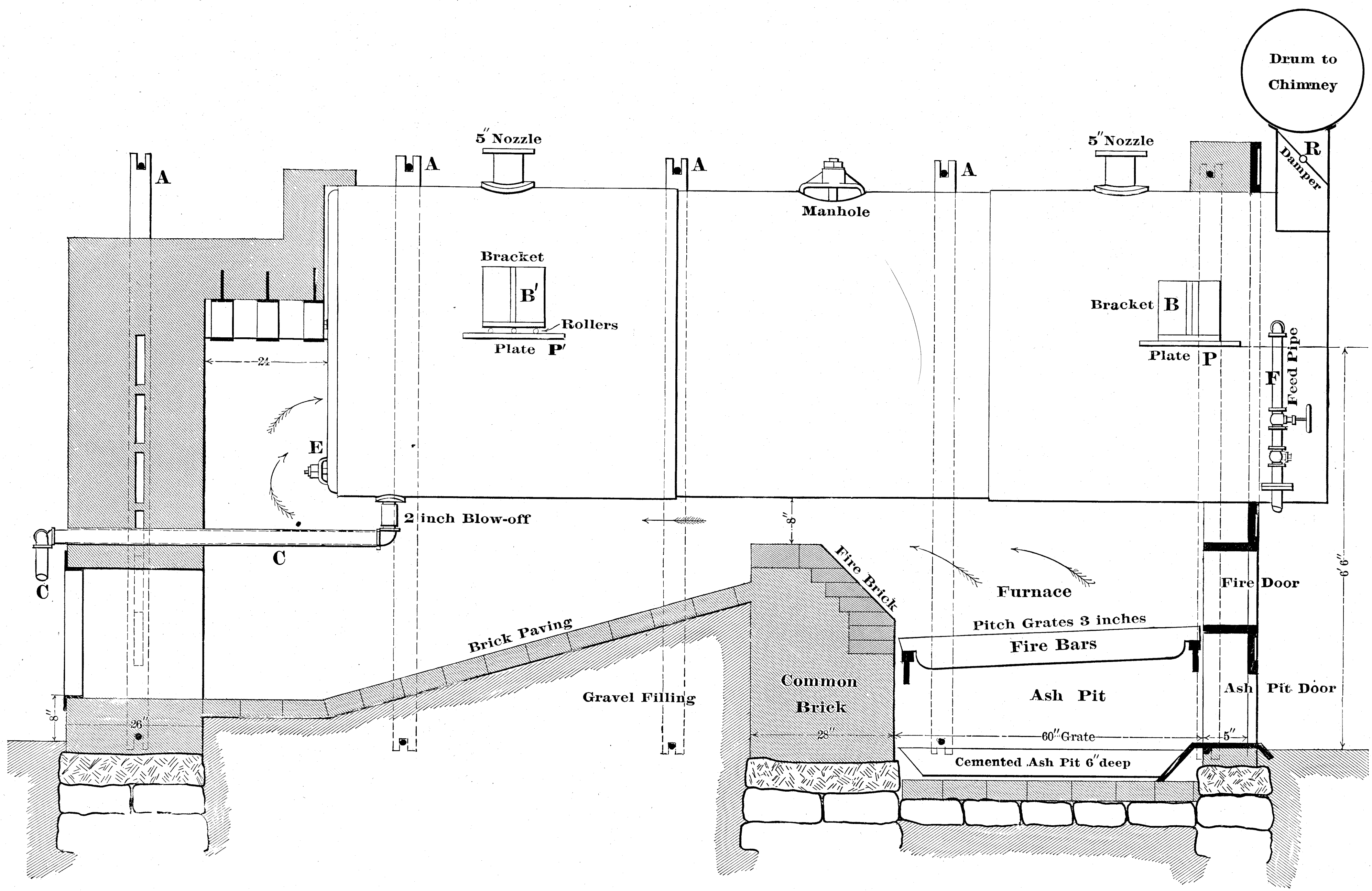

Drum to Chimney
R
Damper
5″ Nozzle
5″ Nozzle
A
A
A
A
Manhole
Bracket
B′
Rollers
Plate P′
Bracket B
Plate P
F
Feed Pipe
24
E
2 inch Blow-off
C
C
8″
6′ 6″
Fire Brick
Furnace
Fire Door
Pitch Grates 3 inches
Fire Bars
Brick Paving
Common Brick
Ash Pit
Ash Pit Door
Gravel Filling
8″
26″
28″
60″ Grate
5″
Cemented Ash Pit 6″ deep

Fig. 3271.

the tubes to the front end of the boiler, whence it passes through the drum to the chimney.

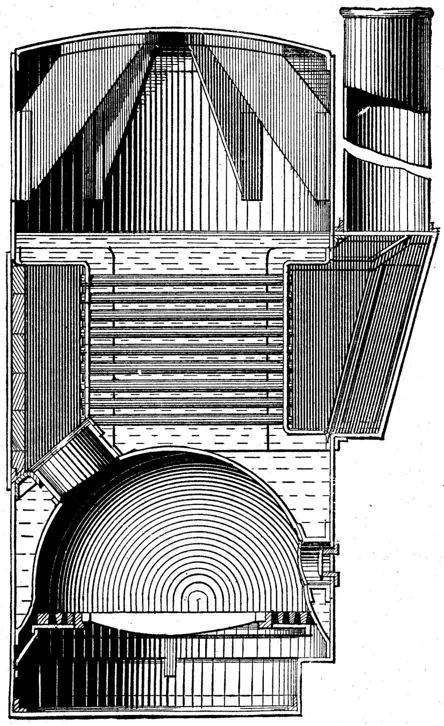


Fig. 3269.

The boiler is supported on the brackets B, B′, the front one, B, resting on an iron plate imbedded in the brickwork, and the back ones on rollers which rest on the plates P′ imbedded in the brickwork. This allows the boiler to expand

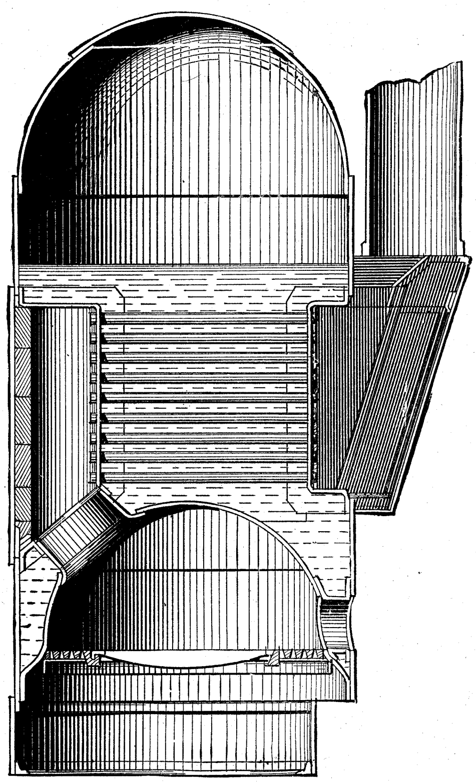


Fig. 3270.

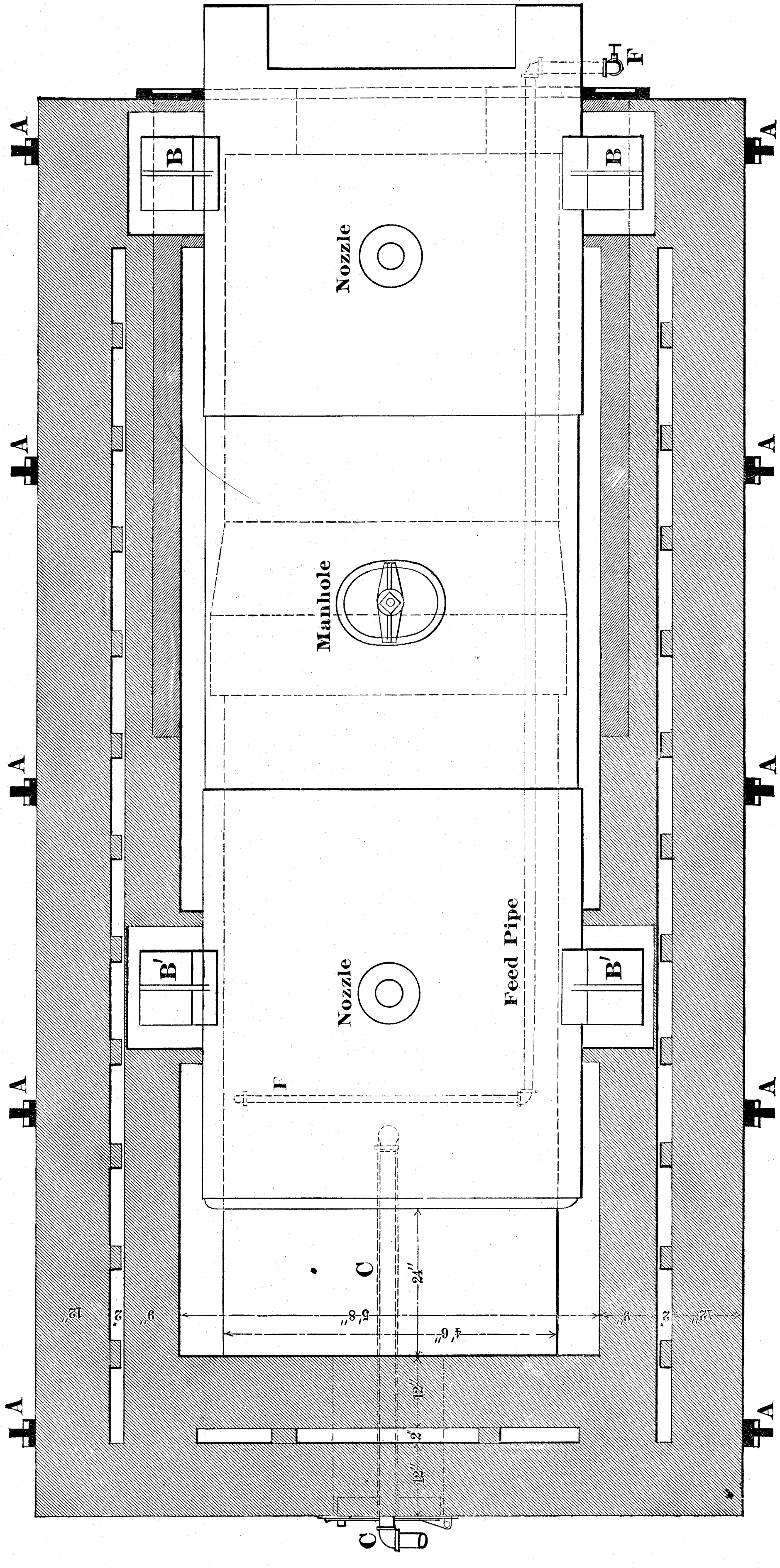


Fig. 3272.

and contract endways under variations of temperature without racking the brickwork.

A, A, etc., are for holding the brickwork together. The blow-off pipe C is for emptying or blowing down the boiler. The feed-pipe F enters the front end of the boiler, passes along it, and then crosses over. A pipe H from the steam space of the boiler supplies steam to the steam gauge G, and to the upper end of the prevent cracking of the brickwork. The tubes are arranged in vertical and horizontal rows and are equally spaced throughout.

Fig. 3274 represents the front end, and Fig. 3275 a longitudinal sectional view of the front end of a boiler of this class. In this case, however, the pipes for the water gauge pass direct into the boiler.

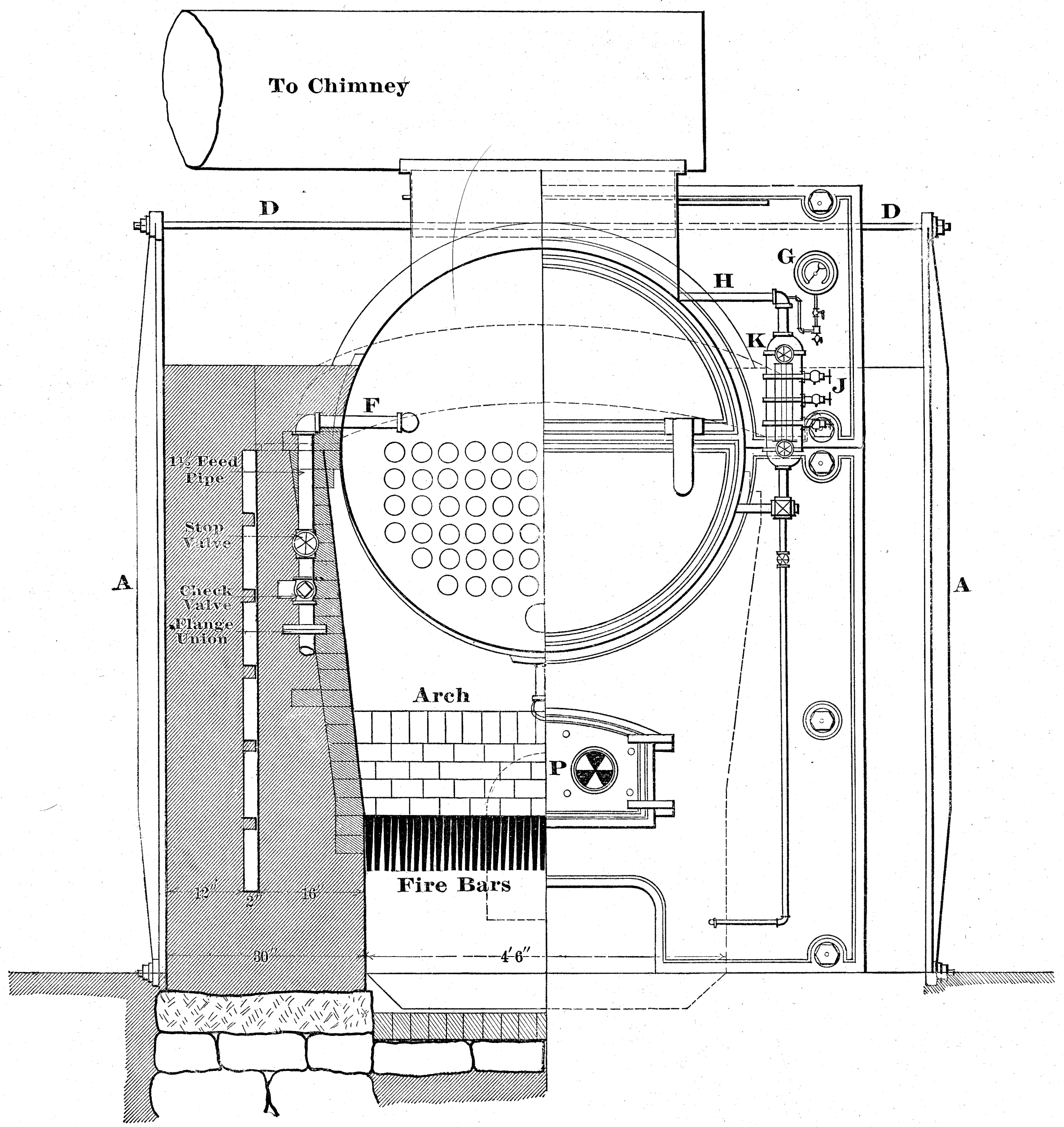


Fig. 3273.

gauge glass, which is on the casting K. The lower end of the gauge glass receives water from a pipe which passes into the water space of the boiler; at J are the three gauge cocks for testing the height of the water in the boiler.

The manhole affords ingress into the boiler for inspecting and for scaling or cleaning it, the nozzles being for a safety valve. At E is a hand-hole for washing out and cleaning the boiler. P is a damper in the fire door for admitting air above the fire bars, and R is a damper for regulating the draught.

In the brick walls that support the boiler there are air spaces to prevent the conduction of the heat through and

In some practice the tubes are arranged as in Fig. 3276, being wider pitched or spaced in the middle of the boiler to increase the circulation of the water in the boiler.

Another arrangement is shown in Fig. 3277, the tubes being *staggered* or arranged zigzag. This permits of the employment of a greater number of tubes, but does not afford such free circulation of the water.

Fig. 3278 represents an arrangement where the tubes are in rows both vertically and horizontally.

Fig. 3279 represents a boiler by the Erie Iron Works, the details of the setting being as follows:

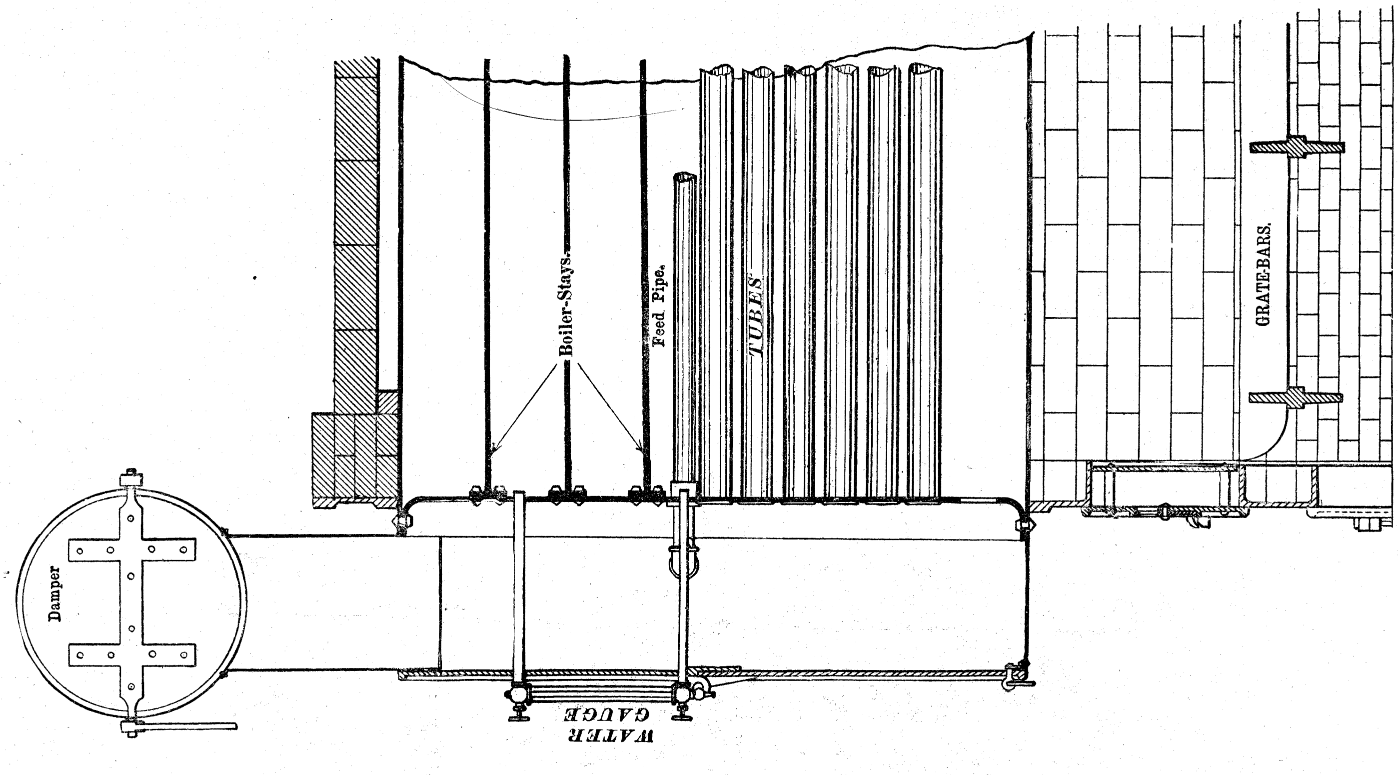


Fig. 3275.

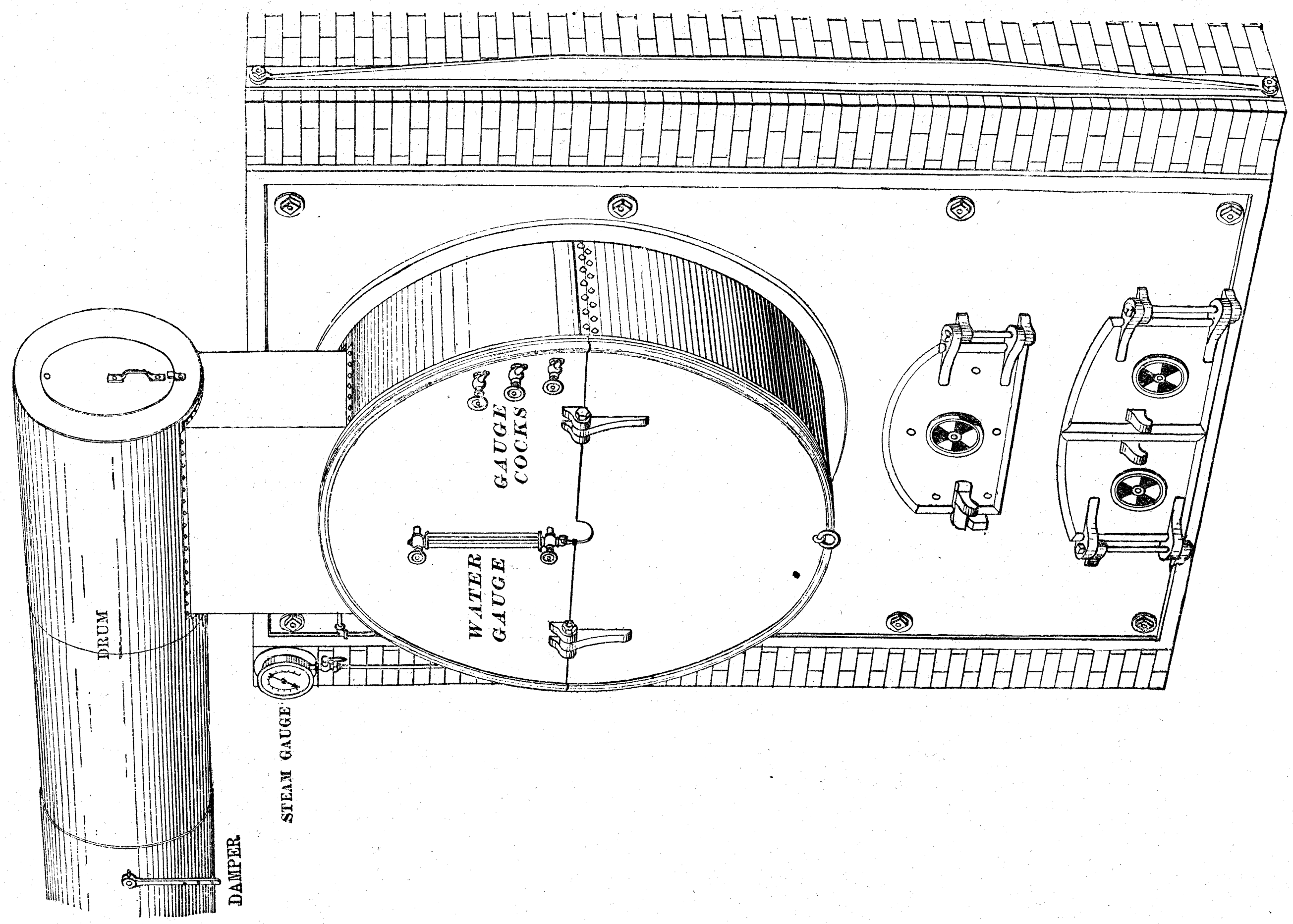


Fig. 3274.

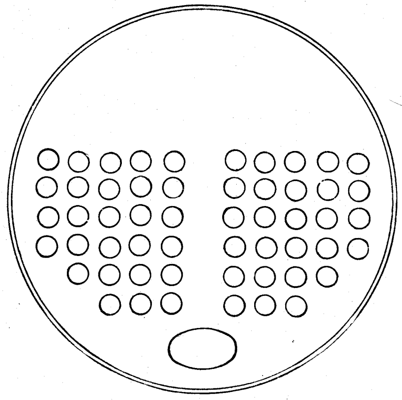


Fig. 3276.

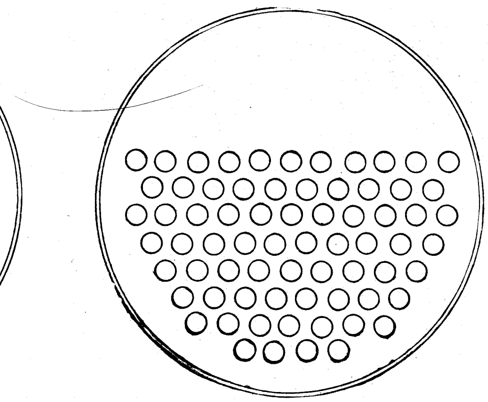


Fig. 3277.

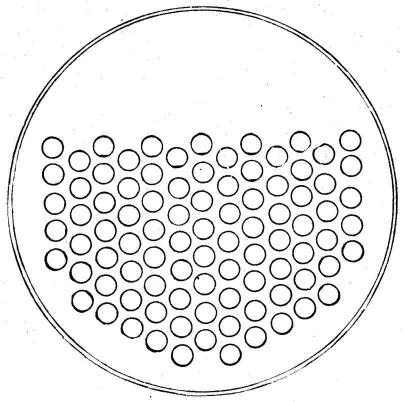


Fig. 3278.

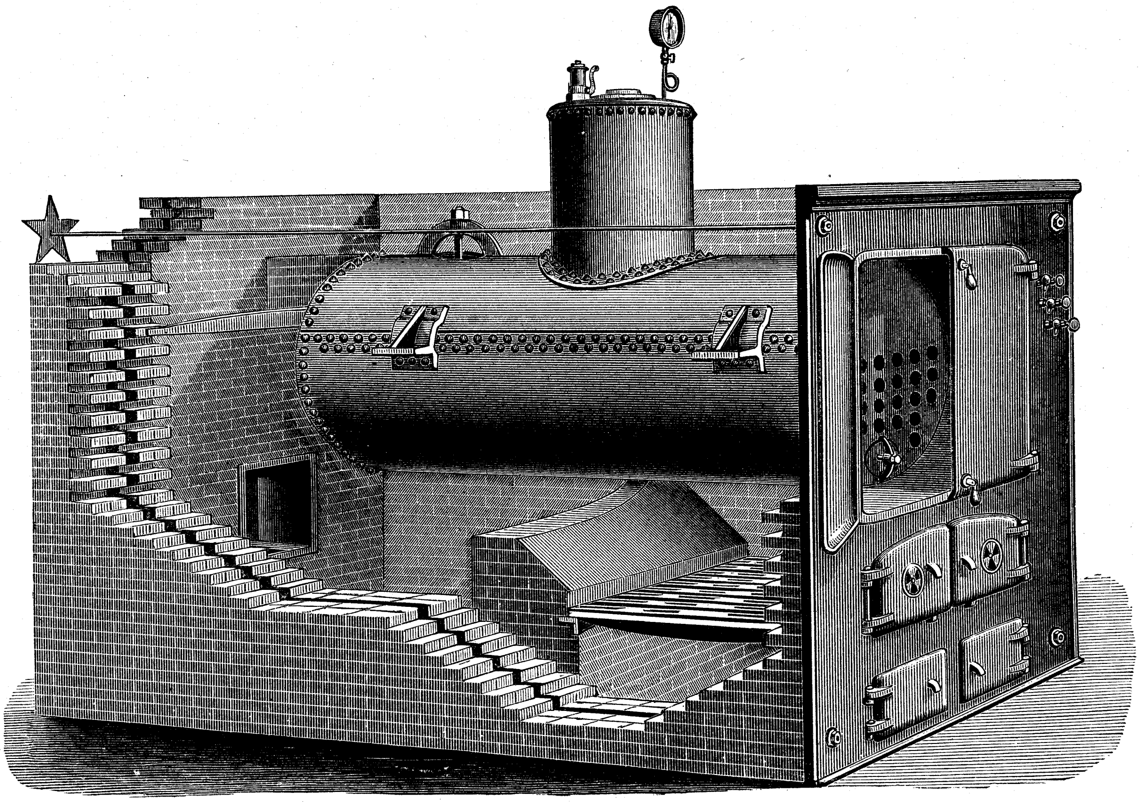


Fig. 3279.

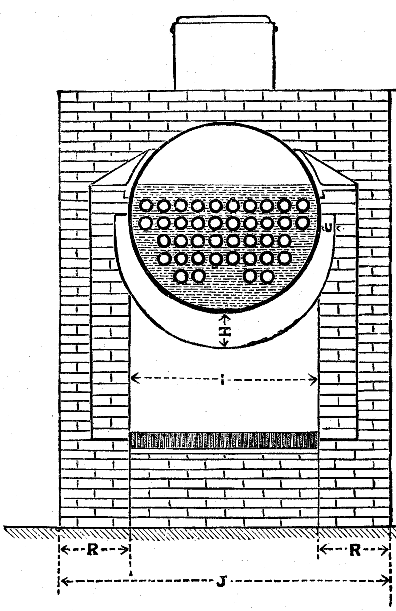


Fig. 3280.

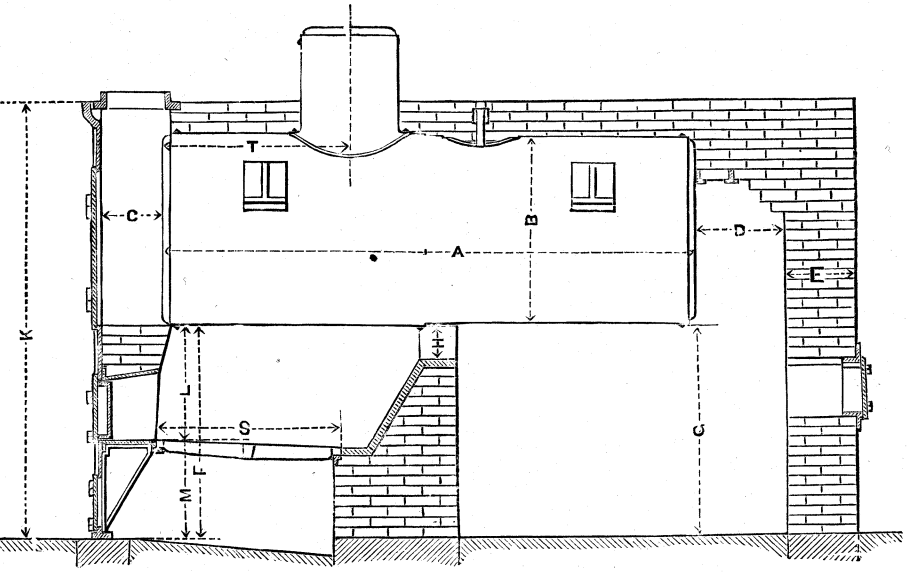


Fig. 3281.

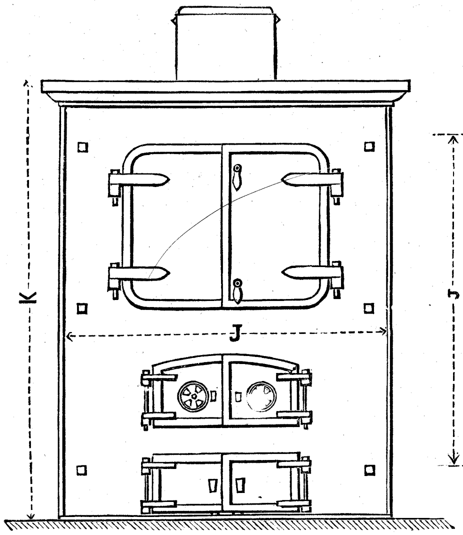


Fig. 3282.

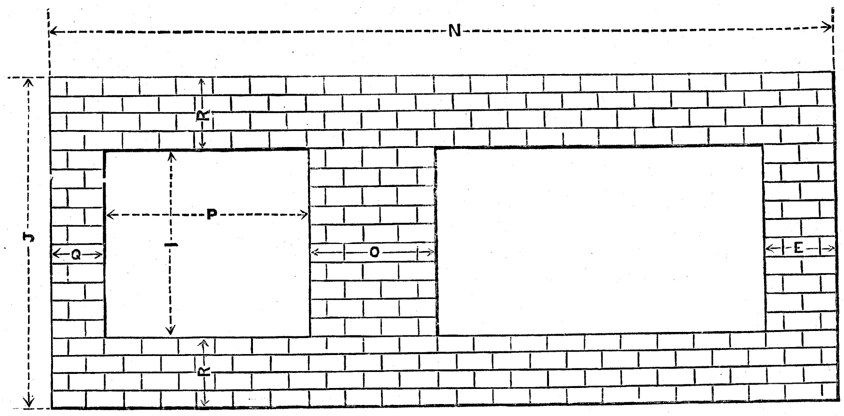


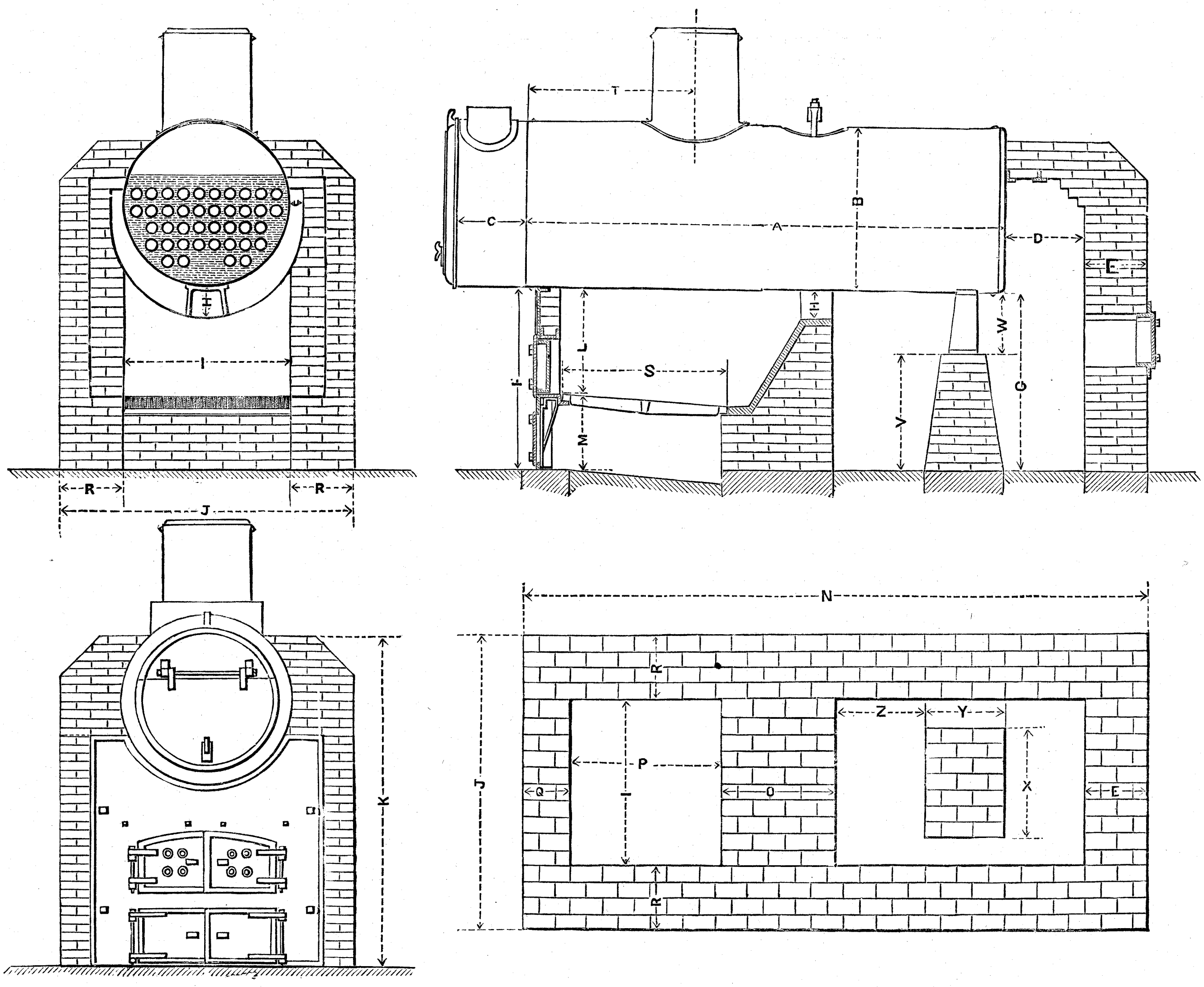
Fig. 3283.

Fig. 3280 is an end view of the setting with the brickwork in section.

Fig. 3281 side view of the boiler and setting.

Fig. 3282 a front end view of the boiler, and Fig. 3283 a ground plan of the brickwork. When the front plate of the boiler setting extends above the middle of the boiler, as in Fig. 3279, it is said to have a "full arch front." Whereas when this plate or casting extends to the middle only of the boiler, it is said to have a half arch front.

Figs. 3284, 3285, 3286, and 3287 show the setting for a half arch front boiler, the dimensions of the settings of both these boilers being given in the following tables:



Figs. 3284, 3285, 3286, 3287.

MEASUREMENTS FOR SETTING TUBULAR STATIONARY BOILERS WITH FULL ARCH FRONTS.

REFERENCE LETTERS ON DIAGRAMS.

No.	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	NO. OF FIRE BRICK.	NO. COMMON BRICK ABOVE FLOOR LEVEL.
	Ft.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ft. Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.		
1	7	32	12	20	16	45	44	7	32	64	85	26	19	11-6	20	40	12	16	36	34	4	600	6800
2	7	34	12	20	16	48	47	8	34	66	90	26	22	11-6	20	40	12	16	36	34	4	600	7500
3	8	36	12	20	16	48	47	8	36	68	92	26	22	12-6	24	40	12	16	36	34	4	650	7700
3½	10	36	12	20	16	48	47	8	36	68	92	26	22	14-6	28	46	12	16	42	42	4	720	8500
4	8	42	14	20	16	48	47	8	42	74	98	27	21	12-8	24	40	12	16	36	34	4	730	8500
5	10	42	14	20	16	48	47	8	42	74	98	27	21	14-8	28	46	12	16	42	42	4	770	9600
6	10	44	14	24	16	48	47	10	44	76	100	27	21	15-0	28	46	12	16	42	42	4	880	10500
7	12	44	14	24	16	48	46½	10	44	76	100	27	21	17-0	32	52	12	16	48	49	4	940	10800
7½	14	44	14	24	16	47	45½	10	44	76	99	26	21	19-0	36	58	12	16	54	84	4	1120	11500
8	12	48	16	24	16	47	45½	10	48	88	103	26	21	17-2	32	52	12	20	48	49	4	1120	13600
9	14	48	16	24	16	47	45½	10	48	88	103	26	21	19-2	36	58	12	20	54	84	4	1140	15700
10	12	54	16	24	20	50	48½	10	54	94	112	26	24	17-6	32	52	12	20	48	49	4	1160	16200
10½	15	54	16	24	20	50	48½	10	54	94	112	26	24	20-8	36	56	16	20	54	90	4	1270	17500
11	12	60	18	24	20	50	48½	12	60	108	118	26	24	17-10	32	50	16	24	48	49	4	1400	20500
12	14	60	18	24	20	50	48½	12	60	108	118	26	24	19-10	36	56	16	24	54	84	4	1500	23000
13	16	60	18	26	20	50	48	12	60	108	118	26	24	22-0	40	56	16	24	54	96	4	1540	25300
14	15	66	18	28	20	50	48½	12	66	114	124	26	24	21-2	36	56	16	24	54	90	4	1590	26000
15	16	66	18	28	20	50	48	12	66	114	124	26	24	22-2	40	56	16	24	54	96	4	1620	27000
16	16	72	20	30	20	50	48	12	72	120	130	26	24	22-6	40	56	16	24	54	96	4	1750	30000

NOTE.—In setting "Standard" boilers, the side walls should be so built that the longitudinal seams of the shell will be protected from the fire.

MEASUREMENTS FOR SETTING TUBULAR STATIONARY BOILERS WITH HALF ARCH FRONTS.

REFERENCE LETTERS ON DIAGRAMS.

No.	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W	X	Y	Z	NO. OF FIRE BRICK.	NO. OF COM. BRICK ABOVE FLOOR LEVEL.
	Ft.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins	Ft. Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins	Ins.	Ins.	Ins.	Ins.	Ins	Ins.		
1	7	32	14	20	16	46	45	7	32	64	73	26	20	10-3	20	33	12	16	36	34	4	36	9	24	12	7	600	6150
2	7	34	14	20	16	46	45	8	34	66	75	26	20	10-3	20	33	12	16	36	34	4	36	9	28	12	7	600	6200
3	8	36	14	20	16	46	45	8	36	68	77	26	20	11-3	24	33	12	16	36	34	4	36	9	28	12	15	650	6700
3½	10	36	14	20	16	46	45	8	36	68	77	26	20	13-3	28	39	12	16	42	42	4	36	9	28	12	25	720	7050
4	8	42	18	20	16	46	45	8	42	74	83	27	19	11-3	24	32½	12	16	36	34	4	32¾	12¼	32	16	11	730	7700
5	10	42	18	20	16	46	45	8	42	74	83	27	19	13-3	28	38½	12	16	42	42	4	32¾	12¼	32	16	25	770	8700
6	10	44	18	24	16	46	45	10	44	76	85	27	19	13-7	28	38½	12	16	42	42	4	32¾	12¼	36	16	25	880	8800
7	12	44	18	24	16	46	44½	10	44	76	85	27	19	15-7	32	44½	12	16	48	49	4	32¼	12¼	36	20	35	940	9300
7½	14	44	18	24	16	46	44½	10	44	76	85	27	19	17-7	36	50½	12	16	54	84	4	32¼	12¼	36	24	45	1120	9500
8	12	48	19	24	16	50	48½	10	48	88	93	26	24	15-7	32	48	12	20	48	49	4	36¼	12¼	36	20	35	1120	11100
9	14	48	19	24	16	50	48½	10	48	88	93	26	24	17-7	36	54	12	20	54	84	4	36¼	12¼	36	24	45	1140	12900
10	12	54	19	24	20	50	48½	10	54	94	99	26	24	15-11	32	48½	12	20	48	49	4	34	14½	42	20	35	1160	13200
10½	15	54	19	24	20	50	48½	10	54	94	99	26	24	19-1	36	52½	16	20	54	90	4	34	14½	42	24	57	1270	14200
11	12	60	21	24	20	46¾	45½	12	60	108	101¾	26	20¾	16-1	32	47	16	24	48	49	4	31	14½	48	20	37	1400	16700
12	14	60	21	24	20	46¾	45	12	60	108	101¾	26	20¾	18-1	36	53	16	24	54	84	4	30½	14½	48	24	45	1500	19200
13	16	60	21	26	20	46¾	45	12	60	108	101¾	26	20¾	20-3	40	53	16	24	54	96	4	30½	14½	48	24	65	1540	21500
14	15	66	24	28	20	47	45½	12	66	114	108	26	21	19-5	36	52½	16	24	54	90	4	31	14½	54	24	57	1590	22100
15	16	66	24	28	20	47	45½	12	66	114	108	26	21	20-5	40	52½	16	24	54	96	4	31	14½	54	24	65	1620	23100
16	16	72	24	30	20	48	46½	12	72	120	115	28¼	19¾	20-7	40	52½	16	24	54	96	4	27½	19	54	24	65	1750	26000

NOTE.—In setting "Standard" boilers, the side walls should be so built that the longitudinal seams of the shell will be protected from the fire.

THE EVAPORATIVE EFFICIENCIES OF BOILERS.

*"Many tests have been undertaken to ascertain the evaporative power of different classes of boilers in actual work; but few of these are of any value, owing to the unreliable means usually employed to measure the quantity of water evaporated. The easiest method, and consequently the one most frequently adopted, is to measure the quantity by the difference of its height in the water-gauge glass at the beginning and end of the trial, and also at intermediate stages. This method is very rude and uncertain, since there can be little doubt that in many boilers at work the surface of the water is not level, but is usually higher over the furnace, or where the greatest ebullition occurs. The difference in height at any moment will greatly depend upon the intensity of the ebullition, which is ever varying during the intervals between firing. With mechanical firing the difference of height is probably reduced to a minimum.

"The meters employed for measuring the water are sometimes not trustworthy. The only sure method of ascertaining the quantity of water evaporated is by actual measurement with a cistern or vessel whose cubic contents are accurately known. The quantity of water in the boiler before and after the trial should be measured at the same temperature, which should not exceed 212° to insure accuracy. But even when the amount of water introduced and the quantity passed off from the boiler are accurately ascertained, there yet remains a doubt as to how much has been actually evaporated, and how much may have passed off in priming, unless the trial has been conducted with the boiler open to the atmosphere, which appears to be the only condition under which accuracy can be insured, unless a suitable apparatus can be provided for accurately measuring the weight and temperature of all the steam and water given off, when the boiler is working above atmospheric pressure.

"There are very few boilers that do not prime more or less, and the quantity of water passed off in this manner is sometimes very considerable, and has led to the impossible results of 16 and 17 lbs. of water evaporated per lb. of ordinary coal in locomotive and water-tube boilers being seriously recorded. Externally fired boilers, that have given the moderate result of 5 lbs. of water per

* From "*A Treatise on Steam Boilers*," by Robert Wilson.

lb. of coal at atmospheric pressure, have shown the unexpected result of 10 and 12 lbs. of water evaporated at 40 lbs. pressure. In fact, unless the amount of water passed over with the steam by priming or foaming, when working under pressure, can be accurately ascertained, the evaporative results are not to be relied upon, however careful in other respects the trial may have been conducted. It is customary to give the quantity of water evaporated from and at a temperature of 212°, or the boiling point at atmospheric pressure, to which the results of evaporation are reduced."

The quantity corresponding to any temperature of feed water and working pressure can readily be found with the aid of the annexed table, taken from *The Encyclopædia Britannica*, wherein are presented the relations of the properties of steam, as now accepted by the best authorities.

TABLE GIVING THE PRESSURE, TEMPERATURE, AND VOLUME OF STEAM.

Total pressure per square inch measured from a vacuum. lbs.	Gauge pressure or pressure above atmosphere. lbs.	Sensible temperature in Fahrenheit degrees.	Total heat in degrees from zero of Fahrenheit.	Weight of one cubic foot of steam. lbs.	Relative volume of steam compared with the water from which it was evaporated.
1	—	102.1	1144.5	.0030	20582
2	—	126.3	1151.7	.0058	10721
3	—	141.6	1156.6	.0085	7322
4	—	153.1	1160.1	.0112	5583
5	—	162.3	1162.9	.0138	4527
6	—	170.2	1165.3	.0163	3813
7	—	176.9	1167.3	.0189	3298
8	—	182.9	1169.2	.0214	2909
9	—	188.3	1170.8	.0239	2604
10	—	193.3	1172.3	.0264	2358
11	—	197.8	1173.7	.0289	2157
12	—	202.0	1175.0	.0314	1986
13	—	205.9	1176.2	.0338	1842
14	—	209.6	1177.3	.0362	1720
14.7	0	212.0	1178.1	.0380	1642
15	.3	213.1	1178.4	.0387	1610
16	1.3	216.3	1179.4	.0411	1515
17	2.3	219.6	1180.3	.0435	1431
18	3.3	222.4	1181.2	.0459	1357
19	4.3	225.3	1182.1	.0483	1290
20	5.3	228.0	1182.9	.0507	1229
21	6.3	230.6	1183.7	.0531	1174
22	7.3	233.1	1184.5	.0555	1123
23	8.3	235.3	1185.2	.0580	1075
24	9.3	237.8	1185.9	.0601	1036
25	10.3	240.1	1186.6	.0625	996
26	11.3	242.3	1187.3	.0650	958
27	12.3	244.4	1187.8	.0673	926
28	13.3	246.4	1188.4	.0696	895
29	14.3	248.4	1189.1	.0719	866
30	15.3	250.4	1189.8	.0743	838
31	16.3	225.2	1190.4	.0766	813
32	17.3	254.1	1190.9	.0779	789
33	18.3	255.9	1191.5	.0812	767
34	19.3	257.6	1192.0	.0835	746
35	20.3	259.3	1192.5	.0858	726
36	21.3	260.9	1193.0	.0881	707
37	22.3	262.6	1193.5	.0905	688
38	23.3	264.2	1194.0	.0929	671
39	24.3	265.8	1194.5	.0952	655
40	25.3	267.3	1194.9	.0974	640
41	26.3	268.7	1195.4	.0996	625
42	27.3	270.2	1195.8	.1020	611
43	28.3	271.6	1196.2	.1042	598
44	29.3	273.0	1196.6	.1065	595
45	30.3	274.4	1197.1	.1089	572
46	31.3	275.8	1197.5	.1111	561
47	32.3	277.1	1197.9	.1133	550
48	33.3	278.4	1198.3	.1156	539
49	34.3	279.7	1198.7	.1179	529
50	35.3	281.0	1199.1	.1202	518
51	36.3	282.3	1199.5	.1224	509
52	37.3	283.5	1199.9	.1246	500
53	38.3	284.7	1200.3	.1269	491
54	39.3	285.9	1200.6	.1291	482
55	40.3	287.1	1201.0	.1314	474
56	41.3	288.2	1201.3	.1336	466
57	42.3	289.3	1201.7	.1364	458
58	43.3	290.4	1202.0	.1380	451
59	44.3	291.6	1202.4	.1403	444
60	45.3	292.7	1202.7	.1425	437
61	46.3	293.8	1203.1	.1447	403
62	47.3	294.8	1203.4	.1469	424
63	48.3	295.9	1203.7	.1493	417
64	49.3	296.9	1204.0	.1516	411
65	50.3	298.0	1204.3	.1538	405
66	51.3	299.0	1204.6	.1560	399
67	52.3	300.0	1204.9	.1583	393
68	53.3	300.9	1205.2	.1605	388
69	54.3	301.9	1205.5	.1627	383
70	55.3	302.9	1205.8	.1648	378
71	56.3	303.9	1206.1	.1670	373
72	57.3	304.8	1206.3	.1692	368
73	58.3	305.7	1206.6	.1714	363
74	59.3	306.6	1206.9	.1736	359
75	60.3	307.5	1207.2	.1759	353
76	61.3	308.4	1207.4	.1782	349
77	62.3	309.3	1207.7	.1804	345
78	63.3	310.2	1208.0	.1826	341
79	64.3	311.1	1208.3	.1848	337
80	65.3	312.0	1208.5	.1869	333
81	66.3	312.8	1208.8	.1891	329
82	67.3	313.6	1209.1	.1913	325
83	68.3	314.5	1209.4	.1935	321
84	69.3	315.3	1209.6	.1957	318
85	70.3	316.1	1209.9	.1980	314
86	71.3	316.9	1210.1	.2002	311
87	72.3	317.8	1210.4	.2024	308
88	73.3	318.6	1210.6	.2044	305
89	74.3	319.4	1210.9	.2067	301
90	75.3	320.2	1211.1	.2089	298
91	76.3	321.0	1211.3	.2111	295
92	77.3	321.7	1211.5	.2133	292
93	78.3	322.5	1211.8	.2155	289
94	79.3	323.3	1212.0	.2176	286
95	80.3	324.1	1212.3	.2198	283
96	81.3	324.8	1212.5	.2219	281
97	82.3	325.6	1212.8	.2241	278
98	83.3	326.3	1213.0	.2263	275
99	84.3	327.1	1213.2	.2285	272
100	85.3	327.9	1213.4	.2307	270
101	86.3	328.5	1213.6	.2329	267
102	87.3	329.1	1213.8	.2351	265
103	88.3	329.9	1214.0	.2373	262
104	89.3	330.6	1214.2	.2393	260
105	90.3	331.3	1214.4	.2414	257
106	91.3	331.9	1214.6	.2435	255
107	92.3	332.6	1214.8	.2456	253
108	93.3	333.3	1215.0	.2477	251
109	94.3	334.0	1215.3	.2499	249
110	95.3	334.6	1215.5	.2521	247
111	96.3	335.3	1215.7	.2543	245
112	97.3	336.0	1215.9	.2564	243
113	98.3	336.7	1216.1	.2586	241
114	99.3	337.4	1216.3	.2607	239
115	100.3	338.0	1216.5	.2628	237
116	101.3	338.6	1216.7	.2649	235
117	102.3	339.3	1216.9	.2674	233
118	103.3	339.9	1217.1	.2696	231
119	104.3	340.5	1217.3	.2738	229
120	105.3	341.1	1217.4	.2759	227
121	106.3	341.8	1217.6	.2780	225
122	107.3	342.4	1217.8	.2801	224
123	108.3	343.0	1218.0	.2822	222
124	109.3	343.6	1218.2	.2845	221
125	110.3	344.2	1218.4	.2867	219
126	111.3	344.8	1218.6	.2889	217
127	112.3	345.4	1218.8	.2911	215
128	113.3	346.0	1218.9	.2933	214
129	114.3	346.6	1219.1	.2955	212
130	115.3	347.2	1219.3	.2977	211
131	116.3	347.8	1219.5	.2999	209
132	117.3	348.3	1219.6	.3020	208
133	118.3	348.9	1219.8	.3040	206
134	119.3	349.5	1220.0	.3060	205
135	120.3	350.1	1220.2	.3080	203
136	121.3	350.6	1220.3	.3101	202
137	122.3	351.2	1220.5	.3121	200
138	123.3	351.8	1220.7	.3142	199
139	124.3	352.4	1220.9	.3162	198
140	125.3	352.9	1221.0	.3184	197
141	126.3	353.5	1221.2	.3206	195
142	127.3	354.0	1221.4	.3228	194
143	128.3	354.5	1221.6	.3250	193
144	129.3	355.0	1221.7	.3273	192
145	130.3	355.6	1221.9	.3294	190

Total pressure per square inch measured from a vacuum. lbs.	Gauge pressure or pressure above atmosphere. lbs.	Sensible temperature in Fahrenheit degrees.	Total heat in degrees from zero of Fahrenheit.	Weight of one cubic foot of steam. lbs.	Relative volume of steam compared with the water from which it was evaporated.
146	131.3	356.1	1222.0	.3315	189
147	132.3	356.7	1222.2	.3336	188
148	133 3	357 2	1222.3	.3357	187
149	134.3	357.8	1222 5	.3377	186
150	135.3	358 3	1222.7	.3397	184
155	140.3	361.0	1223.5	.3500	179
160	145.3	363.4	1224.2	.3607	174
165	150.3	366.0	1224.9	.3714	169
170	155.3	368.2	1225.7	.3821	164
175	160.3	370.8	1226.4	.3928	159
180	165.3	372.9	1227.1	.4035	155
185	170.3	375.3	1227.8	.4142	151
190	175.3	377.5	1228.5	.4250	148
195	180.3	379.7	1229.2	.4357	144
200	185.3	381.7	1229 8	.4464	141
210	195 3	386.0	1231.1	.4668	135
220	205.3	389.9	1232.3	.4872	129
230	215.3	393.8	1233 5	.5072	123
240	225.3	397 5	1234.6	.5270	119
250	235.3	401.1	1235 7	.5471	114
260	245.3	404.5	1236.8	.5670	110
270	255.3	407.9	1237.8	.5871	106
280	265.3	411.2	1238.8	.6070	102
290	275.3	414.4	1239.8	.6268	99
300	285.3	417.5	1240.7	.6469	96

Here we see that at 212° the total quantity of heat in the steam is 1178.1°, which gives a difference of 966.1°. This heat, usually termed latent, is absorbed in performing the work of expanding the particles of water from the liquid to the gaseous state. Now, suppose the water is evaporated at 60 lbs. pressure, the steam will have a temperature of 307°, and a total heat of 1207°. If the feed has been introduced at 60°, it is evident that 1147° of heat have been imparted. As the amount evaporated is inversely proportional to the quantity of heat required, we have $1147 \div 966 = 1.2$. Multiplying by this factor, the quantity evaporated at 60 lbs. pressure from 60°, we obtain the amount that would be evaporated at 212° by the same quantity of fuel.

By the same table can be ascertained the comparatively small increase of heat required to evaporate water at higher pressures. Suppose we take water evaporated at 45 lbs. pressure from a feed temperature of 60°, then each lb. of water will require $1202.7 - 60 = 1142.7°$ for its conversion into steam. If we take the pressure at 100 lbs., we shall have $1216.9 - 60 = 1156.9°$ as the quantity required. The difference between these two total quantities is only 14.2°, and is so small as to be scarcely worth considering. Leaving out of account the loss due to the slight reduction of the conducting power of the material, the increased amount of heat required for the higher pressure will be only $\frac{1}{80}$ of the total heat required at 60 lbs. With an evaporation of 7 lbs. of water from 1 lb. of coal, it will be obtained by using $\frac{1}{563}$ more fuel, or about 1 lb. in about 556 lbs., a quantity not appreciable to the ordinary modes of weighing coal. The economy is then manifest of using steam of high pressures when at the same time advantage is taken of the facilities it offers for working the steam more expansively to the engine cylinders.

The saving that may be effected by heating the feed water may be shown as follows:

If we take the normal temperature of the feed water at 60°, the temperature of the heated water at 212°, and the boiler pressure at 20 lbs., the total heat imparted to the steam in one case is

$$1192.5° - 60° = 1132.5°$$

and in the other case $1192.5° - 212° = 980.5°$

the difference being 152°, or a saving of $\frac{152°}{1132.5}$

which is 13.4 per cent. If the pressure be taken at 120 lbs., instead of 20 lbs., the saving will be 13.1 per cent. showing a slight diminution in the economy effected by heating the feed water when a high boiler pressure is employed.

THE CARE AND MANAGEMENT OF STATIONARY ENGINE BOILERS.

The first thing to do in taking charge of a stationary engine boiler is to know from personal inspection that the safety fittings and the boiler-feeding apparatus are in good order.

The safety valve is the first thing to inspect, as it is liable to stick in its seat, especially in cases in which it is set at a greater pressure than is got up in the boiler, because in that case it is not lifted from the seat and in time sticks fast there.

In such cases it is proper to lift the valve at least once a day while steam is on. For this purpose a cord may be attached to the lever, passing over a pulley directly above the lever, and thence to some convenient place near the boiler, but where it is not liable to get caught and pulled accidentally.

Before lighting the fire, see that there is sufficient water in the boiler. If there is a gauge glass on the boiler, it should show three-quarters full, or three-quarters of a glass, as it is called.

The gauge glass may show a false water level, and to be sure that such is not the case, open the top gauge cock and the cock at the bottom of the gauge glass, letting the water run through the gauge glass. Open and close the cock below the gauge glass two or three times to see that the water comes to the same level each time.

If the steam pressure has been allowed to fall in the boiler without any of the cocks being opened, there will be a partial vacuum in the boiler, and air must be let in before the true water level will be shown either by the gauge glass or by the gauge cocks.

Opening the upper gauge cock will let in the air, and it should not be closed again until enough steam has been got up in the boiler to expel the air again, or in other words, until steam begins to issue from it.

The grate bars and ash pit should be cleaned of clinker, ashes, etc., and it should be seen that the tubes are clear of ashes, etc., before the fire is laid; if the grate is a shaking one, the lever should be applied to see that the grate will shake properly.

TO LIGHT THE FIRE.—In the case of anthracite or hard coal, as it is sometimes termed, first cover the bars with a thin layer of coal and then put in pieces of lighted greasy waste (if it is at hand) distributed about the furnace, taking especial care to light the fire at the fire-door end and in the corners, because the fire will spread from the front end towards the back easier than it will from the back end towards the front.

The fire should light from the bottom and not from the top, hence the thinnest pieces of the wood should be put in first.

If there is any soft coal at hand, a small quantity of it will accelerate lighting the fire, as it burns easier and quicker than hard coal.

Before putting on the coal the wood should be well lighted, the bottom portion of it having ceased flaming.

This causes the lighted wood to spread over the bars and the fire to light evenly.

Charge the coal lightly, first covering the places that have burned up the most.

FIRING.—The fire door should be kept open as little as possible, as it admits cold air that is detrimental to the combustion, as well as to the draught, hence firing should be done quickly.

A good fireman will maintain as even a temperature as possible in the fire box by charging the coal lightly and quickly.

Some firemen will, after the fire is at its proper depth all over the grates, charge the fire in the front end, that is, at the fire-door end, and push it back as it burns up, to keep up the thickness of the fire at the back.

The thickness of the fire depends upon the size and kind of coal. With small coal a fire from 4 to 6 inches deep will answer, while, if the lumps are five or six inches in diameter, a fire from a foot to 15 inches deep may be maintained, as is done in some locomotives.

The object is to have the fire thick enough to prevent it from burning through in spots or letting cold draughts of air pass through it.

The sides of the furnace require particular attention, not only because cold air is more likely to get through there, but also in boilers having fire boxes the cool sides of the box keep the temperature of the fuel down, hence a thicker fire is necessary around the sides than in the middle of the furnace or fire box.

Three things are to be considered in cleaning a fire—first, that the boiler pressure will fall during, and for a short time after, the cleaning; second, that the depth of fire will be diminished by the cleaning; and third, that the temperature of the fire will fall during the cleaning.

SHAKING GRATE BARS.

When a furnace has shaking grate bars, the cleaning of the fire is greatly facilitated, and with bars that shake singly (and good coal) the fire is often not disturbed during the day, except to shake the bars occasionally, passing the poker through it and using the hoe to keep it evenly spread.

If the grate shakes in sections, more cleaning will be required to break up the clinker, while, if the bars do not shake, the cleaning assumes greater importance.

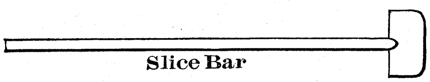


Fig. 3288.

Before cleaning, therefore, see that there is sufficient water in the boiler, that it need not be fed while cleaning, nor just after cleaning the fire.

Prepare for cleaning by having a thick fire on the grate, so that after cleaning it will burn up quickly, and let the cleaning be done as quickly as possible.

The tools used for cleaning the fire are the slice bar, Fig. 3288, which is pushed along the top of the fire bars to loosen up the fire, and let the ashes fall through the bars.

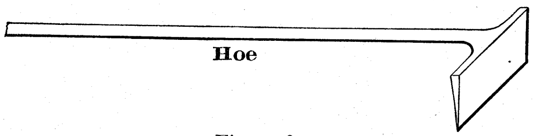


Fig. 3289.

The hoe, Fig. 3289, which is used to push the fire to the back of the furnace and to pull it forward. The poker, Fig. 3290, which dislodges any clinker that may be between the bars, and lets the ashes fall through.

The clinker hook or devil's claw, Fig. 3291, which is used to haul clinker out of the fire, and the rake, Fig. 3292, which is used to spread the fire evenly over the bars after it is cleaned.

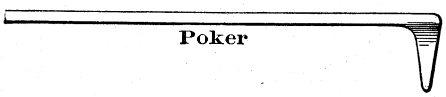


Fig. 3290.

In cleaning a fire, first use the slice bar to loosen up the fire and let the ashes fall through, and also dislodge clinkers from the surface of the bars. Then push the fire to the back of the furnace. Next use the poker to clean out clinker from between the exposed part of the bars. Then with the hoe pull a part of the fire forward and pull out the clinker that may be in this part, doing so with the hoe as far as possible, as that will save time, but if it should be necessary, use the clinker hook.

Then pull forward a second portion of the fire, and spread it on the bars, removing the clinker as before. When all the fire has thus been cleaned, use the rake to spread it evenly over the bars, and put on a light charge of coal, covering the brightest parts of the fire first, and taking care that no part of the fire bars is left uncovered.

The cleaning should be done quickly.

DRAUGHT.—The draught should be decreased while the fire is being cleaned, but the damper should never be entirely closed, as this might cause an explosion in the fire box and tubes.

During a temporary interruption, as in the case of the engine stopping, partly close all the dampers, as it is wasteful to make steam and blow it off through the safety valve.

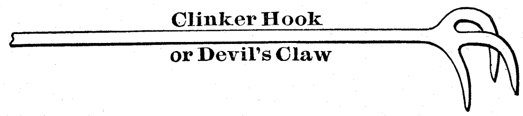


Fig. 3291.

COMBUSTION.—A blue flame is evidence of incomplete combustion, but there may be a blue flame and imperfect combustion at the back end of the furnace, and a white flame and perfect combustion at the other end.

This is likely to occur with heavy firing near the fire door, and a thin fire at the tube sheet end of the fire box. In this case the unconsumed gases produced near the fire door (as evidenced by the blue flame) are consumed in passing over the bright fire at the tube plate end of the furnace.

AT NIGHT.—Always leave plenty of water in the boiler when leaving it for the night, not only to allow for any leak, but also because it gives a fair start in the morning and more time to remedy any defect in the feed pump if it arise.

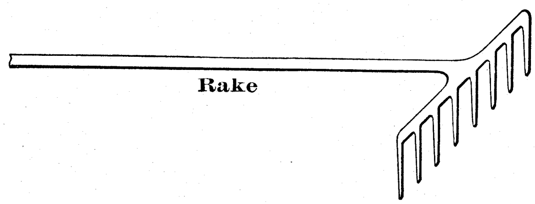


Fig. 3292.

By plenty of water, very nearly a full gauge is meant, or if there is no gauge glass to the boiler, let the water stand above the second or middle cock.

The usual method of leaving the fire for the night is to bank it. There is an element of danger, however, in banking a fire, unless it is done to suit the circumstances, because steam may generate very rapidly, and perhaps more rapidly than the safety valve can carry it off.

A safe method is to clean the fire, leaving the clinker and ashes covering the front half of the grate and the fire piled up on the back half.

The damper and ash pit door should be closed tight, the fire door open, and the fire well covered with fresh coal, choosing small rather than large coal.

If this method is found not to keep up the fire sufficiently, the same plan may be employed, except that the ashes and clinker may be removed, and if this still leaves too cold a boiler, and too poor a fire in the morning, the fire may be left spread over the grate, but heavily covered with fresh coal, the draught being stopped as much as possible by closing the dampers and opening the furnace door.

To further insure safety, the weight on the safety valve lever should be pushed towards the valve, so as to cause the safety valve to blow off at a less pressure than during the day.

IN THE MORNING.—In starting up a banked fire in the morning, first close the fire door and open the damper, so as to give the fire all the draught possible, and let it burn up a little; then, if it has been piled up at the back of the furnace, clean out the ashes by passing the T bar beneath the fire, and spread it over the grate, letting it burn up a little before making up a fire.

BOILER-FEED.—The fireman should endeavor, if possible, to so regulate the boiler feed that it is kept going as nearly continuously as possible while maintaining a uniform quantity of water in the boiler, and this, with uniform firing, will give the greatest economy.

When pumps are used to feed with, the amount of the lift of the valves can be regulated by a screw, so as to vary the amount of water the pump will deliver, and in this case it is comparatively easy to set them so that the pump may be kept going without putting too much water in the boiler.

When injectors are used, however, the feed will be intermittent, and a uniform quantity of water in the boiler is best obtained by feeding at short intervals, stopping the feed when the fire door is opened much, as when cleaning the fire.

If the feed water is dirty, the gauge glass should be kept clean by first shutting off the upper cock and opening the lower one, so as to let the water blow through the lower cock, and then shutting off the lower cock from the boiler, and opening the upper one, which will let the steam blow all the water out of the glass. This should be done two or three times a day, so as to keep the holes in the boiler and those in the cocks from closing up with fur or scale.

If the water falls in the glass, or if the gauge cocks show the water to be falling, notwithstanding that the feed pump has been started, it is evident that the pump is not working.

This may occur from a stuck valve, a leak in the suction pipe, from the feed water being too hot, or from the pump failing to start in action from leaky or choked valves.

A stuck valve may generally be relieved by striking a few blows on the outside of the pump with a hammer and a block of wood, or if this does not answer, with the hammer only. Check valves are the ones most likely to stick.

If a pump fails to work by reason of the feed water being too hot, the remedy is to open the pet cock to let the steam out of the pump, but if this does not succeed, cold water may be poured on the outside of the pump, which will start it, after which, in most cases, the pump will keep going and the pet cock may be closed.

If the suction pipe has a joint, a leak there will impair the action of the pump, and, if the leak is great enough, will stop it; the remedy is to make the joint tight.

Plunger pumps sometimes fail to act because the plunger has worn so small in diameter that there is sufficient air between the plunger and the pump barrel to expand and compress without lifting the valve; the remedy is obviously a new plunger of as large diameter as the pump gland will admit of, boring the gland out to admit the new plunger.

All the impurities in the water are left in the boiler when the water has evaporated, and it is obvious these impurities must be blown off or they will form scale on the internal surface of the boiler and the external surface of the tubes or flues.

This scale obstructs the passage of the heat from the iron to the water, and if let get thick enough will cause the iron to rapidly burn out.

To prevent the formation of scale, two principal methods are employed, one being to purify the feed water, and the other to occasionally blow the impurities out of the boiler.

Feed-water heaters generally serve also as purifiers, and their effectiveness is increased in proportion as the water can pass quietly through them, and has a large area on which the impurities can settle. Horizontal heaters have the advantage that they have a large settling area, and a less distance for the impurities to fall through. The water-gauge glass and the lower gauge cock are usually set so as to have a margin of about three inches of water above the tubes or crown sheet of the fire box, hence if it is known that the water is but just below the bottom of the gauge glass or gauge cock, there is no positive danger, although it is improper to let it get so low.

If the water is out of sight, and it is not known exactly how low it is, then it is dangerously low, and every minute is of vital importance.

Should the water get dangerously low in the boiler, the most dangerous thing to do is to lift the safety valve or pump in cold water, especially if it is not known how much water there is in the boiler.

As quickly as possible cover the fire with ashes, coal, earth, sand, or anything that is at hand that will smother the fire, then close the draught to the fire, leaving the fire door and the chimney damper open.

Leave all the steam outlets just as they are, and also the feed.

PRIMING.—Priming, which is also called "foaming," is that the steam carries up water into the steam space. This may arise from several causes, but it is well known that what will stop priming in some cases will cause it in others.

The known causes of priming are—first, too little room for the steam in the boiler, and it follows that a high water level may cause priming; second, it may be caused by a difference of temperature between the water and the steam in the boiler. Suppose, for example, that the pressure of the steam and water in the boiler is 160 lbs. by gauge, and its sensible temperature will be 370 degrees. Suppose then that enough steam is permitted to escape from the boiler to reduce the steam pressure to 140 lbs., and its temperature will be reduced to 361 degrees. But the water will remain at 270 degrees, and the result will be that it will pass into steam so rapidly that it will carry up the water and hold it in suspension among the steam. The water will pass with the steam into the engine cylinder, and the boiler will be said to "prime," "foam," or "work water." The same thing may happen if the water is heated very rapidly.

Priming is wasteful because it rapidly empties the boiler of its water, and dangerous because it may cause the piston to knock out the cylinder head or cover.

When the safety valve blows off, priming may be induced, especially if the engine is at work, because in this case the boiler is being forced, or, in other words, is making steam more rapidly than it is designed to do, and the passage of so large a body of steam through the water is apt to lift it.

Muddy water will sometimes cause foaming or priming, as will also insufficient circulation of the water in the boiler or sometimes the presence of grease or oil.

Priming may be detected from the discharge of water with the steam when the gauge cock is opened, the steam looking white and fluttering as it escapes, and also by violent motion of the water in the gauge glass, or by a thump or pound at the ends of the piston stroke.

To stop priming, the steam from the boiler should be decreased by slackening the speed of the engine, or if necessary, by stopping it. The true water level can then be seen, and if there is too much water in the boiler some of it may be blown off, while if the quantity of water in the boiler will permit it, the feed may be put on.

If the boiler has a surface blow-off cock, or a mechanical boiler cleaner, it is best to blow off from that, as it carries off the scum at the same time as relieving the boiler.

To prevent priming, a steady and uniform rate of boiler feed, the use of pure water, a clean boiler, and steady firing are the best means, turning on the steam slowly so as not to violently disturb the water in the boiler.

The engine as well as the boiler requires attention when the boiler primes. Thus the cylinder cocks should be opened to let out the water from the cylinder and prevent breakage of the cylinder cover.

SCALE IN BOILERS.—The steam leaves behind it all the impurities that the water contained, and these impurities deposit in form of mud and scale, which must be got rid of because it causes a loss of fuel, and if allowed to get thick enough will cause the boiler to burn.

The use of boiler compounds or scale preventatives may be resorted to with advantage, providing they are of a nature to suit the water, but mechanical cleaning must also be resorted to at periods determined by the nature of water.

Boilers are cleaned in two ways—first, by blowing off the impurities before they have formed into scale; and second, by removing at certain intervals whatever scale has formed.

Blowing down may be done in two ways—first, from the surface of the water by means of mechanical cleaners; and second, by blowing out from the bottom of the boiler.

The first draws off the impurities as they are thrown to the surface, the second draws them off after they have become more condensed and sink to the bottom.

How often a boiler should be blown down depends upon the kind of water fed to the boiler; where purifiers are used, less blowing down is obviously needed.

It is best to blow off from the bottom of the boiler when no steam is being used, as during dinner time, letting the water blow down about a quarter of the glass, or from the upper to the middle gauge cock.

As no steam is being used, the feed can then be put on to restore the quantity of water without reducing the temperature of the boiler so much. The feed should be gradual and the fire regulated to keep the steam pressure even.

How often a boiler should be washed out and cleaned depends upon the quality of the water it uses, and varies from about once a week to once a month, according to whether bad and unpurified water or purified water is used.

The first thing to do is to draw the fire, leaving the chimney damper open and closing all the other dampers so that as little cold air as possible can get into the boiler, while the heat can pass away up the chimney.

Let the steam and water all remain in the boiler until there is a gauge pressure of about 5 lbs. in the boiler.

Then open the blow-off cock and let out the water. If the water is blown off under a high pressure, then after the waste is all out the iron is hot enough to dry up the scale, making it hard and very difficult to remove.

After all the water is blown off, take out all the mud plugs and the man-hole and hand-hole covers, and wash out the boiler under as much water pressure as can be had, directing the hose so to reach all parts of the boiler and tubes, and continuing the washing until the water leaves the boiler clean.

Then with a wooden hoe on a piece of gas-pipe of small diameter for a handle, and small enough to pass through the hand-hole, draw all the loose scale to the hand-hole and remove it, letting the water run slowly, so as to carry the small pieces of scale towards the hand-hole as fast as the hoe disturbs it.

Then get inside the boiler, and a few blows with a light ball-pened hammer will loosen the scale, and a steel scraper will remove more, which must be washed down and drawn out with a hoe.

After the cleaning and scaling are complete, the engineer, with lamp in hand, should carefully examine the interior of the boiler and of the fire box, paying especial attention to the stays to see that they are not broken.

The hammer test should also be applied. It consists of sounding the boiler by light blows given by a light ball-pened hand hammer, the sound indicating defective places.