THE POWER HANDBOOKS

BOILERS

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COMPILED AND WRITTEN

BY

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1908

HILL PUBLISHING COMPANY

505 PEARL STREET, NEW YORK

6 BOUVERIE STREET, LONDON, E.C.

American Machinist — Power — The Engineering and Mining Journal

Hill Publishing Company, New York, U.S.A.

INTRODUCTION

This volume endeavors to furnish the reader with much new and valuable material on an old subject, together with much standard information which every engineer likes to have at his hand. A glance at the chapter headings will show the scope of the book. It will be seen that the subject is pretty fully covered from the working conditions inside of a boiler to simple talks on the various phases of boiler practice. It also covers the design of boiler furnaces for wood burning, and much other useful material.

One very important feature is the portion on the safety valve based on Mr. Fred R. Low's supplement to Power on that subject. The author is indebted to Mr. Low for permission to incorporate this material in the book, and to various other contributors, whose articles have been used as a whole or in part in the work.

Hubert E. Collins.

New York, *November*, 1908.

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I

WATCHING A BOILER AT WORK[1]

If we take a test-tube filled with water nearly to the top and hold it over a Bunsen flame, the water boils violently and overflows the tube. This violent over-boiling is due to the conflicting action of the ascending and descending currents of steam and water in the tube. On the other hand, if we take a tube shaped like a U, the arms of which are connected together at the top, fill it with water and place one leg of the U in the flame, a direct circulation soon commences. The water passes along in one direction and the steam is liberated at the surface. In this case there is very little violent ebullition, because there are no counter currents and the steam is discharged quietly over a liberal surface.

Desiring to ascertain just how nearly a boiler could be designed to work upon the U-tube principle of circulation, after several trials the model boiler shown in Fig. 1 was produced.

This model was built entirely of brass. It contained three drums four inches in diameter and 120 brass tubes one-quarter of an inch in diameter. The tubes were connected into headers and into the circum-

[1] Contributed to Power by C. Hill Smith.

ferences of the drums. The heads of all three drums were made of plate glass, for observation of the interior of the boiler when making steam. It will be noted that the design of the boiler closely resembles

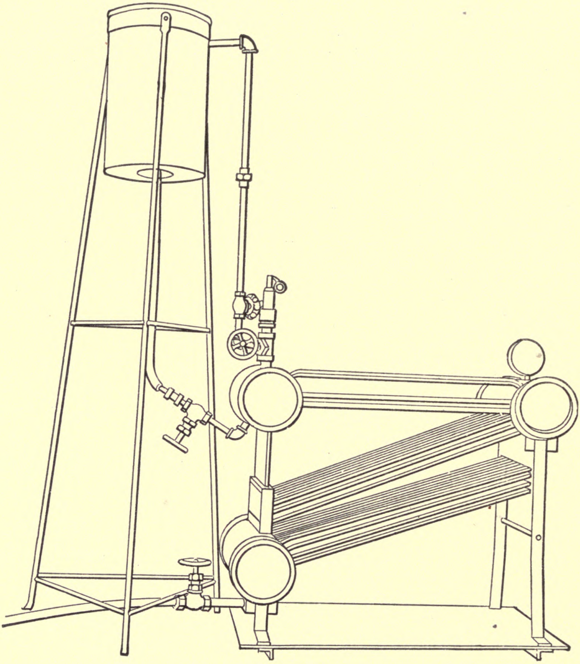

FIG. 1.

the U shape, only that one leg is considerably longer than the other, and there are two legs on the side of the U where the heat is applied.

Each of the three drums serves a special function which will be noted from the description of the experiments. The two legs, instead of being connected together at the top, as was the case in the U-tube, are connected by two separate passages, one for the water to pass through and the other for the steam.

In preparing for the tests the boiler was mounted on a stand, so that the tubes inclined from the horizontal 20 degrees, and the whole was enclosed on all sides by brass plates. Alcohol lamps were placed inside the casing at a point to correspond with the regular location of grates, or at about one-fourth of the distance between the front headers and the rear drum. The steam outlet was located on the rear drum, as was the safety valve, for experimental reasons, although in actual practice the safety valve would be located on the front drum. The feed-pipe was introduced in the rear drum, while the blow-off entered the lowest point of the lower drum, which we will call the mud-drum. The boiler was attached to an open condenser.

The boiler being ready for test, it was filled with cold water until the upper drums were filled to one-half their volume. Candles were placed behind one head of each of the three drums for the purpose of lighting the inside. The alcohol lamps were then lighted and the boiler interior was ready to observe through the glass heads of the drums.

The first action noted was in the front drum, which served as a discharge chamber for all the steaming tubes. The tubes of the lower bank discharged into it through headers, while those of the upper bank discharged into it independently. Many faint, oily-white streamers were seen to rise from the nipples connecting the headers to the front drum, passing upward to the surface of the drum. They resembled little streamers of white smoke. On reaching the surface of the water they passed into the horizontal circulating tubes which connected the two upper drums. These little streamers were heated water, which, being lighter than the water in the drum, rose to the surface. This same action soon appeared from the ends of the upper bank of tubes, the little streamers rising in a similar manner and passing into the horizontal tubes.

By observing the rear drum, the little streamers could be seen coming into this drum from the front drum. Here they turned downward into the vertical tubes which connected the rear drum to the rear headers and the mud-drum. No action could be noted in the mud-drum, which fact seemed to indicate that these currents of water passed into the upper tubes and thence into the front drum again, as the action from these tubes appeared very much more decided than the action from the nipples, notwithstanding the fact that the lower tubes were nearer to the flame than the upper ones. This was undoubtedly due to the fact that the heated currents of water remained as near the surface as possible, while the colder water passed to the bottom of the boiler, having greater specific gravity.

Particles of sediment could be seen coming down the vertical circulating tubes into the mud-drum, evidently precipitated from the water that was being heated. This sediment passed to the bottom of the drum, where it remained. A very gradual action was now noted in the mud-drum in the nature of similar currents of water coming down the vertical tubes. These currents acted strangely on entering the drum; they spread out on coming in contact with the colder and denser water lying at the bottom. By placing the finger on the upper portion of the glass head and then on the lower, quite a difference in temperature was noted. Little streamers of heated water soon commenced to pass into the lower bank of streaming tubes which were connected into this drum. They passed across the drum with a sort of shivering motion. A new and very interesting phenomenon was now noticed. Occasionally there would appear from the ends of the steaming tubes little rings of heated water, which shot across the drum with considerable velocity.

The action in the front drum became very much more pronounced and air bubbles appeared from the nipples and tubes. The boiler was circulating water with great rapidity in the same direction and it was noticed, by placing the hand on different parts of the boiler, that all parts were of the same temperature. The air bubbles now discharged in great quantities from the tubes and nipples and rising to the surface disturbed the water level considerably. It was noted that they floated along under the surface of the water before they broke.

Gradually these air bubbles ceased to appear and a new kind took their places. The latter were steam bubbles and they discharged into the drum with greater velocity than the former. On reaching the surface of the water they broke immediately, but they agitated the water level to a much greater extent. Fountains of water would shoot up into the drum for quite a distance and showed very vividly the conditions present in the shell type of boiler, where there are no defined paths for the water and steam to travel and nothing to prevent their conflict with each other. This also shows the cause for wet steam, and the great danger of entraining water with steam, as is the case where the steam is removed from the same place where violent ebullition is present.

While the water level in the front drum was violently agitated, the water level in the rear drum remained perfectly calm. No steam was generated in this drum, as the horizontal tubes connecting it to the front drum only circulated water that had been freed of its steam.

As the steam gage soon registered a pressure of 9 pounds, the main stop-valve was opened to allow the steam to flow to the condenser. The abrupt release of pressure caused the water to expand suddenly and the water level rose about one-quarter of an inch. This was evidently due to the sudden generation of steam caused by the drop in the pressure. This increased ebullition caused a very violent action in the front drum and the circulation of water through the boiler increased greatly in velocity. The nipples and tubes in the front drum discharged great quantities of bubbles. The water level in the rear drum during this increase in ebullition showed but a few ripples, which were evidently due to the vibration of the steam passing into the steam main, or the discharge into the water of the open condenser.

The sudden generation of steam caused by the opening of the steam valve and subsequent reduction in pressure, it is believed, explains how the partial rupture of the shell of a return-tubular boiler is advanced to a disastrous explosion by the unexpected increased generation of steam due to the lowered pressure.

The steam main was now closed sufficiently to allow the boiler to operate on a constant pressure of about 6 pounds. It operated very smoothly under these conditions and made a very interesting sight with the steam generating in the front drum, where the nipples and tubes discharged great quantities of bubbles.

The action in the mud-drum had in the meantime become well worth watching. In the other two drums the water showed clear in the candle light, but the color of the water in the mud-drum was very murky. Particles of sediment were noted settling to the bottom. The withdrawal of water from this drum by the steaming tubes did not appear to draw this sediment into the tubes, as the drum was of ample size so that the suction was not felt at the bottom where the sediment deposited. This emphasized clearly the advantage of a very large mud-drum to allow of the thorough settling of the sediment.

The condition of the front drum was thought to be too violent for good practice, because the ebullition indicated restriction of circulation. The boiler was put out of operation for the purpose of making changes in this drum to prevent extreme ebullition. The glass heads were removed and other nipples inserted over the nipples that connected the headers into the drum, it being here that the most violent discharge of steam was discernible. These new nipples were cut long enough to reach to the water level, or just a trifle below it.

The glass heads were replaced and the boiler put in operation again. The circulation was similar to that in the first test, and no real difference was noted until the boiler commenced to make steam. Then it was seen that the ebullition in this drum was considerably reduced, the agitation that remained being caused by the discharge from the tubes of the upper bank. This reduction was evidently due to having provided a channel through which the water and steam from the nipples might flow to the surface of the water and so prevent contact with the water in the drum. As the steam and water no longer had to force their way to the surface, the disturbance of the water level was naturally reduced entirely in this direction. The water rose from the nipples in little fountains, the steam disengaging from it in the upper part of the drum.

The boiler was operated under very severe conditions to try the value of this addition of nipples. The main stop-valve was suddenly opened after a considerable steam pressure was obtained. It had very little effect on the water level in this drum, only causing the nipples to discharge fountains of water quite a distance into the drum. No water was thrown into the superheating tubes, as the fountains of water discharged vertically and fell back immediately to the water level.

The value of this attachment being proved, the boiler was blown down, and after the water was all withdrawn from the boiler considerable sediment was found in the bottom of the lower or mud-drum. Nowhere else was sediment found, as the drums offered no opportunities for the sediment to settle, being pierced at their lowest points by tubes and nipples. The tubes were inclined 20 degrees, which insured thorough draining of the boiler.

From the foregoing experiments many points of great value for improvement in design of water-tube boilers can be derived. The violent ebullition in the front drum shows conclusively that steam should not be withdrawn from the boiler at a point where ebullition is present, on account of the danger of getting water entrained with the steam. It also shows that any sudden reduction of the pressure causes violent ebullition and priming. The front-drum conditions show that this is a good place to locate the safety valve, as the sudden opening of it would cause no liability of priming if the steam is not withdrawn from this drum.

The total lack of any ebullition in the rear drum shows that this is an ideal spot to remove the steam. It was noted that, owing to the large amount of separating surface provided, the opening of the steam valve caused no priming in this drum. Another feature to be noted is the value of a large mud-drum to provide ample opportunity for the sediment to settle, and also to provide a large supply of water for the bottom tubes. It would be impossible to force the boiler hard enough to drain this drum of water, so the danger of burning out these tubes is eliminated.

The provision of the long nipples in the front drum proved the advantage of providing separate passages to allow the steam and water to reach the surface of the water, thus obviating the necessity of their forcing their way to the surface through the large body of water in this drum and so cause violent ebullition.

II

SIMPLE TALK ON EFFICIENCY OF RIVETED JOINTS

MATTER is conceived to be composed of myriads of tiny molecules separated from each other by distances which are very considerable as compared with their diameters, and held in fixed relation to each other in solid bodies, by such an attraction as holds the earth to the sun or the moon to the earth. When we tear a piece of boiler sheet apart it is the attraction of these molecules which we are overcoming, and if the metal is uniform the force required to separate it will depend upon the surface which we expose. It will take twice as much force to pull the larger of the two bars in Fig. 2 apart as it will the smaller, because there is twice as much surface exposed at B as at A, and the attraction of twice as many molecules to overcome.

The force tending to pull a body apart in this way is called a "tensile" force, and the resistance to the force necessary to pull a piece apart is called its "ultimate tensile strength." This is usually given in pounds per square inch, and is for boiler iron around 45,000 and for boiler steel around 60,000 pounds. It should be found stamped on the sheets of which boilers are made. Suppose we have a single riveted joint like Fig. 3. We

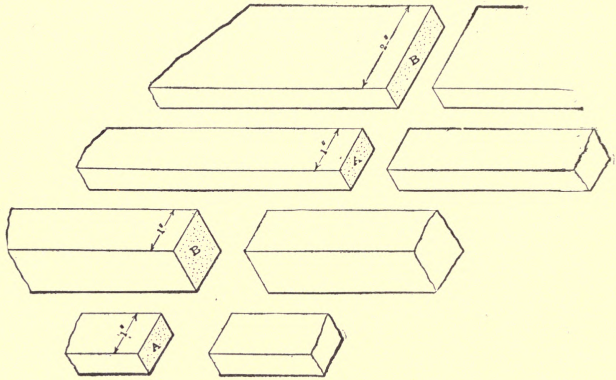

FIG. 2.

can divide it into strips as by the dotted lines half-way between the rivets, and consider one of these strips, for since they are all alike, what is true of one will be true of all. The width of each strip will be the same as

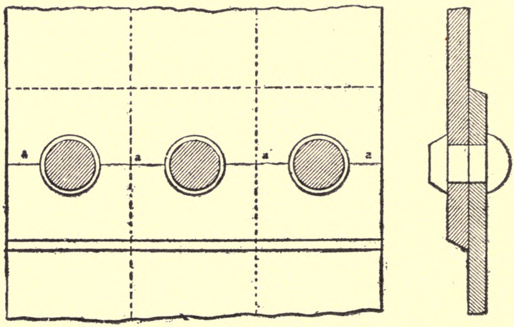

FIG. 3.

the distance from center to center of the rivets. This is called the "pitch." Let us suppose the pitch to be $2\frac{1}{4}$ inches, the diameter of the rivet 1 inch, the thickness of the plate $\frac{1}{2}$ inch, the tensile strength of the plate 60,000 and the shearing strength of the rivets 43,000 pounds.

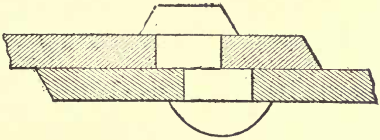

FIG. 4.

There are two ways in which this joint can fail: by tearing the sheet apart where there is the least of it to break, as at *a a a a*, Fig. 3, or by shearing the rivet as in Fig. 4.

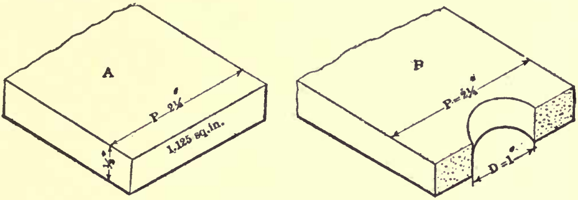

FIG. 5.

If the strip were whole as at *A* in Fig. 5, it would have

$$2\tfrac{1}{4} \times \tfrac{1}{2} = 1.125 \text{ square inches}$$

of section, and since it takes 60,000 pounds to pull one square inch apart it would take

$$1.125 \times 60{,}000 = 67{,}500 \text{ pounds}$$

to separate it.

But 1 inch of the sheet has been cut out for the rivet, so that there are left only

$$2\tfrac{1}{4} - 1 = 1\tfrac{1}{4} \text{ inches}$$

of width to be separated, and

$$1\tfrac{1}{4} \times \tfrac{1}{2} = 0.625 \text{ square inch}$$

of area. This would stand a pull of only

$$0.625 \times 60{,}000 = 37{,}500 \text{ pounds.}$$

Whether the joint will part by tearing the sheet or shearing the rivet depends, of course, on which is the stronger. The rivet has

$$1 \times 1 \times 0.7854 = 0.7854 \text{ square inch}$$

of area, and it takes 49,000 pounds to shear each square inch, so that it would take a pull of

$$0.7854 \times 49{,}000 = 38{,}484.6 \text{ pounds.}$$

to shear the rivet.

It is evident that the rivets would go, then, long before the plate, and that the strength of the joint would be

$$38{,}484.6 \div 67{,}500 = 0.57$$

or 57 per cent. of the strength of the full plate.

But we can add to the rivet strength without reducing the plate strength by putting in another row of rivets behind the first row. In Fig. 6 two rivets have to be sheared, doubling the rivet strength without reducing the plate strength, for the holes for these extra rivets do not reduce the plate section along any one line if there is space enough between the rows. In Fig. 7 the sheet is no more apt to part along the line *a a a a* than it would be if the second row of rivets were not there, and no more likely to part on the line *b b b b* than on the other. Any strip of a width equal to the pitch will

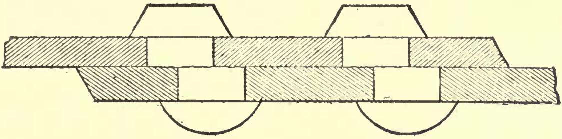

FIG. 6.

contain two rivets, whether we take it through the rivet centers, as at *A*, Fig. 7, or at equal distance to either side of one rivet in each row, as at *B* in the same figure. In the first case it includes one full rivet and two halves, and in the latter case two full rivets.

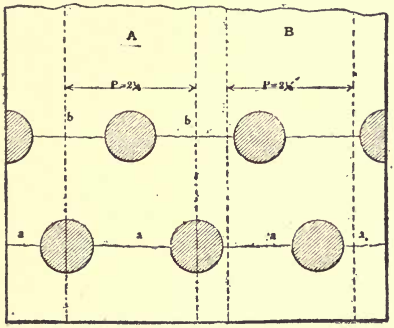

FIG. 7.

To find the efficiency of this joint, then, we calculate the efficiencies of the plate and use the lowest efficiency.

To calculate the plate efficiency, divide the difference *between the pitch and the diameter of the rivets by the pitch.*

This is simpler than the operation which we went through above, which was

$$\frac{(\text{pitch-diam.}) \times \text{thickness} \times \text{tensile strength}}{\text{pitch} \times \text{thickness} \times \text{tensile strength}}$$

the numerator being the pull required to separate the sheet with the rivet holes cut out, and the denominator the pull required to separate the full sheet. As the thickness and tensile strength appear in both numerator and denominator, they cancel out.

To find the rivet efficiency, multiply the diameter of the rivet by itself, by 0.7854, *by the shearing strength per square inch and by the number of rows, and divide by the product of the pitch, thickness and tensile strength per square inch of section.*

These rules are applicable only to lap joints where the rivets are in single shear.

III

SIMPLE TALKS ON THE BURSTING STRENGTH OF BOILERS

THERE are two ways that a shell, such as is shown in the sketches herewith, might break under internal pressure. The sheets might tear lengthwise, letting the shell separate, as in Fig. 8, or they might tear across, letting it separate endwise, as in Fig. 9.

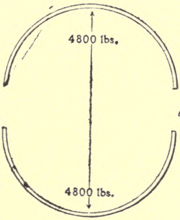

FIG. 8.

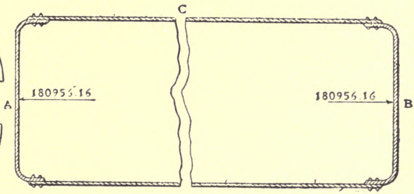

FIG. 9.

Which is it the more likely to do?

To push it apart endwise, as in Fig. 9, we have the force acting on the heads. This force is the pressure per square inch multiplied by the number of square inches in the head. The area of a circle is the diameter multiplied by itself and by 3.1416 and divided by 4; or since 3.1416 divided by 4 is .7854, the area is the square of the diameter multiplied by .7854.

Suppose the internal diameter of the shell to be 48 inches, and the pressure 100 pounds per square inch, the pressure on each head would be

$$48 \times 48 \times .7854 \times 100 = 180{,}956.16 \text{ pounds},$$

or over 90 tons. This pressure would act on each head, and the effect would be the same as though two weights of 180,956.16 pounds each were pulling against each other through the boiler, as in Fig. 10.

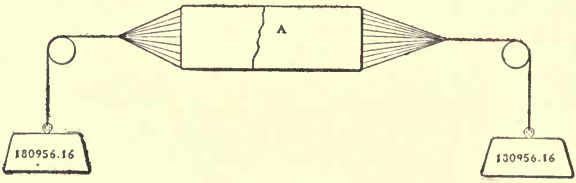

FIG. 10.

If the shell were not heavy enough to stand the strain, it would tear apart along the line where the metal happened to be the weakest, as at *A*. At first sight it looks as though the metal had to sustain both these forces or weights, and that the stress upon the shell would be twice 180,956.16 pounds; but a little consideration will show that this is not so. One simply furnishes the equal and opposite action with which every force must be resisted. A man pulling against a boy on a rope (Fig. 11) can pull no harder than the boy pulls against him. If he does he will pull the boy off his feet, and the strain on the rope will be only what one of them pulls, not the sum of both pulls. In order that the man may pull with a force of 50 pounds, the boy must hold against him with a force of 50 pounds. Both are pulling with a force of 50

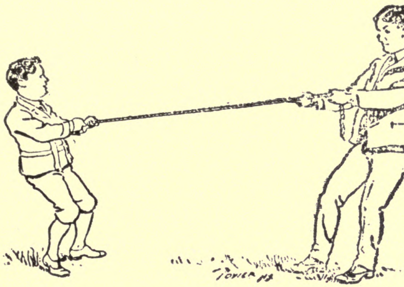

FIG. 11.

pounds, but the tension on the rope is 50 pounds, not 100. The boy might be replaced with a post (Fig. 12). Now, when the man pulls with a force of 50 pounds

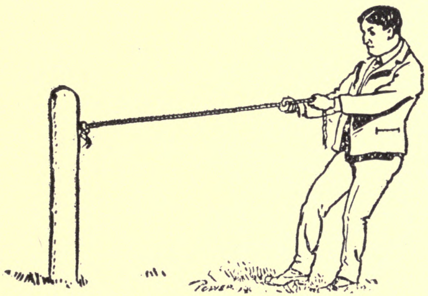

FIG. 12.

against the post, you would not say that there was 100 pounds tension on the rope; yet the post is pulling or holding against him with a force of 50 pounds, just as the boy did. In Fig. 13 it is easily seen that the tension on the cord is 50 pounds. You would not say that it was 100, if the pull of the weight were resisted by another weight of 50 pounds, as in Fig. 14, instead of by the floor.

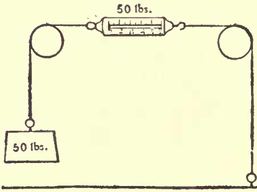

FIG. 13.

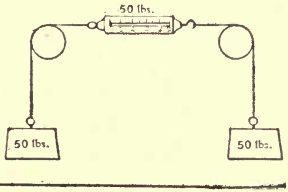

FIG. 14.

The shell is therefore in the case which we have imagined subjected to a force of 180,956.16 pounds, which tends to pull it apart endwise, as in Fig. 10.

To resist this there are as many running inches of shell as there are inches in the circumference.

The circumference is 3.1416 times the diameter, so that to pull the boiler in two

$$48 \times 3.1416 = 150.7968 \text{ inches}$$

of sheet would have to be pulled apart.

The force exerted upon each running inch of sheet would be the pressure acting endwise divided by the circumference, or

$$180{,}956.16 \div 150.7968 = 1{,}200 \text{ pounds.}$$

The area is

$$\frac{\text{diam.} \times \text{diam.} \times 3.1416}{4}$$

The circumference is

$$\text{Diam.} \times 3.1416.$$

Dividing the area by the circumference we have

$$\frac{\text{diam.} \times \text{diam.} \times 3.1416}{4 \times \text{diam.} \times 3.1416} = \frac{\text{diam.}}{4}$$

or the strain on each running inch of sheet per pound of pressure is one-fourth the diameter.

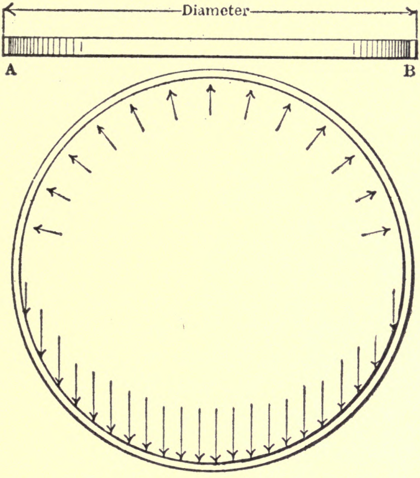

FIG. 15.

Now let us see what it would be in the other direction.

If we consider the pressure acting in all directions as in the upper half of Fig. 15, we should, to get the total pressure on the area, have to multiply the pressure per square inch by the whole area, which would be the circumference for a strip 1 inch wide; but if we are considering the effect of pressure in one direction only, we must consider only the area in that direction. If we are studying the effect of the pressure in forcing the shell in the direction of the arrows in the lower half of Fig. 15, we must consider only the area which comes crosswise to that direction, the "projected area," as it is called; the area which the piece would present if we were to hold it up and look at it in the direction of the arrows or of the shadow which it would cast in rays of light running in the direction of the pressure. This, it will be easily recog-

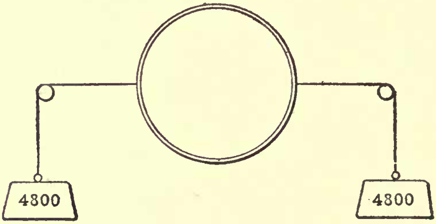

FIG. 16.

nized, is the diameter of the boiler wide and 1 inch high, as shown in Fig. 15, so that the number of square inches upon which the pressure is effective in one direction is equal to the diameter for a strip 1 inch wide. There is therefore a force tending to pull each 1-inch ring of the shell apart, as in Fig. 16, of $48 \times 100 = 4800$ pounds, and as this force is resisted by two running inches of metal, one at A and one at B (Fig. 15), the stress per inch will be 4800 ÷ 2 = 2400 pounds. This is just twice what we found it to be in the other direction; and it is plain that this should be so, for the stress per pound of pressure tending to burst the boiler, as in Fig. 8, is, as we have just seen,

$$\frac{\text{diam.}}{2}$$

which is just twice the $\frac{\text{diam.}}{4}$ which we found it to be in the other direction. It is for this reason that boilers are double riveted along the side or longitudinal seams, while single riveting is good enough for girth seams.

IV

SIMPLE TALKS ON THE BURSTING STRENGTH OF BOILERS

IN the preceding chapter we found that a cylinder equally strong all over, will split lengthwise with one-half the pressure which would be needed to tear it apart endwise.

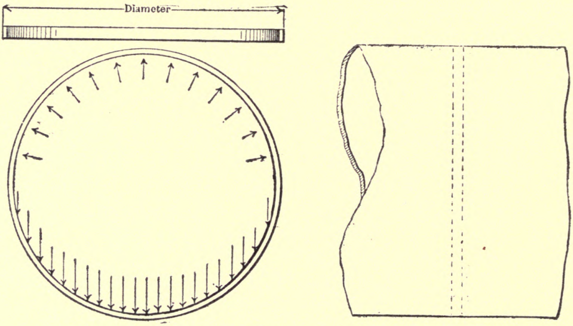

FIG. 17.

Let us see how much pressure it would take to burst a shell of this kind. We will consider a strip 1 inch in width, as in Fig. 17, for the action upon all the similar strips into which the boiler can be imagined to be spaced off will be the same. We see that the pressure tending to pull the ring, 1 inch in width, apart is equal to the pressure per square inch multiplied by the diameter of the ring. The total pressure *in all directions*, acts on the circumference as shown by the radial arrows at the upper portion of the cut, but when we come to consider the force acting in *one* direction we must take the projected area in that direction; the area of the shadow, as explained before, cast by rays of light flowing in that direction, and that area would be that of the strip as we see it at the top of Fig. 17, 1 inch wide and the diameter of the boiler long.

It is sometimes hard for one to see why the diameter is used here, instead of the circumference, and a further illustration is here given.

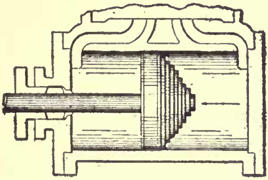

FIG. 18.

Suppose you had a piston in an engine cylinder made in steps like Fig. 18. This would have a good deal more surface to rust or to condense steam than would a flat piston, but it would have no more effective area for the production of power, would it? One hundred pounds behind it in the cylinder would push no harder on the crosshead with this than with a perfectly flat piston; because the sidewise pressure against the steps is balanced by an equal pressure from the opposite side; only the pressure on the flat rings effective to move the piston forward, and the area of all these rings added together, is just the same as that of a flat surface of the same external diameter, as seen by the projection at the right.

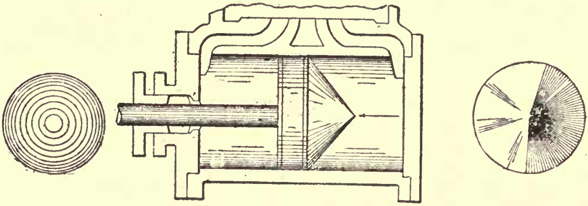

FIG. 19.

This would be just as true if the steps were a millionth or a hundred-millionth of an inch wide and high instead of an inch or more, so that it is just as true of a conical surface, like Fig. 19, as of Fig. 18, or of a concave

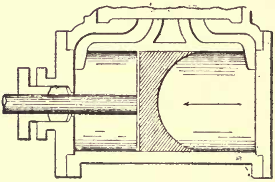

FIG. 20.

surface, like Fig. 20, as of either; and it is evident that it is the flattened-out area which one sees in looking at the object in the line of the force considered, the projected area in that direction as it is called, and not the real superficial area which is effective.

We have then a force equal to the pressure per square inch multiplied by the internal diameter of the shell tending to pull each inch in length of it apart, and we have *two* sections, *A* and *B*, Fig. 15, where the sheet must part.

The force tending to tear *each* of these is the pressure per square inch multiplied by the *radius*, or half the diameter of the shell. The resistance that the piece of shell will offer to being torn apart is the tensile strength per square inch multiplied by the number of square inches to be torn apart.

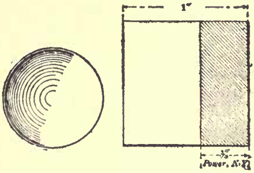

FIG. 21.

This area is one inch long and the thickness of the sheet in width. The area in square inches is therefore the same as the thickness in inches. If the plate were $\frac{3}{8}$ of an inch thick, for example, its section per inch of length would be $\frac{3}{8}$ of a square inch, as shown in Fig. 21.

The two opposing forces, which must be equal, not only at the point of fracture, but at all times, are:

Pressure per square inch $\times$ radius, and pull per square inch $\times$ thickness.

The pull on the sheet is called the tensile force. If we want to find the tensile force on the sheet for any pressure per square inch, we multiply that pressure by the radius and divide by the thickness of the sheet in inches.

If we want to find the pressure per square inch necessary to get up a given tensile force per square inch, we multiply the given pull per square inch by the thickness of the plate and divide by the radius in inches.

We can find the pressure per square inch necessary to rupture the sheet by multiplying the ultimate tensile strength, that is, the tensile force required to pull a square inch of it apart by the thickness and dividing by the radius.

Example. — What pressure would be required to burst a tank 48 inches in diameter, made of steel $\frac{1}{4}$ of an inch in thickness, having a uniform tensile strength of 60,000 pounds per square inch?

$$\frac{\text{Tensile strength} \times \text{thickness}}{\text{radius}} = \text{pressure.}$$

$$\frac{60{,}000 \times .25}{24} = 625 \text{ lbs. per sq. in.}$$

But we cannot or do not in boiler practice get a shell of uniform strength. There have to be joints and these joints are not so strong as the plate itself. We will have a talk later about how to figure the strength of a riveted joint. Suppose the riveted joint was only 70 per cent. of the plate strength, then it would take only 70 per cent. of the force to pull it apart, and the result just found must be multiplied by .70 if the tank, instead of having a "uniform tensile strength of 60,000," has a sheet strength of 60,000 and a longitudinal seam of 70 per cent. efficiency.

The complete operation of finding the bursting strength of a boiler shell is

$$\frac{\text{Tensile strength} \times \text{thickness} \times \text{efficiency of joint}}{\text{radius}}.$$

RULE. — *Multiply the tensile strength of the weakest sheet in pounds per square inch by the least thickness in inches and by the efficiency of the longitudinal riveted joint, and divide by the inside radius of the shell in inches. The result is the pressure per square inch at which the shell should split longitudinally.*

The safe working pressure is found by dividing the above by the desired "factor of safety," usually from 3.5 to 5.

This, it must be noted, is the pressure at which the shell should fail in the manner described. The boiler may be weaker somewhere else, as upon some of the stayed surfaces, so that all these points should be considered before the allowable pressure is fixed upon.

V

SIMPLE TALKS ON THE BRACING OF HORIZONTAL RETURN TUBULAR BOILERS

In former chapters we have discussed the strength of a boiler so far as the parting of the shell is concerned, but even if the shell is heavy enough and the joint well proportioned the boiler may be weak in other respects.

The head of a 60-inch boiler has an area of

$$60 \times 60 \times 0.7854 = 2827 \text{ square inches.}$$

At 100 pounds per square inch there would be a pressure against the head of

$$2827 \times 100 = 282700 \text{ pounds.}$$

or over 140 tons.

Besides its tendency to pull the shell apart endwise, this pressure tends to bulge the heads, as shown in Fig. 22. In the case of a tank, or of the drums of water-tube boilers where there are no tubes in the heads, they can be made safe against change of shape under pressure by giving them in the first place the shape that the pressure tends to force them into; but the tube sheet of a horizontal tubular boiler, for instance, must be flat to allow the tubes to enter square with its surface. The tubes themselves act as stays to the lower part, but the pressure on the part above the tubes tends to bulge the head and might cause the central tubes to pull out.

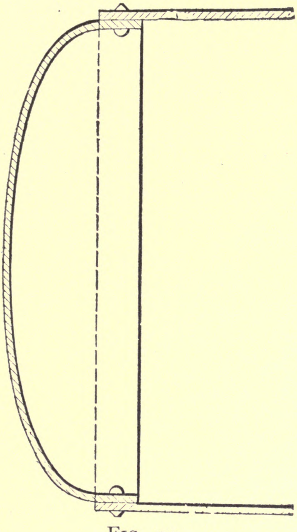

FIG. 22.

This is prevented by bracing the unsupported part of the head either by "through braces," as in Fig. 23, or by "diagonal braces," as in Fig. 24.

In order to find how many braces are required, or if a given boiler is sufficiently braced, the area to be braced must be computed. This area may be taken as that included within lines drawn 2 inches above the top line of tubes and 2 inches inside of the shell,

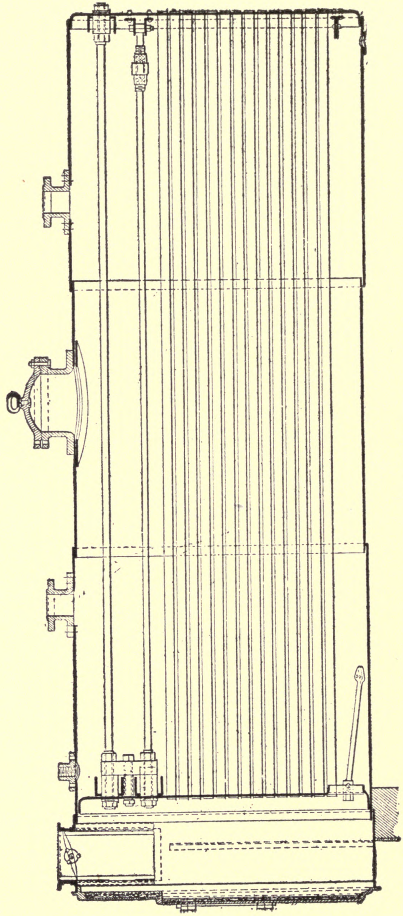

Fig. 23.

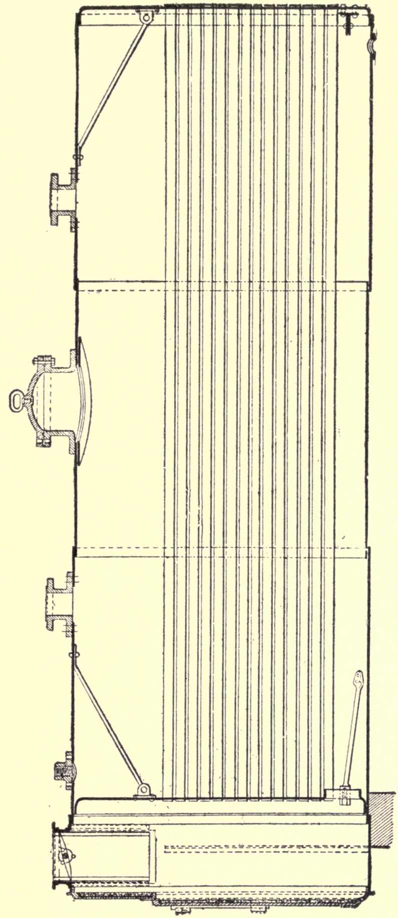

FIG. 24.

as in Fig. 25, the area outside of these lines being considered to be sufficiently braced by the shell and tubes. This figure is a "segment" of a circle and its area is found by dividing its height *h* by the diameter of the circle, of which it is a part, finding the quotient in the column of "versed sines" of the accompanying table, and multiplying the segmental area as given opposite that quotient in the next column by the square of the diameter.

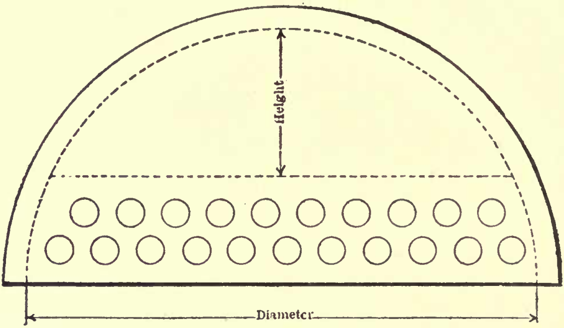

FIG. 25.

For example, suppose the hight *h* in Fig. 25 to be 18 inches and the diameter of the boiler 60 inches. The diameter of the circle of which the segment is a part is 56 inches, because we go 2 inches inside the shell on both ends of the diameter.

Following the rule we divide the height by the diameter

$$18 \div 56 = 0.3214.$$

The values in the table are given to only three places of decimals, but the division should be carried out to four places. If the last figure is less than 5, drop it off. If it is 5 and the quotient comes out even, *i.e.*, there is no remainder after the 5, drop it off, also. If the last figure is greater than 5, or in case it is 5, and the division did not come out square, drop it off, but raise the third figure one.

In the case in hand the quotient, 0.3214, comes between the 0.321 and 0.322 of the table, and is nearer the 0.321, being but $0.3214 - 0.321 = 0.0004$ off; while it is $0.322 - 0.3214 = 0.0006$ off from the higher value. If it were 0.3216, however, it would be nearer 0.322 than 0.321.

The segmental area corresponding with 0.321 in the table is 0.2176. Multiplying this by the square of the diameter gives $56 \times 56 \times 0.2176 = 682.39$ square inches as the area of the segment, and the force to be braced against is this number of square inches multiplied by the pressure per square inch.

A through-brace which pulls squarely on the plate has an effect in keeping it from bulging equal to the tensile strain in the brace itself — *i.e.*, if the brace were under a strain of 6000 pounds, it would tend to pull the head in and keep it from bulging with an equal force; but if a diagonal brace, as in Fig. 26, were under a strain of 6000 pounds, it would tend to pull the lug on the head in its own direction with that force, but would resist a force in a direction at right angles with the head of only .91 as much, or 5,460 pounds. This figure is found by dividing the length of the line *b c* by the length of the line *a b*.

In order to find if a boiler is sufficiently braced;

Find the smallest cross-section of each brace in square inches.

Multiply the cross-section of each diagonal brace by the quotient of the distance of its far end from the head in a line perpendicular to the head (*b c*, Fig. 26), divided by the length of the brace.

Add all these results together.

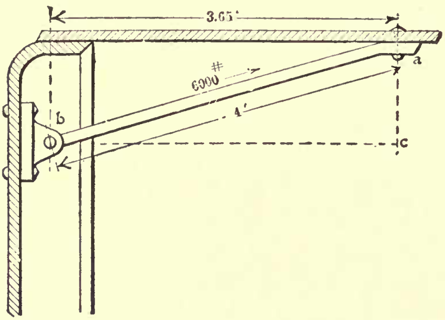

FIG. 26.

For such braces as are all alike, as for through-braces of the same diameter of cross-section, you can of course compute one and multiply it by the number of similar ones.

Divide the product of the area to be braced and the pressure per square inch by the sum of all these values, and you will have the strain on the braces per square inch of section.

The rules of the United States Board of Supervising Inspectors allow a strain of 6000 pounds per square inch on the braces. If the computed stress does not exceed this amount, the boiler is sufficiently braced.

To determine what pressure a boiler will stand, so far as its bracing is concerned, multiply the minimum cross-section of each brace by the quotient of the distance of its far end from the plate perpendicularly divided by the length of the brace. Add the results and multiply by 6000. Divide the produce by the number of square inches in the segment, and the quotient will be the pressure per square inch that the bracing is good for.

AREAS OF SEGMENTS OF CIRCLES

Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.
.1	.04087	.121	.05404	.142	.06892	.163	.08332
.101	.04148	.122	.05469	.143	.06892	.164	.08406
.102	.04208	.123	.05534	.144	.06962	.165	.0848
.103	.04269	.124	.056	.145	.07033	.166	.08554
.104	.0431	.125	.05666	.146	.07103	.167	.08629
.105	.04391	.126	.05733	.147	.07174	.168	.08704
.106	.04452	.127	.05799	.148	.07245	.169	.08779
.107	.04514	.128	.05866	.149	.07316	.17	.08853
.108	.04575	.129	.05933	.15	.07387	.171	.08929
.109	.04638	.13	.06	.151	.07459	.172	.09004
.11	.047	.131	.06067	.152	.07531	.173	.0908
.111	.04763	.132	.06135	.153	.07603	.174	.09155
.112	.04826	.133	.06203	.154	.07675	.175	.09231
.113	.04889	.134	.06271	.155	.07747	.176	.09307
.114	.04953	.135	.06339	.156	.0782	.177	.09384
.115	.05016	.136	.06407	.157	.07892	.178	.0946
.116	.0508	.137	.06476	.158	.07965	.179	.09537
.117	.05145	.138	.06545	.159	.08038	.18	.09613
.118	.05209	.139	.06614	.16	.08111	.181	.0969
.119	.05274	.14	.06683	.161	.08185	.182	.09767
.12	.05338	.141	.06753	.162	.08258	.183	.09845

AREAS OF SEGMENTS OF CIRCLES — *Continued*

Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.
.184	.09922	.216	.12481	.248	.15182	.28	.18002
.185	.1	.217	.12563	.249	.15268	.281	.18092
.186	.10077	.218	.12646	.25	.15355	.282	.18182
.187	.10155	.219	.12728	.251	.15441	.283	.18272
.188	.10233	.22	.12811	.252	.15528	.284	.18361
.189	.10312	.221	.12894	.253	.15615	.285	.18452
.19	.1039	.222	.12977	.254	.15702	.286	.18542
.191	.10468	.223	.1306	.255	.15789	.287	.18633
.192	.10547	.224	.13144	.256	.15876	.288	.18723
.193	.10626	.225	.13227	.257	.15964	.289	.18814
.194	.10705	.226	.13311	.258	.16051	.29	.18905
.195	.10784	.227	.13394	.259	.16139	.291	.18995
.196	.10864	.228	.13478	.26	.16226	.292	.19086
.197	.10943	.229	.13562	.261	.16314	.293	.19177
.198	.11023	.23	.13646	.262	.16402	.294	.19268
.199	.11102	.231	.13731	.263	.1649	.295	.1936
.2	.11182	.232	.13815	.264	.16578	.296	.19451
.201	.11262	.233	.139	.265	.16666	.297	.19542
.202	.11343	.234	.13984	.266	.16755	.298	.19634
.203	.11423	.235	.14069	.267	.16844	.299	.19725
.204	.11503	.236	.14154	.268	.16931	.3	.19817
.205	.11584	.237	.14239	.269	.1702	.301	.19908
.206	.11665	.238	.14324	.27	.17109	.302	.2
.207	.11746	.239	.14409	.271	.17197	.303	.20092
.208	.11827	.24	.14494	.272	.17287	.304	.20184
.209	.11908	.241	.1458	.273	.17376	.305	.20276
.21	.1199	.242	.14665	.274	.17465	.306	.20368
.211	.12071	.243	.14751	.275	.17554	.307	.2046
212	.12153	.244	.14837	.276	.17643	.308	.20553
213	.12235	.245	.14923	.277	.17733	.309	.20645
.214	.12317	.246	.15009	.278	.17822	.31	.20738
.215	.12399	.247	.15095	.279	.17912	.311	.2083

AREAS OF SEGMENTS OF CIRCLES — *Continued*

Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.	Versed Sine.	Segmental Area.
.312	.20923	.343	.23832	.374	.26804	.405	.29827
.313	.21015	.344	.23927	.375	.26901	.406	.29925
.314	.21108	.345	.24022	.376	.26998	.407	.30024
.315	.21201	.346	.24117	.377	.27095	.408	.30122
.316	.21294	.347	.24212	.378	.27192	.409	.3022
.317	.21387	.348	.24307	.379	.27289	.41	.30319
.318	.2148	.349	.24403	.38	.27386	.411	.30417
.319	.21573	.35	.24498	.381	.27483	.412	.30515
.32	.21667	.351	.24593	.382	.27580	.413	.30614
.321	.2176	.352	.24689	.383	.27677	.414	.30712
.322	.21853	.353	.24784	.384	.27775	.415	.30811
.323	.21947	.354	.2488	.385	.27872	.416	.30909
.324	.2204	.355	.24976	.386	.27969	.417	.31008
.325	.22134	.356	.25071	.387	.28067	.418	.31107
.326	.22228	.357	.25167	.388	.28164	.419	.31205
.327	.22321	.358	.25263	.389	.28262	.42	.31304
.328	.22415	.359	.25359	.39	.28359	.421	.31403
.329	.22509	.36	.25455	.391	.28457	.422	.31502
.33	.22603	.361	.25551	.392	.28554	.423	.316
.331	.22697	.362	.25647	.393	.28652	.424	.31699
.332	.22791	.363	.25743	.394	.2875	.425	.31798
.333	.22886	.364	.25839	.395	.28848	.426	.31897
.334	.2298	.365	.25936	.396	.28945	.427	.31996
.335	.23074	.366	.26032	.397	.29043	.428	.32095
.336	.23169	.367	.26128	.398	.29141	.429	.32194
.337	.23263	.368	.26225	.399	.29239	.43	.32293
.338	.23358	.369	.26321	.4	.29337	.431	.32391
.339	.23453	.37	.26418	.401	.29435	.432	.3249
.34	.23547	.371	.26514	.402	.29533	.433	.3259
.341	.23642	.372	.26611	.403	.29631	.434	.32689
.342	.23737	.373	.26708	.404	.29729	.435	.32788

VI

CALCULATING THE STRENGTH OF RIVETED JOINTS[1]

In calculations relative to the strength of steam boilers and vessels of a similar character for withstanding high pressures, one of the most important points to be considered is the strength of the seams where the plates are joined. This is not only important to the designer of such vessels, but also to the operating engineer, who is often required to fix the limit of pressure which should be carried on the boilers under his charge, and frequently, owing to increased output without corresponding addition to the boiler capacity, it becomes necessary to carry the pressure as high as safety will permit, and in such cases it is important for the engineer to be able to fix this safe limit.

It is the purpose in this chapter to show how the strength of the various types of joint generally used in boiler construction may be calculated, and as only simple arithmetic is required for the calculations, any reader should find no difficulty in understanding how it is done, and applying the principles to calculate the strength of the particular joints which may be of interest to them. To avoid the use of formulas, which

[1] Contributed to Power by S. F. Jeter.

are confusing to many, numerical examples will be used to illustrate the methods of making the calculations, and for the sake of uniformity the tensile strength of the sheets (which is the strength to resist being pulled apart) will be assumed as 55,000 pounds per square inch; the shearing strength of the rivets (which represents their resistance to being sheared through by the plates at right angles to their length) will be assumed as 42,000 pounds per square inch in single shear, as represented in Fig. 31, and 78,000 pounds per square inch in double shear, as represented in Fig. 34. The resistance of the rivets to crushing will be assumed at 95,000 pounds per square inch. For modern construction consisting of steel plates and steel rivets, the above values are average figures.

It is customary to express the strength of a riveted joint as a percentage of the strength of the plates which are riveted together. Thus, if the joint illustrated in Fig. 35 has an efficiency of $62\frac{1}{2}$ per cent., it would mean that any portion of its length that divides the rivet spaces symmetrically would be 0.625 times as strong as a section of the same length through the solid plate.

Possible Modes of Failure

Before proceeding to calculate a practical boiler joint, the different ways in which two pieces of plate riveted together might fail should be noted. If a piece of boiler plate, $\frac{3}{8}$ inch thick and $2\frac{1}{2}$ inches wide, is placed in the jaws of a testing machine, as illustrated in Fig. 27, and pulled apart, it would separate at some section as *A A*. If the tensile strength was 55,000 pounds per square inch, the force that would have to be applied to the jaws would be 55,000 times the area separated in square inches, which in this case is

$$2\tfrac{1}{2} \times \tfrac{3}{8} = \tfrac{15}{16} = 0.9375$$

square inch, so that the pull would be

$$55{,}000 \times 0.9375 = 51{,}562.5$$

pounds.

If another piece of plate be taken, identical in every

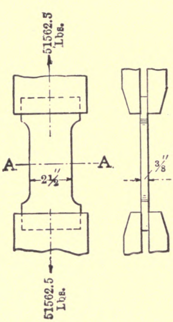

FIG. 27.

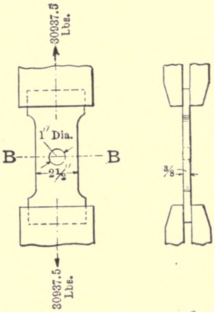

FIG. 28.

respect to the first, except that a hole 1 inch in diameter is drilled through it as illustrated in Fig. 28, and the plate be pulled apart in the testing machine as before, it is evident that it would fail along the line *B B*, as the area of the reduced section caused by drilling the hole would be only

$$(2.5 - 1) \times 0.375 = 0.5625$$

square inch, and the force necessary to pull it apart would be

$$55{,}000 \times 0.5625 = 30{,}937.5$$

pounds, the strength of the metal being the same in both instances. Now if the relation between the strength of the solid plate and the drilled plate be expressed by dividing the latter by the former, the result would be

$$\frac{30{,}937.5}{51{,}562.5} = 0.6,$$

or, in other words, the drilled plate is capable of sustaining 60 per cent. of the load that could be carried by the solid plate.

If, instead of using a single piece of plate, two plates are drilled with 1-inch holes in the ends and are joined together by a rivet, as shown in Fig. 29, and an attempt

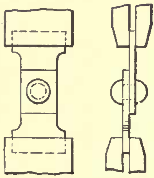

FIG. 29.

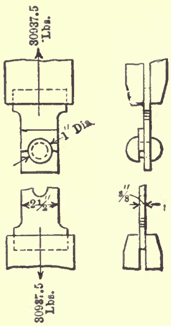

FIG. 30.

should be made to pull them apart as before, there would be four probable ways in which failure might take place, all of which are considered in the calculation and design of riveted joints. First, the section of plate each side of the rivet hole might break, leaving the ends as shown in Fig. 30. Again, the plates might shear the rivets off, as illustrated in Fig. 31. Thirdly, it has been found by practical tests of joints that steel rivets cannot be subjected to a pressure much greater than 95,000 pounds per square inch of bearing surface without materially affecting their power to resist shearing, and therefore the joint might fail, as shown in Fig. 31, due to an excess crushing stress on the rivet.

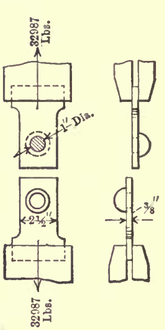

FIG. 31.

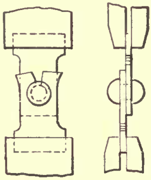

FIG. 32.

A fourth possible method of failure would be for the metal in the sheet in front of the rivet to split apart or pull out, as illustrated in Fig. 32. This latter mode of failure is erratic, and cannot be calculated, but it has been practically demonstrated in tests of joints, that if the distance from the edge of the plate to the center of the rivet hole is $1\frac{1}{2}$ times the diameter of the hole, this mode of failure is improbable, and in the following calculations of joints it will be assumed that they are properly designed to render such failure impossible.

To determine the actual strength or efficiency of such a joint as is illustrated in Fig. 29, the force required to produce rupture must be calculated for each of the first three ways mentioned, and the weakest mode of failure taken as the maximum strength of the joint.

To rupture the plate as illustrated in Fig. 30, the pull required would be the same as to rupture the drilled plate illustrated in Fig. 28, which was found to be 30,937.5 pounds. To shear the rivet off, as in Fig. 31, would require a force equal to the area to be sheared in square inches, times the shearing strength per square inch; or since the area of a 1-inch rivet is 0.7854 square inch, the force required would be

$$0.7854 \times 42,000 = 32,987$$

pounds. The pressure required to cause failure by crushing was stated to be 95,000 pounds per square inch, and in calculating the area exposed to pressure for pins and rivets, it is figured as equal to the diameter of the pin or rivet, times the thickness of the plate; therefore, we have

$$1 \times 0.375 = 0.375$$

square inch of area to withstand crushing, or

$$0.375 \times 95,000 = 35,625$$

pounds would be required to produce rupture of the joint in this manner.

From these figures it is evident that the method of failure first considered is the weakest of the three, and, therefore, determines the efficiency of the joint, which would be 60 per cent. as found for the drilled plate.

If one plate is riveted between two other plates, as illustrated in Fig. 33, the several methods of failure are calculated in the same way, except for the shearing of the rivet, which would occur as shown in Fig. 34, and

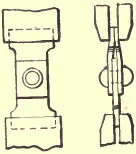

FIG. 33.

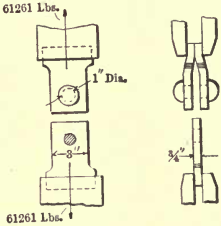

FIG. 34.

is described as double shearing. While the metal sheared in this case would be just twice as much as in single shear, it has been found by test that the force required is not exactly twice as much, but 1.85 to 1.90 times the amount in single shear; so as stated at the beginning, 78,000 pounds per square inch is assumed for rivets in double shear and 42,000 pounds per square inch when in single shear.

Calculating the strength of a joint with the dimensions as illustrated in Fig. 34, the strength of the solid plate would be

$$3 \times 0.75 \times 55{,}000 = 123{,}750$$

pounds. The strength of the center plate through the rivet hole, the failure being assumed similar to that illustrated in Fig. 30, would be

$$(3 - 1) \times 0.75 \times 55{,}000 = 82{,}500$$

pounds. The crushing strength of the rivet would be

$$1 \times 0.75 \times 95{,}000 = 71{,}250$$

pounds. The shearing strength of the rivet, or failure assumed as in Fig. 34, would be

$$0.7854 \times 78{,}000 = 61{,}261$$

pounds.

From these figures it is evident that failure would most likely occur as shown in Fig. 34, and the relative strength of the joint as compared with the solid plate is

$$\frac{61{,}261}{123{,}750} = 0.495,$$

or $49\frac{1}{2}$ per cent. As will be shown later, the foregoing simple calculations are all that are required to estimate the strength of the most complicated joints.

The Unit Section

In calculating the strength of a practical boiler joint, the strength for the entire length of a sheet could be estimated, but this would be laborious owing to the number of figures involved in the calculations, and the same result can be obtained by considering any length that divides the rivets symmetrically. For convenience, the shortest length that thus divides the rivets is the one used in such calculations, and this length is called a unit section of the joint. When the lines dividing the joint into unit sections pass through a rivet, only one-half of the rivet is considered in the calculation, and when rivets thus divided are lettered for reference, the two halves on opposite sides will be lettered the same, so that referring to the letter will indicate a whole rivet. Thus, if the rivet *A*, in Fig. 43, is spoken of, it would mean the combined halves of the two rivets on the outer row.

In measuring joints already constructed to obtain the length of a unit section, or the pitch, it should be remembered that rivet heads do not always drive fairly over the center of the rivet holes, and the rivet holes themselves are sometimes irregular distances apart; so it is more accurate to measure a number of pitches and divide the distance by the number measured to obtain the average pitch. It will be found most convenient, where space permits, to measure ten pitches, and then placing the decimal point one figure to the left will give the average unit length.

Single-riveted Lap-joint

First to be considered is the single-riveted lap-joint illustrated in Fig. 35. In a unit section of 2 inches one rivet is in single shear and $\frac{3}{4}$ inch has been cut out of the plate by the rivet hole. The calculation for strength is the same as has been made for Fig. 38, and the three methods of failure to be considered are:

(1) Breaking of the section of plate between the rivet holes, which is called the net section.

(2) Shearing of a $\frac{3}{4}$-inch rivet in single shear.

(3) Resistance of one rivet to crushing.

Using the numerical values given in Fig. 35 the following results are obtained:

(1) $(2 - 0.75) \times 0.25 \times 55{,}000 = 17{,}187.5$ pounds.
(2) $0.4418 \times 42{,}000 = 18{,}556$ pounds.
(3) $0.75 \times 0.25 \times 95{,}000 = 17{,}812.5$ pounds.

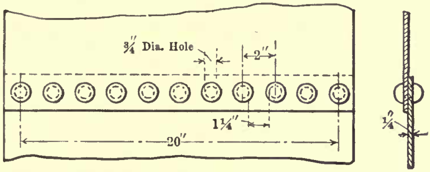

FIG. 35.

Of the three methods of failure, the first is seen to be the most probable, and since a unit section length of the solid plate would have a strength of

$$2 \times 0.25 \times 55{,}000 = 27{,}500$$

pounds, the efficiency of the joint would be

$$\frac{17{,}187.5}{27{,}500} = 62.5$$

per cent.

DOUBLE-RIVETED LAP-JOINT

Next in line is the double-riveted lap-joint illustrated in Fig. 36. There is one feature connected with this joint which should be considered before proceeding with the calculation of its strength. It would evidently be possible to have the two rows of rivets forming this joint so close together that the combined net sections between rivets *A B* and *B C* would be less than between rivets *A C*. It has been found in practical tests of joints that it is necessary to have the combined area of these two sections 30 to 35 per cent. in excess of that between rivets *A* and *C* in order to be sure that the joint will fail along line *A C*. This would

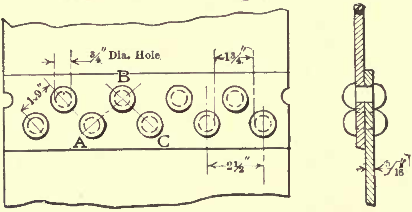

FIG. 36.

correspond to a diagonal pitch of two-thirds of the pitch from *A* to *C* plus one-third of the diameter of the rivet hole, or 1.9 inches in the joint shown in Fig. 36. Ordinarily, if the rows are much closer than this, the joint has an abnormal appearance which would be noted at once. In further calculations it will be assumed that the joints are proportioned so that this method of failure will not be possible.

Proceeding with the calculation of the strength of the joint illustrated in Fig. 36, the methods of probable failure to be calculated are the same as for the single-riveted joint:

(1) Failure of net section between rivet holes.

(2) Shearing of two rivets in single shear.

(3) Crushing strain on two rivets.

Using the values given in Fig. 36, we have for the above:

(1) $(2.5 - 0.75) \times 0.3125 \times 55{,}000 = 30{,}078$ pounds.

(2) $2 \times 0.4418 \times 42{,}000 = 37{,}112$ pounds.

(3) $2 \times 0.75 \times 3.5120 \times 95{,}000 = 44{,}531$ pounds.

It is evident that the first method of failure is the most probable, and since the strength of the solid plate is

$$2.5 \times 0.3125 \times 55{,}000 = 42{,}969$$

pounds, the efficiency of the joint will be

$$\frac{30{,}078}{42{,}969} = 70$$

per cent.

Triple-riveted Lap-joint

In Fig. 37 is illustrated a triple-riveted lap-joint. Here the length of unit section is 3 inches, and the different probable modes of failure are identical with those of the single- and double-riveted lap-joints except in rivet strength. It will be noted that in this case there are three rivets contained in each unit section, which are subjected to shear and crushing. The several methods of probable failure to be investigated are as follows:

(1) Failure of net section between the rivet holes of outer rows.

(2) Shearing of three rivets in single shear.

(3) Crushing strain on three rivets.

Using the numerical values specified in Fig. 37, we would have:

(1) $(3 - 0.75) \times 0.375 \times 55{,}000 = 46{,}406$ pounds.
(2) $3 \times 0.4418 \times 42{,}000 = 55{,}667$ pounds.
(3) $3 \times 0.75 \times 0.375 \times 95{,}000 = 80{,}156$ pounds.

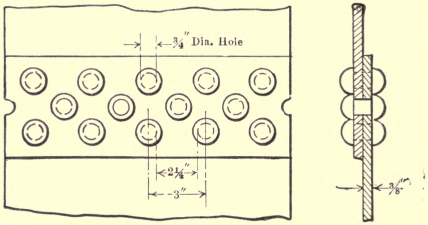

FIG. 37.

The first method of failure assumed is the most likely, and as the strength of the solid plate for a unit section of length is

$$3 \times 0.375 \times 55{,}000 = 61{,}875$$

pounds, the efficiency of joint is

$$\frac{46{,}406}{61{,}875} = 75$$

per cent. The triple-riveted joint represents about the maximum strength that can be obtained in practice from simple lap-riveted joints, as in this form the maximum pitch distance that permits proper calking of the edge of the plates is reached, and still leaving the net section of metal between the rivet holes the weakest portion of the joint, so that further addition of rivets would not add to its strength.

Chain Riveting

Joints illustrated in Figs. 36 and 37 have the rivets arranged so that the rivets in one row come opposite the

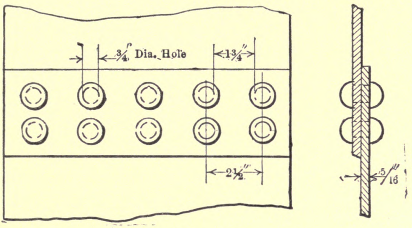

Fig. 38.

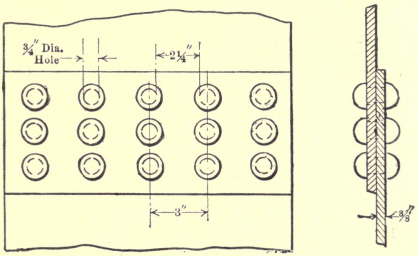

Fig. 39.

spaces in the adjacent rows, and this arrangement is termed staggered riveting. The same forms of joint are sometimes made with the rivets placed in straight rows across the joint, is illustrated in Figs. 38 and 39, which is known as chain riveting. The calculations for joint efficiency in chain-riveted joints are identical in every respect to those for staggered riveting, and with equal diameters and spacing of rivets and equal thicknesses of plate, the efficiencies are the same for either type.

Lap-riveted Joint with Inside Strap

While the lap-riveted joint with inside strap is not extensively used in the manufacture of new boilers, it affords a ready means of strengthening simple lap

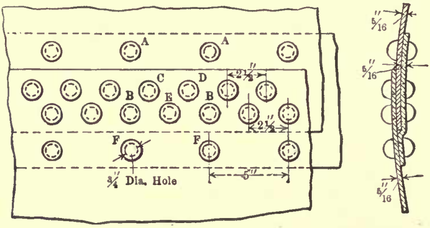

Fig. 40.

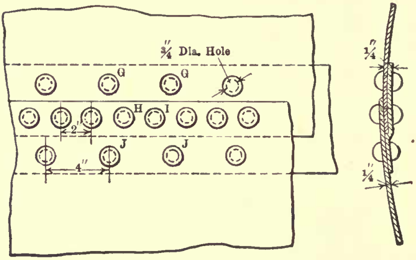

Fig. 41.

seams on boilers already constructed, and it is quite extensively used for this purpose. This joint is illustrated in Figs. 40 and 41, and it will be seen that it is equally applicable to single- or double-riveted lap-joints; it could also be applied to triple-riveted joints. The joint illustrated in Fig. 40 is identical in every respect with the one shown in Fig. 36, excepting the addition of the $\frac{5}{16}$-inch cover strip and the outer rows of rivets, these dimensions being selected to facilitate comparison between the strengths of the two joints. A unit section of this joint is 5 inches long, and five methods of failure present themselves for consideration in determining the strength of the joint:

(1) Breaking of the plate along the section between the rivet holes *A A*.

(2) The separation of the plate along the section on line of rivets *C D* and shearing the rivet *A*.

(3) Separation of the plate along the section on line of rivets *C D* and the crushing of rivet *A*.

(4) Crushing of rivets *A B C D E* by the shell.

(5) Shearing of rivets *B C D E F* in single shear.

The pulling out of the upper plate, which would shear rivet *A* single, and *B C D E* double, need not be considered, since it would evidently be stronger than the first method considered above. Calculating the value of the possible methods of failure by using the dimensions given in Fig. 40, we have:

(1) $(5 - 0.75) \times 0.3125 \times 55{,}000 = 73{,}040$ pounds.

(2) $[5 - (2 \times 0.75)] \times 0.3125 \times 55{,}000 + 0.4418 \times 42{,}000 = 78{,}726$ pounds.

(3) $[5 - (2 \times 0.75)] \times 0.3125 \times 55{,}000 + 0.3125 \times 0.75 \times 95{,}000 = 82{,}436$ pounds.

(4) $0.3125 \times 0.75 \times 95{,}000 \times 5 = 111{,}330$ pounds.

(5) $0.4418 \times 42{,}000 \times 5 = 92{,}780$ pounds.

Evidently the next section between the rivet holes $A\ A$ is the weakest portion of the joint, and since a section of the solid plate 5 inches long has a strength of

$$5 \times 0.3125 \times 55{,}000 = 85{,}937$$

pounds, the efficiency of the joint is

$$\frac{73{,}040}{85{,}937} = 85$$

per cent.

Calculation of the joint illustrated in Fig. 41 is proceeded with in the same manner as for Fig. 40. It will be noted that to aid comparison the dimensions have been assumed the same as in Fig. 35 with the strap added. The methods of possible failure to be compared are:

(1) Separation of the plate along net section $G\ G$.

(2) Separation of plate along section $H\ I$ and shearing of rivet G in single shear.

(3) Separation of plate along section $H\ I$ and crushing of rivet G.

(4) Crushing of rivets $G\ H\ I$.

(5) Shearing of rivets $H\ I\ J$ in single shear.

According to the dimensions given in Fig. 41 the numerical values would give the following results.

(1) $(4 - 0.75) \times 0.25 \times 55{,}000 = 44{,}687$ pounds.

(2) $[4 - (2 \times 0.75)] \times 0.25 \times 55{,}000 + 0.4418 \times 42{,}000 = 52{,}931$ pounds.

(3) $[4 - (2 \times 0.75)] \times 0.25 \times 55{,}000 + 0.75 \times 0.25 \times 95{,}000 = 51{,}187$ pounds.

(4) $0.75 \times 0.25 \times 95{,}000 \times 3 = 53{,}436$ pounds.

(5) $0.4418 \times 42{,}000 \times 3 = 55{,}668$ pounds.

Since the strength of the solid plate is

$$4 \times 0.25 \times 55{,}000 = 55{,}000$$

pounds, the efficiency would be

$$\frac{44{,}687}{55{,}000} = 81.25$$

per cent.

It is thus apparent that by adding a strap to the joint illustrated in Fig. 35 and making it like Fig. 41, the efficiency has been increased from 62.5 per cent. to 81.25 per cent., which would permit an increase in steam pressure of 30 per cent. on the boiler after such change.

Single-riveted Double-strapped Butt-joint

In describing all forms of butt-joints it is customary to refer to the rivets on one side of the butt only; thus, in Fig. 42 there are actually two rows of rivets, but

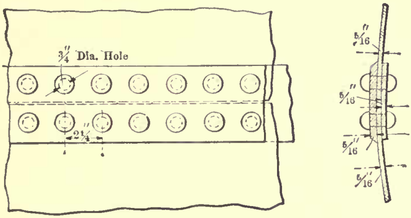

Fig. 42.

the joint is only single-riveted, for the strength of the joint along either row is in no wise dependent on the other row. If the two rows should not be riveted alike, it would be necessary to consider each side as a separate joint to find which was the weaker, in order to determine the strength of the combination. This, however, is not necessary in practical boiler joints, since they are constructed alike on each side of the butt.

In the joint illustrated in Fig. 42 it will be noted that all of the rivets are in double shear, and only three methods of possible failure are presented for calculation:

(1) Breaking the net section.
(2) Shearing of one rivet in double shear.
(3) Crushing of a rivet by the shell.

With the dimensions given in the figure we have:

(1) $(2.25 - 0.75) \times 0.3125 \times 55{,}000 = 25{,}781$ pounds.

(2) $0.4418 \times 78{,}000 = 34{,}460$ pounds.

(3) $0.75 \times 0.3125 \times 95{,}000 = 22{,}230$ pounds.

The strength of the solid plate is

$$2.25 \times 0.3125 \times 55{,}000 = 38{,}672$$

pounds, and since the weakest portion of the joint is the resistance to crushing of the rivets, the efficiency is

$$\frac{22{,}230}{38{,}672} = 57.5$$

per cent.

Double-riveted Double-strapped Butt-joint

Double-riveted butt-joints can be made in two forms, the one generally used being illustrated in Fig. 43. The calculations for the efficiency of this joint are the same as for the single-riveted joint, except that there are two rivets to be considered in each unit section of the joint instead of one. The three methods of possible failure are:

(1) Pulling apart of the sheet along net section *A A*.
(2) Shearing of rivets, *A B*, in double shear.
(3) Crushing of rivets *A B*.

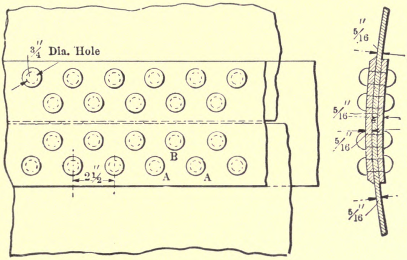

FIG. 43.

Substituting the values given in Fig. 17, we have:

(1) $(2.5 - 0.75) \times 0.3125 \times 55{,}000 = 29{,}085$ pounds.

(2) $0.4418 \times 78{,}000 \times 2 = 68{,}920$ pounds.

(3) $0.75 \times 0.3125 \times 95{,}000 \times 2 = 44{,}532$ pounds.

The strength of the solid plate is

$$2.5 \times 0.3125 \times 55{,}000 = 42{,}969$$

pounds, and the weakest portion of the joint is the net section between rivets *A A*. Therefore, the efficiency is

$$\frac{29{,}085}{42{,}969} = 67.6$$

per cent.

In Fig. 44 is illustrated the second type of double-riveted butt-joint. This form of joint, if proportioned properly, can be made considerably stronger than the one illustrated in Fig. 43. There are six methods of possible failure to be considered:

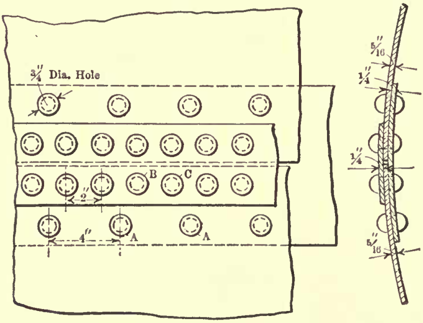

FIG. 44.

(1) Pulling apart of the sheet along net section *A A*.

(2) Pulling apart of the sheet along section *B C* and shearing rivet *A*.

(3) Pulling apart of sheet along section *B C* and crushing of rivet *A*. (Note that in calculating the crushing of rivet *A* the thickness of the strap is to be used instead of the plate, owing to the strap being thinner than the plate.)

(4) Shearing of rivet *A* single and *B*, *C* double shear.

(5) Crushing of rivets *B C* in the plate and *A* in the strap.

(6) Crushing of rivets *B C* in the plate and shearing of rivet *A*.

Substituting the numerical values from Fig. 44, we have:

(1) $(4 - 0.75) \times 0.3125 \times 55{,}000 = 55{,}859$ pounds.

(2) $[(4 - 1.5) \times 0.3125 \times 55{,}000] + (42{,}000 \times 0.4418) = 61{,}525$ pounds.

(3) $[(4 - 1.5) \times 0.3125 \times 55{,}000] + (0.75 \times 0.25 \times 95{,}000) = 60{,}781$ pounds.

(4) $(42{,}000 \times 0.4418) + (78{,}000 \times 0.4418 \times 2) = 87{,}476$ pounds.

(5) $(0.75 \times 0.3125 \times 95{,}000 \times 2) + (0.75 \times 0.25 \times 95{,}000, = 62{,}272$ pounds.

(6) $(0.75 \times 0.3125 \times 95{,}000 \times 2) + (42{,}000 \times 0.4418) = 63{,}087.25$ pounds.

From these figures it will be seen that the net section between the rivet holes *A A* is the one most likely to fail, and since the strength of a unit section of the solid plate is

$$4 \times 0.3125 \times 55.000 = 68{,}750$$

ponuds, the efficiency of the joint is

$$\frac{55{,}859}{68{,}750} = 81.25$$

per cent.

Triple-riveted Double-strapped Butt-joint

The joint illustrated in Fig. 45 is known as the triple-riveted butt-joint. The methods of failure to be investigated are the same as those in Fig. 44, and are as follows:

(1) Pulling apart of sheet at net section *A A*.

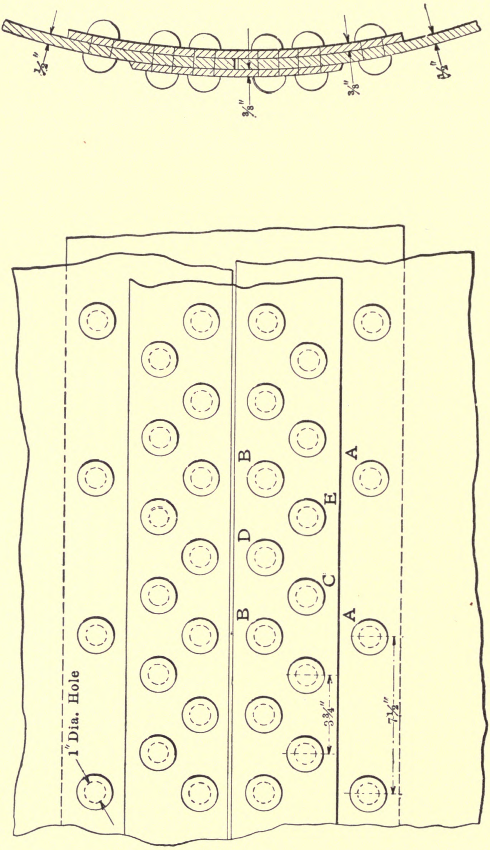

Fig. 45.

(2) Pulling apart of sheet along section *C E* and shearing rivet *A*.

(3) Pulling apart of sheet along section *C E* and crushing rivet *A*.

(4) Shearing rivet *A* single and *B C D E* double.

(5) Crushing of rivets *B C D E* in the plate and *A* in the strap.

(6) Crushing of rivets *B C D E* in the plate and shearing of rivet *A*.

Substituting the values given in Fig. 45:

(1) $(7.5 - 1) \times 0.5 \times 55{,}000 = 178{,}750$ pounds.

(2) $[(7.5 - 2) \times 0.5 \times 55{,}000] + (42{,}000 \times 0.7854) = 184{,}237$ pounds.

(3) $[(7.5 - 2) \times 0.5 \times 55{,}000] + (1 \times 0.5 \times 95{,}000) = 198{,}250$ pounds.

(4) $(0.7854 \times 42{,}000) + (0.7854 \times 78{,}000 \times 4) = 278{,}027$ pounds.

(5) $(1 \times 0.5 \times 95{,}000 \times 4) + (1 \times 0.375 \times 95{,}000) = 225{,}625$ pounds.

(6) $(1 \times 0.5 \times 95{,}000 \times 4) + (0.7854 \times 42{,}000) = 222{,}987$ pounds.

For a unit length the strength of the solid plate is

$$7.5 \times 0.5 \times 55{,}000 = 206{,}250$$

pounds. The net section between rivets *A*, *A* is the weakest portion of the joint, so that the efficiency is

$$\frac{178{,}750}{206{,}250} = 86.7$$

per cent.

Quadruple-riveted Double-strapped Butt-joint

The last type of joint to be considered is the quadruple-riveted butt-joint illustrated in Fig. 46. This joint is now used on nearly all high-grade boilers of the horizontal return-tubular type, and it marks about the practical limit of efficiency for riveted joints connecting plates of uniform thickness together. The methods of failure to be considered are practically the same as in the two preceding joints, except that there are more rivets concerned in the calculations:

(1) Pulling apart of the sheets along net section *A A*.

(2) Pulling apart of the sheet along section *D E F G* and shearing rivets *A B C*.

(3) Pulling apart of sheet along section *D E F G* and crushing of rivets *A B C* in the strap.

(4) Shearing rivets *A B C* in single shear and *D E F G H I J K* in double shear.

(5) Crushing of rivets *D E F G H I J K* in plate and *A B C* in the strap.

(6) Crushing of rivets *D E F G H I J K* in the plate and shearing rivets *A B C*.

Using the numerical values of Fig. 46, we have:

(1) $(15.5 - 1) \times 0.5625 \times 55{,}000 = 448{,}580$ pounds.

(2) $[(15.5 - 4) \times 0.5625 \times 55{,}000] + (3 \times 42{,}000 \times 0.7854) = 454{,}739$ pounds.

(3) $[(15.5 - 4) \times 0.5625 \times 55{,}000] + (3 \times 0.4375 \times 1 \times 95{,}000) = 480{,}465$ pounds.

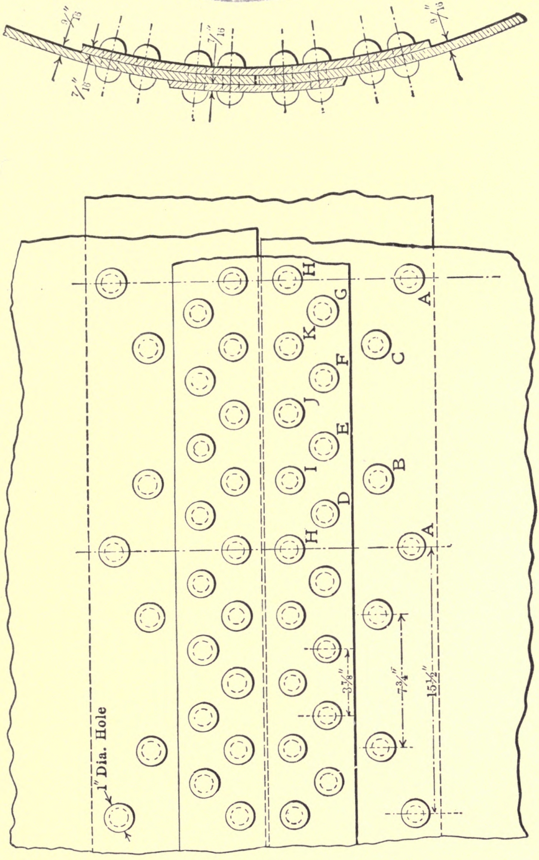

FIG. 46.

(4) $(3 \times 42{,}000 \times 0.7854) + (8 \times 78{,}000 \times 0.7854) = 589{,}050$ pounds.

(5) $(8 \times 0.5625 \times 1 \times 95{,}000) + (3 \times 0.4375 \times 1 \times 95{,}000) = 552{,}187$ pounds.

(6) $(8 \times 0.5625 \times 1 \times 95{,}000) + (3 \times 42{,}000 \times 0.7854 = 526{,}461$ pounds.

The strength of the solid plate is

$$15.5 \times 0.5625 \times 55{,}000 = 479{,}528$$

pounds, and the failure of the sheet by pulling apart along the net section *A A* is the one that determines the efficiency of the joint, which is

$$\frac{448{,}580}{479{,}528} = 93.55$$

per cent.

From the foregoing calculations it may be observed that estimating the efficiency of riveted joints, while very simple, is a rather tedious process, particularly if many joints are to be calculated

VII

TO FIND THE AREA TO BE BRACED IN THE HEADS OF HORIZONTAL TUBULAR BOILERS

For the purpose of determining the number of braces to be used, it is not necessary to figure the area of a boiler head to a fraction of a square inch, and a simple rule, the reason for which is so plain that it can never be forgotten, will be helpful to the candidate before the examiner, or when a table of circular segments is not to be had.

The diameter of the boiler and the hight above the top row of tubes are the only measurements which are ordinarily given. The flange is considered good for three inches around the outside, and the tubes for two inches above their top edges, so that the area to be braced is a part of a circle having a diameter six inches less than the given diameter of the boiler and a hight 5 inches less than that of the undiminished segment, which area is represented by the shaded area in Fig. 47.

The area of a circle is the diameter multiplied by itself and by 0.7854. It is easy, then, to find the area of the circle of which the shaded area is a part. Suppose we are dealing with a 72-inch boiler. Allowing for 3 inches on each end of the diameter, the diameter of the circle of which the segment to be braced is a part would be

$$72 - 6 = 66 \textit{ inches},$$

and its area would be

$$66 \times 66 \times 0.7854 = 3421 \text{ square inches};$$

and the area of the half circle *a b c d e* would be one-half of this, or 1710 square inches.

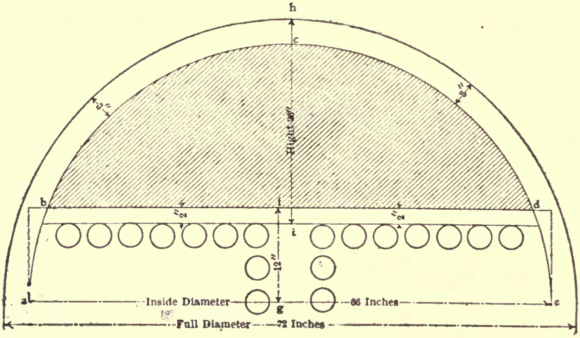

FIG. 47.

Now, if from this area the area *a b d e* is subtracted, the remainder will be the required area of the (shaded) portion to be braced. The hight *f g* is the radius, or one-half the given diameter less the given hight plus 2, and it will be near enough if we consider its length equal to the diameter, as the length of the chord *b d* is not usually given. Suppose the hight *h i* to be 26 inches, then the hight *f g* of the portion to be subtracted would be

$$\frac{72}{2} - 26 + 2 = \text{inches},$$

and if its length be taken at 66 inches its area will be

$$12 \times 66 = 792 \text{ square inches.}$$

This is too great by the area of the two little dotted triangles at *a b* and *d e*, but this is so small a proportion of the total area that it may be neglected, especially if it is borne in mind when deciding upon the number of braces that the area as determined is a little small.

Subtracting this area from that of one-half the 66-inch circle, as above found, we have

$$1710 - 792 = 918 \text{ square inches}$$

as the area to be braced.

If the pressure is 100 pounds per square inch, the force to be braced against is

$$918 \times 100 = 91,800 \text{ pounds,}$$

and if the braces used are good for 8000 pounds apiece, it will take

$$91,800 \div 8000 = 11.5 \text{ braces.}$$

We should have to use 12 braces, anyway, and these would be good for

$$\frac{12 \times 8000}{100} = 960 \text{ inches,}$$

while the actual area is 936, instead of 918, as the above approximate method made it. Unless the number of braces comes out very nearly square in the calculation, there will be enough leeway in using a whole brace for the fraction to make up for the shortness of the area. When this fraction exceeds, say, 0.9, safety would be insured by putting in an extra brace.

VIII

GRAPHICAL DETERMINATION OF BOILER DIMENSIONS [1]

The variables entering into the design of a steam boiler shell are the working pressure, the diameter of the shell, the thickness and tensile strength of the plate, the diameter, spacing and shearing value of the rivets, the efficiency of the joints and the factor of safety.

The usual working pressures are 80, 100, 125, and 150 pounds per square inch.

The standard diameters of shell are 44, 48, 54, 60, 66, 72, 78, 84, 90 and 96 inches.

The tensile strength of the plate is 52,000 to 62,000 pounds per square inch for flange steel and 55,000 to 65,000 pounds per square inch for fire-box steel. The average assumed for calculations is 60,000 pounds per square inch. The shearing value of steel rivets is 38,000 to 42,000 pounds per square inch. Until recently 38,000 pounds per square inch was used for all calculations, but this value has been gradually increasing with the improved quality of steel rivets, until 42,000 pounds per square inch is now the more generally accepted value.

[1] Contributed to Power by N. A. Carle.

This has resulted in an increased spacing of rivets, together with an increase in the efficiency of the joints, and a consequent reduction in the thickness of plate.

Rivet holes are usually punched $\frac{1}{16}$-inch larger than the rivets and calculated as $\frac{1}{8}$-inch larger than the rivets.

In marine practice, holes are specified as drilled or punched $\frac{1}{16}$-inch small, the shell assembled and the holes then reamed to full size.

The shearing value of the rivet is calculated for the stock size before driving.

The crushing value of steel rivets has been practically eliminated from the problem, because in practice the sizes selected give values in excess of the shearing value.

No consideration is given to the friction of the joint, it being assumed that this is all destroyed before rupture, so that it is not a factor of the ultimate strength.

The kind of joints and size and spacing of the rivets are governed by accident insurance companies' requirements and shop practice.

The size of rivets and spacing used necessary to insure good calking usually make the horizontal joint the weakest point in the boiler and therefore the governing factor.

It is desirable to get a high efficiency of the joint for high pressures and thick plates. Different types of joints are designated as single lap-riveted, double lap-riveted, triple lap-riveted, double butt-strap-riveted, triple butt-strap-riveted and quadruple butt-strap-riveted.

The single lap-riveted joint is used on girth seams generally, as the stress is only one-half that on the horizontal joint, and on the horizontal seams only for very small diameters and pressures.

The quadruple butt-strap-riveted joint is used only on very heavy plate, large diameters and high pressures.

The efficiencies depend upon the rivet spacing, diameter of rivets and the allowances and assumptions made.

Design conditions reduce the problem to the efficiency of the joint based on tearing between the outer row of rivets.

The usual efficiencies used in calculations in the shell formula are double lap, 70 per cent.; triple lap, 75 per cent.; double butt-strap, 80 per cent., and triple butt-strap, 86 per cent.

The factors of safety ordinarily used are 4, $4\frac{1}{2}$ and 5, with 6 sometimes specified in marine practice.

The shell formula is

$$D. \times W.P. \times F.S. = 2 \times S. \times E. \times t.$$

$D.$ = Diameter of shell in inches.
$W.P.$ = Working pressure in pounds per square inch.
$F.S.$ = Factor of safety.
$E.$ = Efficiency of horizontal joint.
$t.$ = Thickness of plate in inches.

These are shown graphically in the calculating diagram (Fig. 48). The use of this diagram is probably best illustrated by an example:

Given. — The boiler shell 66 inches in diameter for a working pressure of 125 pounds with a factor of safety of 5. What thickness of plate is required for the shell?

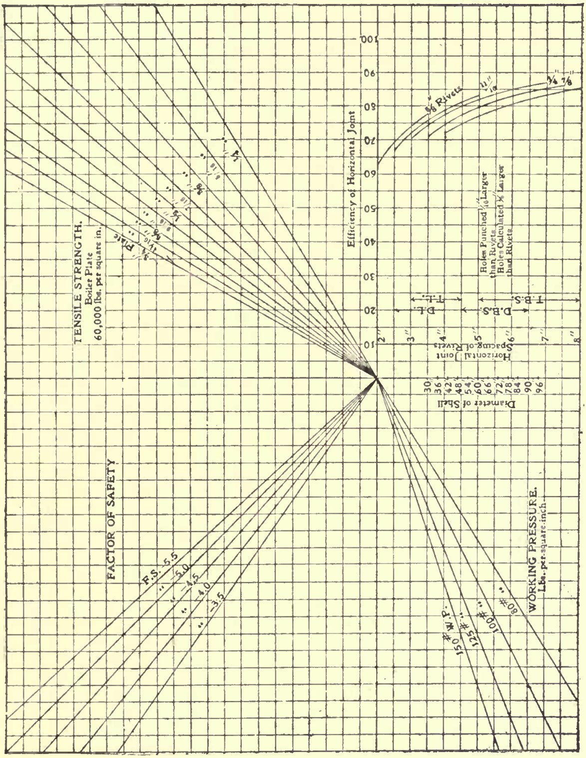

FIG. 48.

Assume that a double butt-strap joint will be used with an efficiency of 80 per cent. Starting with 66 inches "diameter of shell," read across to 125 pounds "working pressure," then up to a "factor of safety" of 5, and then across to its intersection with a vertical line through 80 per cent. "efficiency of joint." This gives a value slightly less than $\frac{7}{16}$ inch for "thickness of plate."

Hence use $\frac{7}{16}$ inch and by reading back it will be found that this gives about 5.1 as factor of safety.

Usually the designer has shop practice to follow, so that instead of using approximate values for the efficiency, the usual spacing and diameter of rivets can be selected and the actual efficiency obtained. As an example, assume that for a double butt-strap-riveted joint the shop spacing was $4\frac{1}{2}$ inches and $2\frac{1}{4}$ inches, using $\frac{11}{16}$-inch rivets. Read across from $4\frac{1}{2}$ inches "spacing of rivets" to $\frac{11}{16}$-inch rivets and then up to 82 per cent. "efficiency of horizontal joint." The boiler heads are made $\frac{1}{16}$ inch thicker than the shell, as the metal is decreased about this amount in dishing and flanging the head. The spacing of the girth-seam rivets is according to shop practice and does not require a high efficiency, as the stress is only one-half that of the shell. It is, therefore, a dependent factor in the design.

IX

THE SAFETY VALVE

THE study of the safety valve has been the first step of many a man in scientific engineering. Induced to its study by the necessity of solving its problems before the examiner, his consideration of this simple device has led him into the computation of areas, into a study of the principle of the lever, of moments of forces, of the velocity of flow of steam and other fundamental principles of mechanics. This applies to those who have studied the subject intelligently, not to those who have attempted to get over it by learning a rule by rote, simply to be confounded when confronted by another rule, or a case to which their rule would not apply. The whole subject is so simple that an hour's study will put a man in possession of the fundamentals so that he can make his own rules or solve any problem without a rule, from a sheer understanding of the principles involved.

PRESSURE PER SQUARE INCH

A cubic foot of water weighs, in round numbers, 62 pounds. If you can imagine ten cubic feet packed one above the other, as in Fig. 49, they would make a column weighing some 206 pounds, supported on a

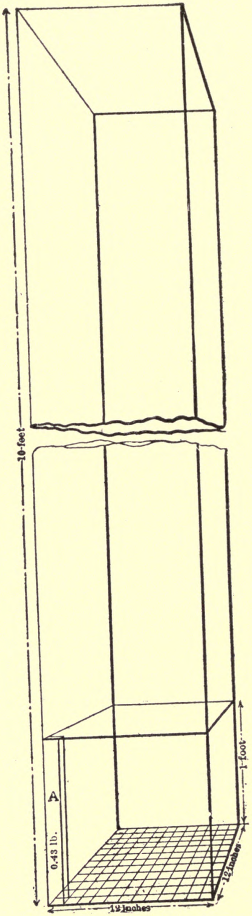

FIG. 49.

base one foot square, so that the pressure would be 620 pounds *per square foot.* The water in a tank or pond may be conceived to be divided into columns of this kind, and it will be seen that there will be a pressure on the bottom of 62 pounds per square *foot* for every foot of depth. But, in the square foot supporting this weight there are 144 square inches; and as the pressure is evenly distributed, each square inch carries:

$$62 \div 144 = 0.43 \text{ of a pound.}$$

for each foot in depth, and the pressure in the case of the column 10 feet in hight would be 620 pounds per square foot, or 4.3 pounds *per square inch.*

Just as the tank or pond could be conceived to be divided into columns of one square foot section, each square foot can be conceived to be divided into 144 columns of one square inch section, as shown in Fig. 49, and each foot in hight of such column, like the piece marked A, would weigh $\frac{1}{144}$ of the whole weight of the cubic foot of which it is the $\frac{1}{144}$ part, and press upon its square inch of base with a pressure of:

$$62 \div 144 = 0.43 \text{ of a pound.}$$

As this pressure in a liquid or gas is exerted in all directions, it is evident that the pressure on the horizontal piston in Fig. 50 would be 4.3 pounds *per square inch*, and if it has an area of 30 square inches there would be a force of:

$$4.3 \times 30 = 129 \text{ pounds.}$$

forcing the piston to the right; and since there is at a

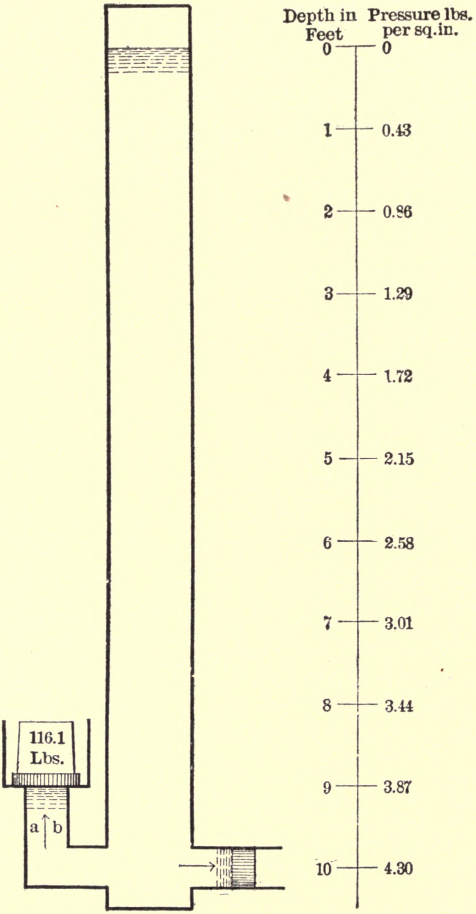
Depth in Feet
Pressure lbs. per sq.in.
0
0
1
0.43
2
0.86
3
1.29
4
1.72
5
2.15
6
2.58
7
3.01
8
3.44
9
3.87
10
4.30
116.1 Lbs.
a
b

FIG. 50.

depth of 9 feet a pressure of 3.87 pounds *per square inch* the valve at the left would have 3.87 pounds pushing upward on *each square inch* of its exposed area, *i.e.*, the area corresponding with the diameter *a b*, and if that area were 30 square inches it would take:

$$30 \times 3.87 = 116.1 \text{ pounds}$$

to hold the valve closed against that pressure.

The steam gage shows the pressure *per square inch*. If the gage points to the 100 mark it indicates that if the pressure existing in the boiler were exerted upon one square inch of area, Fig. 51, it would push with a force of one hundred pounds. If exerted upon an

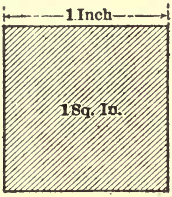

FIG. 51.

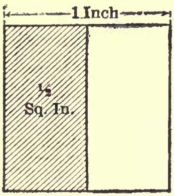

FIG. 52.

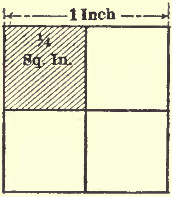

FIG. 53.

area of one-half a square inch, Fig. 52, it would push with a force of 50 pounds; upon an area $\frac{1}{2}$ inch square, or $\frac{1}{4}$ of a square inch, Fig. 53, 25 pounds; upon an area of one square foot, or 144 square inches, 14,400 pounds, etc.

The force exerted by the steam to lift a safety valve depends then upon the area of the valve as well as upon the intensity of the pressure.

To Find the Area of a Circle

The area of a 1-inch circle is 0.7854 of a square inch, the difference, 0.2146, between this and the full square of the diameter being taken up by the corners, Fig. 54. If the side of the square is *doubled* the area

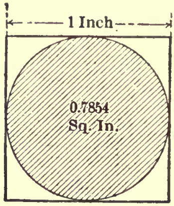

Fig. 54.

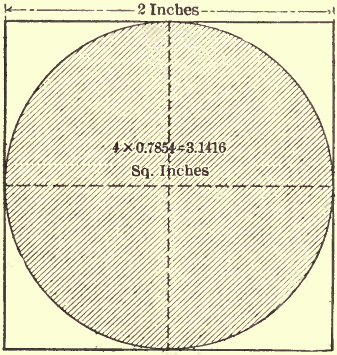

Fig. 55.

of the square will be multiplied by *four*, as is plainly shown by Fig. 55, which obviously contains *four* squares of the area of that shown in Fig. 54, although its side is but *twice* as long; and it is equally evident that the inclosed circle bears the same proportion to the total area in both cases and that the shaded area of the circle in Fig. 55 is *four* times that in Fig. 54, although its diameter is but *twice* that of the smaller circle. If we *treble* the length of the sides the area of the square will be multiplied by *nine*, always the *square* of the side, *i.e.*, the side multiplied by itself.

The area of any circle may be found by multiplying the area of a 1-inch circle (0.7854) *square inch by the square of the given diameter.*

In Fig. 55 the diameter is 2 inches and the area is:

$$2 \times 2 \times 0.7854 = 3.1416 \text{ square inches.}$$

The area of a 4-inch circle would be:

$$4 \times 4 \times 0.7854 = 12.5664 \text{ square inches.}$$

It may aid in remembering the factor 0.7854 to know that it is one-fourth of 3.1416, the number by which the diameter is multiplied to get the circumference.

The area of a triangle is obviously one-half the product of its base and hight. In Fig. 53*a* the product

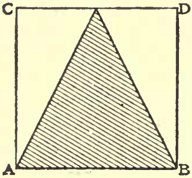

FIG. 53*a*.

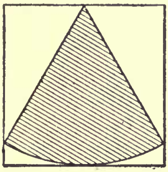

FIG. 53*b*.

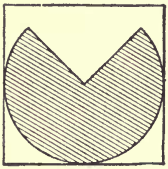

FIG. 53*c*.

of the base *A B* and the hight *A C* would be the area of the rectangle *A B C D* and the shaded area of the triangle is obviously one-half of this, for the two unshaded portions put together would make a similar triangle. This is just as true if the base is an arc of a circle as in Fig. 53*b*, and just as true if the base incloses the apex as in Fig. 53*c*. The circle is therefore a triangle with a circular base 3.1416 times the diameter or 2×3.1416 times the radius, and with a hight equal to the radius, and its area (one-half the product of hight and base) is:

$$\text{Area} = \frac{\text{radius} \times 2 \times 3.1416 \times \text{radius}}{2} = 3.1416 \text{ radius}^2,$$

so that the area equals 3.1416 times the square of the radius, and since the radius is one-*half* the diameter, the square of the radius is the square of the diameter divided by *four:*

$$\text{Area} = 3.1416\, r^2 = 3.1416\, \frac{D^2}{4} = D^2 \times \frac{3.1416}{4}$$
$$= 0.7854\, D^2.$$

Effect of Pressure in Lifting a Valve

Suppose the 3-inch valve in Fig. 56 to be loaded with six weights of 100 pounds each and that the valve and steam weighed 30 pounds, what would the pressure per square inch have to be to lift it?

The total weight to be lifted is 630 pounds. The total upward pressure must equal this, and if 630 pounds is exerted on 7.0686 square inches (the area of a 3-inch valve, see table) the pressure on each square inch will be:

$$630 \div 7.0686 = 89.1 \text{ pounds.}$$

How much load would have to be put upon the same valve to allow it to blow off at 75 pounds per square inch?

If the pressure exerts 75 pounds on one square inch, it would exert on the 7.0686 square inches of the valve which is exposed to it:

$$75 \times 7.0686 = 530 \text{ pounds,}$$

which must be the combined weight of the valve and the weights with which it is loaded.

Fig. 56 does not show a practicable valve, but is sufficient to illustrate the point that the force tending

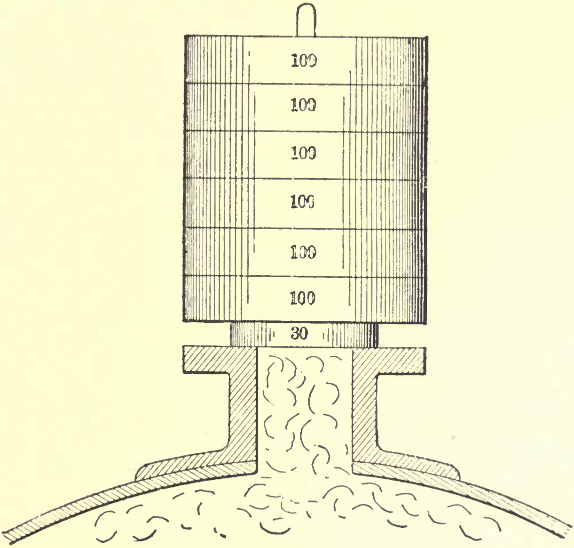

FIG. 56.

to lift the valve must equal that holding it to its seat (in this case the dead weight of the valve itself and the weights with which it is loaded), and that this upwardly acting force is the area of the valve in square inches, multiplied by the pressure per square inch. Such a dead-weight valve is ponderous and impracticable and the usual practice is to use a lighter weight, increasing its effect by leverage, or to hold the valve to its seat with a spring.

THE PRINCIPLE OF THE LEVER

Suppose a strip of board balanced over a sharp edge as in Fig. 57. If equal weights be placed upon it at

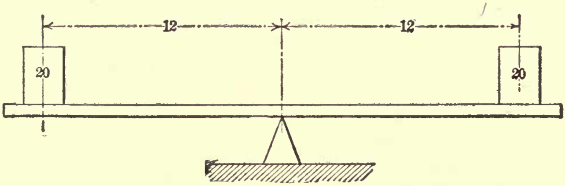

FIG. 57.

equal distances from the center it will still be in balance. If one of the weights be moved in *half* of the distance to the point at which they are balanced, as in Fig. 58, the other weight will have to be *halved* to

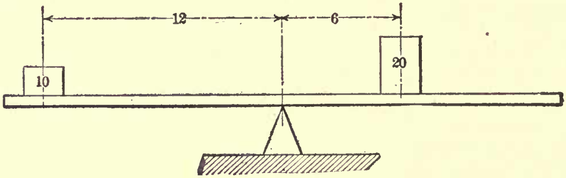

FIG. 58.

preserve the equilibrium. If one of the weights be moved to *one-third* of its distance from the balancing point, as in Fig. 59, the other weight will have to be reduced to *one-third* of its original magnitude to preserve the balance at the original distance.

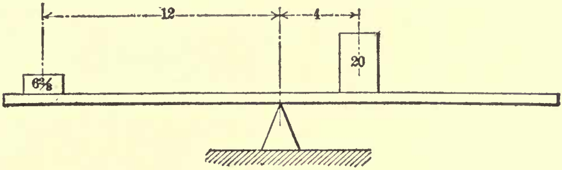

FIG. 59.

Notice that in each case the product of a weight by its distance from the point over which they balance is the same as the product of the weight which balances it and its distance from the same point. Suppose the weights in Fig. 57 to be each 20 pounds, each at 12 inches from the center. Here obviously the weights and distances being the same their products are equal:

$$20 \times 12 = 240 \quad \textit{and} \quad 20 \times 12 = 240.$$

When the right-hand weight is moved in to 6 inches from the center the other had to be reduced to 10 pounds:

$$10 \times 12 = 120 \quad \textit{and} \quad 20 \times 6 = 120.$$

When the left-hand weight was moved in to 4 inches from the center the other had to be reduced to $6\frac{2}{3}$:

$$6\tfrac{2}{3} \times 12 = 80 \quad \textit{and} \quad 20 \times 4 = 80.$$

The same principle applies in Fig. 60, where the force exerted by the man, multiplied by the distance *A B*, must, if he lifts the machine, equal the pressure with which the load bears on the bar at the point *C*, multiplied by the distance *B C* of that point from the point *B* around which the lever turns. In mechanics, this point, the *B* of Fig. 60, is called the "fulcrum" and the product of the load, weight or force by its distance from the fulcrum is called its "moment." In the case described by Fig. 57 the moment of each

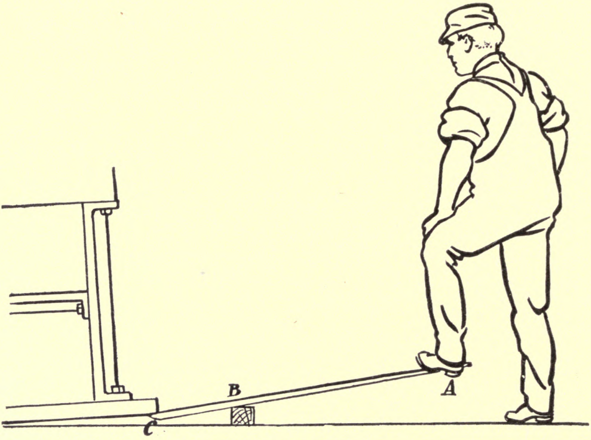

FIG. 60.

weight is 240; in that of Fig. 58, 120; in that of Fig. 59, 80; in that shown in Fig. 60 the moment of the load is the weight or force with which the load bears on the point *C*, multiplied by its distance from the fulcrum *B*, and the moment of the force is the force which the man exerts upon the bar at *A*, multiplied by the distance of that point from the fulcrum.

Notice that in Fig. 61 the fulcrum is at one end of the lever instead of between the load and force as in the other examples. The principle is the same. The fulcrum is the *stationary* point about which the load and the force move. In Figs. 60 and 61 it is evident that the shorter the distance between the load and the fulcrum the less the man will have to exert himself.

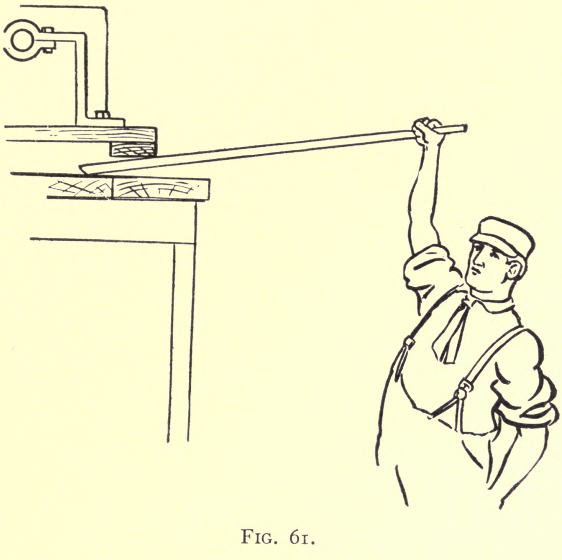

FIG. 61.

The point to grasp and remember is that the *moments must be equal* in order for the force to balance or lift the load.

Equal Moments Produce Equilibrium

There are four important things about a lever:

L = the load.
F = the force applied to balance or overcome the load.
D_l = distance of the load from the fulcrum.
D_f = distance of the force from the fulcrum.

If any three of these are known the third can be easily determined, for, as has been just explained,

Force × distance of force = load × distance of load.

$$F \times D = L \times D_l$$

Moment of force = moment of load.

To find the force required to lift a given load:
FORMULA:

$$F = \frac{L \times D_l}{D_f}.$$

RULE. — *Multiply the load by its distance from the fulcrum, and divide by the distance at which the force is applied from the fulcrum.*

To find the distance at which a given force must be applied from the fulcrum to balance a given load:
FORMULA:

$$Df = \frac{L \times D_l}{F}.$$

RULE. — *Multiply the load by its distance from the fulcrum and divide by the given force.*

To find the load which may be lifted with a given force:
FORMULA:

$$L = \frac{F \times D_f}{D_l}.$$

RULE. — *Multiply the given force by the distance of its point of application from the fulcrum and divide by the distance of the load from the fulcrum.*

To find the distance at which a given weight or load must be placed from the fulcrum to balance a given force:

FORMULA:
$$D_l = \frac{F \times D_f}{L_l}.$$

RULE. — *Multiply the given force by the distance of its point of application from the fulcrum and divide by the load.*

THE LEVER SAFETY VALVE

Effect of the Leverage of the Ball

Suppose the weight instead of setting directly upon

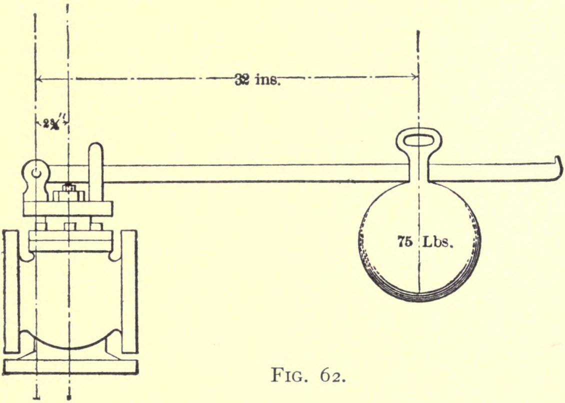

FIG. 62.

the valve, as in Fig. 56, is applied through a lever, as in Fig. 62. From what has preceded it will easily be seen that the weight multiplied by its distance from the fulcrum will equal the force which it will exert upon the valve stem multiplied by the distance of its point of application from the fulcrum.

$$\left\{\begin{matrix}\text{Weight}\\ \text{of}\\ \text{ball}\end{matrix}\right\} \times \left\{\begin{matrix}\text{Distance}\\ \text{of ball}\\ \text{from}\\ \text{fulcrum}\end{matrix}\right\} = \left\{\begin{matrix}\text{Pressure}\\ \text{of ball}\\ \text{on}\\ \text{stem}\end{matrix}\right\} \times \left\{\begin{matrix}\text{Distance}\\ \text{of stem}\\ \text{from}\\ \text{fulcrum}\end{matrix}\right\}$$

Let the weight equal 75 pounds,
distance of weight from fulcrum 32 inches,
distance of stem from fulcrum $2\frac{3}{4}$ inches,
what will be the force exerted by the ball to hold the valve to its seat?

$$\frac{\text{Weight of ball} \times \text{Distance of ball from fulcrum}}{\text{Distance of stem from fulcrum}} = \text{Pressure of ball on stem.}$$

Then the moment of the ball is:

$$75 \times 32 = 2400 \text{ inch-pounds,}$$

and:

$$2400 \div 2.75 = 872.727 \text{ pounds}$$

will be the pressure on the valve stem due to the ball and the moments will be equal:

$$75 \times 32 = 2400 \quad \text{and} \quad 872.727 \times 2.75 = 2400.$$

Suppose this to be a 4-inch valve, the area of which is 12.5666 square inches. The pressure per square inch upon the under side of the valve necessary to balance the effect of the ball would be:

$$872.727 \div 12.5666 = 69.4 \text{ pounds.}$$

This is the pressure at which the valve would blow off if nothing but the ball were holding it to its seat. It takes a little additional pressure to lift the valve and to overcome the weight of the lever, as will be explained later, but this is a comparatively small affair and in usual approximate calculations is not taken into account. Neglecting these we can make the following simple

Rules for lever safety valve, neglecting weight of valve, stem and lever:

Let W = weight of the ball,
D = distance of ball from fulcrum,
A = area of valve in square inches,
d = distance of stem from fulcrum,
P = pressure per square inch on valve which will balance ball.

To determine the pressure on a valve of given diameter required to balance a ball of given weight at a given distance from the fulcrum.

FORMULA:

$$P = \frac{W \times D}{A \times d}.$$

RULE. — *Multiply the weight of the ball by its distance from the fulcrum. Multiply the area of the valve in square inches by the distance of its stem from the fulcrum. Divide the first product by the second and the quotient will be the pressure per square inch required to overcome the weight of the ball.*

EXAMPLE. — The stem of a 4-inch safety valve is $2\frac{3}{4}$ inches from the fulcrum. Supposing the valve will blow when the gage shows 7 pounds without any weight upon the lever (*i.e.*, that it takes 7 pounds per square inch on the area of the valve to overcome its own weight, that of the stem and the bearing effect of the empty lever), at what pressure would it blow with a weight of 75 pounds (Fig. 62) 32 inches from the fulcrum?

BY THE FORMULA:

$$P\frac{W \times D}{A \times d} = \frac{75 \times 32}{12.5666 \times 2.75} = 69.4 + 7 = 76.4 \text{ pounds.}$$

BY THE RULE:

$$\begin{array}{lrrl}
\text{Area of valve} & 12.5666 & 75 & \text{Weight of ball} \\
\text{Distance of stem} & \underline{2.75} & \underline{32} & \text{Distance of ball} \\
 & 628330 & 150 & \\
 & 879662 & 225 & \\
 & \underline{251332} & & \\
\end{array}$$

$34.558\cancel{150})2400.00)69.4$ pounds.

This is the pressure required to lift the ball. Adding the 7 pounds required to blow the valve without the ball, the answer would be 76.4 pounds. Scratching out the last three figures of the first product saves handling large numbers and does not materially affect the result. If we called this 34.6 (nearer right than 34.5 because the 58 rejected is over one-half) the quotient would still be 69.36.

To find the weight required to hold a given pressure on a given valve:

FORMULA:

$$W = \frac{A \times P \times d}{D}.$$

RULE. — *Multiply the area by the pressure and by the distance of the stem from the fulcrum and divide by the distance of the ball from the fulcrum. The quotient will be the weight of ball required to balance the steam pressure on the valve.*

EXAMPLE. — What weight of ball would be required to allow the valve in the above example to blow off at 80 pounds?

The ball must provide for 73 pounds per square inch, the lever valve and stem taking care of the other seven, so that $P = 73$ pounds.

BY THE FORMULA:

$$W = \frac{A \times P \times d}{D} = \frac{12.5666 \times 73 \times 2.75}{32} = 78.8 \text{ pounds.}$$

BY THE RULE:

$$\begin{array}{rl}
12.5666 & \text{Area} \\
73 & \text{Pressure} \\
\hline
376998 & \\
879662 & \\
\hline
917.3618 & \\
2.75 & \text{Distance of stem} \\
\hline
45868090 & \\
64215326 & \\
18347236 & \\
\hline
\end{array}$$

Distance of ball, $32)\overline{2522.744950}(78.8$ pounds.

To find the position of the weight in order that it may exert a given pressure on the stem:

FORMULA:
$$D = \frac{A \times P \times d}{W}.$$

RULE. — *Multiply the area by the pressure and by the distance of the stem from the fulcrum and divide by the weight of the ball. The quotient will be the distance at which the ball must be from the fulcrum in order to produce a given pressure on the stem.*

EXAMPLE. — If the original 75-pound weight had been used, at what distance from the fulcrum would it have had to have been placed to have allowed the valve to blow off at 80 pounds?

BY THE FORMULA:

$$D = \frac{A \times P \times d}{W} = \frac{12.566 \times 73 \times 2.75}{75} = 33.6 \text{ inches.}$$

BY THE RULE. — The product of the factor in the numerator is 2522.74495 as before, and dividing this by 75, the weight of the ball:

$$75 \,)\, 2522.74495 \,(\, 33.6 \text{ inches.}$$

These simple rules will serve all practical purposes, especially if it is borne in mind that P represents the pressure with which the *ball only* bears upon the stem, not including the weight of the valve, lever, etc., and an allowance be made for these other effects as has been done in the examples. A general idea of what the pressure per square inch required to lift the valve, stem and lever may be is given in column 8 of the table on page 119. It is well, however, to know how to make these allowances accurately, and they will now be considered.

Effect of the Weight of the Valve and Stem

The pressure acts directly upon the valve and stem without leverage, and must exert a force to balance their weight equal simply to that weight, just as was the case in Fig. 56.

Suppose the valve and stem of a 3-inch valve to weigh 1.5 pounds, how much pressure per square inch would be required to lift the valve from its seat?

Comparing Figs. 56 and 63, it will be seen that this case is the same as the first example given in describing the earlier cut. The total pressure on the valve must be 1.5 pounds, and if 1.5 pounds is to be exerted on 7.0686 square inches, the pressure per square inch will be:

$$1.5 \div 7.0686 = 0.212 \text{ pound.}$$

Column 3 of the table on page 119 gives roughly the weights of valve and stem used on valves of the standard diameters of three makers, and in connection with column 4, which gives the pressure per square inch required to lift the valves of the given weights, serves to indicate the relative importance of this factor of the problem.

The Effect of the Lever

The weight of the lever tends to hold the valve upon its seat. It is evident that it would take a considerable pull to lift the lever of a large safety valve with a cord attached at the point at which the pin bears, as in Fig. 64, and this pull as measured upon a scale would be the force which the valve would have to exert to push the lever up. Every successive particle in the length of the lever is acting with a different leverage, so that it

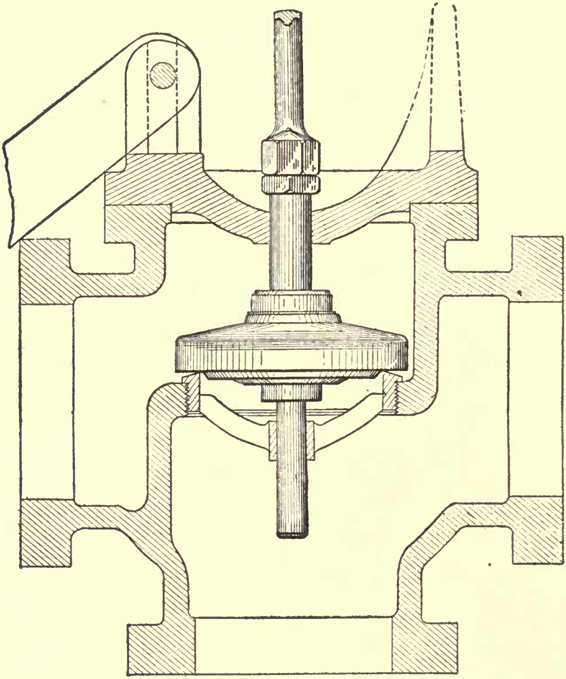

Fig. 63.

would at first appear a complicated process to calculate this force; but a body acts in this respect just as though its whole mass were concentrated at its center of gravity and this makes the problem very simple.

If the lever be taken off and balanced over an edge, as in Fig. 65, the center of gravity will be at the point

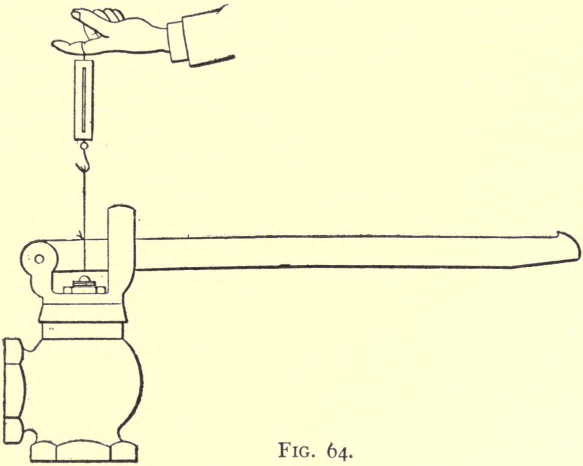

Fig. 64.

above the knife edge when the lever is balanced, and the effect of the lever would be the same as if all the mass were concentrated at that point.

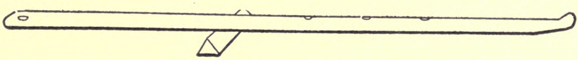

Fig. 65.

Now find the distance of the center of gravity from the fulcrum, from the point around which the lever turns. This will be from the center of the hole when it turns upon a pin, as in Fig. 66, or from the point where it bears if a knife edge is used, as in Fig. 67; the distance *a c* in each case.

In measuring for moments the distances must be taken on a line passing through the fulcrum and at

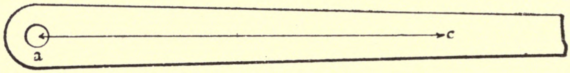

FIG. 66.

right angles to the direction of the force. In the case of the lever safety valve the holding-down force is

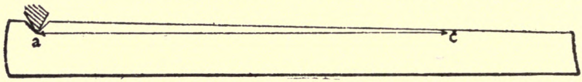

FIG. 67.

gravity, which acts vertically. A line at right angles to the vertical is horizontal, so that distances should

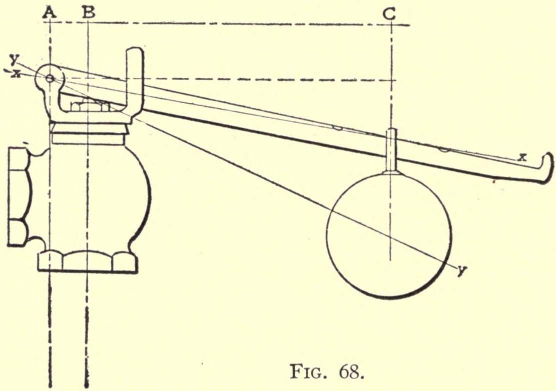

FIG. 68.

be measured in a horizontal direction as through *A B C*, Fig. 68, and not on the lines *x x* or *y y*.

In determining the distance *a c*, Figs. 66 and 67, do not get bothered about the piece of lever which extends back of the fulcrum. The more metal there is back of this point the nearer the center of gravity is to the fulcrum. If there were as much weight to the left of the pin in Fig. 65 as to the right, the center of gravity would be at the pin; the lever would balance over the pin as it did over the knife edge and not bear on the stem at all.

To apply this, suppose that the lever of a 3-inch valve weighed six pounds, that the distance *a c*, Fig. 66, between the fulcrum and the center of gravity was found to be 15 inches, and the distance *a b* from the fulcrum to the point at which the pin bears $2\frac{1}{4}$ inches. The moment of the lever must be:

$$6 \times 15 = 90 \text{ inch-pounds.}$$

The moment of the lifting force must equal this, and that moment is $2\frac{1}{4}$ times the force. Then the force must be:

$$90 \div 2\frac{1}{4} = 40 \text{ pounds,}$$

$$2\frac{1}{4} \times 40 = 90 \quad \textit{and} \quad 15 \times 6 = 90.$$

Since a force of 40 pounds is to be exerted upon 7.0686 square inches, the force per square inch would be:

$$40 \div 7.068 = 5.66 \text{ pounds.}$$

The combined effect of the valve and stem and of the lever of the 3-inch valve in question would be:

$$0.212 + 5.66 = 5.87 \text{ pounds.}$$

Columns 5 and 6 of the table already referred to give the weights of levers and the distances of their centers of gravity from the fulcrum as ordinarily found, and column 7 gives the pressure per square inch on the valve necessary to lift such levers. Column 8 gives the sum of the respective values in columns 4 and 7, *i.e.*, the pressure per square inch required to lift the valve and stem and the lever. It will be seen that the values run fairly even for all sizes of valves, and that by using seven or eight pounds as an allowance as in the above examples, results can be attained with the simple rules which will be within a pound or two of right.

Spring-loaded or Pop Safety Valves

A rule for calculating the pressure at which a spring-loaded valve will blow off is sometimes asked for. There are none reliable that do not involve the determining by experiment of the force required to compress the spring, and if you are going to do this you may as well determine by experiment at what pressure the valve will blow off. In practice nobody thinks of computing the spring-loaded valve. If they want it to blow off at 120 pounds they procure a suitable spring from the makers and turn down upon the binding nut until the valve will blow experimentally at the desired pressure. The pressure at which a spring will yield depends not only upon the shape and size of the material of which it is made, the diameter, number, and pitch of the coils, all of which are measurable and determinable, but upon the nature and condition of the material itself. You can readily appreciate that a spring of brass would compress with less pressure than one of steel, similar in every other respect, and that there is such a wide difference in steels that there will be a great deal of difference in the action of steel springs according to the kind of metal, degree of temper, etc. The best rule known is the following:

To find at what pressure a valve will lift with a spring of given dimensions and compression:

Multiply the compression in inches by the fourth power of the thickness of the steel in sixteenths of an inch, and by 22 for round or 30 for square steel. Product I.

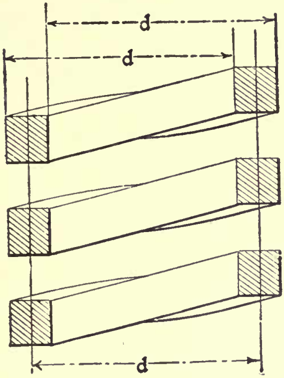

FIG. 69.

Multiply the cube of the diameter of the spring, measured from center to center of the coil (as on the line d, in Fig. 69) in inches, by the number of free coils in the spring, and by the area of the valve in square inches. Product II.

Divide Product I by Product II and the quotient will *be the pressure per square inch at which the valve will blow off.*

The weight of the valve and of the spring should in strictness be added to Product I, when the construction is such that the valve supports the spring; but inasmuch as the values 22 and 30 are guessed at it will not pay to go into refinements in other directions. The result of this rule has never been compared with an actual valve. It is based on a formula adopted by a committee of Scotch engineers and shipbuilders. Correspondence with the manufacturers of pop safety valves as to the accuracy of the formula brings out the fact that they proportion and calibrate their springs only by experience and experiment. However, this rule is given for what it is worth. If you have a spring-loaded valve calculate it by this rule and see how nearly it comes to the point at which the valve will blow off.

With a dead weight or a lever-loaded valve the force required to lift it remains the same, no matter how high the valve lifts. The weights weigh no more if they are raised an inch or two, and the leverage does not change, but with the spring-loaded valve the more the valve lifts, the more the spring is compressed, and the more force is required to compress or hold it. It follows then that if an ordinary valve were loaded with a spring it would simply crack open and commence to sizzle when the pressure equaled the force at which the spring was set, and that if this were not enough to relieve the boiler the pressure would have to increase, opening the valve more and more until the steam blew off as fast as it was made.

Complete Saftey-valve Rules

It is evident that any complete rule for the safety valve must include the separate treatment of the valve and stem, the lever and the ball as factors in holding the valve to its seat.

$$\left.\begin{array}{l}\text{Moment of the}\\ \text{lifting force}\end{array}\right\} = \left\{\begin{array}{l}\text{moment of ball}\\ \qquad\qquad + \\ \text{moment of lever}\\ \qquad\qquad + \\ \text{moment of valve and stem.}\end{array}\right.$$

The lifting force consists of the pressure per square inch into the area of the valve, and its moment is the product of the force by its distance from the fulcrum. Expanded, then, the above becomes:

$$\left.\begin{array}{l}\text{Pressure}\\ \qquad\times\\ \text{Area}\\ \qquad\times\\ \text{Distance of stem}\\ \quad\text{from fulcrum}\end{array}\right\} = \left\{\begin{array}{l}\text{Weight of ball} \times \text{distance of}\\ \quad\text{its center of gravity from the}\\ \quad\text{fulcrum}\\ \qquad\qquad + \\ \text{weight of lever} \times \text{distance of}\\ \quad\text{its center of gravity from}\\ \quad\text{fulcrum}\\ \qquad\qquad + \\ \text{weight of valve and stem} \times\\ \quad\text{distance of their center of}\\ \quad\text{gravity from the fulcrum.}\end{array}\right.$$

In order to find one of these qualities we must know all the rest, and consequently since the missing quantity can be but on one side of the equal mark we can figure the combined value of the quantities on one side of the equation (that is, in one set of brackets). Then we can work out the operation indicated on the other side as far as we can go. If the missing quantity is on the left-hand side of the equation it can be found by dividing the value of the other side of the equation by the product of the two known factors on the left-hand side.

To find the pressure at which a certain valve will blow off:

Multiply the weight of the ball, of the valve and stem and of the lever, each by the distance of its center of gravity from the fulcrum and add the products. Multiply the area of the valve by the distance of its center from the fulcrum and divide the sum above found by the product. The quotient will be the pressure required.

Or more briefly:

Divide the sum of the moments of the valve, lever and ball by the product of the area of the valve and distance from the fulcrum.

EXAMPLE.—At what pressure will a 3-inch valve blow off with stem $2\frac{1}{4}$ inches from the fulcrum, valve and stem weighing $1\frac{1}{2}$ pounds, lever weighing 6 pounds, having its center of gravity 15 inches from the fulcrum and weighted with a 48-pound ball 24 inches from the fulcrum?

$$\left.\begin{array}{l}\text{Pressure}\\ \qquad\times\\ \text{Area} = 7.0686\\ \qquad\times\\ \text{Distance } 2\frac{1}{4}\\ \text{Product} = 15.90435\end{array}\right\} = \left\{\begin{array}{rr} 48 \times 24 = & 1152\\ 6 \times 15 = & 90\\ 1.5 \times 2\frac{1}{4} = & 3.375\\ \text{Sum of moments} = & 1245.375\end{array}\right.$$

$$1245.375 \div 15.90435 = 78.3 \text{ pounds.}$$

This is all that we shall be likely to wish to find on this side of the equation, for the distance of stem is fixed and the area determined by other considerations.

The other two things that interest us are the weight of the ball and its distance from the fulcrum.

To find weight of ball or its distance from fulcrum:

Multiply the pressure by the area and by the distance of the stem from the fulcrum. The product is the moment of the force.

Multiply the weight of the valve by the distance of the stem from the fulcrum; multiply the weight of the lever by the distance of its center of gravity from the fulcrum, and add the products.

Subtract the sum of the products just found from the moment of the force, and the difference is the moment of the ball.

Divide the moment of the ball by the weight of the ball and the quotient is its distance from the fulcrum.

Divide the moment of the ball by the distance from the fulcrum and the quotient is the weight of ball required.

EXAMPLE — What weight of ball at the same distance would be required to allow the valve given in the previous example to blow at 75 pounds, and at what distance would the 48-pound ball there given have to be placed from the fulcrum to produce the same result?

75 × 7.0686 × Moment $2\frac{1}{4}$ of force 1192.826 93.375 1099.451 moment of ball	Weight × distance of ball + 6 × 15 = 90.000 + 1.5 × 2.25 = 3.375 93.375 sum of moments, valve and lever.

$$\frac{1099.451}{48} = \text{inches, distance of ball.}$$

$$\frac{1099.451}{24} = \text{pounds, weight of ball.}$$

But the ideal valve should stay on its seat until the pressure reaches the desired limit, then open wide and discharge the excess. This result is accomplished by the construction shown in Fig. 70. With the first opening of the valve the steam passes into the little "huddling chamber" made by the cavity near the overhanging edge of the valve and a similar cavity surrounding the seat. The pressure which accumulates here, acting on the additional area of the valve, raises it sharply with the "pop" which gives the valve its name, and it is sustained by the impact and reaction of the issuing steam until the pressure has subsided sufficiently to allow the spring to overcome these actions.

The outside edge of the lower trough in the valve shown is composed of an adjustable ring which may be

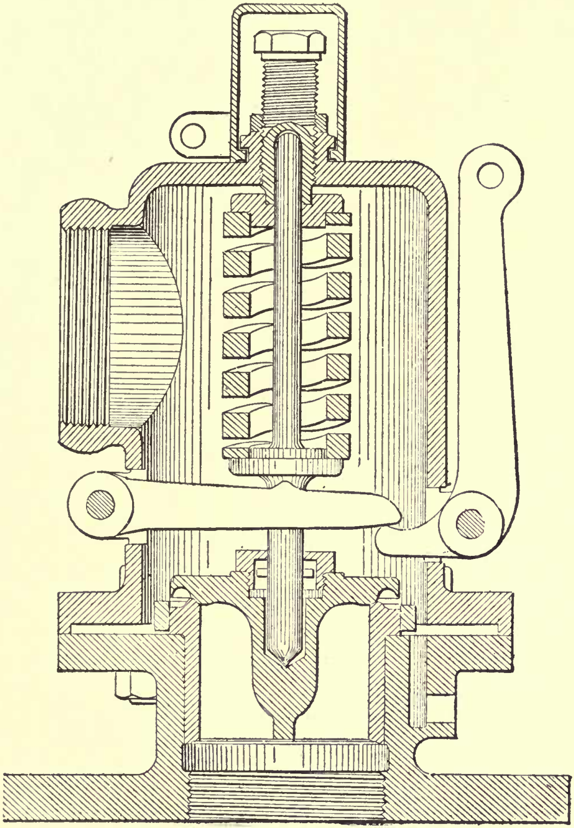

FIG. 70.

screwed up or down so as to diminish or increase the distance between the overhanging lip of the valve and its own inner edge, controlling the outlet from the chamber; and diminution of pressure or the "blow back" required to allow the valve to seat so that the valve opens wide at a given pressure and seats promptly without sizzling or chattering when the pressure has been reduced a certain amount depending upon the adjustment of the ring. The various makers have adopted different devices for adjusting the ring or other device for controlling the outflow from the huddling chamber.

The Capacity of Safety Valves

Let us next consider the capacity of valves; how large a valve is required for a given boiler. Most of the rules deal with grate surface and the area of the valve; the rule adopted by the U. S. Board of Supervising Inspectors being one square inch of valve area for each two feet of grate area. That the valve should be proportioned to the grate surface seems proper because it is the grate surface, and not the heating surface, which determines and limits the capacity of a boiler. To a given grate surface, however, we should apportion a sufficient amount of area of opening, and this area of opening is not proportional to the area of the valve but to the diameter and lift. A valve 1 inch in diameter has an area of 0.7854 of a square inch, but that does not mean that there will be an opening of 0.7854 of a square inch for the steam to escape. If the valve is flat, as in Fig. 71, the area opened for the discharge of steam will be the circumference of the valve multiplied by the lift. The circumference is

$$\text{Diameter} \times 3.1416 \qquad (1)$$

and the area of the complete circle is

$$\text{Diameter} \times 3.1416 \times \frac{\text{Diameter}}{4} \qquad (2)$$

and the area for the escape of steam is

$$\text{Diameter} \times 3.1416 \times \text{Lift}. \qquad (3)$$

When the lift is one-quarter the diameter, or

$$\frac{\text{Diameter}}{4},$$

the area for the escape of steam is the same as the area of the circle; formula 3 is the same as formula 2.

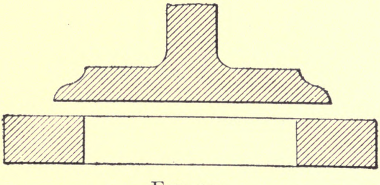

FIG. 71.

When a flat valve has lifted a quarter of its diameter it has reached the limit of its capacity to discharge steam. It doesn't do any good to lift higher, for the area around the edge of the valve is already as large as the area of the valve itself and the capacity of the valve is proportional to the area or the square of the diameter.

In practice, however, the lift of valves is much less than one-quarter of their diameter, and for a given lift the area for the escape of steam is proportional to the circumference or the diameter rather than to the area. Most of the rules, however, as above stated, allow a given amount of valve area to a square foot of grate surface, and make the allowance liberal enough to include all conditions. For instance, the rule of the U. S. Board of Supervising Inspectors calls for one-half a square inch of valve area for each square foot of grate surface. A 4-inch valve has about 12 square inches of area and would thus take care of 24 square feet of grate. It would not be possible to burn over 25 pounds of coal per square foot of grate per hour with natural draft, nor to evaporate over 12 pounds of water with a pound of coal, so that the boiler could not possibly make more than

$$25 \times 12 \times 24 = 7200 \text{ pounds of steam per hour,}$$

or

$$7200 \div (60 \times 60) = 2 \text{ pounds of steam per second.}$$

Now the weight of the steam which will escape through a given aperture per second is given by the following formula:

$$\text{Wt.} = \frac{\text{Pressure} \times \text{Area}}{70}$$

that is, the weight in pounds which will escape in a second is equal to the absolute pressure in pounds per square inch multiplied by the area in square inches and divided by 70.

On the other hand, the area required to discharge a given weight is

$$\text{Area} = \frac{\text{Weight} \times 70}{\text{Pressure}}$$

that is, the weight in pounds to be discharged per second multiplied by 70, and divided by the absolute pressure equals the required area. Now we have found that with a rate of combustion practically impossible, with natural draft, and a practically unattainable evaporation per pound of coal, the most steam that the boiler with 24 square feet of grate suface could furnish is 2 pounds per second. The area required to discharge this at 70 pounds pressure, absolute, is

$$\frac{2 \times 70}{70} = 2 \text{ square inches.}$$

The 4-inch valve which this boiler would require would have a circumference of practically 12 inches, and would need to lift only one-sixth of an inch to furnish the two square inches of opening necessary to discharge the steam, for

$$12 \times \tfrac{1}{6} = 2.$$

One-sixth of an inch is only one twenty-fourth of the diameter of the valve. You see that this simple rule gives an ample margin, requiring but a small lift to discharge more steam than the boiler can possibly make. It is altogether useless and nonsensical to figure the areas of opening to four places of decimals involving with beveled seats complicated operations with sines and cosines, in a calculation which involves no accuracy but which requires simply a result which shall be amply large to cover any emergency likely to be encountered in practice. It is like trying to measure the distance to the next town in feet and inches, in order to answer a man who would be abundantly satisfied to know that it was about three-quarters of a mile. You may be sure that a valve which has a square inch of area for each two square feet of grate surface will liberate all the steam that can be made by the coal that you can burn on that grate surface, so long as the valve is free and in good condition. It is quite probable that a smaller valve would do, but in a matter of this kind we want to provide not the smallest that will possibly do but enough capacity to be absolutely safe. For all purposes of ordinary practice, therefore, divide the grate surface by 2, which will give you the valve area required and you can find the corresponding diameter by multiplying the square root of the area by 1.128. Don't carry your decimals out too far because you will have to take the nearest commercial size after all.

Here is a rule which will give you the diameter of the valve in inches at once:

Multiply the square root of the grate surface by 0.8.

This would be particularly handy when the grate is square, or nearly so, for then the length would be the square root of the area.

You can see how the rule is made, or rather, makes itself.

By the supervising inspector's rule the valve area required equals the grate surface divided by 2.

$$\text{Area} = \frac{\text{grate surface}}{2}.$$

The diameter is the square root of the quotient of the area divided by 0.7854.

$$\text{Diameter} = \sqrt{\frac{\text{area}}{0.7854}}.$$

And since in this case the area equals one-half the grate surface the diameter will be the square root of one-half the grate surface divided by 0.7854.

$$\text{Diameter} = \sqrt{\frac{\text{grate surface}}{2 \times 0.7854}};$$

or,

$$\text{Diameter} = \sqrt{\frac{\text{grate surface}}{1.5708}}.$$

We can get rid of the square root in the denominator by finding it once for all. It is 1.25 very nearly. So our formula becomes

$$\text{Diameter} = \frac{\sqrt{\text{grate surface}}}{1.25}.$$

Dividing by 1.25 is just the same as multiplying by $\frac{1}{1.25}$, and as $\frac{1}{1.25} = 0.8$, the multiplication is easier, so we have

$$\text{Diameter} = \sqrt{\text{grate surface}} \times 0.8.$$

The grate surface will never be so large that the square root cannot be easily determined with sufficient accuracy mentally. If it is between 25 and 36 the root is between 5 and 6. The square of 7 is 49, of 8, 64, etc., so that by trial the root can be determined approximately. Here is an easy trick to get the square of a number with two figures ending in 5:

Multiply 1 *plus the left-hand figure by the left-hand figure, and annex* 25 *to the product.*

What is the square of 35?

The left-hand figure is 3. Three plus 1 is 4, and $4 \times 3 = 12$. Annex 25 and get 1225.

This rule works just the same when the 5 is a decimal, only in that case the annexed 25 is a decimal too, and will enable you to determine instantly by inspection the nearest number advancing by halves to the square root. As the sizes of safety valves advance by half inches, the nearest root determined in this way will be sufficiently accurate, as we have to take the nearest commercial size anyhow.

What is the square of 6.5?

Six plus $1 = 7$; $7 \times 6 = 42$; add 25, which in this case will be a decimal fraction, there being two places to point off, and get 42.25.

In this way you can square 1.5, 2.5, 3.5, etc., and this is as near as it is ever necessary to get a root in the above formula. Suppose, for instance, you had 58 square feet of grate surface. What is the square root? Seven times $7 = 49$, and $8 \times 8 = 64$. It must be between 7 and 8; $7.5 \times 7.5 = 56.25$.

That is near enough to 58. The square root of 58 is really 7.615. Multiplying this by 0.8 we get $7.615 \times 0.8 = 6.092$, which is practically a 6-inch valve. We should have got at the same result if we had taken the square root as 7.5, for $7.5 \times 0.8 = 6$.

When the grate surface is over 30 or 40 feet it is better to get the required capacity by putting on two valves than by using one large one. In fact it is a pretty good plan to have two safety valves anyway. There is a great deal of responsibility on that little appliance, and many of the most destructive of boiler explosions would have been avoided by an operative safety valve of sufficient capacity. So many little things can occur to make it hold against a destructive pressure, even when the attendant follows the usual directions to raise it from its seat daily, that prudence dictates the use of an auxiliary valve. It would be a remarkable coincidence if both stuck at the same time without criminal negligence.

The amount of opening of an ordinary lever safety valve is determined by the amount of surplus steam to be delivered. If the boiler is making more steam than is to be taken out of it the pressure will increase, and when it reaches an amount sufficient to overcome the weight of the ball, etc., the valve will be raised a little from its seat and the steam will escape. If the opening thus afforded is sufficient with the other drafts on the boiler (such as the supply to the engine, etc.) to allow all the steam the boiler is making to escape, the valve will not open any wider, but if not the pressure will continue to increase and force the valve open until the steam can escape as fast as it is made. As the surplus production of steam decreases, as by closing the dampers or a greater demand by the engine, the valve gradually settles down to its seat again.

On account of its greater lift and effective discharging area the pop valve is allowed by the Board of Supervising Inspectors three square feet of grate surface per inch of area instead of two, as with the ordinary lever valve.

We have seen that the escape of steam through an opening of given size is proportional to the absolute pressure. Twice as much steam will go out of an inch hole in a minute with 190 pounds behind it as with 95 pounds. It is presumed, for this reason, that the inspectors only require a square inch of valve area for every 6 feet of grate surface on boilers carrying a steam pressure exceeding 175 pounds gage.

It has been said that although the area effective for the escape of steam is not proportional to the area due to the diameter of the valve, and although the latter area is that used in the formula for capacity, the allowance is so liberal that it is practically useless to figure the former. It may be interesting, however, to know how to figure it, and a treatise on the safety valve would hardly be complete without directions for so doing.

With a flat valve we have already seen that the area for the escape of steam is the lift of the valve multiplied by its circumference. With a bevel-seated valve in which the valve does not lift out of the seat the area $A A$, Fig. 72, is that of a frustum of a cone, Fig. 73. Now to find this area the rule is to add the circumference of the greater circle to the circumference of the lesser $C D$; divide by 2, and multiply by the slant hight $C A$. In other words, to multiply the average length of the strip which would be made by flattening this surface out by the width of that strip. To work this rule out would take us too far into trigonometry, but the rule follows:

(1) *Multiply the diameter of the valve by the lift, by the stine of the angle of inclination and by* 3.1416.

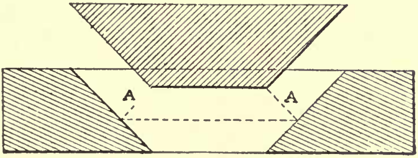

FIG. 72.

(2) *Multiply the square of the lift by the square of the sine of the angle of inclination, by the cosine of this angle and by* 3.1416.

(3) *Add these two products.*

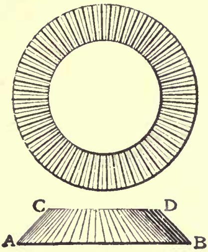

FIG. 73.

The U. S. rules require a bevel of 45 degrees, and most valves are made with seats of that degree of inclination. For such a valve the rule becomes:

(1) *Multiply the diameter of the valve by the lift and by* 2.22.

(2) *Multiply the square of the lift by* 1.11.

(3) *Add these two products.*

When a valve with a beveled seat lifts clear of the seat as a valve with a slight bevel may, the area of the opening is computed by the above rule for a lift which would raise it to the upper level of the seat, and to this is added the circumference of the valve multiplied by the lift above the seat level.

1	2	3			4		
Diam'r of Valve	Area of Valve	Weight of Valve and Stem			Pressure Required to Lift Valve and Stem		
In.	Sq. In.	Pounds			Pounds per Sq. In.		
$\frac{3}{8}$	0.1104	0.125			0.131		
$\frac{1}{2}$	0.1963	0.156		0.14	0.7947		0.713
$\frac{3}{4}$	0.4418	0.187		0.23	0.423		0.521
1	0.7854	0.187		0.34	0.238		0.432
$1\frac{1}{4}$	1.2272	0.312		0.60	0.254		0.488
$1\frac{1}{2}$	1.7671	0.437		0.75	0.247		0.424
2	3.1416	0.542	1.5625	0.97	0.172	0.497	0.308
$2\frac{1}{2}$	4.9087	0.8395	2.75	1.69	0.171	0.560	0.344
3	7.0686	1.339	3.50	2.33	0.189	0.495	0.329
$3\frac{1}{2}$	9.6211	1.8	4.75	2.60	0.187	0.494	0.270
4	12.5666	2.371	5.75	4.12	0.189	0.458	0.327
$4\frac{1}{2}$	15.9043	3.0	6.75	5.18	0.189	0.424	0.326
5	19.6350	4.125	9.75	6.45	0.210	0.497	0.324
6	28.2744	5.87	11.875	8.62	0.208	0.420	0.305

5			6			7			8		
Weight of Lever			Distance of Center of Gravity from Fulcrum			Pressure Required to Raise Lever			Pressure Required to Raise Valve, Stem and Lever		
Pounds			Inches			Pounds per Sq. In.			Pounds per Sq. In.		
0.125			3.25			5.89			6.003		
0.140		0.20	3.0		6.25	2.85		8.49	3.6447		9.203
0.343		0.38	4.812		9.0	4.98		10.32	5.403		10.841
1.0		0.48	7.75		9.0	8.32		5.50	8.558		5.932
1.125		0.65	7.312		8.8125	5.65		3.73	5.904		4.218
0.875		0.87	6.875		10.125	2.87		3.63	3.117		4.054
2.5	3.0	1.12	13.312	14.56	11.50	7.37	7.42	2.43	7.542	7.92	2.738
3.5	3.562	4.0	15.25	16.37	15.0	5.79	5.59	7.24	5.961	6.15	7.584
4.75	5.812	6.0	18.625	17.37	19.25	6.67	6.35	7.07	6.859	6.84	7.399
5.75	8.25	10.50	21.75	19.0	19.25	5.78	6.52	9.08	5.967	7.01	9.350
6.0	12.875	13.0	23.0	22.0	23.25	4.78	8.20	8.75	4.969	8.66	9.077
7.0	13.125	18.0	22.125	23.0	26.75	3.89	6.33	11.09	4.079	6.75	11.416
10.75	18.25	20.0	25.25	25.5	31.25	5.53	7.22	10.62	5.74	7.72	10.944
15.0	21.25	32.0	27.5	29.62	37.25	5.56	6.36	12.05	5.768	6.78	12.355

9			10					
Length of Lever			Distance of Stem from Fulcrum			Weight of Ball Ordinarily Furnished		
Inches			Inches			Pounds		
6.31			0.62			1.56		
5.62		12.5	0.75		0.75	3.2		2.0
9.50		18.0	0.75		0.75	5.5		2.6
14.87		18.0	1.19		1.0	8.12		4.8
14.4		17.625	1.19		1.25	9.62		11.0
13.4		20.25	1.19		1.37	15.37		14.0
26.1	29.50	23.0	1.44	1.87	1.687	19.0	30	24.0
30.0	33.0	30.0	1.87	2.12	1.687	29.0	45	34.5
37.12	35.0	38.5	1.87	2.25	2.312	38.0	63	50.5
43.1	38.375	38.5	2.25	2.50	2.312	48.5	88	67.0
45.5	44.5	46.5	2.37	2.75	2.75	70.0	110	86.5
43.75	46.125	53.5	2.5	3.0	2.75	83.0	140	86.5
49.87	51.50	62.5	2.5	3.25	3.0	98.0	168	103.0
54.5	59.50	74.5	2.625	3.50	3.5	139.0	220	139.0

X

HORSE-POWER OF BOILERS[1]

IN a recent catalog of a well-known maker of engineering specialties the following approximate rules for calculating the horse-power of various kinds of boilers were noticed and copied. The rules are intended for use in determining the proper sizes of injectors and other apparatus when the exact dimensions or heating surface of the boiler is unknown or hard to obtain:

KIND		H. P.
Horizontal Tubular	=	$\text{Dia.}^2 \times \text{Length} \div 5$
Vertical Tubular . .	=	$\text{Dia.}^2 \times \text{Hight} \div 4$
Flue Boilers	=	$\text{Dia.} \times \text{Length} \div 3$
Locomotive Type	=	$\text{Dia. of Waist}^2 \times$
Length over all	$\div$ 6.	

All dimensions to be in feet.

In the first and third cases the length is the length of the tubes or that of a "flush-head" boiler and does not include the extended smoke-box. In the second case, the hight is that of a plain vertical boiler in which the upper part of the tubes is above the water line; it is not the hight of a boiler with submerged tubes.

[1] Contributed to Power by C. G. Robbins.

The extreme simplicity of the rules aroused curiosity as to their accuracy, and comparisons were made between manufacturers' ratings and ratings calculated by the formulas above. The results are given in the accompanying table. They agree very closely, except in a few of the larger sizes of tubular boilers, where the calculated rating falls below that of the manufacturer. And in these sizes it will be noticed that the heating surface per horse-power is less than in the smaller sizes where the two ratings practically agree.

It is quite possible that the ratings of other manufacturers would show a better or worse agreement. In any event, the rules prove to be valuable for just what is intended and will save considerable trouble in measuring up and calculating the power of existing boilers when ordering injectors, feed pumps, and the like.

HORIZONTAL TUBULAR

Diameter (inches) × Length (feet)	36×8	36×10	42×10	42×12	48×12	54×13	54×15	60×14
Heating Surface	232	290	370	429	568	684	800	944
Manufacturers' Rating	15	20	25	30	40	50	60	70
Rating by Rule	14.4	18	24.5	29.4	38.4	52.7	60.8	70

Horizontal Tubular (Concluded)

Diameter (inches) × Length (feet)	60×15	60×16	66×15	66×16	72×16	72×18	78×18	84×18
Heating Surface	1010	1080	1249	1350	1674	1829	2115	2511
Manufacturers' Rating	75	80	90	100	125	150	175	200
Rating by Rule	75	80	90.8	96.8	115.2	130	152.1	176.4

VERTICAL TUBULAR

Diameter (inches) × Hight (feet)	24×5	30×5	30×6	36×7	40×7½	44×8	48×8	48×9	54×9
Heating Surface	58	84	100	180	238	292	358	480	578
Manufacturers' Rating	5	7	8	15	20	25	30	40	50
Rating by Rule	5	7.8	9.4	15.6	20.6	26.8	32	36	45.6

TWO-FLUE BOILERS

Diameter (inches) × Length (feet)	36×18	36×22	42×20	42×24	44×26	48×26	40×30	50×32	54×34	60×33
Heating Surface	206	253	285	338	393	415	478	542	610	663
Manufacturers' Rating	15	20	25	30	35	40	45	50	60	70
Rating by Rule	18	22	23.3	28.7	31.8	34.7	40	44.5	51	55

LOCOMOTIVE TYPE

Waist Diam. (in.) × Length (ft.)	40×14	42×14⅓	44×15	44×16⅓	48×17½	54×18½	56×19½	58×19¾	60×21	62×22¼
Heating Surface	268	309	350	400	503	596	710	809	898	1050
Manufacturers' Rating	25	30	35	40	50	60	70	80	90	100
Rating by Rule	25.9	29.3	33.6	36.6	46.7	62.5	72	77	87.5	90

XI

BOILER APPLIANCES AND THEIR INSTALLATION

In this paper it is my aim to briefly point out a few of the deficiencies which not only exist in so many of the less modern plants located in isolated places, but which are too often found in the large and perhaps otherwise well-equipped plants.

First in importance is the safety valve, which in some instances can be called such in name only, for in their neglected or overloaded condition they would not in any sense answer the purpose for which they are intended. We still find a few engineers who persistently stick to the old-style lever and weight safety valve — for what reason, we cannot say, unless it is because they can be overloaded more easily than the more modern spring-loaded pop valve. Certainly everything else is in favor of the pop valve, especially in the hands of incompetent persons, for the most successful design of any steam appliance is that one which is absolutely fool-proof. From the fact that they can be locked and made fool-proof, that they are much more reliable in their action and so much less wasteful of steam, it is believed they will soon be used universally, and that the lever valve will be a thing of the past. But with all their advantages they will not relieve the boiler of over-pressure unless properly installed and kept in operative condition.

Boilers have been seen equipped with these valves ample in size to take care of all the steam the boiler could generate, and then the discharge opening reduced to one-third the area of the valve and piped up through the roof. Again, as many as four 72×18 boilers have been seen all equipped with 4-inch pop valves, and all piped to blow into one continuous 4-inch header, which extended through the wall. There is no serious objection to piping the waste steam from a safety valve out of the building, when properly done, but the better plan would be to have a suitable ventilator in the roof, and let them discharge in the building. There are, however, many plants where this cannot be done. If the waste pipe is run out of the building, it should never be smaller than the valve itself, and if it is necessary to carry it any great distance, 20 feet or more, the pipe should be increased one size and connected up with as few turns as possible. It is also a very dangerous plan to run a waste pipe direct from the safety valve horizontally some distance, and then run a vertical pipe up through the roof, unless the pipe is properly supported to not only sustain its own weight, but to carry the downward thrust due to the reaction of the steam, which would in turn throw a severe strain on the casing of the valve and the flange bolts. The amount of pressure so exerted is of course a matter of conjecture, for the full boiler pressure could hardly be expected to be realized on the waste pipe. However, serious accidents are known to have happened from just such construction, therefore they are not mere possibilities. It is also quite necessary that the waste pipe be supplied with the proper opening for free and continuous draining, and not depend altogether on the drip opening on the valve itself.

Next in importance to the safety valve is the water column and its connections. There are probably more accidents to boilers traceable to defective water columns than to any other one cause. On a recent visit to a new plant where three 150 horse-power boilers were being installed, the water columns were found piped up with $\frac{3}{4}$-inch pipe, with several turns in the lower connection and with no blow-off pipe. With some feed waters this would probably answer the purpose, but water used for boiler purposes is often found which would close up the lower connection in a very few days' run. In this case, as in many others, the boiler makers were at fault, as they were furnishing the attachments. Water columns should never be connected up with pipes smaller than $1\frac{1}{4}$ inches, and in general practice $1\frac{1}{2}$-inch pipe is better, but in every instance the lower connection should be provided with a $\frac{3}{4}$-inch blow-off, and for convenience should be piped to discharge into the ash-pit. This blow-off may be provided with any good valve or cock, but if the latter is used, a closed end wrench should be provided, as an adjustable wrench is too apt to be carried away and the blowing out neglected. The removable disk Y-valves now on the market have been found very serviceable and reliable for boiler blow-offs, and no doubt they would be equally as good for water columns. There is quite a difference of opinion among engineers as to the advisability of placing stop valves in the column connection, but there is no real good reason advanced why they should be so equipped. Many plants with valves in both the lower and upper connections are found, but it is not a misstatement to say that one-half the lower valves can be found in an inoperative condition, owing to the accumulation of scale on the seat and valve. The less the number of attachments which may prove a source of danger the better. In connecting up water columns it is a good plan to use crosses in the lower connections, plugging the unused openings with gun-metal or brass plugs. These will be found very convenient for removing deposit which may accumulate and cannot readily be blown out. Water columns are often too small to give the best results. The chamber should be at least 3 inches, and preferably 4, in diameter, internally.

Ignorance is also often displayed in placing water-columns. A column placed too high is fully as dangerous in the hands of some men as one placed too low. In plants with the columns so placed it has been observed that when the water was just visible in the bottom of the glass there would be 6 inches of water above the top of upper row of tubes. The fireman, knowing this fact, will carry the water low in the glass, and if by chance the water should disappear from view altogether, he will tell you that it just went out of sight, and will then proceed to speed up the boiler feed pump. A better and safer plan is to set the column with the bottom of the glass just level with the top of the upper row of tubes, then pull or cover the fire before the water leaves the glass, in case of the failure of the water supply. Gage-glass valves and try-cocks are also too often neglected. While it is believed there is no better way of ascertaining the hight of water in boiler than by blowing out the water column and then noting the rapidity with which water returns to the glass, do not neglect the try-cocks, for the water glasses will break and at times when it is not convenient to replace them; then the try-cocks will come in handy, and should be found in working order.

The steam gage is next in importance, but, as a rule, receives little attention. The dial on the factory clock is kept clean, so there will be no mistake in reading the time when the whistle should be blown; but with the gage it is different; it has no such important duties to perform. A steam gage is not the delicate instrument that some would believe. However, their accuracy is easily destroyed, if not properly connected up. They should not be attached to the breeching or boiler front, unless protected from the heat, and they should be provided with a water trap to protect the Bourdon spring from the heat of the live steam; otherwise they will not give correct readings, and may be ruined altogether. The best form of trap is one made up of nipples and fittings, with a small drain cock placed in the lowest point of trap for the purpose of blowing out and of draining, to prevent freezing in winter. A trap of this kind will be found much easier to clean, in case of stopping up, than the bent pipe.

Notes

The use of cast-iron flanged nozzles connecting boilers to steam pipes is being superseded by dropped forge steel flanges. The former are objectionable, for the rivet holes must be accurately drilled and the curve must be a neat fit to the boiler plate or a calking gasket be provided to make the joint tight. Then such flanges will fail by fracture in riveting after some time and money has been spent on the job. If the cast nozzle is flanged to receive the steam pipe, the holes for bolts must be drilled out and bolts go with the nozzle. Many buyers object to superfluous flanges as that much more for the engineers to care for and pack. Then if the plant be a one or two boiler one the best plan is to have no flanges between the boiler and the main steam valve, for if a flange blows out the packing one can readily shut the main steam valve and repack it, while on the other hand one will wait until the boiler cools off. With regard to strength of material, while cast iron can be made, and undoubtedly is, to run as high as 25,000 pounds tensile strength, the fact is it may run less than that, and in calculating averages must be taken, which doubtless will not exceed 15,000 pounds per square inch. In addition allowance must be made for shrinkage strains in such castings; usually an unknown amount, but allowance should be given. If the flange be threaded the strength of such threads will of course not equal threads in forged steel. The expansion of the boiler shell sets up strains on these flanges, and while cast iron resists bending or flexure well, that is of no special value, as the strains are continuous and unavoidable. On the contrary dropped forged steel flanges furnished in all pipe sizes and to fit practically any required circle are now made and kept in stock by boiler supply houses, ready for attaching and tapped to size.

The tensile strength equals flange steel. Granting cast iron 15,000 tensile strength and forgings 60,000, note the wide difference in strengthening by reinforcement of the hole cut in the boiler shell. The steel flange may have rivet holes punched to within $\frac{1}{8}$-inch of size and reamed out without damage. It will "give" in fitting and riveting to the shell and no gasket is needed. A Fuller calking tool will quickly close any leak at the seam, but such rarely occurs, as it is forged on a smooth die. The threads are $1\frac{1}{2}$ inches deep on a 6-inch size, and are very strong. From all of the above, proved by experience, the steel flange is vastly better than the cast-iron one. Nevertheless many old men will not accept anything other than the latter, doubtless due to their opinions having formed and "froze in" years ago.

The foregoing applies to a large extent to cast-iron man-head frames, but as such are of larger size it is clear that the strains due to shrinkage and expansion under working pressure must be kept in mind as the casting is narrower at the cross-section of the minor ellipse than at major. Presuming the casting projects upward and inward to receive the plate, the usual type, it is likely shrinkage strains occur at the corners of the angles. Considering the large amount of the shell cut away for an 11×15 inch man-head, and allowing 15,000 tensile strength for castings, it is often the case that the weakest link in the chain composing the strength of a boiler is at this point. In my opinion engineers, designers and boiler makers should abandon cast-iron man-head frames, if for no other reason than to strengthen the boiler. Such frames should be, as high-class shops now use, weldless flange steel, forged into shape and double-riveted to the shell plate. No one thus far has seen a cast man-head frame double-riveted to the shell. Please note that point.

With an elliptical steel frame to save packing and also profanity, a $\frac{1}{2}$-inch ring should be shrunk on to the inner flange and planed off to give a seat one inch wide. But if packing costs money, then in the man-head plate have a groove provided so the packing cannot be forced out, and a piece of asbestos $\frac{3}{4}$ rope with plumbago and oil when the joint is opened will last two years. In one case with two boilers washed out every two weeks it cost $3.00 each opening for gaskets. The old plates were on my advice replaced with new ones grooved and the cost for gaskets reduced from $78 per annum to $2, a thrifty saving by the way.

Returning to the strength of the steel frame, it is so superior to the cast one in every manner that no comparison can be made. Nor indeed is it necessary to buy one particular type, as while these frames are patented, several being of about equal merit, prices are not kept at a high point. In view of the failures in riveting castings and including drilling it is doubtless true the steel frames like the steel nozzles are cheaper to the builder, while in every way each should be more desirable to the buyer, the engineer and the insurance companies.

To a large extent the above applies with equal truth to pressed steel lugs or brackets supporting the boiler, but in addition in shipping a boiler, while a steel lug may by transit be bent, it can easily be straightened and without injury — a valuable quality. When a boiler is set, the lugs being out of sight are out of mind. As the walls transmit heat it is clear the lugs become quite warm, for it is usual to protect them by the thickness of only one brick, and as the lug is covered outside, the heat is not lost but retained. More protection should be given under a lug, at least two courses of brick and an air space be open above the lug.

Reverting to strength, note that it is usual to have a space of 4 inches between the boiler and the side wall, and by properly carrying the brick out to the boiler the weight is transmitted over the entire seat of the lug. On the other hand, if this is not done, then one must make the lug stronger to allow for the 4-inch span. When of cast iron, we cannot, as stated, accept more than 15,000 pounds tensile strength per square inch, while if of steel we can take 60,000 tensile strength as the ultimate strength. Hence a $\frac{1}{4}$-inch steel equals a 1-inch casting; but the lugs are made, if of steel, of equal width with ordinary cast-iron lugs, and in addition the pressed ribs make it much wider. It is usual to have such lugs of $\frac{3}{8}$-inch plate, equaling castings of $1\frac{1}{2}$ inches in strength. Steel lugs are easily punched and fitted to boiler shells. There is life in them, as before reaching a breaking point through overload the give would be noticeable, while the casting would fail without warning.

From all of the above you will doubtless agree that in the modern steam boiler steel is winning its fight over cast iron through its superiority in strength and its adaptability for these purposes. In the up-to-date shops these arguments plus actual reduction in costs have led to its adoption.

XII

CARE OF THE HORIZONTAL TUBULAR BOILER[1]

ALTHOUGH the boiler room is the very heart of every steam plant, it is frequently the subject of the grossest neglect, and the instances in which it receives the care and thought to which it is entitled are very rare. Under the very best of conditions it is wasteful, but in a great many, in fact in the majority of cases, it is much more so than there is any necessity of. It must be admitted that the boiler room is necessarily a rather dirty and uninviting place, but that is no excuse for neglecting it.

In the engine room every possible economy is practised. Every foot of steam pipe is covered; the best grade of oil is used; the engine valves are set with the greatest care; belts are run as slack as possible; and many other points are watched in order to keep the steam consumption down to the lowest possible point. This is all very proper and good, and should be encouraged as much as possible, but in many cases far more serious losses are permitted in the boiler room, and it is these which can and should be stopped.

[1] Contributed to Power by M. Kennett.

A Common Source of Loss

A very common source of loss is the leakage of cold air through cracks in the settings. When flat plates from one side wall to the other are used over the rear of the combustion chamber, it is not unusual to find a space of from $\frac{1}{2}$ to 1 inch between the rear boiler head and the plate. This admits a large quantity of cold air to pass directly through the upper tubes, which are the most valuable for generating steam.

As a rule these openings are not caused by faulty setting of the plate, as these are usually well set, making a tight joint, before the boiler is fired up. But when the boiler becomes heated and expands, the plate is forced back, and when the boiler cools, a small space is left between it and the plate. Pieces of mortar and chips of brick lodge here, and when the boiler is again fired up and expands, the plate is forced still farther back. It is practically impossible to prevent these openings with this style of plate, but matters can be greatly improved by packing the crack loosely with waste which has been filled with soft fire-clay, as this forms an elastic packing which will not readily burn out.

The style of arch shown in Fig. 74 is practically free from this objection as it rests against the boiler head and follows its movements. This plan is open to the objection that the angle iron on the boiler head finally burns out, and in order to replace it the studs have to be removed from the head, and new ones put in, with the attendant trouble of making the job tight. Bearing bars are sometimes built into the side walls as a substitute for these angle irons, but they soon burn out, also. All trouble from this source may be overcome by using an extra heavy pipe as a bearing bar, and making it part of the feed line, so that water is being constantly pumped through it. Or, as in one case in mind, a small open tank may be provided for this purpose, the water circulating by gravity.

Numerous other cracks are constantly developing in various parts of the settings, and should be kept well filled with clay.

The combustion chamber back of the bridgewall should not be allowed to become filled with ashes to such an extent as to impede the draft.

Proper Depth of Combustion Chamber

There is a great difference of opinion concerning the proper depth of this chamber, some engineers claiming that it should be filled up level with the bridgewall, while others claim it should be quite deep. Much depends upon the kind of fuel used, the writer believing that when soft coal is used, it should be fairly deep to allow the unburned gases to become thoroughly mixed with the air and burn. An excellent plan is to slope the flame bed from almost the hight of the bridgewall to the ground in the rear, and pave it with firebrick. This brick paving becomes incandescent and ignites the gases and also reflects the heat upward toward the boiler. This style of chamber is easily cleaned out through a door in the rear wall at the level of the floor, and the ashes should be raked out every day.

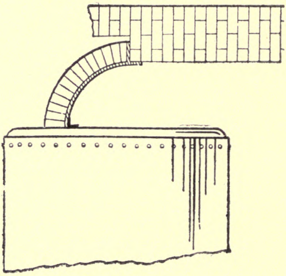

Fig. 74.

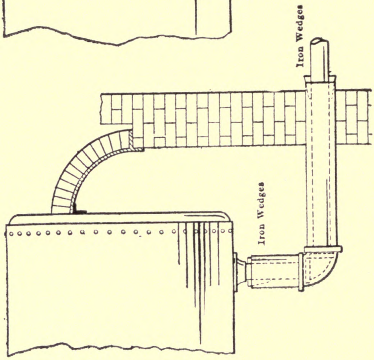

Fig. 75.

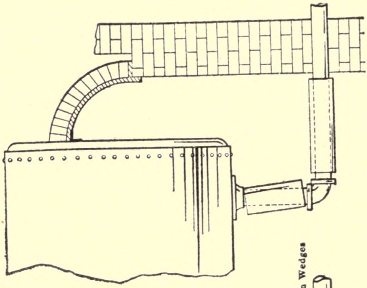

Fig. 76.

Do not make the mistake of placing this door one or two feet above the ground, as is frequently done, thus making it necessary to enter the chamber to clean it.

The hight and form of the bridgewall are also worthy of consideration. The principal object of the bridgewall is to keep the fire on the grates, and it should not be built up too close to the boiler, nor should it be curved to conform to the circle of the boiler shell. Such walls tend to concentrate the intense heat in the fire-box. This burns out the fire-door linings, and increases the danger of burning the fire-sheet, in case scale or sediment collects on it. They also prevent the free passage of a sufficient quantity of air into the combustion chamber to burn the gases.

We believe that all bridgewalls should be straight and not closer to the shell than 12 inches for 48-inch boilers, and 20 inches for 72-inch boilers. In many cases we believe these distances may be increased with advantage.

The necessity of keeping the tubes free of soot is pretty well understood, and this point usually receives proper attention. In addition to the usual blowing out with the steam blower, however, they should be thoroughly scraped at least once a week. The steam from the blower condenses to a great extent before reaching the rear end of the tubes; a great deal of the soot is simply moistened and left adhering to the tubes and soon burns into a hard scale which can only be removed by a thorough scraping.

Importance of Cleaning Boilers

Generally speaking, there are few operations about a steam plant which are so badly neglected as the cleaning of the boilers. The operation too often consists simply of letting the water out, removing the lower man-head, and washing the mud out with a hose. The natural result is that the heating surfaces, especially the tubes, become heavily coated with scale. This accumulates most rapidly at the rear head, and the space between the tubes soon becomes entirely choked for a short distance, preventing the free access of the water to the tube-sheet. This causes the tube ends to become overheated and they begin to leak. The only remedy is to remove the scale and reroll the tubes, but in order to remove the scale it is usually, or at least frequently, necessary to cut out some of the tubes.

The bagging of boilers due to the accumulation of scale and dirt is of such common occurrence as to require no discussion, other than to say that it is the result of improper cleaning.

Of course there are many cases in which, even with the best possible care, a great deal of scale will form, or where it is impossible to keep the boiler out of commission long enough to clean it properly. But there are also many cases in which a pretty thorough cleaning could be given if the engineer really wanted it done, and realized its importance sufficiently to see that it was done.

The number of boiler compounds which are guaranteed to keep boilers clean is legion, but still it will be hard to find the one which will remove the scale and hand it to the engineer, although this seems to be what some men expect of it. Most of them will do all that can be expected of them. They will soften and loosen the scale and considerable of it will drop off. After it is loosened the boiler cleaner should scrape it off by entering the top and bottom with suitable tools. Boiler compounds, like many other things, should be mixed with a good deal of common sense, then results will be obtained. When a boiler is badly scaled great care must be exercised in the use of a scale solvent, as it may cause considerable scale to drop off and bag a sheet. The action can be watched by frequent cleanings and, if there seems to be danger of such trouble, more frequent cleaning may be resorted to, or less solvent may be used.

An excellent plan is to use a scale pan, which is a shallow pan about four to six feet long and as wide as can be passed through the manhole. It is supported by light legs about three inches long, and is placed on the fire-sheet directly over the grates. As the scale falls it is caught by this pan and is thus kept off the sheet, preventing the bagging of the latter.

Oil a Source of Difficulty

Probably the most difficult thing to cope with in a boiler is oil. There are many different kinds of oil. Genuine crude petroleum is oil, but when properly used, it is difficult to find anything which excels it for keeping boilers free from scale. Kerosene is frequently used for the same purpose, and neither causes any trouble. The oil which we refer to, and which causes the most trouble, is the cylinder-oil carried over by the exhaust to an open heater or hot-well, and from thence into the boilers. This first appears at about the water line, and on the top tubes, where it gives no trouble, but it soon spreads over the entire heating surface, and it is surprising how little it takes to cause a very serious bulge on a fire-sheet.

A bulge caused by oil is different from one caused by scale or mud in that it usually covers considerable area, while the latter is not often over a foot or 18 inches in diameter, but is much deeper in proportion to its size. When a bulge is from three to five feet long, as those caused by oil usually are, there is nothing to be done but to put in a new sheet. This is an expensive repair, as it necessitates tearing down the brickwork in addition to the boiler work.

The best method of removing the oil from the feed-water is to filter it through a bed of coke and excelsior. This must be renewed from time to time, as it soon gets coated with the oil and becomes useless. There are numerous separators on the market guaranteed to extract the oil from the steam and water, but invariably better results have been obtained from the filter.

Faulty Blow-off Pipes

The records of a large boiler-insurance company show that there are more claims due to the failure of blow-off pipes than from any other single cause. A volume might be written on this, for the blow-off pipe certainly is a very troublesome, although necessary evil. When the feed-water is not introduced through the blow-off pipe, there is practically no circulation in it, and mud and sediment are very apt to collect. If the pipe is not protected from the direct action of the fire, this is very liable to cause it to burn and burst. Even though this results in no damage, it necessitates cutting out the boiler, and this may happen at a very inopportune time. If, however, the boiler is fed through the blow-off, the danger of such accident is reduced to the minimum, but even then it is better to protect the pipe from the direct action of the flame and gases.

It is a very common practice to incase the pipe in a sleeve formed of a pipe one or two sizes larger. This is of no value unless the sleeve is arranged as in Fig. 75, so as to allow a circulation of air between it and the blow-off pipe. When the sleeve is simply slipped over the pipe and allowed to hang loose, as in Fig. 76, it is of no value whatever.

In order to make it effective, the sleeve should inclose the entire pipe from the outside of the setting to within an inch or so from the boiler, and should be held in position by iron wedges, as shown in Fig. 75. This allows the cool air to traverse the entire pipe, being drawn in by the draft. While this is an excellent plan theoretically, it is open to the very serious objection that the sleeve rapidly burns out, and in order to renew it, the entire blow-off pipe has to be taken down. There is a cast-iron split sleeve made for this purpose, which can be replaced at any time without disturbing the pipe.

Perhaps as good a plan as any, all things considered, is to run the blow-off pipe straight down to the bottom of the combustion chamber and build a V-shaped fire-brick pier in front of it, just far enough away to allow removing the pipe and replacing it without disturbing the pier.

When necessary to use fittings in the combustion chamber, they should be of either cast steel or malleable iron, as cast iron is too liable to crack when exposed to high temperature.

Best Method of Feeding a Boiler

The method of feeding boilers has had a great deal of discussion, some advocating feeding through the blow-off, and some as strongly advising the top feed with a certain type of heater, the water passing through a length of pipe in the steam space of the boiler, and thus becoming heated to more nearly the temperature of the water in the boiler before discharging.

It is the general opinion the top feed is, generally speaking, the proper method; but circumstances must necessarily determine the best method for each particular case, and the writer has seen many cases where he has advised feeding through the blow-off. The objection to this method is that the comparatively cool water from the heater is discharged on the hot sheets. The water from the heater is hot, it is true, but when compared to that in the boiler there is considerable difference in temperature. It is seldom that the ordinary exhaust heater, except the most modern open heaters, raises the water to more than 175 degrees, while the temperature of the water in the boiler at 100 pounds pressure is 337 degrees. This is a difference of 162 degrees, or about the same difference as between boiling water and a block of ice. Now if this water is passed through twelve or fourteen feet of pipe in the steam space before it is discharged, its temperature will be raised, perhaps not very much, but at the end of this pipe it is discharged in the body of water in the boiler, and cannot come in contact with the sheets until it has mingled with and attained the temperature of this water. If an injection is used, or if there is no heater used in connection with the feed-pump, the top feed should be used by all means.

It is fully realized that with some waters this internal pipe soon chokes up, especially at the end, but usually this is readily cleaned, when the boiler is cleaned, and it may easily be made of sufficient area to run three or four weeks without giving any trouble.

The point of discharge for this pipe is largely a matter of personal preference, but it should be remembered that the sediment will collect worst at the point of discharge. A good plan is to have the pipe enter the front head just above the tubes and at one side of the boiler, carrying it to within two or three feet of the back head, and supporting it by brackets from the braces. From here let it run across to the middle space between the tubes, using a union near the end of this piece. Then drop two pipes between these tubes to the level of the lower tubes. This makes it very easy to clean these down pipes, by opening the union and removing them.

If there is no manhole below the tubes, so that the scale and sediment cannot be scraped from the shell and tubes at this point, it may be found better to discharge at the side near the rear and just below the water line.

If for any reason the blow-off pipe cannot be arranged so that it can be properly protected from the fire (and occasionally this is the case), and if a good heater is used, there is no great objection to feeding through the blow-off if that is the preference of the engineer. It is always well to have both systems installed so that if one fails the other may be used.

XIII

CARE AND MANAGEMENT OF BOILERS[1]

THERE has been a great deal written by different authors on the subject of care and management of boilers. Valuable advice has been given, yet boiler explosions and accidents still occur. Therefore, too much cannot be said to impress upon the mind of the stationary engineer the importance of taking care of boilers.

The first and most important thing to begin with is a good, sound boiler, for if the boiler is an old and dilapidated concern the best and most skilful engineer cannot make it safe and reliable, and the only advice in any case like this would be to have nothing to do with it, as not only his reputation as an engineer would be at stake but also his life and the lives of others.

When taking charge of a plant that has been run for some time the engineer should lose no time in ascertaining as far as possible the exact condition of the boilers, and at the first opportunity he should make an internal and external examination and see that they are free from scale and incrustation. If they are not, he should see that they are thoroughly cleaned both inside and outside of the shell. When a boiler is once

[1] Contributed to Power by John McConnaughy.

thoroughly cleaned the competent engineer will always resort to the proper means of keeping it so far as conditions will allow.

The accumulation of scale can be in a measure avoided by blowing small quantities of water from the bottom and surface blow-off, as all minerals held in suspension become of greater specific gravity than the water. When heated, the tendency by specific gravity is to settle toward the bottom while the lighter portions remain upon the top and float in the form of a scum. It has been found that by frequently blowing from the surface and bottom blow-off, much of the mineral substance which forms scale will be carried out before it can settle sufficiently to attach itself to the iron. By so doing, much of the trouble from scale may be avoided.

Notwithstanding all the care that may be taken, in some localities where the water is largely impregnated with minerals a certain amount of scale will accumulate in spite of the efforts of the most careful and experienced engineer. There are various devices and compounds on the market which have proved effective and in a measure beneficial for preventing this scale. Others are of a doubtful character; it is advisable before using a compound to have a chemical analysis made of the feed-water, as the nature of the supply receives too little attention.

Some engineers having charge of boilers with manholes under the tubes do all their cleaning from below the tubes and do not open the boiler on top. As it is impossible to wash all the dirt down from the top by washing from the under side of the tubes, the boiler is in bad condition above the tubes before they know it and they will tell you that the boilers are in good shape inside.

In cleaning boilers, all manholes and hand-hole plates should be taken out and the washing should be done from above and below the tubes. The engineer should then go inside the boiler and clean between them, so that any scale that has been lodged between the tubes can be taken out. On the outside, all seam heads and tube ends should be examined for leaks, cracks, corrosions, pitting and grooving. The condition of stays, braces and their fastenings should be examined. The shell of the boiler should be thoroughly cleaned on the outside, as soot is a bad conductor of heat, holds dampness and is liable to cause corrosion. All valves about the boiler should be kept clean and in good working condition. The pumps or injectors should be in the best working order. The connections between the boiler and water column and also the gage glass should receive the closest attention, but they are sadly neglected by some engineers. The brickwork should be kept in good condition and all air holes stopped, as they decrease the efficiency of the boiler and are liable to cause injury to the plates by burning.

There should be a good heater in connection with the boiler and the feed-water as hot as you can work it, for feeding cold water causes too much contraction and expansion. This causes vibration in the seams and makes them weak at those points. For example, if one hundred pounds of steam will do your work; never carry any more nor less, as the rise and fall in pressure causes expansion and contraction of the plates.

Never open the fire doors to cool your boiler. Close the ash-pit doors and open the smoke-box doors in case you get too much steam, as opening the fire door causes too much contraction by the cold air cooling the furnace. It would be better to allow steam to blow off from the safety valve, which will not in any way injure the boiler.

The safety valve should be raised from its seat every day to make sure it does not stick from any cause, and observe from the steam gage if the valve blows off at the pressure it is set for.

It is of the highest importance to keep the blow-off pipe free from sediment of any kind, as the pipe is liable to fill up and burn off, and the only way to keep it free is to open the blow cock often enough to keep everything flushed out.

The best time to blow off is in the morning before the fires have been started up, as a good deal of sediment in the boiler will then have settled to the bottom of the shell and much of it will pass out when the cock is opened. Noon is also a good time, after the fires have been banked for half an hour or more, so that the water in the boiler has been quiet long enough to deposit the particles that are being whirled about with it through all parts of the boiler.

When a blow-off cock is opened, it must be remembered that it is not to be yanked wide open and then closed the same way. This practice is very dangerous. No valve about a steam system ought to be closed suddenly, except in time of emergency, because the sudden strain on the pipe and fittings is liable to cause a rupture in the pipe or else break the elbow or valve.

The boiler is the life of any plant and my advice to all owners of steam plants is to keep a first-class engineer, one who is strictly temperate, pay him good wages, give him the necessary material, and his plant will get the proper care and management.

XIV

SETTING RETURN TUBULAR BOILERS

A GREAT improvement is made when we discard the old-time setting of return-tubular boilers, in which cast-iron brackets were supported by brick walls which are constantly crumbling away, for the substantial form of setting which is obtained by suspending return-tubular boilers from I-beams supported by cast-iron columns.

The accompanying Figures 77 and 78 show the setting of boilers in single or double batteries. In setting an even number of boilers, as six or eight in one setting, it is best to divide them into pairs so that not more than two boilers will be suspended between supports.

The principal reason for this is that when the large sizes, such as from 150 to 250 horse-power are used, the size I-beam required to safely carry this load between supports is so large that it overbalances the cost of two or more cast-iron columns.

In setting an odd number of boilers, such as three or five, in a battery, columns are usually placed between each boiler with a 2-inch air space all around the column and an air duct at the bottom of the setting which runs through from the front to the back and connects with

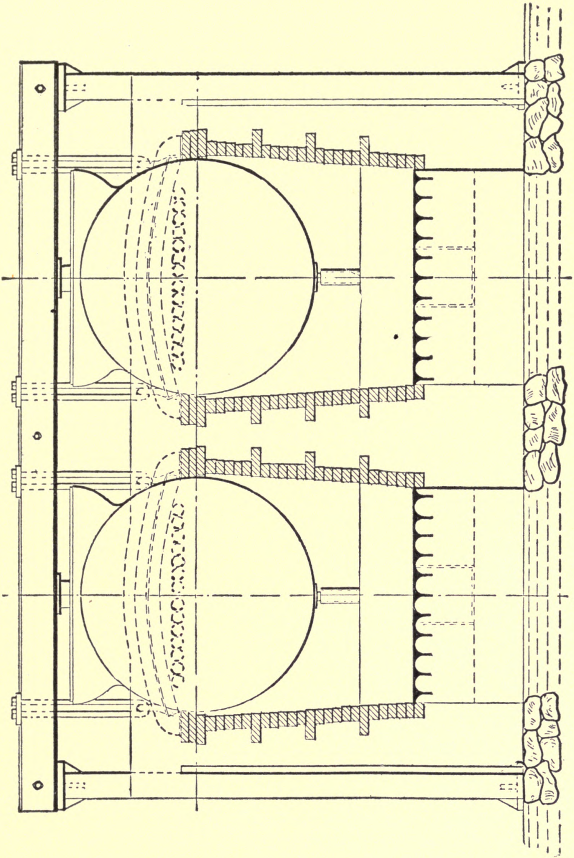

Fig. 77.

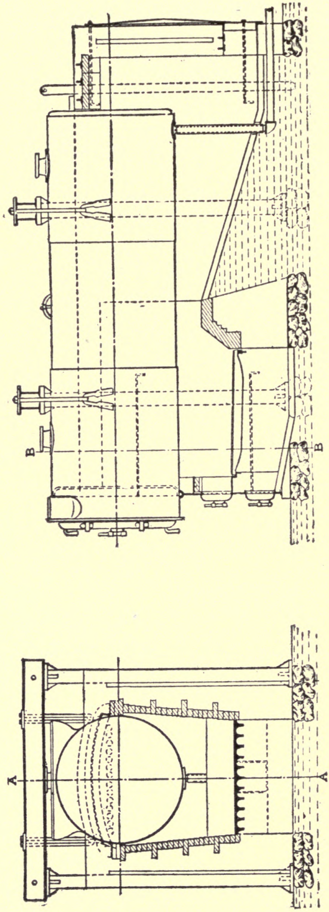

Fig. 78.

each air space around the column. This keeps up a circulation of air and the columns are kept comparatively cool.

In setting boilers in this manner the columns and I-beams are set in position first. Then the boiler is hoisted to the proper hight by means of tackle which is fastened to the I-beams and when the boiler is brought to the proper hight the U-bolts are slipped into place and fastened by nuts and washers to the I-beams. This does away with the use of blocking and barrels which are generally used and leaves all the space clear under the boilers.

The expansion is easily taken care of by the U-bolts and hangers, as is shown in the setting plans, and if the walls are properly set, they should show no cracks as they carry no weight and are entirely free.

The accompanying table has been carefully worked out with a factor of safety of 5 and gives the different lengths and sizes of I-beams and columns required, so that a person estimating on a job of this kind can readily determine the cost of such a setting. It covers boilers from 36 inches in diameter and 8 feet long to 90 inches and 20 feet, giving the total weight to be supported and the sizes, weights and positions of columns and I-beams required.

SIZES AND WEIGHTS OF COLUMNS AND I-BEAMS RE

HORSE POWER	15	20	25	30	35	40	45	50	60
Dia. of boiler in inches	36	36	42	42	44	48	50	54	54
Length of tubes in feet	8	10	10	12	12	12	13	13	15
Length of curtain sheet in inches	11	11	12	12	12	14	14	14	14
Total weight of boiler and water	6500	7500	9400	10500	11500	13300	14200	15300	17800
Rear head to center of hanger	2–0	2–6	2–6	3–0	3–0	3–0	3–3	3–3	3–9
Center to center of hangers	4–0	5–0	5–0	6–0	6–0	6–0	6–6	6–6	7–6
Front head to center of hanger	2–0	2–6	2–6	3–0	3–0	3–0	3–3	3–3	3–9
Distance between C of supports (1 boiler)	6–6	6–6	7–0	7–0	7–2	7–6	7–8	8–0	8–0
Distance between C of supports (2 boilers)	11–8	11–8	12–8	12–8	13–0	13–8	14–0	14–0	14–8
Length of I-beam for 1 boiler	7–3	7–3	7–10	7–10	8–0	8–4	8–6	8–10	8–10
Length of I-beam for 2 boilers	12–6	12–6	13–8	13–8	14–0	14–8	15–0	15–10	15–10
Size of I-beam required for one boiler	4	4	5	5	5	6	6	6	6
Size of I-beam required for 2 boilers	6	6	8	8	9	9	9	10	10
Weight per ft. of I-beam for one boiler	7.5	7.5	9.75	9.75	9.75	12.25	12.25	12.25	12.25
Weight per ft. of I-beam for two boilers	12.25	12.25	18	18	21	21	21	25	25
Length of cast-iron column	8–0	8–0	8–6	8–6	8–8	9–3	9–5	10–0	10–0
Outside Dia. of C. I. col. for one boiler	4	4	4	4	4	5	5	5	5
Outside Dia. of C. I. col. for two boilers	5	5	5	5	5	6	6	6	6
Size of flange on ends of col. for one boiler	9½	9½	10	10	10	10½	10½	10½	10½
Size of flange on ends of col. for two boilers	10½	10½	12	12	12½	12½	12½	13½	13½
Thickness of C. I. col. for one boiler	½	½	½	½	½	½	½	½	½
Thickness of C. I. col. for two boilers	¾	¾	¾	¾	¾	¾	¾	¾	¾

QUIRED IN SETTING RETURN TUBULAR BOILERS.

70	75	80	90	100	125	150	175	200	200	225	225	250
60	60	60	66	66	72	72	78	78	84	84	90	90
14	15	16	15	16	16	18	18	20	18	20	18	20
16	16	16	17	17	18	18	18	18	20	20	22	22
20800	24800	27200	30300	35000	40000	44000	48000	56000	55000	67000	65000	75000
3–6	3–9	4–0	3–9	4–0	4–0	4–6	4–6	5–0	4–6	5–0	4–6	5–0
7–0	7–6	8–0	7–6	8–0	8–0	9–0	9–0	10–0	9–0	10–0	9–0	10–0
3–6	3–9	4–0	3–9	4–0	4–0	4–6	4–6	5–0	4–6	5–0	4–6	5–0
9–0	9–0	9–0	9–6	9–6	10–0	10–0	10–6	10–6	11–0	11–0	11–6	11–6
16–2	16–2	16–2	17–2	17–2	18–2	18–2	19–2	19–2	20–2	20–2	21–2	21–2
10–0	10–0	10–0	10–6	10–6	11–0	11–0	11–7	11–7	12–0	12–0	12–6	12–8
17–4	17–4	17–4	18–4	18–4	19–5	19–5	20–6	20–6	21–6	21–6	22–6	22–6
7	7	7	7	7	8	8	9	9	9	9	9	10
12	12	12	12	12	15	15	15	15	15	15	15	15
15	15	15	15	15	18	18	21	21	21	21	21	25
31.5	31.5	31.5	40	40	42	42	60	60	60	80	80	80
10–8	10–8	10–8	11–2	11–2	12–0	12–0	12–6	12–6	13–0	13–0	13–10	13–10
5	5	5	6	6	6	6	6	6	6	6	6	6
6	6	6	6	6	8	8	8	8	8	8	8	8
11½	11½	11½	11½	11½	12	12	12½	12½	12½	12½	12½	13½
14	14	14	14½	14½	15	15	16	16	16	17	17	17
¾	¾	¾	¾	¾	¾	¾	1	1	1	1	1	1
¾	¾	¾	1	1	¾	¾	¾	¾	1	1	1	

XV

RENEWING TUBES IN A TUBULAR BOILER[1]

While the renewal of boiler tubes is properly the work of the boiler maker, the engineer who knows how to and can do it is just so much more valuable to the employer. The purpose of this article is to describe the method employed, together with the tools required.

First, it is essential to place a distinguishing mark on the front and rear heads to show which tube is to be cut out, using chalk or soapstone for the purpose, and the best way to make sure that the helper at the other end of the boiler marks the same tube that you do is to run through a strip of wood four or five inches longer than the tube. As such a strip is of use farther along in the process it is well to make a length of $\frac{7}{8} \times 2$-inch pine to serve both purposes. Next, with a hammer and a heavy cape chisel having a wide cutting edge, which is less liable to cut or mar the boiler (see Fig. 79), face the beads on both ends of the old tube until they are flush with the heads of the boiler. Then, at the front head, with a diamond-point chisel such as is shown in Fig. 80, cut a slot or channel, $\frac{1}{16}$ inch wide, in the

[1] Contributed to Power by J. E. Sexton.

bottom of the tube, extending inward to about $\frac{3}{8}$ of an inch beyond the inner edge of the head, making sure that the groove is cut in the tube only and that the head

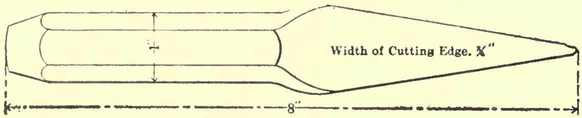

FIG. 79.

is not cut or even marked by the chisel. *Do not drive the chisel clear through the tube, either.*

With an offset chisel, Fig. 81, carefully turn up the

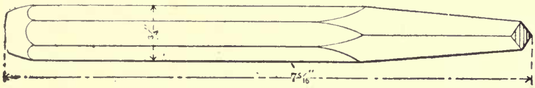

FIG. 80.

edges of the tube at both sides of the cut, until the tube-end resembles the condition shown in Fig. 82, when it will be found that this end of the tube has

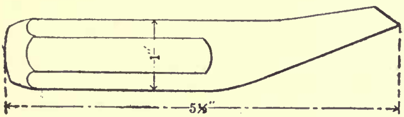

FIG. 81.

been released from the head. In cutting the slot, especially after the cutting edge of the chisel has gone beyond the thickness of the head, if the chisel is allowed to go through the tube it will be the source of considerable trouble, as it will cause the tube to spread. Hence, at this point extreme care must be used.

If a tube is corroded and muddy, it will be harder to remove and the method will have to be changed somewhat. Considerable force is required sometimes to remove such a tube. Instead of one slot in the bottom of the tube, two are cut, about $\frac{3}{8}$ of an inch apart, and

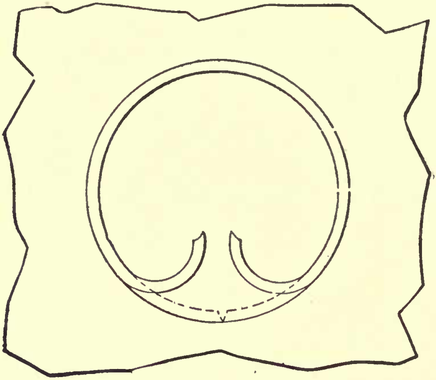

FIG. 82.

the offset chisel is used as before, except that the $\frac{3}{8}$-inch piece is turned up until it looks like the letter C, with its back toward the front of the boiler. Then proceed as before, turning the edges of the cut upward as far as they will go. A hook on the end of a chain or rope may then be inserted in the loop formed by the C-piece. This takes care of the front end.

At the other end of the tube insert the end of a piece of shafting about 10 inches long and a little smaller in diameter than the outside diameter of the tube.

The end of this shafting should be turned so that it will enter the tube about one inch, with an easy fit, and by giving a few taps on the outer end of this improvised mandrel the tube will be loosened at this end. Then, by working the tube backward and forward it can be released altogether.

The next step is to mark the new tube so it can be cut to length. Insert the $\frac{7}{8}$x2-inch piece of pine into the holes the old tube came out of until one end of the strip extends through the rear head about $\frac{5}{32}$ of an inch. Hold it there and proceed to place a mark on the end extending from the front head $\frac{5}{32}$ of an inch from face of the head. This gives the proper length to which to cut the new tube. Then, while the tube is being cut to length, take a half-round second-cut file, or a finish-cut file, and carefully smooth up the heads around the holes, removing any marks or cuts which may have been made in taking out the old tube. This is to prevent future leaks. Next, push the new tubes into place and station the helper at the rear end with a tube expander, being sure that the ends of the tube are equidistant from the heads. It is advisable to insert one end of an 8-foot section of 1-inch pipe in the front end of the tube, for a distance of 12 inches or so, and exert a downward pressure on the lever so provided to prevent the tube from turning while the rear end is being expanded. As soon as the tube is tight at the rear end, proceed to expand the front end.

A self-feeding expander, Fig. 83, will give good results, especially if a ratchet wrench is used to turn the spindle, for one can tell by the feeling just when to stop expanding. A monkey wrench will do, however, if a ratchet wrench is not available.

The beading comes next. This requires a special tool similar to that shown in Fig. 84. Place the long prong of the tool inside the tube, with the short prong pressing against the tube-end. Then bead the tube-end

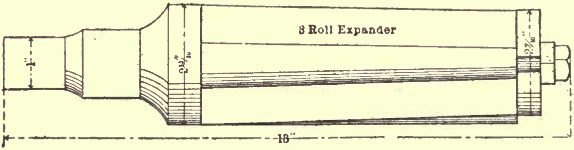

FIG. 83.

thoroughly *throughout the circumference*, for if it is only beaded here and there it will prove very unsatisfactory. When both ends are beaded, use the expander lightly in each end once more, to remove the marks made by the beader.

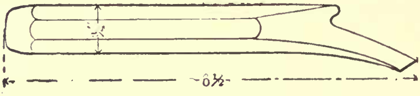

FIG. 84.

If both hand-hole plates are tight and the blow-off valve works O. K., fill the boiler with either hot or cold water, until the tube is covered, and if the tube does not leak water it will hold steam, and the boiler is ready to put into commission. If the tube leaks, re-expand it very lightly. Ordinarily, a man and helper can renew a tube in an hour, with ease.

XVI

USE OF WOOD AS FUEL FOR STEAM BOILERS[1]

In nearly all plants where lumber or wooden articles are the finished products, wood is used as a fuel for the boilers, because it is a refuse and is easily and cheaply disposed of in that manner. In some plants the amount of this refuse is greater than can be burned under the boilers; in others, there is not enough waste to furnish the steam required.

This is the case in a great many wood-working industries, and in some sawmills on the South Atlantic coast. To this class of industries this article is directed.

A certain wood is a good fuel or a poor fuel, depending on (1) the moisture contained and (2) the size of the pieces as fired. Whether it burns well under the boiler depends on the shape of the furnace, the method of firing and the draft of the chimney.

Calorific Value of Various Woods

The main idea to be shown in this section is that the value of all woods is about the same, depending on the amount of moisture contained.

[1] Contributed to Power by J. A. Johnston.

In various works of reference, the weight of one cord of different woods (thoroughly air-dried) is about as follows, the quality of coal not being given:

Hickory or hard maple — 4500 lb. equals 1800 lb. of coal (others, 2000 lb.).

White oak — 3850 lb. equals 1540 lb. of coal (others, 1715 lb.).

Beech, red and black oak — 3250 lb. equals 1300 lb. of coal (others, 1450 lb.).

Poplar, chestnut and elm — 2350 lb. equals 940 lb. of coal (others, 1050 lb.).

Average pine — 2000 lb. equals 800 lb. of coal (others, 925 lb.).

Referring to the figures last given in each case in connection with "others," it is said:

"From the above it is safe to assume that $2\frac{1}{4}$ pounds of dry wood are equal to 1 pound of average quality soft coal, and that the fuel value of different woods is very nearly the same, that is, a pound of hickory is about equal to a pound of pine, assuming both to be dry."

It is important that the woods be dry in the comparison, as each 10 per cent. of water or moisture in the wood will detract about 12 per cent. from its fuel value.

Take an average wood of the chemical analysis: Carbon, 51 per cent.; hydrogen, 6.5 per cent.; oxygen, 42 per cent.; ash, 0.5 per cent. If perfectly dry, its fuel value per pound, according to Dulong's formula,

$$V = \left[14{,}500C + 62{,}000\left(H - \frac{O}{8}\right)\right]$$

is 8170 B.t.u. The calorific value of carbon equals 14,500 B.t.u., and the calorific value of hydrogen equals 62,000 B.t.u.

The hydrogen in the fuel being partly in combination with the oxygen, only that part not in such combination can be counted on as a fuel, hence the factor

$$\left(H - \frac{O}{8}\right).$$

If this wood as ordinarily dried in air contains 25 per cent. moisture, then the heating value of a pound of such wood is $8170 \times 0.75 = 6127$ B.t.u., less the heat required to raise the $\frac{1}{4}$ pound of water from atmospheric temperature to steam, and to heat this steam to chimney temperature. Say, for instance, it takes 150 B.t.u. to heat the water to 212 degrees and 966 B.t.u. to evaporate it to steam, and 100 B.t.u. to raise the temperature of the steam to chimney temperature; in all 1216 B.t.u. per pound or 304 B.t.u. per $\frac{1}{4}$ pound. The net value of the wood as a fuel would then be $6127 - 304 = 5824$ B.t.u., or about 0.4 that of 1 pound of carbon. This method can be applied to any wood, knowing its chemical analysis and its percentage of moisture as burned.

The Moisture Content

As nearly all woods have about the same chemical analysis, the heat value of woods depends, as before mentioned, almost entirely on the moisture contained in the wood when burned. When newly felled wood contains a proportion of moisture which varies much in different kinds and different specimens, ranging between 30 and 50 per cent., and averaging about 40 per cent. Perfectly dry wood contains about 50 per cent. of carbon, the remainder consisting almost entirely of hydrogen and oxygen in the proportion which forms water. The coniferous (pines) family contains a small quantity of turpentine, which is a hydrocarbon. The proportion of ash in wood is from 1 to 5 per cent. The total heat of combustion in all woods is almost exactly the same, and is that due to the 50-per cent. carbon.

American woods vary in percentage of ash from 0.3 to 1.2 per cent., and the heat value ranges from 6600 B.t.u. for white oak to 9883 for long-leaf pine, the fuel value of 0.5 pound of carbon being 7272 B.t.u.

In the absence of any method of determining the heating value of a certain wood, the following are averages of the analyses of beech, oak, birch, poplar, and willow:

Carbon, per cent	49.70
Hydrogen, per cent	6.06
Oxygen, per cent	41.30
Nitrogen, per cent	1.05
Ash, per cent	1.80

These can be used in the foregoing formula, and will give an approximate value for nearly all American woods.

A very good and fairly accurate approximation of the amount of moisture in any particular sample can be obtained by weighing the wood (say about 10 pounds of it), and then placing the sample in a closed vessel with a small hole in it to allow the steam to escape. Subjecting the whole to a temperature of about 220 degrees until all the moisture has been driven off, weigh the sample again, and the percentage of moisture in the original can be computed easily. With this percentage known, the subtraction for moisture present can be made, as before shown, and an approximate value of the sample is obtained.

Nearly all woods will give a heat value, dry, of about 8200 B.t.u. Having obtained the percentage of moisture present, the heat value of the fuel is 8200 multiplied by (100 per cent. — per cent. moisture) less (heat required to raise water contained to evaporative point) less (heat required to evaporate water) less (heat required to heat steam made by this water to chimney-gas temperature). All of the latter quantities can be obtained from steam tables.

Easiest Method for Getting at the Heat Value

Probably the easiest and most accurate of all methods of obtaining the heat value of a certain specimen of wood is not to inquire into the chemical analysis, but to take a sample of the wood just in the condition in which it is burned, place it in a closed, air-tight vessel, and keep it there until it is brought to a calorimeter. This instrument should be used by one who is familiar with its use. It will give the heat value expressed as B.t.u. per pound, dry. The percentage of moisture being found, the correction for moisture is made as before.

A case of this kind, taken from a report by the writer, may be mentioned and calculated. The fuel was sweet gum refuse from a veneer mill, run through a hog and ground into chips approximately the size of a man's little finger. The logs were brought to the mill by rafting down a river, so the chips as fed to the boilers were not out of the water over three-quarters of an hour. A sample of chips was weighed wet, then dried in a closed vessel and weighed again, giving a moisture percentage of 47.50. A sample of the dried wood was then ground and tested in a calorimeter, giving a heat value of 8208 B.t.u. per pound, dry.

For every pound of the wood fired there was only $8208 \times 0.525 = 4309$ B.t.u. given up by the wood in burning, for there was but $1.00 - 0.475 = 0.525$ pounds of dry wood fired for 1 pound of fuel.

One pound of water requires 966 B.t.u. to evaporate it at the pressure in the furnace. There was 0.475 pound of water in the 1 pound of fuel fired, so that $966 \times 0.475 = 461$ B.t.u. were required.

The flue-gas temperature was 340 degrees and $340 - 212 = 128$ B.t.u. required to bring the steam from the boiling point to the chimney temperatures. The available heat in the wood was, then, $4309 - 461 - 128 = 3702$ B.t.u.

Probably the greatest chance of error in estimating the value of a wood as fired is to neglect the above calculation, because the difference between its heating value dry and its heating value as fired is often as high as 50 per cent., while a similar calculation for coal would give a comparatively small difference.

After computing by either of the methods given the heat value of the fuel to be burned, it is easily computed how much water can be evaporated per pound of fuel, and knowing the amount of available fuel, the power to be generated at any plant under consideration may be estimated.

Referring again to the same case of wet gum chips, this fuel was brought to the boiler house by a conveyer, the average capacity being measured at 100 pounds per minute, or 6000 pounds per hour brought to the boilers.

A pound of water requiring 966 B.t.u. to evaporate it, and each pound of the fuel having 3702 B.t.u., $3702 + 966 = 3.83$ pounds of water per 1 pound of fuel, with 6000 pounds of fuel per hour, the maximum quantity of water that could be evaporated by the boilers at 100 per cent. efficiency would be $6000 \times 3.83 = 22{,}980$ pounds per hour.

If the boiler were 70 per cent. efficient, $22{,}980 \times 0.70 = 16{,}100$ pounds of water per hour evaporated from and at 212 degrees is all that could be expected, and as $34\frac{1}{2}$ pounds of water per hour evaporated from and at 212 degrees is the equivalent of one boiler horse-power, the evaporation given would represent $16{,}100 + 34\frac{1}{2} = 465$ boiler horse-power. Under test the boilers gave 450 boiler horse-power.

From all that goes before, it appears that wood as a fuel has been allowed a little too high a value, inasmuch as it is rarely if ever fed to the boilers perfectly dry. It is generally green, and in cases of sawmills located on the banks of navigable streams is soaked with water.

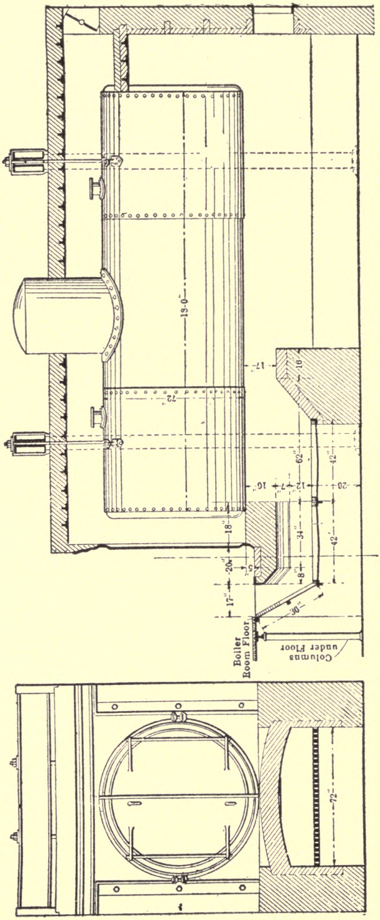

FIG. 85.

Air-drying of wood extracts about one-half of the moisture in a year. Wood perfectly dried, and then exposed to the air, will absorb about the same amount of moisture that it would contain after being thoroughly air-dried. However, when wood is to be used as a fuel, it is almost out of the question to contemplate drying it, so the proposition is to burn the fuel available in the best manner.

Fuel Available

Of course there cannot be given any even approximate method of calculating the amount of fuel that will be available in the refuse from any contemplated plant, for each and every one is to work under different conditions.

In plants already built, an estimate can be made by weighing the fuel brought to the boiler room, and by foregoing methods determining heat value, the available horse-power can be computed.

Most sawmills furnish enough refuse in slabs to run the boilers required to operate the plant. Woodworking plants, sash, blind and door manufactories, furniture factories, etc., depend entirely on the kind of product, as to the amount of scrap.

This will also depend largely on the plant at which the installation is contemplated. Furniture factories, woodworking plants, etc., generally work the kiln-dried lumber up so closely that the refuse as it comes to the boiler is already in an easily burnable condition, that of sawdust, shavings, or small strips or blocks. These can be fed directly to the furnace without further preparation.

In most sawmills where the slabs come off of the logs in long pieces, it is not possible to get the fuel to burn easily if fed as slabs, so it is often and generally in the sawmills on the Atlantic coast fed through a hog which grinds the slabs into chips varying in size from a man's three fingers to one finger or smaller.

This is undoubtedly the best way in which to introduce this fuel to the boiler, for it is then easily handled by conveyers, and can be dumped directly into the fire without any manual work, while slabs will generally require handling, unless some extra design is prepared to meet the case.[1]

The various forms in which wood is fed to the furnace may be summarized as: Cordwood, shavings, sawdust, dust from a hog, strips and blocks from a factory, and tan-bark.

Kind of Furnace Required

Efficient burning of wood requires a large combustion chamber, and grates arranged to prevent admission of a surplus of air. This cannot be obtained to good advantage in the usual coal-burning furnace, so the dutch oven has been developed to meet requirements. This is an extension of the fire-box in front of the boiler, as shown in Fig. 86, with a firing hole in the top through which the ground fuel or sawdust can be fed directly from the conveyer or chute to the grate.

As wood fuel is generally wet, or contains a large amount of moisture, the conditions of success, as pointed out by Thurston, are: To surround the mass so completely with heated surfaces and with burning

[1] A case of this kind is mentioned in Power, November, 1907.

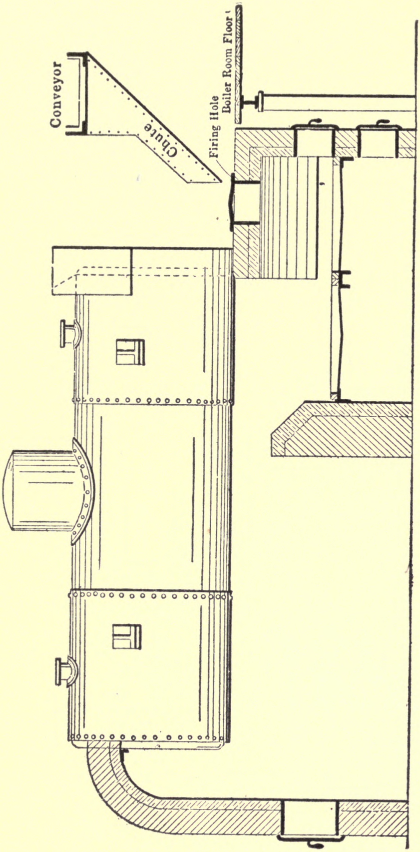

FIG. 86.

fuel that it may be rapidly dried, and so arranging the apparatus that thorough combustion may be then secured, the rapidity of combustion being precisely equal to, and never exceeding, the rapidity of drying. If the proper rate of combustion is exceeded, the dry portion is consumed completely, leaving an uncovered mass of fuel which refuses to take fire.

These conditions are met in the dutch oven, because of the fact that the fire is completely surrounded by fire-brick walls, which become heated to a very high temperature, especially in the case of burning pine shavings. This condition of good burning has been so well met in some cases that the fire-brick lining could not withstand the high temperature longer than a month.

The dutch oven has a firing door and an ash door on the front; the firing door may be used to fire any large pieces of wood, but results are best where the fire door on the front is never opened except for cleaning.

Fig. 87 shows an arrangement where the fuel consisted almost entirely of kiln-dried refuse from a woodworking plant, coming to the boilers in short sticks from about $\frac{1}{2} \times \frac{1}{2} \times 12$ inches to blocks $1 \times 3 \times 10$ inches, all mixed with sawdust and shavings from planers.

In the boiler room the floor is on an exact level with the top of the dutch oven. The fuel is dumped from a conveyer on this floor and shoved by hand into the holes on top of the ovens, and as the holes are kept full of fuel all the time, the doors over them are never closed. The boilers are of the Heine water-tube type,

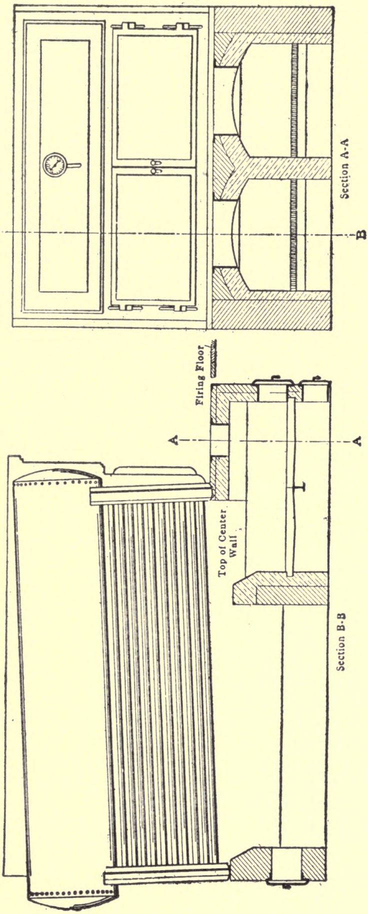

FIG. 87.

arranged in a battery of three and each rated at 300 horse-power. This installation gives perfect satisfaction.

Another form of combustion chamber, shown in Fig. 85, is very satisfactory for burning sawdust with a small mixture of shavings. The grate must be kept covered all the time, or too much air will get through, thereby decreasing the efficiency of the boiler. In this case the fuel is fed in a constant stream from a chute and is shoved back over the grate by a man on the firing floor.

For ordinary air-dried cordwood, a good grate is one placed at the firing-floor level, the area of grate being reduced to about two-fifths the amount required for coal by sloping the furnace walls inward, beginning just under the arch. The grate is, of course, at the bottom, and the cordwood can be carried to a depth of 30 to 36 inches, so that the freshly fired wood will crowd down that which is partly burned, filling the large interstices at the bottom with burning coals, and preventing leakage of air past the fire.

Miscellaneous Points

In handling any kind of wood fuel, it is better, even in small installations, to have the fuel brought by some carrier, as a conveyer, chute or air blast, to the furnace. With dry wood in small pieces, as dust from a hog, or shavings, the fuel being brought to the fire-room, one man can care for about 300 horse-power of boilers. If it is brought right over the firing hole to a dutch oven by an overhead carrier, he can care for, in some cases, 500 horse-power.

Sawdust and dry shavings are very extensively handled by blowers, the suction of the blower being connected to the saw frame or planer, and the refuse being blown into a receptacle over the boiler room. It is then dropped by chutes directly into the fire, or may be blown directly in by the blast furnishing air for the fire.

A chimney could be designed from theoretical calculations involving the chemical composition of the wood to be burned, but as a plant burning wood is rarely or practically never run on a weight basis, this would not be a practical method.

It has been borne out by practice that a chimney designed for a certain horse-power for bituminous coal will work well for wood. The accompanying curves were calculated from Kent's formula:

$$\text{H. P.} = 3\tfrac{1}{2}\ \text{E.}\sqrt{\text{H.}}$$

In Figs. 88 and 89, with any boiler horse-power and any suitable hight, the area of stack can be found. In Fig. 90 this area is expressed for round or square stacks.

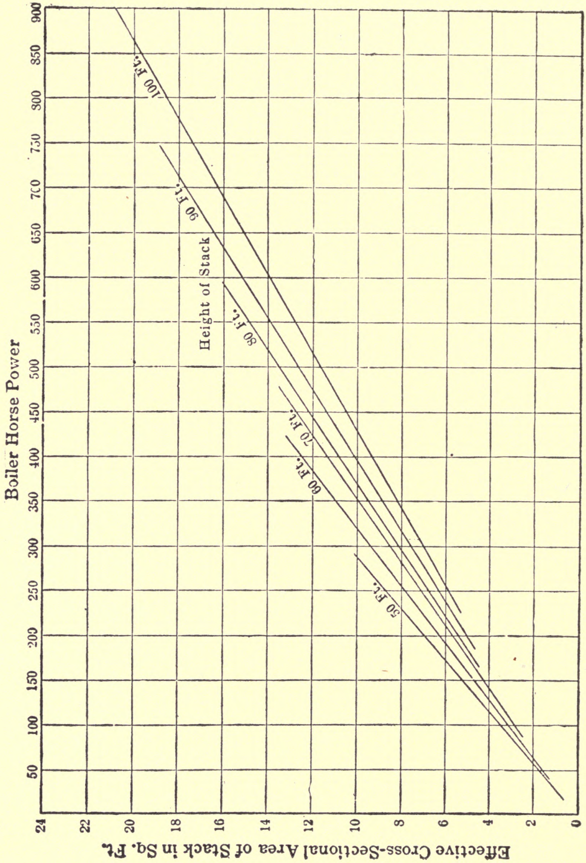
Boiler Horse Power
50
100
150
200
250
300
350
400
450
500
550
600
650
700
750
800
850
900
Effective Cross-Sectional Area of Stack in Sq. Ft.
0
2
4
6
8
10
12
14
16
18
20
22
24
Height of Stack
100 Ft.
90 Ft.
80 Ft.
70 Ft.
60 Ft.
50 Ft.

FIG. 88.

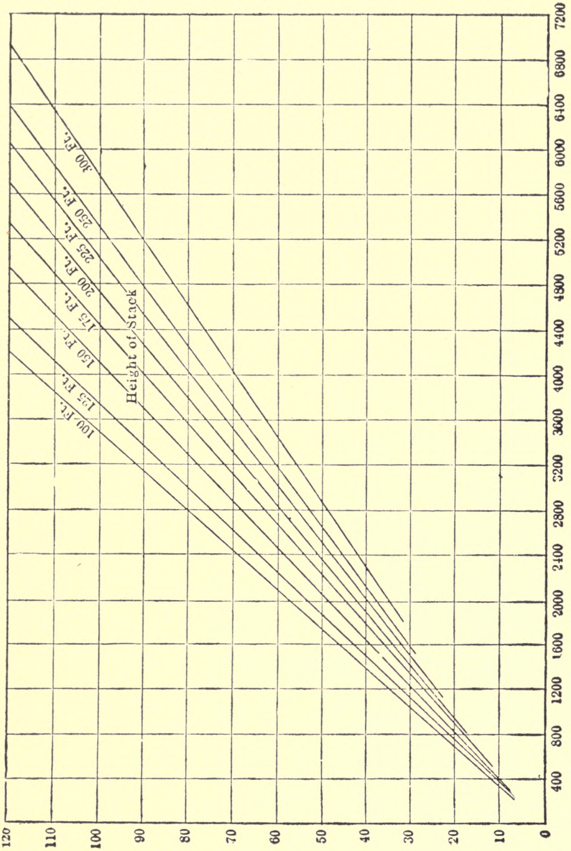
Height of Stack
300 Ft.
250 Ft.
225 Ft.
200 Ft.
175 Ft.
150 Ft.
125 Ft.
100 Ft.
7200
6800
6400
6000
5600
5200
4800
4400
4000
3600
3200
2800
2400
2000
1600
1200
800
400
120
110
100
90
80
70
60
50
40
30
20
10
0

FIG. 89.

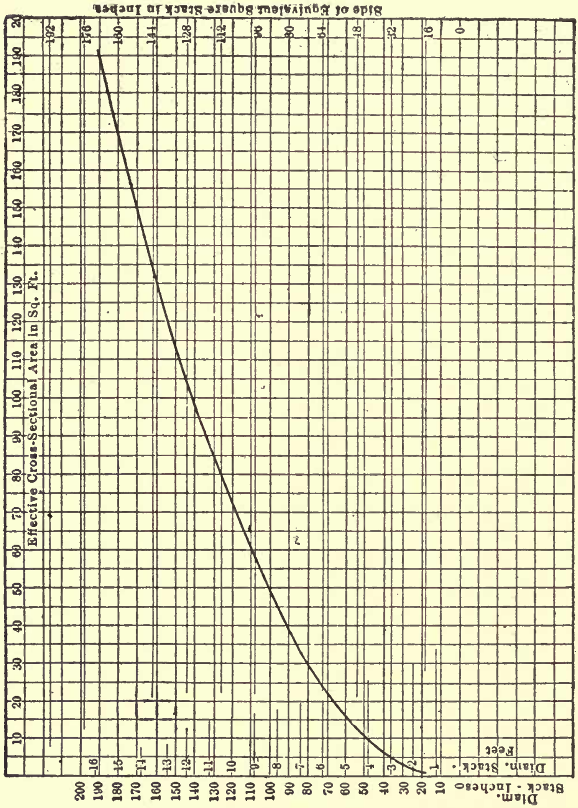

FIG. 90.

XVII

BOILER RULES

The Board of Boiler Rules appointed under a recent act of the Massachusetts legislature has adopted the following regulations:

Section I — Maximum Pressure on Boilers

1. The maximum pressure allowed on any steam boiler constructed wholly of cast iron shall not be greater than twenty-five (25) pounds to the square inch.

2. The maximum pressure allowed on any steam boiler the tubes of which are secured to cast-iron headers shall not be greater than one hundred and sixty (160) pounds per square inch.

3. The maximum pressure allowed on any steam boiler constructed of iron or steel shells or drums shall be calculated from the inside diameter of the outside course, the percentage of strength of the longitudinal joint and the minimum thickness of the shell plates; the tensile strength of shell plates to be taken as fifty-five thousand pounds per square inch for steel and forty-five thousand pounds per square inch for iron, when the tensile strength is not known.

Shearing Strength of Rivets

4. The maximum shearing strength of rivets per square inch of cross-sectional area to be taken as follows:

Iron rivets in single shear	38,000 lb.
Iron rivets in double shear	70,000 lb.
Steel rivets in single shear	42,000 lb.
Steel rivets in double shear	78,000 lb.

Factors of Safety

5. The lowest factors of safety used for steam boilers the shells or drums of which are directly exposed to the products of combustion, and the longitudinal joints of which are of lap-riveted construction, shall be as follows:

(*a*) Five (5) boilers not over ten years old.

(*b*) Five and five-tenths (5.5) for boilers over ten and not over fifteen years old.

(*c*) Five and seventy-five hundredths (5.75) for boilers over fifteen and not over twenty years old.

(*d*) Six (6) for boilers over twenty years old.

(*e*) Five (5) on steam boilers the longitudinal joints of which are of lap-riveted construction, and the shells or drums of which are not directly exposed to the products of combustion.

(*f*) Four and five-tenths (4.5) on steam boilers the longitudinal joints of which are of butt and strap construction.

SECTION 2

Section 2 sets forth the standard form of certificate of annual inspection.

SECTION 3 — FUSIBLE PLUGS

1. Fusible plugs, as required by section 20, chapter 465, Acts of 1907, shall be filled with pure tin.

2. The least diameter of fusible metal shall not be less than one-half ($\frac{1}{2}$) inch, except for working pressures of over one hundred and seventy-five (175) pounds gage or when it is necessary to place a fusible plug in a tube; in which cases the least diameter of fusible metal shall not be less than three-eighths ($\frac{3}{8}$) inch.

3. The location of fusible plugs shall be as follows:

(*a*) In Horizontal Return-tubular Boilers — In the back head, not less than two (2) inches above the upper row of tubes, and projecting through the sheet not less than one (1) inch.

(*b*) In Horizontal Flue Boilers — In the back head, on a line with the highest part of the boiler exposed to the products of combustion, and projecting through the sheet not less than one (1) inch.

(*c*) In Locomotive Type or Star Water-tube Boilers — In the highest part of the crown sheet, and projecting through the sheet not less than one (1) inch.

(*d*) In Vertical Fire-tube Boilers — In an outside tube, placed not less than one-third ($\frac{1}{3}$) the length of the tube above the lower tube-sheet.

(*e*) In Vertical Submerged-tube Boilers — In the upper tube-sheet.

(*f*) In Water-tube Boilers, Horizontal Drums, Babcock & Wilcox Type — In the upper drum, not less than six (6) inches above the bottom of the drum and over the first pass of the products of combustion, projecting through the sheet not less than one (1) inch.

(*g*) In Stirling Boilers, Standard Type — In the front side of the middle drum, not less than six (6) inches above the bottom of the drum, and projecting through the sheet not less than one (1) inch.

(*h*) In Stirling Boilers, Superheated Type — In the front drum, not less than six (6) inches above the bottom of the drum, and exposed to the products of combustion, projecting through the sheet not less than one (1) inch.

(*i*) In Water-tube Boilers, Heine Type — In the front course of the drum, not less than six (6) inches from the bottom of the drum, and projecting through the sheet not less than one (1) inch.

(*j*) In Robb-Mumford Boilers, Standard Type — In the bottom of the steam and water drum, twenty-four (24) inches from the center of the rear neck, and projecting through the sheet not less than one (1) inch.

(*k*) In Water-tube Boilers, Almy Type — In a tube directly exposed to the products of combustion.

(*l*) In Vertical Boilers, Climax or Hazelton Type — In a tube or center drum not less than one-half ($\frac{1}{2}$) the hight of the shell, measuring from the lowest circumferential seam.

(*m*) In Cahall Vertical Water-tube Boilers — In the inner sheet of the top drum, not less than six (6) inches above the upper tube-sheet.

(*n*) In Scotch Marine Type Boilers — In combustion-chamber top, and projecting through the sheet not less than one (1) inch.

(*o*) In Dry-back Scotch Type Boilers — In rear head, not less than two (2) inches above the top row of tubes, and projecting through the sheet not less than one (1) inch.

(*p*) In Economic Type Boilers — In the rear head, above the upper row of tubes.

(*q*) In Cast-iron Sectional Heating Boilers — In a section over and in direct contact with the products of combustion in the primary combustion chamber.

(*r*) For other types and new designs, fusible plugs shall be placed at the lowest permissible water level in the direct path of the products of combustion, as near the primary combustion chamber as possible.

XVIII

MECHANICAL TUBE CLEANERS

THE Hartford Inspection and Insurance Company, in a recent issue of *The Locomotive*, sounds a note of alarm anent the damage which may be inflicted upon a boiler by the improper use of mechanically operated tube cleaners. Coming, as it does, from so high an authority, this warning has produced unnecessary alarm among the present or prospective users of such devices, and the statement that the dangers pointed out are incident to them when improperly handled, and that "many of them give very good results when used judiciously and intelligently," is lost sight of in the light of the stated fact that injury has been produced by their use.

The first instance pointed out is one in which by the use of a cleaner removing external scale by rapidly rapping the internal surfaces of the tubes the latter were stretched to an elliptical section to such an extent that several of them collapsed when subjected to a pressure of ninety pounds. With any of the cleaners as now built by experienced and reputable makers such a result could be produced only by the grossest misuse of the tool and the most flagrant neglect of the directions which are furnished with it. Tests made by Professor Kavanaugh, of the University of Minnesota, with a $3\frac{1}{2}$-inch cleaner prove the energy of the blow when operating under a pressure of 90 pounds to be .106 of a foot-pound and the number of blows per minute 4,560. Only slight local distortion was produced by allowing the hammer to operate continuously in one spot.

It appears, therefore, that the distortion of the tubes in the case mentioned must have been due to a very unskilful use of a very badly designed cleaner rather than to the fact that the tubes were thinned by use but still serviceable. The same remarks will apply to the cases of splitting mentioned.

Another effect is the lengthening of the tubes due to the peening action, causing them either to sag or to project through the head. This action might follow an unduly protracted application of even a good cleaner, but should not be caused by such application as is necessary to remove ordinary scale. Such elongation is liable to crack the cast-iron headers of water-tube boilers, and the makers of at least one of the cleaners of this type discourage for this reason its use in boilers with headers of that material. Such headers are, however, dangerous in themselves and their use is rapidly being discontinued. This peening effect should be present in the mind of the operator of the cleaner, and he should regulate the intensity and time of application of the blows so as to avoid it, and watch carefully for it at the tube sheets.

The unequal expansion caused by discharging the exhaust from steam-operated cleaners through the tubes is also considered, and the use of compressed air for running the cleaner, when available, advised. The makers of the cleaners are alive to this condition, recommend the use of air in preference to steam, and recommend also that the boiler be cleaned while hot.

The conclusions arrived at in *The Locomotive* article are as follows:

(1) That when power-tube cleaners are used they should be kept in motion so that they cannot strike a succession of blows against any one part of the tube; (2) they should be operated by a pressure not exceeding 20 pounds, or, at the most, 30 pounds per square inch; (3) steam should not be permitted to blow through the tubes of a cold boiler for a sufficient time to sensibly heat the tubes; (4) compressed air should be used to operate tube cleaners unless the motive power is entirely external to the tube; (5) in any case, the boiler should be carefully watched during and after the application of a power cleaner, especially around the ends of the tubes and on the headers, and at the first sign of distress of any kind the use of the cleaner should be promptly discontinued; (6) lastly, a power cleaner should never be put in charge of any attendant save one upon whose judgment and skill the owner of the boiler can implicitly rely.

These conclusions commend themselves even to the makers of the devices in question, with the exception that they claim that a pressure of from 40 to 90 pounds is better than the lower pressure recommended as giving more rapid vibrations and of less amplitude, and deny that the heating due to exhaust steam will injure a sound tube. Signs of distress may be evidences of weakness revealed by the cleaner, and point to reforms or repairs rather than the discontinuance of the use of the cleaner. The strictures apply principally to cleaners operating by hammer action and discharging steam through the tube, but do not amount to a condemnation of the type, the successful present use of over 5,000 machines for a single maker evidencing that injury from its use is exceptional and avoidable rather than general and inherent.

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