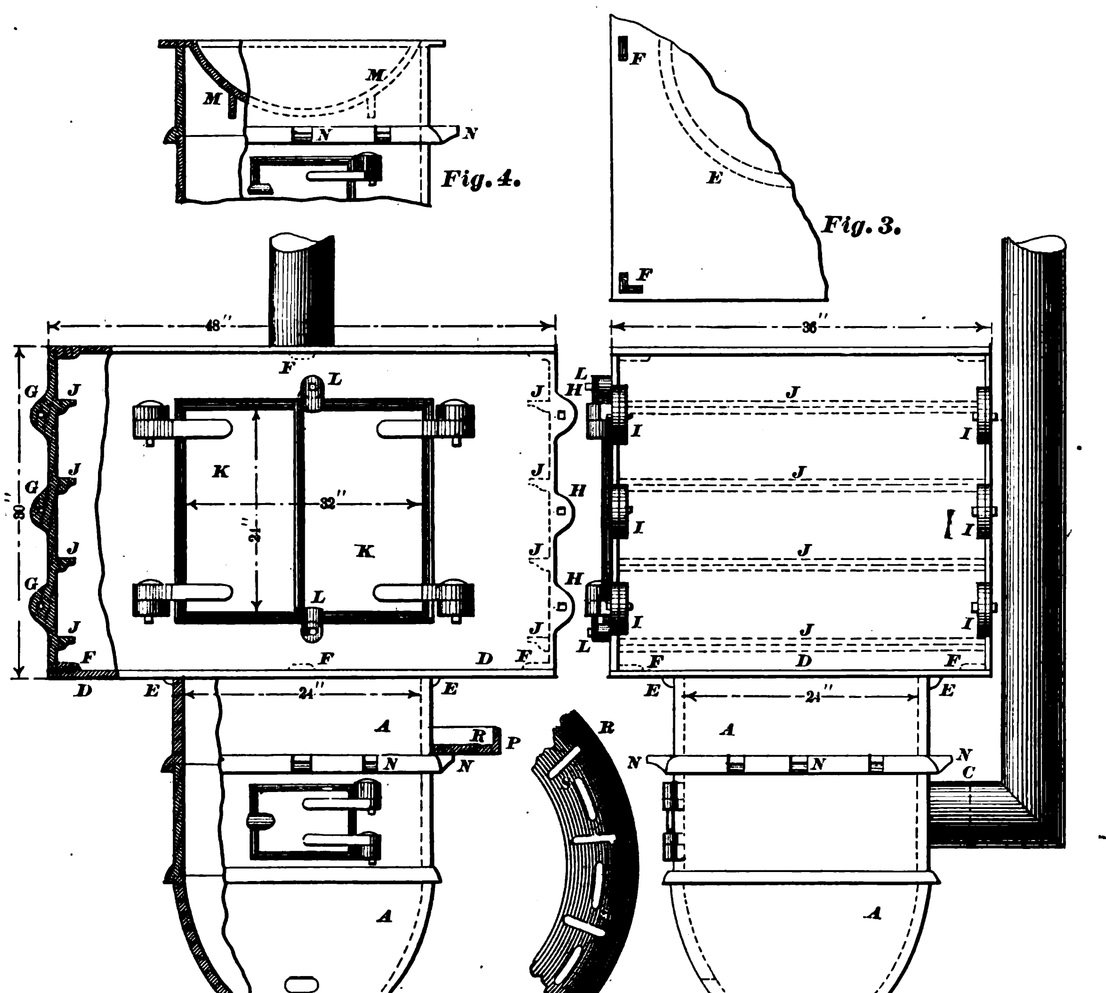
A Handy Contrivance.

By Robert E. Masters.

The drawings show a combined heating stove, core oven and melting pot, used in the brass foundry of a large railroad system. *A*, Figs. 1 and 2, show a plain round heating stove, on a cast-iron bottom plate *B* (no legs are used). This stove was made of the usual style of large dome top heating stoves, used in roundhouses and depots. The dome section is left off, and the pipe connected at *C*, Fig. 2.

It is represented broken away to the right of Fig. 1, to show the inside arrangement. The bottom oven plate *D*, a top view of which is shown at Fig. 3, is cast with a circular rib *E* on the bottom side, to fit around the stove, to prevent sliding. The top plate can be cast from the same pattern, by stopping off the rib *E* in the mould. The angle and straight projections *F* are 1" high, and act as a stay to the side plates. The top and bottom plates are not bolted to the sides. The lugs *G*, on the sides, are cast at right angles with the plates, and fit up against and are bolted to lugs *H* on front and back plates. An edge view of lugs bolted together is seen at *I*, Fig. 2. The brackets *J*, shown in section, and by dotted lines, are to support cross-bars, to set the core plates on. The doors *K* are of medium light wrought-iron, and are fastened when closed with buttons *L*. The back oven plate can be made from the pattern for the front plate by stopping off the lugs and cutting out the sand which forms the opening for the doors in the mould.



A Handy Contrivance.

This oven has a capacity to dry cores for good size brass foundries on certain lines of work, and could be used to good advantage in many iron foundries where they have no core oven, saving the trouble and inconvenience of drying and watching cores on top of furnaces, and in and around boilers. The bars from the center shelves can be removed and a comparatively large core dried, besides utilizing the heat to keep the shop warm in winter. In the summer it could be removed outside of the foundry if desired.

Fig. 4 shows the oven removed, and the kettle *M* set in to melt the metal for lining journal brasses.

At the foundry referred to they make a large number of lead-lined brasses each month. The brass is placed in a former, and held secure by a lever worked with a pedal, the metal is dipped from the kettle and poured with a small babbitt ladle. The mixture of metal used for a lining at this place is 20 pounds of antimony to 80 pounds of lead.

The stove *A*, when used with a dome, has no hole for the stovepipe at *C*, the pipe connecting straight with the top of the dome. It is then used as a sand dryer by railroads having only a few locomotives. A number of projections, *N N*, are cast around the bottom of this section of the stove, for the ring shown in section at *P*, and in part plan Fig. 5, to rest on. A cylinder of heavy wire netting is placed on this, inside of the flange *R*. It is left open at the top and filled with sand over dome and all; as it dries, it falls readily through the slots *S* by starting it with a poker.

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Plug engineers employed in backwoods saw-mills are coming to the front as producers of boiler explosions. There are at least five times as many saw-mill explosions as locomotive boiler explosions, notwithstanding that the former comprises the largest class of boilers to be found in the United States. Poor boilers, incompetent engineers and no insurance makes a good combination for turning mill men into paupers or corpses in short order.—*Lumberman*.

Again of Tapers—A System.

By Jarno.

It would appear that the condition of center tapers is even worse than was shown in the story of the cabinet. One maker has in his lathes ten different rates of taper.

Relatively, the obstacles in the way of changing to a standard have been overstated, because, for every one of these there is an equal or a greater obstacle in the way of continuing the confusion. Waubug asserts that to change now will cost more than all the profit that will ever come from having a uniform standard. The fact that he is the lathe maker having the ten rates of taper may have something to do with his so positive opinion. When made on purpose, has any one ever known of a change that did prove so costly? In a large concern there was a departure from the original standard, in consequence of making new gauges not particularly accurate and of gauges becoming worn. If it is easy to go away from one, it may not, with proper means, be hard to come to a standard. My cabinet acquaintance thinks that, ere long, he will alter or replace half the centers in his cabinet, and, at the same time, take along their spindles.

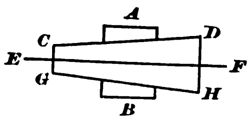

Fig. 1.

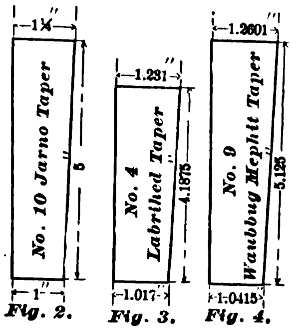

Fig. 2. Fig. 3. Fig. 4.

TAPERS.

Is there a scientific principle involved in establishing a taper? There is this principle, that the angle of the taper must not be so large that a center will not stay, when driven in, and that the angle need not be so small as to make it very difficult to back the center out. These limits are quite indefinite. In Fig. 1 let the piece *A* rest upon the inclined surface *C D*. Now, the greatest angle that *C D* can make with the horizontal line *E F*, before *A* slides upon *C D*, is called the *angle of repose;* it has also been called the *limiting angle of resistance*. This angle varies with the material and the condition of the surfaces. The average angle of repose in metals having smooth surfaces that are oiled, is placed by Unwin at four and a half degrees, and by Willis at five degrees. Rankine gives the same, or a little smaller, angle as Unwin, and adds that in some experiments the angle has been as small as two degrees. Hence, it would appear that if *A* must slide, the angle of *C D* with *E F* should be more than five degrees, and that if *A* must not slip, the angle should be less than two degrees. If *C D G H* be a wedge pressed between two pieces *A* and *B*, the law of its sliding will be the same as that just given, considering each of the two sides of the wedge to be at the same angle with its center line *E F* as we have just considered *C D* to be with *E F*. That is, if the wedge must slide when pressed upon by the two pieces *A* and *B*, the angle that one of its sides makes with the other side should be more than ten degrees, or more than twice the angle of repose. If the wedge must not slip, the angle of one side with the other, which may be called the angle of the taper, should be less than four degrees. Hence, we conclude that if a center be of a taper whose sides make an angle of less than four degrees, it is not likely to slip out, after being driven in, even though it be oiled. An angle of four degrees makes a taper of nearly seven-eighths of an inch to a foot. No center, so far as I have seen, has a taper more than this, which is interesting as indicating that the machinist, in practice, has not exceeded the angle of safety given in the text-books. Gascon says that the interesting thing is that the machinist knew by experience the angle of safety, long before it was recorded in books, though for it he had no special name, and that the books have merely followed the machinist. He says also that the quoting from so many authors looks too much like airing an acquaintance with their books. I reply that the quoting is done to show that it is not always safe to take any one author in a matter of experimental knowledge; the percentage of difference between two and five degrees is considerable. The limit to the smallness of the angle of taper is also indefinite. If a cold center be driven into a warm spindle, one might think that the center is not tapering enough, upon trying to back it out when the spindle is cold. The smallest center angle that I know is something less than two and a half degrees, which in practice has been found to be small enough. We are, therefore, at liberty to choose any angle of taper between two and four degrees.

I find that the best system is the following:

Let the rate of taper be one in twenty, or .05" *to* 1";

Let the tapers be numbered 1, 2, 3, etc.;

Then the number of the taper is the number of tenths of an inch in diameter at the small end, the number of eighths of an inch at the large end, and the number of halves of an inch in length or depth.

Examples.—No. 1 is one-tenth inch at the small end, one-eighth inch at the large end, and half an inch long; No. 3 is .3" at the small end, 3-8" at the large end, and 1.5" long.

Now we have a system in which a number means something; for when we know it we at once know the end diameters and the length. Let us look at a couple of examples of tapers in common use. Before me are six different lists; each of four of them purports to be a list of the same tapers. Neither of these four agrees with any other, and, of course, we cannot tell whether either is correct: confusion increasing—I write these words on the back of my hand, to make sure that I am not dreaming. I avoid the lists altogether, and select two tapers not in them, those of Labrihed and Mephit; their sizes are as given in Figs. 3 and 4. In thus giving correct sizes and real names of the makers, the reader that wishes to examine farther into this subject can have the benefit of my experience. In singling out Labrihed and Mephit, I would say that they are named not because they are pre-eminently bad, but because their tapers are known with some degree of certainty. The sizes in Figs. 3 and 4 can be easily carried in mind, if one has a good memory and nothing else to remember. Most persons cannot spare the time to learn a complicated system, and, therefore, they must consult drawings and tables in dealing with such a system. It gives one but little help to say that the length of a taper is four and five-eighths times the diameter of the large end, because one can consult a table as easily as make a calculation. In the Jarno system all the sizes can at once be derived from the number of the taper. Figs. 3 and 4 are in the vicinity of Jarno No. 10, Fig. 2. As soon as No. 10 Jarno is named, it is known that the small diameter is 1", that the large diameter is 1.25", and that the length is 5". The length can also be derived from the diameter of either end: five times the small diameter, or four times the large diameter, is the length.

I do not expect that Waubug will change. Can any one tell why a boy plays marbles always in the first days of March when the ground shows up through the snow? Whoever can tell this, can tell why Waubug will not change. Perhaps he might, instead of our "Standard Taper," say our *tapers*, if the habit is not too firmly fixed. During the year 1890 some concern will start, be successful, and will be in existence fifty years from to-day. It is this concern, as well as others, that shall start and be successful, in years to come, that will adopt the new system.

In the next paper I purpose to show a way of making and measuring gauges by which any one can conform and adhere to a standard.

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Modern Locomotive Construction.

By J. G. A. Meyer.

ONE HUNDRED AND THIRD PAPER.

The pumps can be attached directly to the frames, as illustrated in Fig. 569, only in eight-wheeled passenger engines. In mogul, ten-wheeled, and consolidation engines there is no room to place them in similar positions, therefore, in the latter classes of engines, we often find them attached to the slides. Figs. 587 and 588 show a pump designed for fastening to the slide and yoke brace; the only difference between this pump and that previously shown is the position of the air chambers, and the position and design of the lugs. The type of slide for which this pump was designed is shown in Figs. 241. The lug near the stuffing-box (Fig. 588) is bolted to the top of the slide, and the lug near the air chamber is bolted to the yoke brace.

The illustrations (Figs. 587, 588) show the air chambers placed in the neighborhood of the center of the barrel; the object aimed at in this design is to concentrate the weight of the pump as much as possible near one of the points of support. As far as the distribution of weight is concerned, the design is correct; but in placing the air chambers at or near the center of the barrel, the direction of the flow of water is interfered with, which is an objection. The proper place for the air chambers, in the class of pumps here shown, is at the end of the barrel; and although they cannot always be placed there, it should be remembered that any change in the direction of the flow of water, such, for instance, as is caused by placing the air chamber in the center of the barrel, has a tendency to affect the efficiency of the pump, and this should be avoided as much as possible.

It may also be well to remark here that, in all pumps, the suction valve should be placed as near as possible to the plunger, leaving only a sufficient water-way between them. Necessary changes in the form and size of the water passages should be made gradually; sudden enlargements and contraction should always be avoided.

We have previously referred to short stroke pumps; sections and other views of this class of pumps are here given in Figs. 589 up to 592. These pumps are placed under the boiler, and are bolted to a cross-brace A extending from frame to frame. The plunger D is worked from an eccentric placed on one of the driving axles. Fig. 593 shows separate views of the pump-rod jaw C, to which the pump-rod B is connected. The valves and cages in this pump are of the same design as those in the full-stroke pumps previously illustrated. This pump is designed for a 12-inch cylinder.

Figs. 594 up to 596 represent another short-stroke pump. The pump is fastened to the draw-bar, and is worked from a pin attached to one of the crank-pins; hence the pump is placed outside of the driving wheels. It will be noticed that the valves L and cages K are made somewhat different from those previously illustrated; this difference is not due to any particular necessity, but is simply due to the fact that the pumps were designed in different locomotive works.

To find the size of pump required, we should estimate the consumption of steam, and this may be approximately obtained by calculating the weight of steam in the cylinder from its pressure and volume at the time of its release. In locomotives, the average pressure of the steam at the time of its release does not vary much, and therefore we may make the pump capacity directly proportional to the cylinder capacity, without finding the amount of steam consumed. Since the stroke of a full-stroke pump is equal to that of the piston in the steam cylinder, we may further simplify the computation and make the cross-sectional area of the plunger directly proportional to the cross-sectional area of the cylinder. Practice indicates that for locomotives having two pumps, good results will be obtained by making the cross-sectional area of each plunger equal to $\frac{1}{72}$d of the cross-sectional area of one cylinder. This proportion is not strictly adhered to by the different builders; indeed, many builders use the same size of pump for two or three different sizes of cylinders; for instance, the pump illustrated in Fig. 568 is used for cylinders 16, 17 and 18 inches in diameter. But the proportion here given we believe to be a good one to adopt in designing a full-stroke locomotive pump. Hence the following:

Rule.—Divide the cross-sectional area of the cylinder by 72; the quotient will be the cross-sectional area of the pump plunger for locomotives in which two pumps are to be used.

Example.—What should be the diameter of the pump plungers for a full-stroke pump in a locomotive having cylinders 17 inches diameter?

The cross-sectional area of a cylinder 17 inches diameter is 226.98 square inches, and $\frac{226.98}{72} = 3.15$ square inches for the cross-sectional area of the plunger; the diameter of a circle containing 3.15 square inches is two inches (very nearly), hence the diameter of the plunger should be two inches.

Let us take another example. Find the diameter of the pump plungers for a full-stroke pump in a locomotive having cylinders ten inches diameter. Here we have $\frac{78.54}{72} = 1.09$ square inches for the cross-sectional area of the pump. plunger, and the diameter will be $1\frac{3}{16}$th, nearly.

In a similar manner we find that the diameter of the pump plungers for full-stroke pumps for cylinders 20 inches in diameter will be nearly $2\frac{3}{8}$ inches. These results agree very closely with the average locomotive practice.

The capacity of a full-stroke pump designed by the foregoing rule will be $\frac{1}{72}$d of the capacity of one steam cylinder; and short-stroke pumps should have the same capacity. We may, therefore, establish the following:

Rule.—Multiply the cross-sectional area of the cylinder in square inches by the length of stroke of piston in inches, and divide the product by 72; the quotient will be the capacity of the pump in cubic inches. Dividing this capacity by the length of stroke of plunger in inches, we obtain the cross-sectional area of the plunger in square inches;

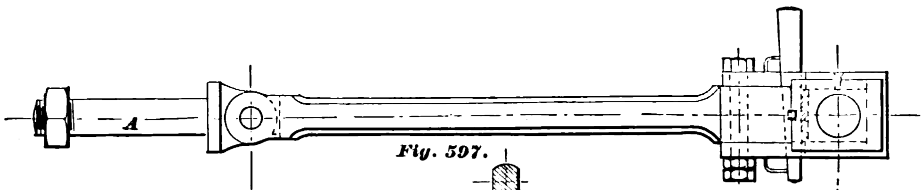


Fig. 597.

Fig. 598.

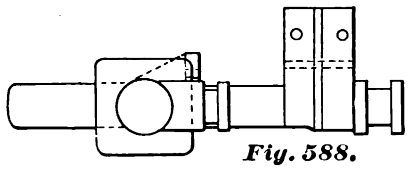


Fig. 588.

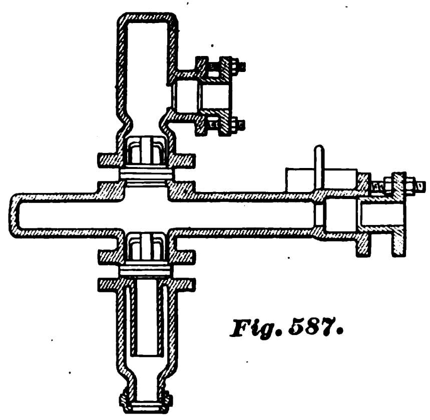
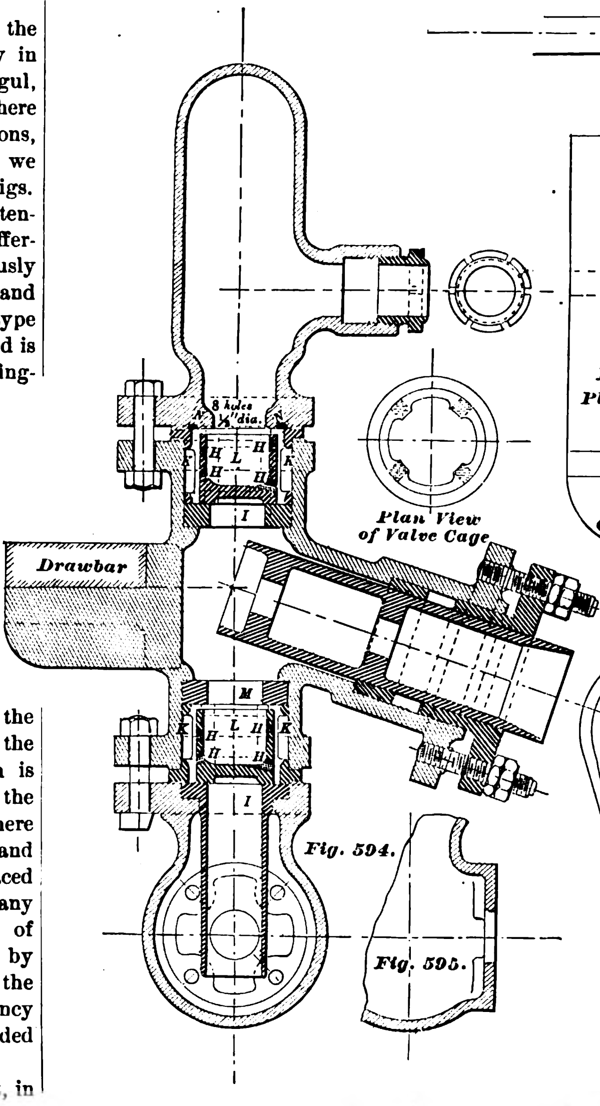


Fig. 587.



Plan View of Valve Cage

Fig. 594.

Fig. 595.

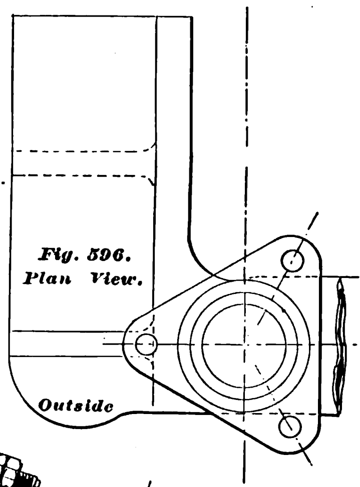


Fig. 596. Plan View.

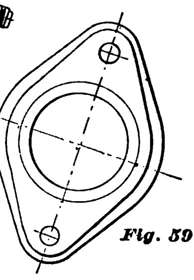


Fig. 594a.

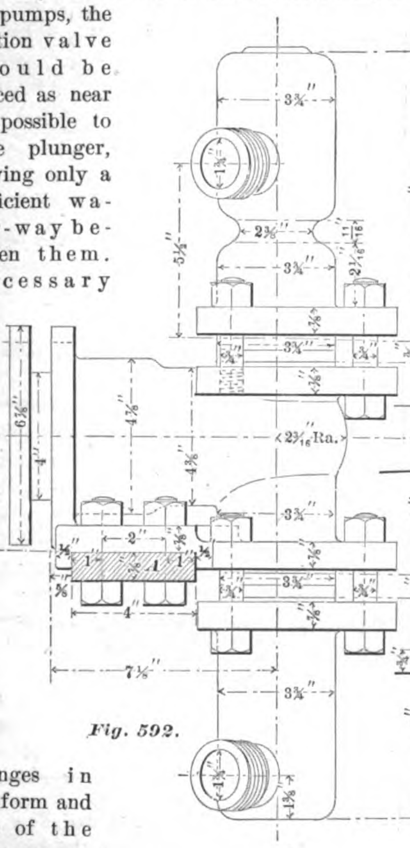


Fig. 592.

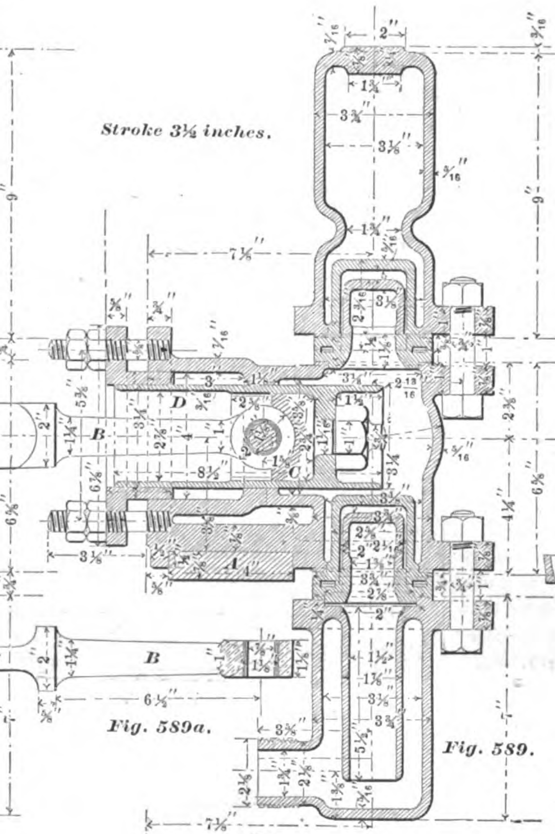


Fig. 589a.

Fig. 589.

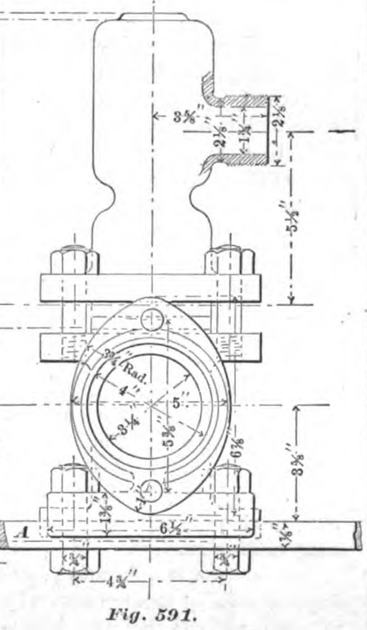


Fig. 591.

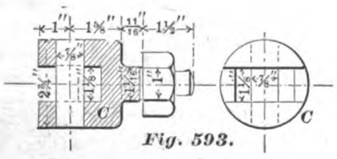


Fig. 593.

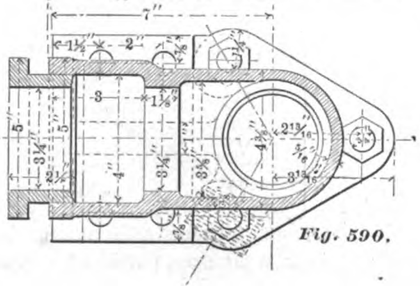


Fig. 590.