THE POWER HANDBOOKS

PUMPS

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PUMPS

TROUBLES AND REMEDIES

COMPILED AND WRITTEN

BY

HUBERT E. COLLINS

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INTRODUCTION

The solution for many of the puzzling troubles with pumps which every engineer is liable to encounter will be found in this work. There are also a number of unusual instances of repairs, which may prove of much value where pumps are seemingly broken down beyond use, and these special instances will often suggest other possibilities of the same sort.

Two chapters are given to the important matter of setting the valves of duplex pumps, and much attention is given to this same subject throughout the book. It is believed that a careful perusal of these pages will give the reader a sufficient number of illustrations in the operation and repairs of pumps to handle any difficulty that may arise in the ordinary operation of a steam-pump plant.

The compiler of this volume is indebted to those contributors of Power whose names appear in connection with the various articles, and to the following special contributors whose suggestions and works have been found useful: Earl F. Webster, W. A. Dow and Samuel S. Murdock.

HUBERT E. COLLINS.

NEW YORK, October, 1908.

I

PUMP TROUBLES[1]

THE first duty of an engineer in a new plant, before an attempt is made to start the pump, is to get acquainted with the discharge and boiler-feed pipes, making sure that all the valves are open that should be open. Of course this is not necessary taking in charge of a plant where the present engineer is going to leave, as in most instances the old engineer will show the new man all he wants to know about the piping arrangement. Where an engineer has been discharged, however, the engineer who takes his place usually has to start up without this friendly aid.

At one small steam-plant, an attempt was made after having the fires started under the boilers, to start the boiler feed-pump, but before doing this what was supposed were the valves on the discharge and feed-pipes were opened in the belief that the arrangement was as is shown in Fig. 1. When the pump was started it was not feeding water to the boilers. Nothing was found wrong. Then the discharge and feed-pipes were inspected and it was found that the valve *A* had been opened the night before. Upon inquiry the information was elicited that the pipe was supplying

[1] Contributed to Power by H. Jahnke.

hot water for the factory and the valve was only opened a few times during the day. The pump received hot water under pressure from a heater, and when hot water was wanted in the factory the feed-valves *B* and *C* were closed and the valve *A* opened for a short time; then the valve *A* was closed and *B* and *C* opened. This appeared to be a bad arrangement, so it was changed by connecting the pipe for the factory directly to the heater, which was of the closed type and received water from the city main.

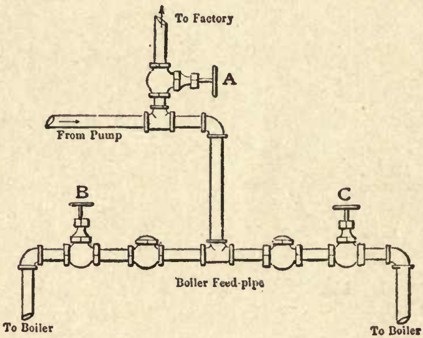


FIG. 1

In some new plants after the discharge and feed-pipes are put in and in use, it is found that the arrangement is not what it should be, and careless engineers, instead of making the needed changes at once, although there has been ample time to think it all out, wait until something goes wrong and then make the changes in a hurry.

Probable Causes of Pump Troubles

If upon starting a pump it is found that it pounds, this may be due to various causes, such as insufficient water supply due to an obstruction in the suction pipe, a leaky foot-valve, a loose water-piston or loose nuts. Also the water-piston may have a tighter fit at some point of the stroke, due to unequal cylinder wear, in which case reboring the cylinder will be in order. Then, again, if the pump is of the duplex type, the pound may be due to improper setting of the steam-valves.

If a pump fails to draw water this may be due to the following causes: If taking water from a well or other supply, either the foot-valve of the suction pipe may leak; the suction pipe may be too small, there may be some obstruction in the suction pipe or in parts leading to the water cylinder; the lift may be too high; the water valves may leak or break; foreign matter may have lodged beneath the suction or discharge valves; the packing on the water-piston may be worn out and leaking; the seats of the suction and discharge valves may be broken, or the water valves may be prevented from lifting because the springs were screwed down too tightly.

If a pump is supposed to receive water under pressure yet fails to get water, it may be due to some valve in the supply pipe not being opened; or the supply pipe may be clogged up; or there is a loose disk or a break in some valve which prevents the full supply going to the pump; the valves may not be wide open;

the supply pipe may be too small; the supply pipe may also furnish water for some other purpose in the factory and may be too small to supply both pump and factory. If a pump receives water under pressure from the city main, the supply pipe should be run direct from the water meter to the pump, and not used for any other purpose, unless the pipe is large enough to supply enough water for both places.

Engineers are often troubled by a groaning noise in pumps; this groaning may be due to any of the following causes: The cylinder oil used may be too heavy (cases are known where the use of lighter oil has cured the trouble); the piston-ring edges may have become so sharp that they scrape the oil from the cylinder walls; if the water-piston packing is too tight, the excessive friction will cause a groaning.

Graphite Mixture Cures Groaning

Groaning in a cylinder can often be cured by the application of graphite mixed with cylinder oil, forced into the cylinder with a hand pump.

Figure 2 shows a good arrangement to place on pumps for feeding graphite and oil to the pumps once a day, or as often as may be necessary. A 2-inch nipple about 5 inches long is provided, with a 2 x ¼-inch reducer on each end; one end is screwed directly into the steam-chest by means of a ¼-inch close nipple, and on the other end is a ¼-inch close nipple with a ¼-inch tee. Then a close nipple and valve are placed in the steam-pipe of the pump above the valve, and a pipe is run from the valve to the feeder, as shown. The reason for placing the valve above the pump valve is to be able to use the full boiler pressure to force the graphite into the cylinder when the pump is throttled down.

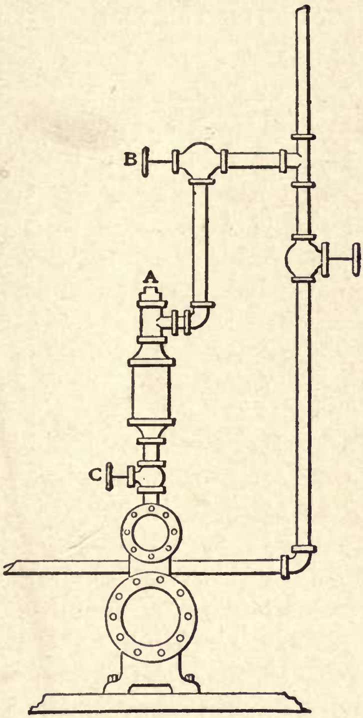


FIG. 2

The operation is as follows: When it is necessary to feed the graphite mixture, the plug *A* is removed from the tee and a supply of graphite and oil placed in the feeder, the plug is replaced and the valves *B* and *C* are opened, when the graphite will be forced into the steam-chest and cylinder.

Correcting Unequal Stroke

One side of a duplex steam-pump used to make a shorter stroke than the other, due to the water cylinder being worn, and no amount of adjusting of the steam valves would remedy the trouble. Then was tried the following method, which readily overcame the difficulty: The piston and rod were removed from the troublesome side of the pump and the walls of the cylinder were covered with graphite mixed with a little engine-oil, well rubbed in; the piston and rod were replaced and the pump was run slowly with no water in the water end for a short time; then the pump was put in service again, when this side ran much better than the other. The other side was treated in like manner, when both pistons made a full, even stroke, and when the pump was not in service the pistons could be moved easily by hand, which showed that there was not much friction in the water cylinder. It was also found that the water-piston packing will last much longer, and that this treatment is likewise excellent for the steam cylinders of pumps and even engines, and they are so treated whenever the pistons are taken out for any reason.

If a pump begins to run more slowly with the steam valve wide open, if the pump is in good condition, the trouble may be due to an obstruction in the feed-pipes, such as scale.

A short time ago a pump began to slow up every day, with the steam-valve wide open. At first it was thought that the steam-piston was running dry, but feeding more oil did not remedy the trouble; then the check-valves on the feed-pipes were examined, when it was found that the checks were so covered with scale that they could not lift sufficiently to admit the water the pump forced into the pipes, and of course the pumps had to slow up. When the checks were cleaned the pump worked as before. Engineers who use very dirty water for the boilers should examine the check-valves in the feed-pipes whenever the boilers are washed out.

Frequently engineers fail to examine the packing in the water-piston until the pump refuses to supply water to the boilers, or otherwise behaves badly, and when looking for the cause they find that most of the water-piston packing has disappeared and may be lodged beneath the water valves or in the feed-pipes. This is obviously bad practice; the cylinder-head should be removed frequently to see what condition this packing is in. If it begins to show signs of giving out, it should be removed at once and replaced by new packing, no matter if the pump has been doing good service, for if this is not done the pump is liable to fail at any moment.

The water valves should be examined in like manner, and if it is found that they are beginning to leak, valves of hard composition, such as are used in pumps handling hot water, should be refaced at once by rubbing the valve on a sheet of fine sandpaper laid on a smooth, flat surface. It is also a good plan to have a good set of new valves in stock, as one can never tell when a valve is going to break or meet with some other accident. Another bad practice is to start a steam-pump in the morning, after it has been standing idle for some hours, without opening the drain-cocks on the bottom of the steam cylinder. These cocks should be opened so that the water of condensation will drain out of the cylinder. They should be closed again, of course, as soon as the condensate is blown out of the cylinder.

A Knowles single-cylinder steam-pump, used for boiler feeding, started to give trouble one day by stopping at one end of the stroke. Adjusting the rocker connection bolt so as to equalize the stroke would not help the trouble.

When the pump was taken apart it was found there was too much lost motion between the lug on the slide-valve and where this lug fits into a slot in the steam-chest piston. To take up the lost motion a piece of heavy sheet iron was riveted on each side of the lug as shown in Fig. 3 at *A* and *B*. After this was done and the stroke was adjusted, the pump ran nicely.

In another case a single-cylinder steam-pump received hot water under pressure. One day this pump failed to furnish enough water, and on examination it was found that two of the water valves had broken. When they were replaced by new valves the pump worked a little better, but had to be run at a higher speed than before. Again examining the water end, it was found that the new valves did not seat properly, due to scale around the valve-studs. Where a pump receives hot water scale will form on the valve-seats and valves and in time will break in some places, when of course the valve will leak and the engineer may look a long time for the cause of the trouble.

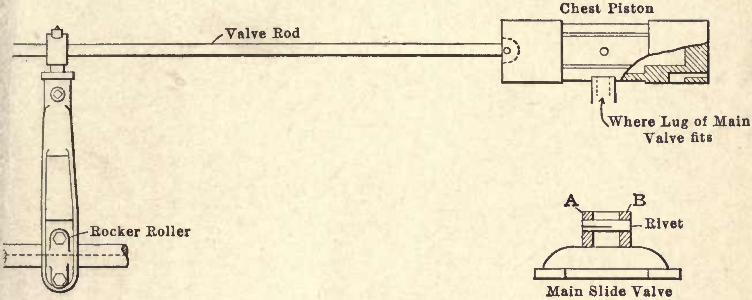


FIG. 3

When putting in new valves, the seats and valve-studs should receive a good cleaning.

It is not advisable to use hot-water valves until they are in bad condition, but they should be taken out occasionally and rubbed over a piece of sandpaper; also overhaul the valve-seats.

A duplex steam-pump which received water under pressure and which is used for boiler feeding, failed to supply enough water. Examination of the water valves and piston showed nothing wrong, so the pump was started again, when it was noticed it was supplying plenty of water. This was a puzzle until it was found that the drain valves from the water cylinder were not closed at night, and when the pump was started after examination, the drain valves were closed, so that the water which at first went into the sewer afterward went to the boilers.

Some engineers have the bad practice of letting a feed-pump run until it fails to furnish water. Some day they may get caught with low water in the boilers and the pump in bad shape and if there is no other means of feeding, it may be a case of shutting down the plant until the pump is repaired.

II

PUMP TROUBLES[1]

THERE is probably no machine which is more generally used in plants of all kinds where steam is used for power than the steam-pump, either for feeding boilers or for elevating water for various other purposes, and it is doubtful if there is anything which can cause more trouble and worry to the engineer in charge than a troublesome pump. Still, it would be hard to find a more simple machine, and it is the object of this chapter to touch briefly upon some of its numerous troubles, their causes and remedies.

In the instruction in the manufacturers' catalogues a statement something like this will often be found: "Be sure the water end is all right before disturbing the steam end." They certainly have good cause for inserting such a statement. It is well-nigh impossible for anything to go wrong with the pistons or valves, and yet what more is there in the steam end? Notwithstanding this, men work almost a whole day changing the setting of the steam-valves to make a pump stroke regularly, when the trouble was manifestly in the water end; and after spending several hours in fruitless work, they would have the valves

[1] Contributed to Power by S. L. Brainerd.

set wrong and would not know how to set them properly. If it is a single pump it is beyond the scope of this article to tell him how to set the valves, as every make of single pump requires special directions; but one way, and a good way, to set duplex pump valves is simply this: Remove the steam-chest cover and set both pistons in the middle of their strokes. Then set the valves in the middle of their strokes; *i.e.*, so they just cover both steam ports. Now there is in all duplex pumps a certain amount of lost motion in the valve-gear; that is, the valve-stem travels a certain distance before it can move the valve. Adjust the valve-stems so that this lost motion is equally divided on each side of the valve, being sure the valve is in its middle position. In other words, to set duplex pump valves set everything in its middle position. The valves are properly set now, and unless something slips, you need never trouble about them again.

A pump is harder on packing than an engine, because it takes steam at boiler pressure the full stroke — provided, of course, the throttle is wide open so it can get boiler pressure — while an engine only takes the steam at boiler pressure up to the point of cut-off, after which the pressure and temperature drop rapidly. The writer has charge of six pumps, and one of them, a compound condensing pump, with cylinders and heads steam jacketed, would burn up enough packing on the high-pressure rods to run the whole plant, engines and all, until a little tube was run to each rod and fed cylinder oil on them very slowly. This was a great help. This same pump had a chronic groan in the low-pressure cylinders, and did some little cutting even with 4 pints of oil in twenty-four hours, until graphite and oil was fed to it, and now it runs six weeks on 4 gallons of oil, without a groan.

It makes no difference whether a pump is of the piston pattern or outside or inside packed plunger pattern; they are all heir to the same troubles, and in a general way the same trouble is remedied in the same way in either style. For instance, if a pump runs smoothly for three strokes of the revolution and jerks back suddenly on the fourth, it always indicates a defective valve in the water end, but there is no way of telling whether it is a discharge valve in the end which the plunger is leaving, or a suction valve in the end which it is approaching, except to remove the hand-hole plates and examine the valves until the one is found which is causing the trouble. Referring to Fig. 4, it is evident that if the suction valve *A* is broken, or if the entire valve-seat is knocked out of the valve deck, as sometimes happens, the water, instead of being forced through the discharge valves *B* against the pressure, will simply surge back through the suction valve *A* and flow into the cylinder through suction valves *C*, thereby relieving the piston of practically all resistance and causing it to jerk back to the head end suddenly. Similarly, if the discharge valve *D* is broken, when the piston moves towards the yoke *Y* it forces the water into the force chamber against the pressure, but when it returns towards the head end *H*, valve *D* fails to hold the water back and it surges around from *B* through *D* into the cylinder, relieving the piston of resistance, the same as in the case of the broken suction valve.

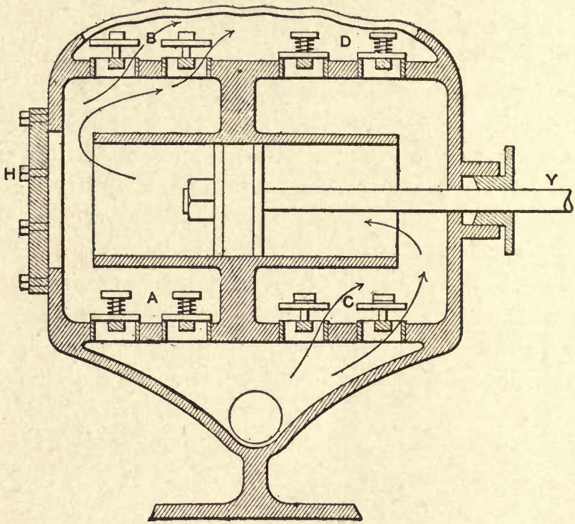


FIG. 4

It is therefore evident that a sudden stroke towards the head end is caused either by a defective suction valve in the head end or by a defective discharge valve in the yoke end, and *vice versa*. A sudden stroke towards the yoke end indicates a defective suction valve in the yoke end or a defective discharge valve in the head end. The degree of suddenness or freedom with which the stroke occurs evidently depends upon the extent of the leak. Frequently a small hole in a valve does not allow enough water to pass to cause any perceptible irregularity in the action of a comparatively large pump, and it can be detected by the noise of the water surging through it, and should be remedied as quickly as possible. Should a serious defect of this kind exist when the pump is started, it will usually fail to create a vacuum in the suction and to pick up its water.

The maximum theoretical lift of a pump is about 33 feet, but in practice 28 feet is about all that can be relied upon, provided the pump is almost directly over the source of supply. When the suction is carried any distance to the pump, a very good rule for a rough estimate is to consider 100 feet horizontal distance equal to 1 foot vertical lift. However, a 28-foot lift is not to be recommended unless absolutely necessary. It is essential that the suction system be perfectly air tight, especially in case of a considerable lift, in order that a good vacuum may be formed and allow the atmospheric pressure to force the water into the pump. A serious leak may be detected either by the refusal of the pump to pick up the water or by a knock at the beginning of the stroke, caused by the cylinder being only partially fitted with water. Remember that a small quantity of air in the suction expands to a considerable volume when under a vacuum of, say, 15 to 20 inches. When this is drawn into the cylinder and is compressed to 100 pounds or more, it allows the piston to travel some little distance before it strikes the water, thus causing the knock. This knock may be avoided by admitting a little water into the suction through the priming pipe. Whenever possible the suction pipe should be brought up to the pump at right angles to the direction in which it enters, and then be brought into it with an ell and a short piece of pipe in order to provide a more flexible connection and take up the vibration of the pump. In this way many a leaky suction will be avoided, as the vibration will not act in a direct line with the pipe; and this will more than offset the slight additional friction caused by the turn, especially if a long radius bend is used. The writer knows of one instance where a 12-inch cast-iron suction was cemented rigidly into a heavy wall; and although the pump did not appear to vibrate more than usual, the flange was broken off within a week. A suction with lead-calked joints is particularly troublesome from leaks caused by this vibration.

In order to cushion the pulsations in the discharge an ample air chamber should be provided, and it should by all means be fitted with a gage glass, so that the amount of air in it may be seen at all times. The makers frequently do not provide for this; but if they do not, the engineer will do well to put one on, because these air chambers very often get filled with water and then are of no service. They should at all times be at least half full of air. Very often a heavy water hammer will occur at about mid-stroke on high-duty pumps, if the air chamber is allowed to fill with water, and this is very hard on the pump in general. If no better provision is made for getting air into it, tap the suction at the pump and put in a $\frac{1}{4}$-inch pet cock, and open this very little until the proper amount of air is obtained, but be careful not to open it too wide, or the pump will knock from air in the suction. Sometimes it is necessary to run with this pet cock open most of the time.

As a general thing pumps are packed entirely too tight, causing excessive wear on the bushings and plungers and consuming an enormous amount of power. Engineers would do well to give more thought and attention to packing generally. A good way to pack a piston pump is to make a light brass or cast-iron bull-ring to fit loosely over the piston and with just spring enough to hold the packing in position, and cut the packing with lap joints, just like snap rings in a steam-piston, allowing a very slight side play between it and the follower. The water will get under it and set it out, insuring a good joint with loose packing. Packing applied in this way wears longer than when clamped tightly between the followers. Sometimes in packing small pumps the packing is so stiff and hard as to make it difficult to bend it and put it in place. By soaking it in hot water a few minutes it becomes quite pliable. Personally, the writer prefers the outside-packed plunger type to all others, for several reasons. In the first place, the heads do not have to be removed for packing and inspection, and on large pumps this is no small item. If the packing is leaking it is seen at a glance, and the glands can be set up, which would not be the case in an inside-packed pump. The soft packing is much easier on the plungers than the hard packing usually used in other styles, and it may be run just barely tight enough to prevent leakage.

It is a mistake to run a pump without lubricating the water end rods and plungers, as well as the steam end, whenever possible. A great many engineers claim the water lubricates them sufficiently, but after trying them with and without oil, it is found that the packing will last much longer and the rods and plungers will be in much better condition when oil is used, to say nothing of the saving in power. Pipe up the water end for oil the same as the steam end, and frequently dust fine graphite on them with an ordinary squirt-can and the surfaces soon become coated with it, and it does not wash off readily.

III

PUMP TROUBLES

FIGURE 5 illustrates one side of a duplex pump fitted with a gland for the stuffing-box that is held in place by nuts on two studs. When the packing in such a stuffing-box begins to leak steam, the engineer proceeds

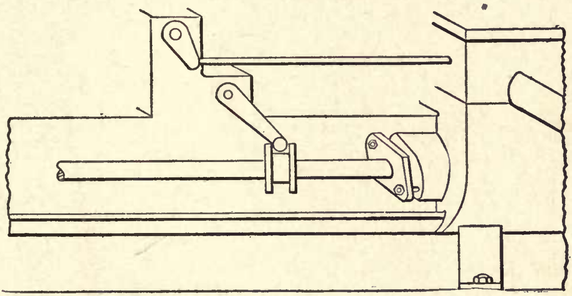


FIG. 5

to screw these nuts on further, but he does not always deem it necessary to turn them both alike, hence the gland binds on the rod, scoring it until it becomes fluted instead of round.

Whenever it is necessary to tighten these nuts, it is a good idea to light a candle and hold it close to the face of gland, then adjust the nuts so that the rod will be exactly central in the gland.

On some of the pumps furnished us these studs are too short, for when packing the stuffing-box the last ring may not go wholly into the box, unless a reasonable pressure is applied to it, and this cannot be done unless the studs are long enough to project through the gland while it is still about ½-inch distant from the box.

Pump manufacturers are beginning to understand this and make the studs accordingly longer, but some of the older ones are defective in this respect. Of course it is possible to get a hardwood stick and a steel hammer and drive the packing into place, but that is an antiquated scheme that should be discarded.

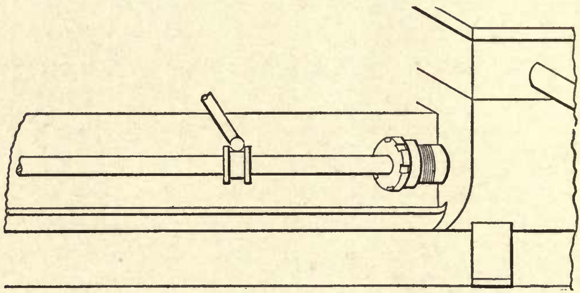


FIG. 6

Figure 6 illustrates another style with which it is impossible to make the above-mentioned mistake, because the gland is much smaller and there are no studs on which to adjust nuts, but instead there is one large nut that is screwed on by means of a spanner wrench.

This nut should be deeper than the length of the gland so that if the packing cannot be pushed entirely into the stuffing-box with the fingers only, the nut will lap over the gland and catch the thread, thus forcing the packing evenly into place.

If the rod is in good order, there is no need of using much force on either kind of gland as a light pressure should answer every purpose. If this does not keep it tight after the packing has been in use for several months, take it out and put in new. It does not pay to use packing until it becomes hard, as it injures the rods more than the value of new packing. When taking out any of the packing, remove all of it, even though it be difficult to get the bottom ring out.

We hear much about pounding in steam-engines, but less about the same trouble with pumps, and the various remedies applied. A pump used for raising cold water, which had previously performed its work very quietly, began to pound in the water cylinder at the end of one stroke, annoying the occupants of a building. Observation taught the engineer that the work of this pump had been increased, and as more water was called for the disagreeable noise grew worse.

He reasoned out the matter as follows. When only a small quantity of water was wanted, the piston did not travel full stroke, therefore the water cylinder was not worn smooth throughout its entire length. When more water was wanted the speed of piston increased and the extra momentum caused a longer stroke; therefore the piston traveled over the comparatively rough part at the end of cylinder, which may have been a trifle smaller than the middle, causing the piston to bind at this point, hence the pound. Taking the cylinder head off, he proceeded to scrape the end of cylinder smooth, and on starting the pump the pound was no longer heard.

It is a good idea to put an air chamber on the discharge pipe of a pump, especially if the water comes to it under pressure, but such a chamber is often worthless, because there is no glass gage on it to enable the engineer to tell where the water level is; consequently the air space fills with water and it is not observed.

A long glass is not necessary because a short one will answer the purpose if set low, as shown in Fig. 7. If the water level is kept low enough to show in this glass, there will be a good body of air above it to act as a cushion for the pump. When the water level rises too high, draw water out of the air chamber and open the pet cock at the top to allow air to flow in. It will be necessary to draw the water down out of sight, because pressure compresses the air, forcing the water level upward.

A duplex pump was used to deliver cold water against a heavy pressure in a mill. One of the steam-pistons struck its cylinder-head at the end of each inner stroke, causing a heavy pound, the cause of which could not be located until the engineer discovered a break in the cylinder-head gasket, as shown in Fig. 8. Both cylinder-heads were cast in one piece and the packing was also in one piece originally, but a small part had broken out, probably when the double head

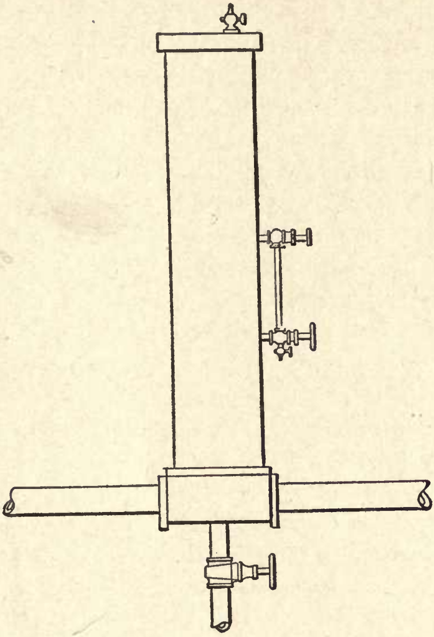


FIG. 7

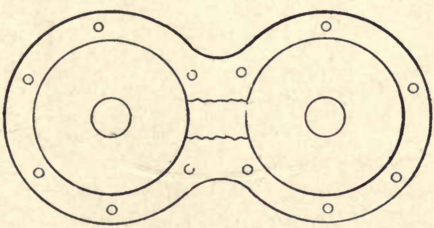


FIG. 8

was removed for inspection, and a new gasket had not been put on, consequently the steam which should have cushioned the piston passed through the small passage to the other cylinder, allowing the piston to strike the head solidly. A new gasket cured the trouble.

A single direct-acting pump began to pound, and the trouble increased as time advanced. The engineer tried to remove the disagreeable noise by adjusting the valve-gear in different ways, without success, after which he had a new valve made with about $\frac{1}{8}$-inch more inside lap, as shown at 2 in Fig. 9, and when it was put in place and steam turned on the pound was no longer heard.

The insurance inspectors came around one day and wished to test the duplex fire pump, and of course they were accommodated. It was run quite fast during the test, taking water from a cistern. After the test was concluded one piston made a stroke much quicker than formerly, denoting a leaky water valve. The water chest of this pump contained forty valves, and under the last one to be examined a stone about the size of a walnut was found. One of the hot-water pumps showed the same defect, and under the first water valve a piece of wood was found. When these were removed the pumps worked perfectly.

A strainer that was packed full of very fine sand caused much trouble. It was at the bottom of a driven well, hence escaped detection for a long time.

In another case the supply of water in an open well was not sufficient; hence the pump sucked air nearly every day, and acted strangely accordingly.

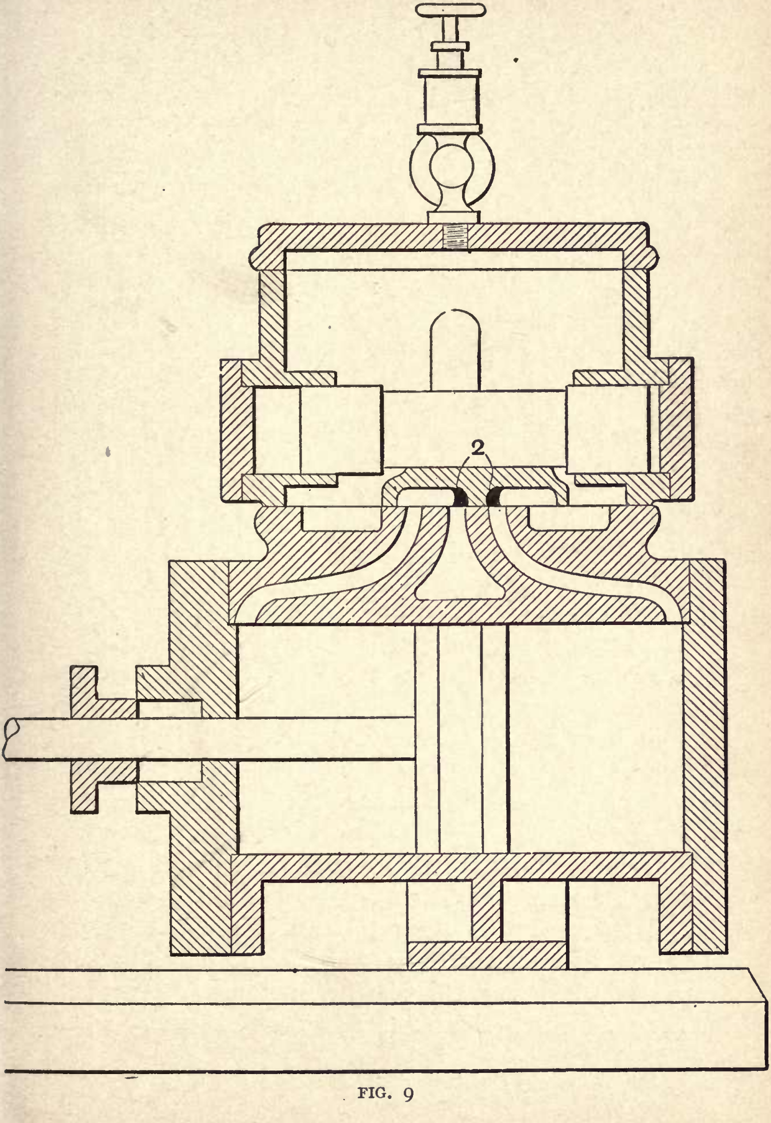


FIG. 9

A crank and fly-wheel pump fitted with brass cylinder, piston, piston-rod and valves complete could not be cured of the habit of groaning caused by pumping very hot and very cold water alternately.

Another pump, used to raise water out of driven wells, could never be made to deliver it against pressure, because it contained a large quantity of air, although no leaks could be found in the suction pipe. When this water was delivered into a cistern, the end of the discharge pipe being below the surface, bubbles of air constantly arose to the surface. After the air had escaped the water was taken out of the cistern and delivered against pressure without trouble.

A pump with no outside valve-gear began to be less and less reliable, and finally refused to work. Careful examination of all the internal parts failed to disclose a defect. As it was a new machine, the manufacturers were notified. They sent a man to remedy the trouble, who brought another steam-chest and auxiliary cylinder with him, which he proceeded to substitute for the defective parts, as it was plain that steam reached both sides of the auxiliary piston, hence it failed to move. He also brought a fine, cup-shaped strainer, which he proceeded to place in the union on the steam pipe.

When steam was again turned on, the pump resumed operations promptly, and has worked well ever since. It is necessary to clean this strainer about once a month, as sediment collects in it, until enough steam to run the pump full speed cannot pass through it. The sediment is sharp and gritty, and is nearly equal to emery for cutting purposes. The strainer is made cup-shaped in order to provide sufficient opening through it to equal the area of the steam-pipe. There is no mystery about the presence of this sediment while the pipes were new; but why should it continue to collect at this point?

One engineer reports that he removed a similar strainer when he found that sediment collected in it and prevented the free passage of steam; but this was evidently a mistake, because it is much better to clean the strainer once a month than to allow the destructive sediment to pass into the steam-chest and cylinder.

IV

PUMP TROUBLES

A SINGLE-CYLINDER boiler feed-pump after being repaired was run for a few months, when it was noticed that the steam piston would not make a full stroke, but stopped about two inches from the cylinder-head. When the drain-cock was opened the piston would finish the stroke. When the cock was closed, after a few strokes the same old trouble appeared. Everything was in good condition at the water end, and the steam piston did not leak.

When the pump was taken apart everything was found O. K., except at the back end of the steam cylinder there was some gummy substance between the piston and the head. The cylinder oil, which was of a very heavy grade, was responsible for the gummy substance, mixing with the water. This matter could not work out of the cylinder, except by way of the drain-cock. When the cock was closed, of course it kept the piston from finishing the stroke. The piston and cylinder were cleaned, a lighter oil was used and there was no more trouble.

This same pump used to give trouble by slacking up or nearly stopping when within 3 inches of the end of the stroke, at each end. After a thorough inspection it was found that the water cylinder was not in line with the steam cylinder, and this caused the piston to bear downward at one end of the stroke and upward at the other end. The bad alignment had been caused by leaving a piece of old gasket on the cylinder face of the water cylinder and the frame when the old one had been replaced with new. The cylinder face was cleaned thoroughly and the parts put together again and the pump ran all right.

Another pump refused to furnish water enough for the boiler. Upon examination, it was seen that the valves at the water end were worn out. New valves were substituted, but the pump did no better then. The water-piston was packed but to no avail. At last it was found that there was scale around the valve-studs and seats, and, the holes in the new valves that had been put in being smaller than those in the old valves, the valves did not properly seat. The scale was removed and plenty of water came after that.

In a good sized station a pump was found with nothing but an ordinary oil cup placed on the steam chest. The pump ran unsteadily and groaned at every stroke. Fig. 10 shows a lubricator as it was fixed up and put on the pump. There is a union at *A*, and another union can be placed between the steam-chest and lubricator, but was not required in this case.

The crack *B* was caused by water freezing in the steam cylinder of the pump. It was patched, as shown in Fig. 11, and then a composition cylinder belonging to the water end of another pump was used and fitted to the inside of the cracked cylinder. The piston was then turned down and new rings fitted to it. At present the pump is giving good service.

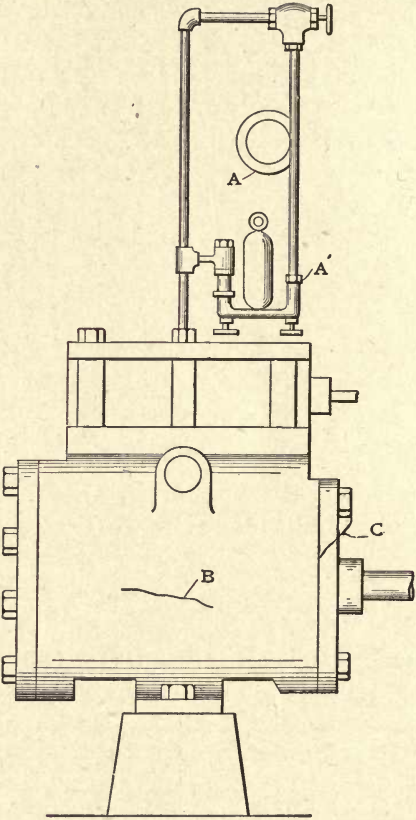


FIG. 10

Once the pump man let the water get below the suction of the pump, with the consequence that the front head was cracked as at *C*, Fig. 10. A clamp shown in Fig. 12 placed around the cylinder proved effective.

Figure 13 shows an old wrinkle, but it may be new

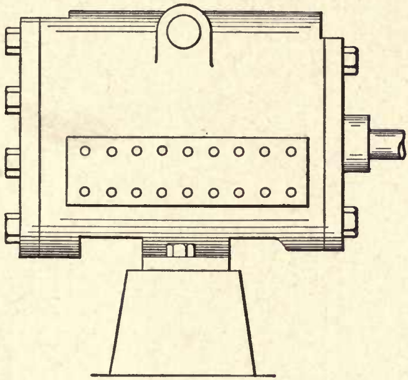


FIG. 11

to some. *A* and *B* are check-valves closed by the atmospheric pressure; *C* is an ordinary globe valve. When starting the pump, air becomes trapped between

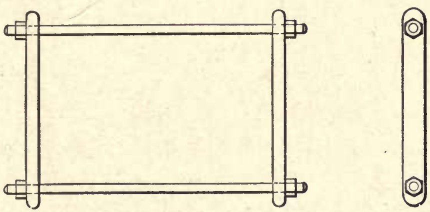


FIG. 12

the delivery valve deck and the suction valves, breaking the vacuum by expanding and contracting in the suction pipe. By opening the valve *C* the air is discharged when starting. As soon as water comes from the valve *C*, close it and the pump will be working properly.

In an isolated plant was found a water supply pump just out of a repair shop giving trouble. On examina-

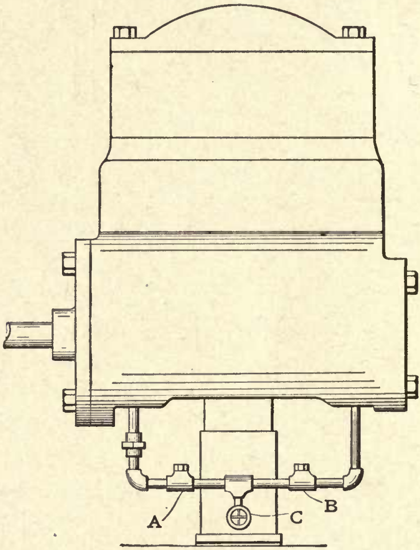


FIG. 13

tion, it was found that the auxiliary valve had not been repaired and was leaking badly (Cameron pump), which prevented its working. The engineer was told what was the matter and that it would be necessary to take the steam-chest back to the shop and bore it and make a new valve. Then as his tank was empty and he needed water if possible to get it, the following plan was executed. Taking the two heads off the steam-chest, the steam was turned on, noting the amount that leaked through each end; then, after removing

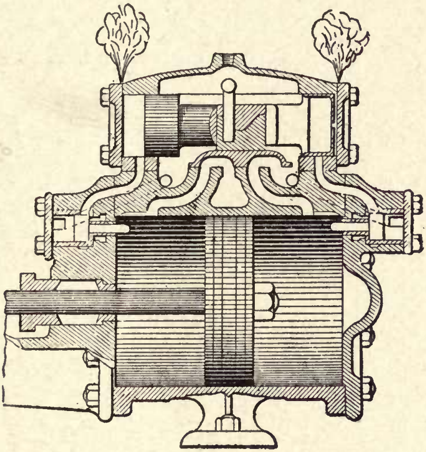


FIG. 14

the gaskets, putting the heads back and screwing the bolts just tight enough to allow the same amount of steam to escape, as in Fig. 14, the pump immediately went to work.

V

SOME PUMP REPAIRS

A CARELESS engineer had started an upright plunger pump in an ice plant with the valves on the delivery line closed, breaking the six-foot cast-iron cylinder into three pieces as shown by the cracks *A–B* and *C–D* in Fig. 15. As the ice plant depended upon this pump for its water, waiting for a new casting from the factory, or the possible longer wait of having a pattern and casting made in one of the local foundries, was out of the question. The warmer city water could have been easily turned in, but every one experienced in ice making and refrigeration knows what a difference of about 20 degrees in the water supply would mean when a plant is pushed constantly to its utmost limit. It would either cut off the greater part of their ice output, or raise the temperature in the cold storage when it was already so high as to be risky. So the engineer was called on to do something.

Two clamps *E* were fitted around the cylinder and four strips *F* fastened the upper pieces to the lower. Then small holes *G* were drilled at intervals all over the cylinder, the outer parts of the holes being countersunk. Then the outside of the cylinder was wrapped with paper and packed around with damp sand in a board casing. A wooden core an inch smaller than the former bore of the pump was centered on the inside of the cylinder and red-hot babbitt poured in. It was expected that it would be a difficult job to get a perfect cast, but the first attempt proved to be a

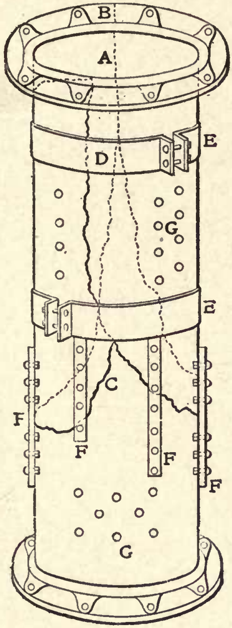


FIG. 15

complete success. It was a simple job to bore this out a little and turn the piston down to fit it. A few burrs of babbitt were cut from the outside and the job was then neat and strong, and the pump gave better service than it had given for years.

Repairing a Pump

Having occasion for the use of another pump, the management decided to buy a new one. Most engineers would have been very willing to have the company go to this expense, but this firm happened to have in their employ one of those careful, saving engineers that are rarely found. This engineer found in the company's pile of junk a pump that had evidently been discarded on account of a crack about 3 inches long in the water cylinder.

Some one had tried calking, but had only opened the crack worse. Somebody, probably the same person, had also made a sort of pocket of clay and attempted the repair by pouring in melted babbitt, but the shrinking of the metal when cooling had been enough to spoil the job.

As the shape of the casting made patching a very difficult undertaking, the engineer decided to try another way. After some difficulty he succeeded in removing the babbitt. He then ground a piece of sheet copper very thin at one end and forced it into the crack. The edge of the copper was then filed off even with the cast iron, and a clamp made of heavy iron was placed around the cylinder and drawn up by the bolt provided for that purpose. The crack was perfectly tight and held for about eight years.

Not long ago this crack again began to leak and no amount of tightening on the clamp would stop it. Another engineer had taken the place of the one who had made the repair. The new man removed the piece of copper and prepared another like the first had been. But after drawing up with the clamp, the crack still leaked, when pumping against more than 80 pounds pressure. As the crack was nearly in a straight line, the new engineer than decided to try another method. A $\frac{3}{16}$-inch hole was drilled along the crack about $\frac{1}{4}$ of an inch. Then a $\frac{1}{4}$-inch copper wire about $3\frac{1}{4}$ inches long was tapered about a thirty-second of an inch and driven into the hole. When the clamp was again tightened, the crack was water-tight against any pressure, and it looks now as though it would hold as long as the pump would last.

After five years' use a set of 150 valve disks needed facing. They were made of hard composition and had been turned over once, so that they were much worn and had ridges in them from $\frac{1}{64}$ to $\frac{1}{16}$ inch deep. A 16-inch bastard file was first used and it was found after cutting three or four, that the teeth of the file were rapidly wearing off. The file was machine cut, and a hand-cut file was procured, but had the same experience with it. Finally the tool shown in the accompanying sketch was made and found to work extremely well.

A piece of scrap cast iron was put in the shaper. With a screw-cutting tool the grooves shown were cut and filled with emery. Extreme care must be used to keep the edges of the grooves exactly even with the top surface of the block. A hole was drilled in the bottom of the block and screwed in a screw-eye, so that the block could be held in a vise. With this arrangement both sides of a valve could be trued up in one minute. The tool, Fig. 16, lasted for the truing of both sides of a full set of valves, and is still good for a dozen more sets.

The emery will get filled up with the granulated rubber which is ground off, but the emery can be saved by heating a piece of iron red hot and putting the emery and rubber on it. The rubber will soon burn up and leave the emery as good as ever.

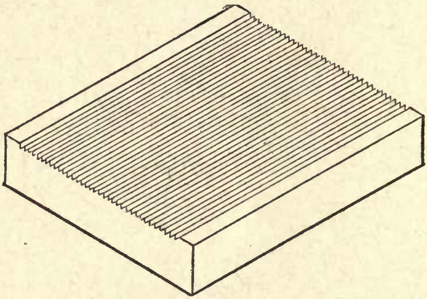


FIG. 16

A duplex pump failed to force water and an examination showed that the valve-seats were badly scored. These seats were expanded into the pump and beaded over so they could not be removed, and to grind them in place was the problem we had to solve.

A $1\frac{1}{2}$-inch coupling was taken, as shown at *J*, Fig. 17, and filled with babbitt and then drilled out for a $\frac{5}{8}$-inch bolt *B*. This bolt was threaded nearly the full length. Four small bolts were driven into holes drilled in the lower end of the coupling as shown at *M*. A bar of iron $\frac{3}{8} \times 1$ inch was shaped as shown in

Fig. 18, bent as in Fig. 19 and placed at *W*, Fig. 17. The nut, Fig. 20, was made from $\frac{7}{8}$-inch iron and shaped to enter the holes *N N* drilled in the bar, Fig. 18. This is shown in place in Fig. 19 and at *C* in Fig. 17. The nut was drilled and tapped out for a $\frac{5}{8}$-inch bolt *Y*, Fig. 19, the outer end fastened to form a handle as

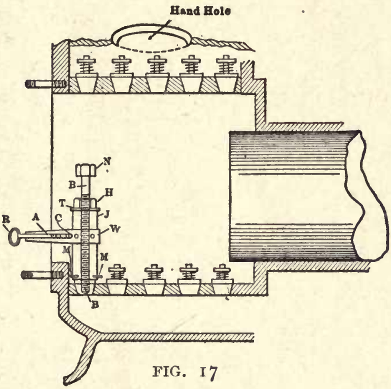


FIG. 17


FIG. 20

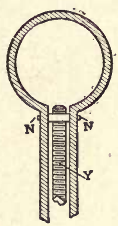


FIG. 19

FIG. 18

at *R*, Fig. 17. Small holes $\frac{3}{8}$ of an inch in diameter were drilled in the side, forming a circle around the coupling. They were spaced $\frac{3}{4}$ inch apart and are shown in the collar *W*, Fig. 17. An old metal valve was taken and ground good and level on the face. Cavities were sawed and chipped out to allow the four pins at *M*, Fig. 17, to drop into them, and by making this connection the body *J* could not turn without turning the valve. The lower end of the bolt *B* was screwed into the hole at the center of the valve-seat where the stud holds the valve in place under working conditions. A good coat of lard oil and ground emery was placed between the valve and the seat to be ground, and the lower end of the body *J* was set up snug enough against the valve to allow the coupling to turn and bring a force on the top of the valve. The strap *W* was turned until the taper end of the bolt *C* came in line with a hole in the body *J* at *W*, when the $\frac{5}{8}$-inch bolt was turned in at the handle *R* and this made it fast to the coupling. The bolt *B* was screwed into the seat tightly so that it could not turn. The handle *N* was wound back and ahead, which moved the body *J* and with it the valve. To change the position of the valve, the bolt *C* was backed out and screwed into another hole to put on more energy. The nut *J* was backed off, the coupling raised and the seat examined. It will be seen that by backing out the bolt *C* and putting it into other holes in the body *J* a full revolution can be made. The job was done and well done too, for the old pump throws water good and fast. Fig. 17 shows the tool in place; the valve chamber being an extension of the water cylinder. To grind the suction-valve seats the head was taken off, but the discharge seats were ground through a hand-hole 12 x 7 inches located as shown in Fig. 17. It is well to mention that the strap *W* is free to move up or down or turn around the body *J* when the bolt *C* is not in one of the holes *W*.

Sometimes the thread on the inside of a stuffing-box gland becomes so burred or broken that it can not be started back on the box thread, and it means the taking apart of the pumps to get the necessary repair made unless some method of doing the work in place

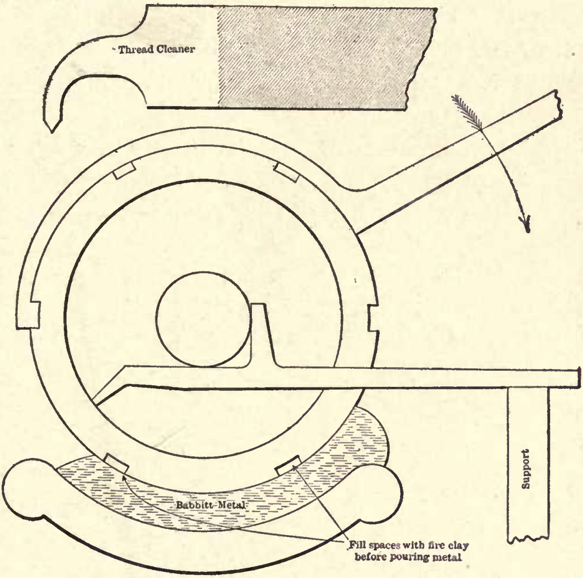


FIG. 21

can be devised. The following method is a good one. Referring to Fig. 21. Move the nut with injured thread out along the rod away from the stuffing-box far enough to get at the thread.

Fill the spaces into which the spanner fits with fire clay and after packing around the rod to hold the nut central, pour babbitt between the nut and frame as shown in figure. Then make a tool such as shown in cut with a V-shaped point to fit the thread. Then clean out the fire clay from the spanner spaces, and using the spanner to turn the nut apply the tool as shown and clean out the thread. After the thread is clear the babbitt can be cleared out of the way and the nut screwed up in place.

VI

SETTING VALVES OF DUPLEX PUMP[1]

As is well known, the slide valves of a duplex pump have neither outside nor inside lap. This is necessary to prevent the pump from stopping should the valves be in a position to cover all ports. By making the length of the valve the exact distance from the outside edge to the outside edge of the steam port, and the exhaust cavity the exact distance from the inside edge to the inside edge of the exhaust port, there is only one point in the travel of the valve where ports are completely closed; and it is not likely, if it ever should happen that both valves were in this position, that the pump would fail to start off, for the leakage of steam past the edges of the valves will never be exactly the same in all four corners, therefore the equilibrium would be destroyed quickly.

By setting the outside edges of the valves "line on line" with the outside edges of the steam ports, the valves will stand in a central position. If, then, both rocker arms are put in a central or vertical position, the clearance on the valve rod must be the same on both ends. In Fig. 22 this clearance is shown inside of the steam-chest and is marked *C*. On larger pumps

[1] Contributed to Power by F. F. Nickel.

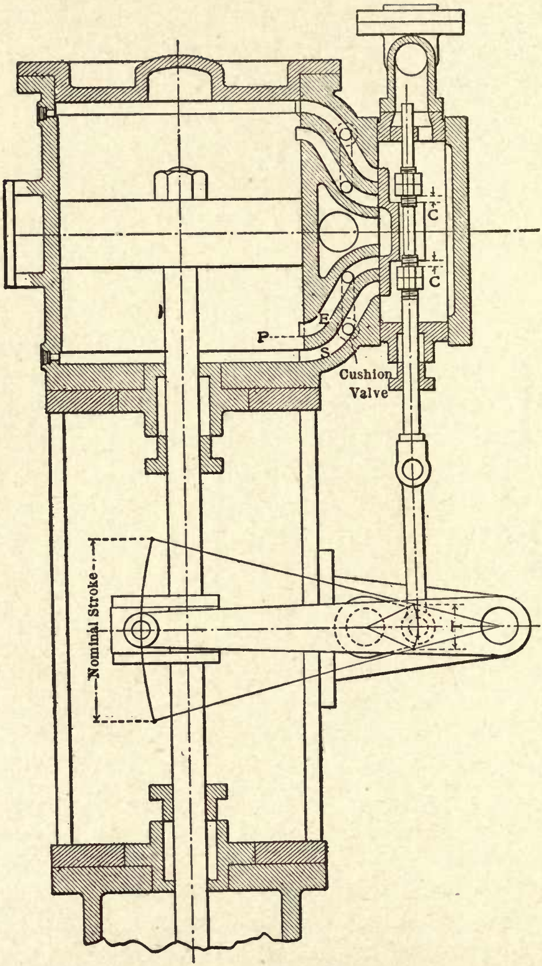


FIG. 22

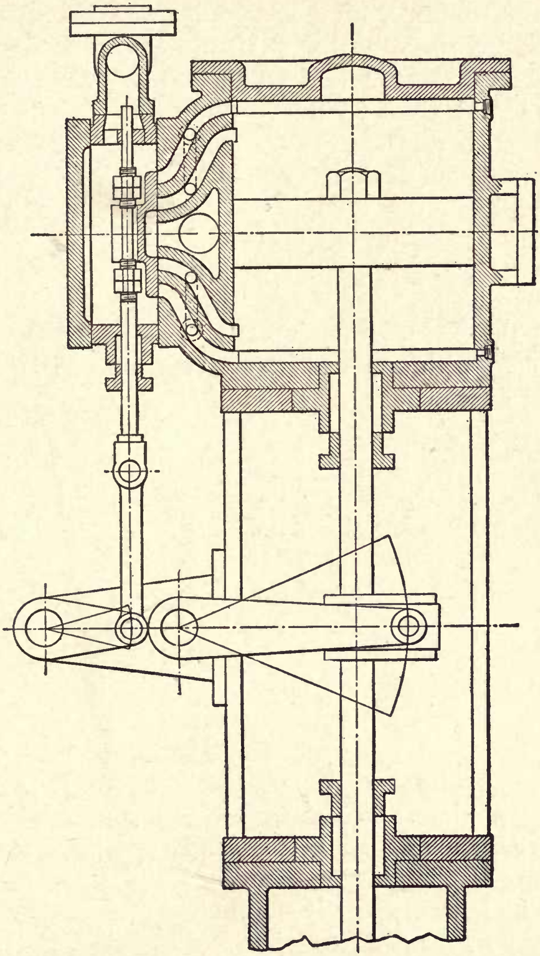


FIG. 23

usually, a lost-motion link is inserted between the crank and the valve-rod clevis, which can be adjusted without taking off the steam-chest cover. No fixed rule can be given for the amount of this clearance, as it must be adjusted to suit the working of the pump.

On a pump of ordinary proportion, such as a boiler feed pump, the total clearance, 2 C, should equal about 25 per cent. of the travel T of the crank-pin at nominal stroke. On a low-service pump (also on a pressure pump for moderate pressure) it is often found that the reciprocating parts are so heavy that the cushion, with the cushion valve shut tight, is not sufficient to stop the motion of the piston at the end of the stroke. In this case the lost motion should all be taken up. If the piston does not make a full stroke, the lost motion may be increased somewhat above the figure given, but it must be kept in mind that this will reduce the travel of the valve and the port opening, and thus may affect the speed of the pump.

The Cross-exhaust Valve

In the case of a compound pump there is still another appliance that can be brought into action to regulate the length of the stroke, and that is a connection, provided with a valve, between the two high-pressure exhaust pipes. The object of this connection is to equalize the pressure in these exhaust pipes and make it more uniform. This is called the cross exhaust, and its influence on the distribution of steam is clearly shown by Figs. 24 to 27 inclusive. Figs. 24 to 27, inclusive, are convenient sectional plans of the steam cylinders of a compound pump, with the pistons in positions that correspond to lines *A B* and *B — C* in the diagram Fig. 29. Fig. 28 represents a diagram with the cross exhaust closed. The steam pressure follows up the full stroke in the high-pressure cylinder,

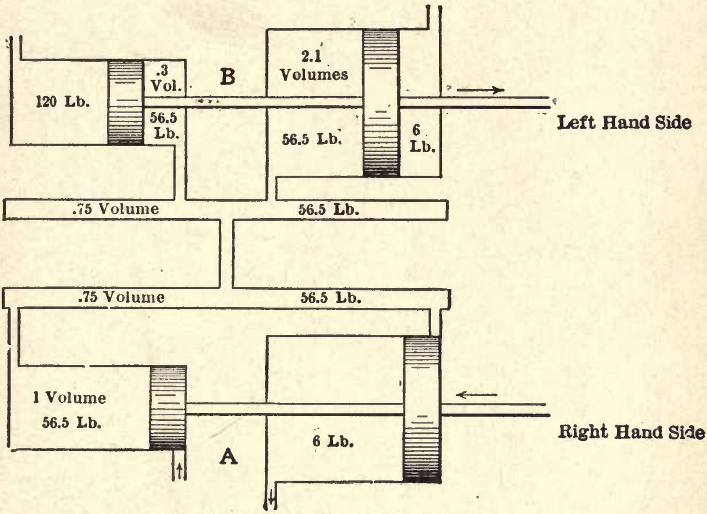


FIG. 24

and when the exhaust valve opens it blows into the intermediate space and mixes with the steam left therein from the preceding stroke.

Assuming the intermediate space to have a volume equal to 0.75 of that of the high-pressure cylinder and a cylinder ratio of 1 to 3, we have the following volumes:

High-pressure cylinder = 1; intermediate space = 0.75; low-pressure cylinder = 3.

Clearances are neglected, as it is only intended to show the action of the cross exhaust. We will also

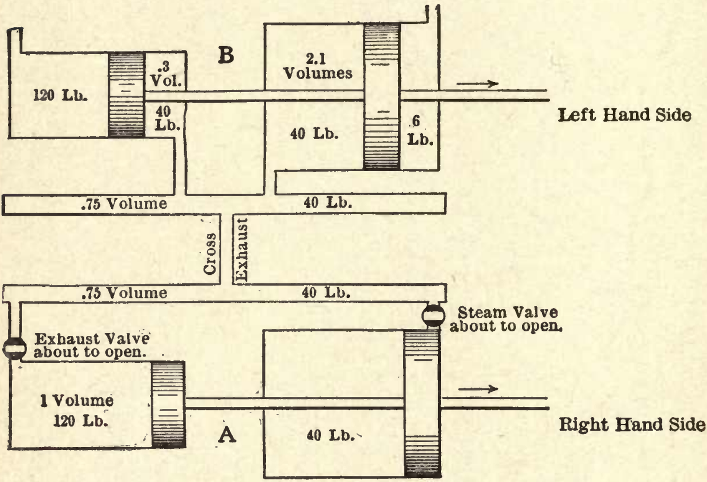


FIG. 25

assume that the steam expands according to Mariotte's law.

$$p \times v = \text{constant},$$

which is sufficiently accurate for our purpose, and assists greatly in getting a clear conception of the behavior of the steam as it passes through the various stages.

The amount of steam passing through one side of the engine is evidently one high-pressure cylinder full at initial pressure. Its measure is $p \times v = 120 \times 1 = 120$ lbs. When the high-pressure exhaust valve opens, this steam flows into the intermediate space, where it meets and mixes with steam that was left there from the preceding stroke. This steam was shut off from

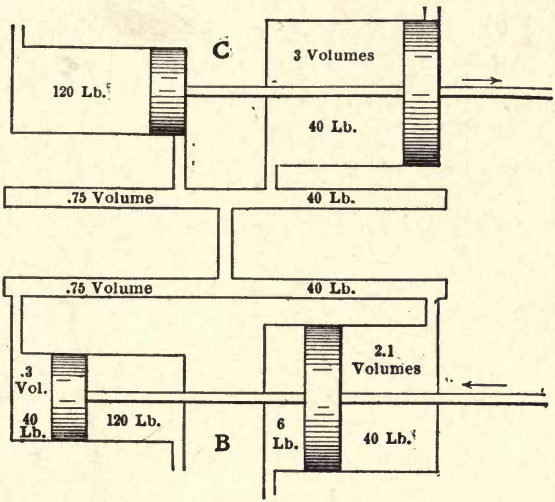


FIG. 26

its communication with the steam in the low-pressure cylinder, when its exhaust valve opened and must be at the same pressure as the steam in the low-pressure cylinder at the point of exhaust. As the ratio of cylinders was assumed to be as 1 to 3, the steam expands three times as it passes from the high-pressure cylinder to the low-pressure cylinder, and the terminal pressure is therefore

$$\frac{120}{3} = 40 \text{ lbs.}$$

It will be noted that 120 is a measure for the steam passing through the engine and this amount is accounted for by the indicator diagram at every point of the stroke. Thus we have:

High-pressure cylinder, $p \times v = 120 \times 1 = 120$.

Low-pressure cylinder, $p \times v = 40 \times 3 = 120$.

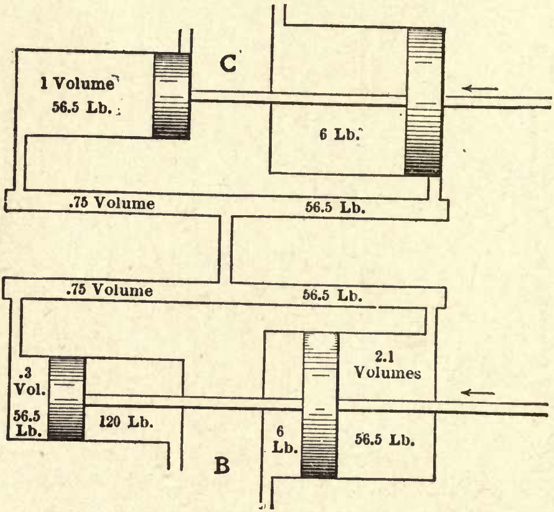


FIG. 27

The amount of steam that is constant and remains in the intermediate space is $0.75 \times 40 = 30$ lbs.; the two combined give $120 + 30 = 150$ lbs., which when distributed over a volume of $1 + 0.75 = 1.75$ results in a pressure of

$$\frac{150}{1.75} = 85 \text{ lbs.}$$

This means that when the high-pressure exhaust valve opens the steam expands from the high-pressure cylinder into the intermediate space from 120 to 85 lbs. without doing any useful work. From 85 lbs. it then expands from the high-pressure cylinder through the intermediate space into the low-pressure cylinder doing useful work upon the low-pressure piston.

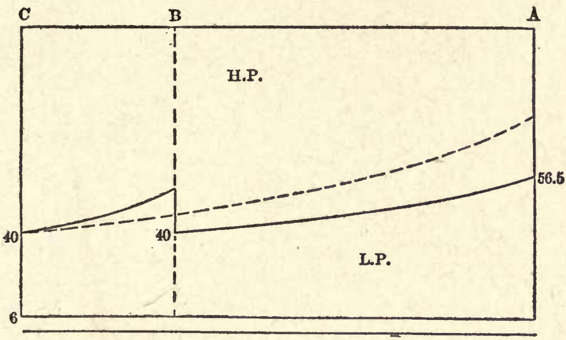


FIG. 28

With two points of the expansion curve, namely, 85 lbs. at the beginning and 40 lbs. at the end of the stroke, it is now easy to construct the remainder of the curve, as it is only necessary to complete the rectangle and draw the diagonal. Where this diagonal meets the line of zero pressure, there is point *o*, the zero point of pressure and volume. Any line drawn through this point *o* will give the volume on the line 85, Fig. 28, and its corresponding pressure on line *A*, Fig. 29.

Under the conditions indicated in Fig. 28, it cannot be expected that an ordinary pump will work satisfactorily, as the following comparison of the steam forces will show.

Beginning of stroke:

$$\begin{array}{ll} \text{H. P., } 120 - 85 = & 35 \\ \text{L. P., } 85 - 6 = 79 \times 3 = & \underline{237} \\ \quad \text{Total steam force} \ldots\ldots & 272 \text{ lbs.} \end{array}$$

End of stroke:

$$\begin{array}{ll} \text{H. P., } 120 - 40 = & 80 \\ \text{L. P., } 40 - 6 = 34 \times 3 = & \underline{102} \\ \quad \text{Total steam force} \ldots\ldots & 182 \text{ lbs.} \end{array}$$

The average of the two, or

$$\frac{272 + 182}{2} = 227 \text{ lbs.,}$$

is a measure of the resistance which, in a pump, is constant throughout the stroke. There is, therefore, at the beginning of the stroke, a surplus of $272 - 227 = 45$ lbs., and at the end a deficiency of $227 - 182 = 45$ lbs. If, however, the cross exhaust is opened, it equalizes these two forces to a certain extent and modifies the diagram, as shown in Fig. 29.

With the assistance of Figs. 24 to 27, inclusive, it is easy to follow the steam through its various stages. In Fig. 24 the pistons of the right-hand side have completed the stroke and are about to return. The cylinders on the other side and intermediate spaces are filled with steam at the low-pressure terminal, or 40 lbs. The total amount of steam is then

$120 \times 1 =$	120
$40 \times 3.9 =$	156
Total	276

which divided by the volume, 4.9, gives a resulting pressure of

$$\frac{276}{4.9} = 56.5 \text{ lbs.},$$

as shown in Fig. 25. This increased pressure gives the low-pressure piston of the left-hand side an additional push and enables it to complete its stroke while

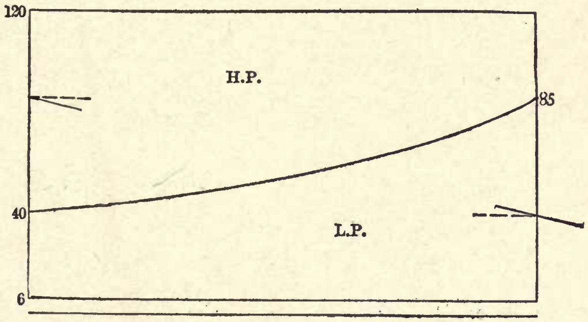


FIG. 29

the steam expands down to 40 lbs. again. Then the steam from the left-hand high-pressure cylinder flows into the intermediate space and raises the pressure to 56.5 lbs., in order to help out the right-hand low-pressure piston.

Fig. 29 shows this action clearly, but in practice the rise in pressure will not be as abrupt as shown there, as the pulsations in the pipes will still more equalize the differences and produce a practically uniform pressure in the intermediate space.

It will also be noted that by opening the cross exhaust, pressure is removed from the low-pressure piston and shifted over to the high-pressure piston which results in a loss of power and reduced speed of the pump.

The cross exhaust should therefore be kept closed whenever the pump runs fairly well in this condition.

VII

ANOTHER METHOD OF SETTING DUPLEX PUMP VALVES

In setting the valves of a duplex pump, first remove the steam-chest cover; next move the piston-rod toward the steam-cylinder head until the steam-piston strikes the head solid, and make a pencil mark on the rod at the face of the steam stuffing-box, as shown in Fig. 30 at *A*. Now move the piston to the opposite

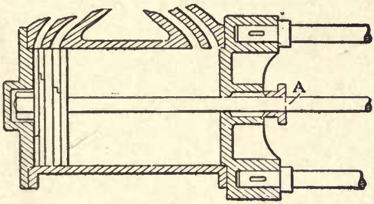


FIG. 30

end until the steam-piston strikes solid, and make another scriber mark exactly half-way between the first mark and the face of the steam stuffing-box, as shown in Fig. 31 at *B*. Then move the piston-rod back until this second mark comes flush with the face of the steam stuffing-box, and now the piston will be at mid-stroke, as shown in Fig. 32. Now take off the steam-chest cover, and place the slide valve exactly in the center or over the steam ports, and set the slide-valve nut exactly in the center between the lugs on the valve, as shown in Fig. 33. Screw the valve-stem

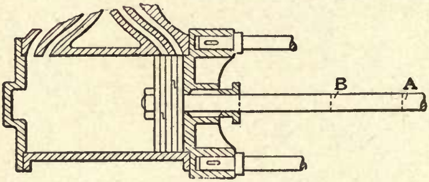


FIG. 31

through the nut until the eye of the knuckle joint is in line with the eye of the link, then slip the link into place. Now we have the valve set on one side, and

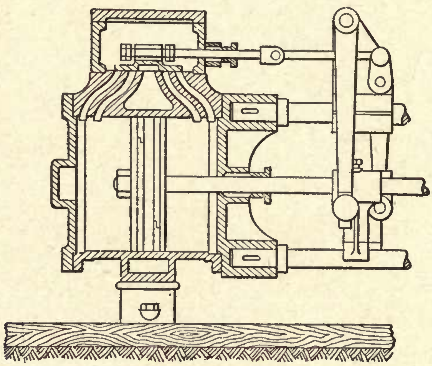


FIG. 32

after repeating this process on the other side of the pump, the valve setting will be completed.

Some pumps are provided with two nuts on the valve-stem on each side of the valve lugs, instead of one nut between the lugs, as shown in Fig. 34. These are set

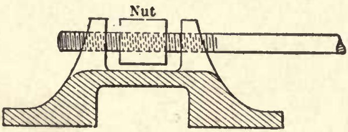


FIG. 33

and locked from the outer faces of the valve lugs, allowing a little lost motion on each side of the lugs. Some make this lost motion equal to half the width of

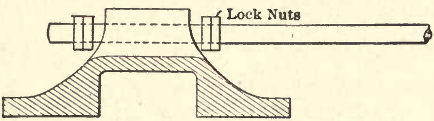


FIG. 34

the steam ports, but sometimes this gives the pump too much or too little stroke, and must be changed accordingly.

VIII

A CENTRIFUGAL PUMP TROUBLE

THE equipment of a centrifugal pumping plant consisted of a 10 x 30 Corliss engine and a 10-inch pump with boilers and accessories, which outfit was to throw 4500 gallons per minute to a hight of 50 feet, 12 feet of which was suction lift.

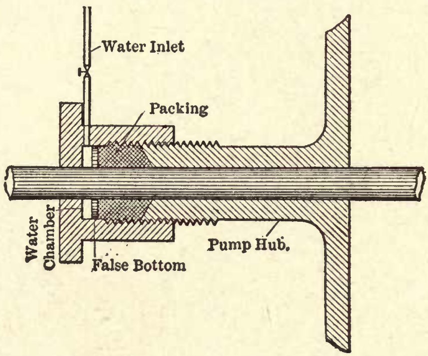


FIG. 35

The pump was speeded so high that the stuffing-box could not be kept tight and cool at the same time. The speed could not be reduced, and the packing burnt out repeatedly. Water and oil applied in the ordinary manner failed to overcome the trouble.

After several days of delay, the gland was taken off and a false bottom inserted on the air side of the packing, leaving a chamber about $\frac{3}{8}$ of an inch deep. This chamber was tapped for a $\frac{1}{4}$-inch pipe, a valve was put on at the gland and the pipe connected into the main discharge of the pump. This practically made a water-packed pump; the packing was left loose and the pump forced to take water instead of air. We had no more trouble with the gland and the packing lasted almost indefinitely.

The change is illustrated in Fig. 35.

IX

BOILER FEED-PUMPS[1]

In selecting a feed-pump two factors enter into consideration, namely, capacity and speed. By capacity is meant the average quantity of water that the boiler which the feed-pump is to supply is capable of evaporating in a certain time, and it is clear that the feed-pump selected should be large enough to supply the maximum quantity of water that can be evaporated in the boiler. At the Centennial Exhibition a standard of 30 pounds per horse-power per hour was adopted and while this is a safe figure to use when calculating the size of boiler required for a steam-engine, it is too low to be considered as a basis for selecting feed-pumps, as the hereinafter considerations will show.

It is general practice among builders to furnish about 12 square feet of heating surface per horse-power, and it has been found that but little decrease of economy will take place if the boiler is forced to evaporate 4 pounds per hour per square foot of heating surface, instead of the $2\frac{1}{2}$ pounds called for by the Centennial standard.

The A.S.M.E. committee on "Trial of Steam Boilers," in 1884 reported as its opinion that a boiler

[1] Contributed to Power by F. F. Nickel.

should be capable of developing its rated horse-power with easy firing, moderate draft and ordinary fuel, and further that it should be capable of delivering at least one-third more than its rated power to meet emergencies.

These considerations led to the adoption of 45 pounds per horse-power per hour as the quantity for which a boiler feed-pump should be calculated. This quantity must be delivered to the boiler at moderate speed, so that in case of low-water level in the boiler the pump can be speeded and the deficiency made up promptly. It is therefore good practice to reduce the speed of the boiler feed-pump to one-half of what the pump would be rated at for regular service.

The accompanying table gives average sizes of feed-pumps as furnished by the various builders, together with the proper speed, capacity and horse-power of the boilers they are intended to supply.

BOILER FEED-PUMPS

Size	Gallons per Revolution	Strokes per Minute	Piston Speed, Ft. per Min.	Gallons per Minute.	Pounds per Hour	Horse-power of Boiler Supplied at 45-lb. Rate
3 × 2 × 3	0.155	80	20	6.2	3,100	70
$3\frac{1}{2}$ × $2\frac{1}{4}$ × 4	0.265	76	25	10	5,000	110
$4\frac{1}{2}$ × $2\frac{3}{4}$ × 4	0.395	76	25	15	7,500	170
$5\frac{1}{4}$ × $3\frac{1}{2}$ × 5	0.805	70	29	28	14,000	310
6 × 4 × 6	1.265	66	33	42	21,000	470
$7\frac{1}{2}$ × $4\frac{1}{2}$ × 6	1.6	66	33	53	25,150	560
$7\frac{1}{2}$ × $4\frac{1}{2}$ × 8	2.15	60	40	65	32,500	720
$7\frac{1}{2}$ × $4\frac{1}{2}$ × 10	2.66	54	45	72	36,000	800
8 × 5 × 10	3.25	54	45	88	44,000	1000
9 × $5\frac{1}{4}$ × 10	3.6	54	45	97	48,500	1100
10 × 6 × 10	4.75	54	45	128	64,000	1400
12 × 7 × 10	6.45	54	45	174	87,000	1900
12 × 7 × 12	7.75	48	48	186	93,000	2100
14 × $8\frac{1}{2}$ × 10	9.55	54	45	258	129,000	2900
14 × $8\frac{1}{2}$ × 12	11.5	48	48	276	138,000	3100
16 × $10\frac{1}{4}$ × 10	14.0	54	45	378	189,000	4200
16 × $10\frac{1}{4}$ × 12	16.7	48	48	401	200,500	4500
$18\frac{1}{2}$ × 12 × 10	19.2	54	45	518	259,000	5800
$18\frac{1}{2}$ × 12 × 12	23.0	48	48	552	276,000	6100
20 × 14 × 10	26.4	54	45	713	356,500	8000
20 × 14 × 12	31.5	48	48	756	378,000	8400

X

HORSE-POWER OF PUMP[1]

A PUMP in doing a certain amount of work is known to consume 5 horse-power. The pump is $1\frac{1}{2}$ x 6 inches, with 15 strokes. The water is discharged into a reservoir, and the work of pumping the water through the pipe line requires 5 horse-power. Required, the pressure per square inch against which the plunger is pumping, all losses to be neglected.

The quantity of water which the pump is delivering must first be found. As the diameter of the plunger is $1\frac{1}{2}$ inches, the area is 1.767 square inches, and as the stroke is 6 inches and there are 15 strokes per minute, $1.767 \times 6 \times 15 = 159.03$ cubic inches of water pumped per minute.

If it were not known just what horse-power the pump was consuming, it could be found from the following formula:

Pounds of water pumped per minute $\times$ head in feet $\div$ 33,000.

But as the horse-power is known in this instance, $33{,}000 \times 5 \div$ pounds of water pumped per minute = head in feet. Since 1 pound of water contains 27.7 cubic inches, $159.03 \div 27.7 = 5.741$ pounds of water

[1] Contributed to Power by Frank L. Ferguson.

discharged per minute, so that by inserting this value in the formula we have: $33{,}000 \times 5 \div 5.741 = 28{,}740$ feet head, which the water is pumped against to consume 5 horse-power.

If it is desired to know the equivalent pressure per square inch acting against the pump plunger, all that is necessary is to multiply 28,740 by $0.434 = 12{,}473.16$ pounds per square inch pressure, as every foot-head equals a pressure of 0.434 pounds per square inch.

XI

TO INDICATE THE AMOUNT OF FEED-WATER PUMPED INTO BOILERS[1]

In no other part of the power plant is there more urgent need of some simple device for indicating the amount of work being done than with the apparatus for feeding water into the boilers. A counter for recording the number of pump strokes, a water meter, or other method of measuring the water, will show the amount pumped during a given period, but will not indicate the rate of flow at any instant. What is wanted is the equivalent of the ammeter on a switchboard. The gage glass cannot be said to be this equivalent, as it only shows the water level, not the rate at which the water is going in. An engineer who knows his plant can judge fairly well how much he is putting in by the speed of the pump, when that is in sight; but in many plants the practice of locating the feed-pumps in a separate room from the boiler room often renders even this unavailable.

It must be understood that these remarks apply only to the normal running conditions of the plant, as of course when tests are being taken suitable provision

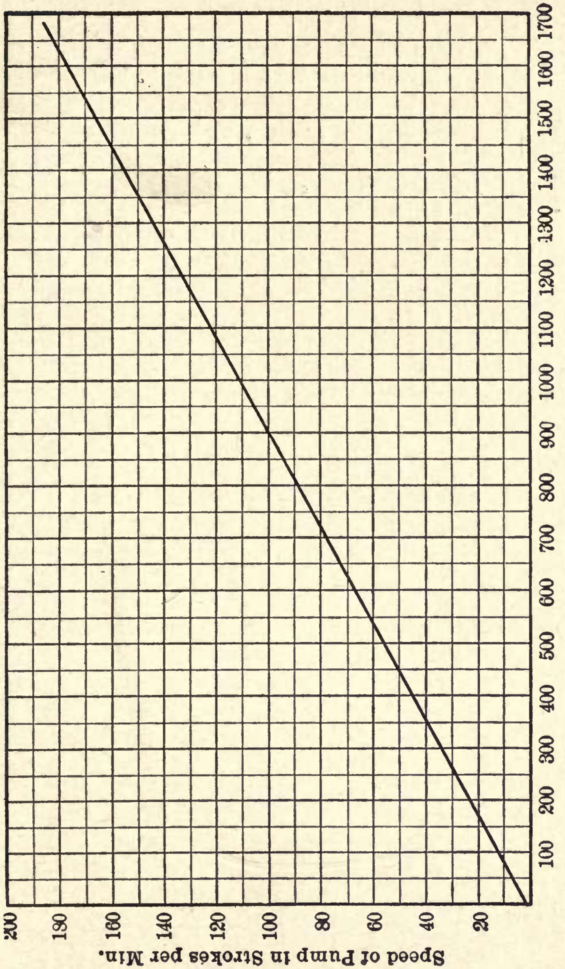
[1] Contributed to Power by F. Sanford.

must be made for accurately measuring the actual amount of feed-water used during a given period.

In a certain plant, consisting of six Babcock & Wilcox boilers, each capable of evaporating 12,000 pounds per hour, duplex steam-driven pumps were installed to supply the necessary feed-water. These were placed in the boiler room, and difficulty was experienced in keeping them in good condition on account of coal dust, which cut the pump plungers badly and caused excessive wear generally.

It was decided to install a motor-driven pump in an engine room at the rear of the boiler house, where it would be under the care of the engineer and free from the dust and grit. The controller was located in the boiler room. Current was obtained from the 3-wire system, making two voltages available, which allowed ample variation in the speed of the pump to meet the varying boiler loads. With this arrangement it was found necessary to provide means of indicating, so the boiler attendant could see whether the pump was running properly and at what speed.

The pump was first tested to ascertain the amounts of water delivered at various speeds against a head equal to the boiler pressure, and from these data a curve was plotted as shown in the accompanying diagram. A small 10-volt, shunt-wound generator, such as is used for gas-engine ignition, was connected to the pump and driven by a belt from the motor spindle. The shunt field was excited from the 125-volt direct-current mains, with a 32-candle-power lamp in series, thus obtaining a practically constant field strength.



Curve showing amounts of water in gallons delivered at various speeds

The voltage obtained from the small generator was then exactly proportional to the speed of the the pump.

The next step was to install a large illuminated-dial voltmeter directly above the controller, in which position it was visible from any part of the boiler room. Instead of showing volts, the voltmeter dial was marked to indicate pounds of water per minute, which were proportional to the speed of the pump, as shown by the curve. To check up the indications of the meter at any time, it was only necessary to take the pump speed and compare the meter reading with the corresponding amount of water shown on the curve. It was not necessary to take the efficiency of the pump into consideration, as the curve was plotted from the actual amount of water delivered at various speeds.

This device was found extremely useful to the boiler attendant, who, being able to read the amount he was pumping at any time, and knowing from long practice the variation of load on his boilers was able to anticipate the demand and keep the water level constant.

XII

PUMPING TAR AND OTHER HEAVY LIQUIDS

In many industries it is necessary to force heavy, viscous liquids through pipes. This involves difficulties not encountered in ordinary pumping, and requires machinery special in design and construction. When the liquid is heavy but not adhesive, as in the case of heavy oils, the action can be made fairly satisfactory and efficient by enlarging the valve openings, making the parts of the pump heavier and so arranging the passages of the pump that there is little liability of choking or clogging. When, however, the liquid is a fluid at high temperatures and a gelatinous adhesive paste or a rubbery solid, clinging to all surfaces and choking openings through which it should pass, as the temperature is lowered, a design differing materially from the ordinary pump must be used.

Tar, molasses and cocoa liquor present more obstacles to pumping than any other substances which it has been found feasible to move in this manner. Each of these liquids thickens into an almost solid mass when cold, rendering it very difficult to start the pump, if some special provision is not made and ample power provided. Another action which must be taken into account is the contraction of the area of the passages and valves as the liquid cools and the consequent throttling which interferes with the liquid's passage and which the pump is forced to overcome. The skin friction of a liquid of this kind creates heat enough to partially alleviate this tendency to throttling when the velocity of the substance is maintained above a certain point and the pipe is not in such a position that the surrounding air will lower the temperature of the liquid below the solidifying point. Although not a common practice, it is well to lag all exposed piping used for conveying heavy oils or other substances of a similar nature.

Gas tar has a number of characteristics rendering it exceptionally difficult to pump. Its condition varies from a solid to a penetrating fluid within a small range of temperature. Two pumps which have proved very efficient in lifting and forcing gas tar were installed a short time ago at the plant of the Maryland Steel Company, of Sparrows Point, Maryland. They are of the standard triplex type, fitted with ball valves peculiarly adapted to this service. The exclusive use of gate valves in the piping system is also interesting. A very flexible power connection is obtained by the use of the Renolds silent chain and a 4-pole alternating-current 3-horse-power motor. The gearing consists of an 18-tooth pinion running at 950 revolutions per minute and a 120-tooth wheel running at 142 revolutions per minute. The chain used has links $\frac{3}{4}$-inch wide and $1\frac{1}{2}$ inches long. It transmits the 3-horse-power generated at 950 feet per minute, giving an excellent efficiency when the service is considered.

A liquid peculiarly difficult to handle is oil-refinery tar, which is usually very hot when it reaches the pump. There is a large percentage of suspended particles of various sizes present in this tar and also a certain amount of unrefined paraffine. The tar is sometimes heated to a temperature of 300 degrees; but quickly cools off if not properly handled, and coats the retaining valves and walls with layers of an adhesive substance closely resembling finely divided particles of coke. To overcome the difficulties the ordinary pump arrangement and design is materially changed.

A special pump for handling oil-refinery tar at the works of the Atlantic Refining Company, in Philadelphia has been designed. By a new arrangement, exceptionally large valve areas are made available, the valves being designed to permit the passage of the substance pumped with the least possible frictional resistance. The suction, discharge and pulsation chambers can be taken apart without unnecessary expenditure of time or labor, and each is in a position where it can be readily reached for cleaning. The pump is of the triplex type, and is fitted with ball valves, which, through test, have proved best adapted for the passage of heavy substances. There are a number of large hand-holes for cleaning the valves.

Machinery which will pump these adhesive oils and other similar substances can be used in many industries, and will save the laborious processes by which this class of work is generally accomplished.

XIII

PUMPING MACHINERY PERFORMANCES

THE table which is on the opposite page was compiled by a prominent engineer in studying the work of municipal pumping machinery, and is a table giving the performances of twenty of the best known and most efficient pumping engines.

The vertical compound engine (No. 2) is notable as holding the record for a compound engine, having shown, on a 6-day test, a duty of 148,655,000 foot-pounds, with a steam consumption of 12.15 pounds per indicated horse-power hour, and having given an average annual duty of about 120,000,000 foot-pounds per 100 pounds of coal, a performance equaling that of many triple-expansion plants.

The Nordberg quadruple-expansion engine (No. 7 in the table) is notable for having established the record for low heat consumption, the figures being 162,132,500 foot-pounds per 1,000,000 B.T.U., or 186 B.T.U. per indicated horse-power per minute, the thermal efficiency being about 22.8 per cent.

The duties usually guaranteed per 1,000 pounds of dry steam are about 60,000,000 for compound condensing, 90,000,000 for triple-expansion condensing, and 110,000,000 for compounds with high-duty attachments, and 130,000,000 for triple-expansion machines with high-duty attachments.

PERFORMANCES OF PUMPING ENGINES, 1893–1903

The engines are of the Triple type, with two exceptions; on line 2 (E. D. Leavitt,) which is Compound; and on line 7 (Nordberg Mfg. Co.) which is Quadruple.

	Builder or Designer	Date of Test	Location of Engine	Capacity, in gallons per 24 hours. (In millions)	Piston Speed, in feet	Head		Steam Pressure, in lbs	Indicated Horse-power	Percentage of Mechanical Efficiency	Steam, I. H. P., hour	B. T. U., I. H. P., min.	Duty, in foot-pounds per 1,000,000 B. T. U.	Duty, in foot-pounds per 1,000 pounds Steam	Percentage of Thermal Efficiency
						Feet	Pounds								
1	Edward P. Allis Co..	1893	Milwaukee, Wis.....	18	203	162.6	70.4	121	574	91	11.68	217.6	137,656,000	154,048,700	19.40
2	E. D. Leavitt	1894	Louisville, Ky	16	371	193.3	83.7	137	643	93	12.15	222.5	137,565,000	148,655,000	19.07
3	E. D. Leavitt	1895	Boston, Mass.	20	607	137.2	59.4	176	576	89.5	[1]11.22	[1]204.3	[1]144,500,000	[1]157,843,000	20.76
4	Lake Erie Eng. Wks.	1897	Buffalo, N. Y.	30	208	199	86	167	1,186	95	12.39	231.2	135,000,000	152,000,000	18.35
5	Snow St. Pump Co. .	1898	Indianapolis, Ind....	20	215	205	89	156	776	95.4	11.26	207.6	150,000,000	167,800,000	20.45
6	Edward P. Allis Co..	1899	Cleveland, Ohio.....	20	200	202	87.5	149	770	93.7	11.45	213	145,220,000	161,990,000	19.9
7	Nordberg Mfg. Co. .	1899	Wildwood, Pa	6	256	604	262	200	712	93	12.26	186	162,132,500	149,500,000	22.8
8	Edward P. Allis Co..	1899	St.Louis, Mo.,No. 7.	10	175	294	127.5	136	549	94.3	11.63	216.8	143,404,000	160,455,000	18.44
9	Edward P. Allis Co..	1899	St. Louis, Mo.,No.8.	10	175	294	127.5	137	545	94.9	11.63	216.5	144,365,000	161,530,000	18.59
10	Edward P. Allis Co..	1900	Hackensack, N. J....	12	211	267	115	173	603	94	11.05	211.9	146,403,400	168,532,800	20.00
11	Edward P. Allis Co..	1900	St. Louis, Mo., No. 9.	15	198	293	127	127	813	96	10.78	204.4	155,237,450	176,419,600	20.78
12	Edward P. Allis Co..	1900	St. Louis, Mo., No. 10	15	197	292	126	126	802	96.8	10.68	202	158,077,320	179,454,250	21.00
13	Edward P. Allis Co..	1900	Boston, Mass.	30	195	140	61	185	748	93.3	10.33	196	163,925,300	178,497,000	21.63
14	E. D.Leavitt	1901	Cambridge, Mass....	21	496	190	82	178	780	93.4	12.17	221	135,816,000	146,173,000	19.43
15	Holly Mfg. Co	1901	Boston, Mass.	35	300	45	19.5	151	323	88	11.10	210	141,532,000	157,349,000	20.50
16	Holly Mfg. Co	1901	Boston (Spot Pond)	20	248	125	54	150	464	96.5	11.01	203.4	156,592,000	173,620,000	20.85
17	E. D. Leavitt	1903	New Bedford, Mass .	10	480	185	80	181	342	95	13.46	223.9	150,010,000	140,000,000	18.95
18	Edward P. Allis Co..	1903	St Louis, Mo.,No.11	15	197	293	126.9	138	800	97.2	10.84	204.8	155,800,000	177,300,000	20.72
19	Edward P. Allis Co..	1903	St.Louis, Mo.,No.12	15	197	293	126.9	138	796	97.7	10.88	205.2	156,900,000	177,200,000	20.67
20	Allis-Chalmers Co. .	1904	St. Louis, Mo. No 1..	20	198	240	104	...	...	...	....	....	....	176,866,000	... ..

[1] Exclusive of auxiliaries.

XIV

GENERAL DIRECTIONS FOR SETTING UP AND OPERATING PUMPS

IN setting up a pump the first requisite is to provide for a full and steady supply of water or other fluid. To accomplish this observe carefully the following points.

The *suction pipe* sizes are given by various pump manufacturers in tables or upon application, and in no case should the size of the pipe be reduced to less size than the manufacturers give. In some cases where the suction pipe is long it must be larger than the size given to overcome friction. Make the pipe connections as short as possible with the fewest number of bends possible and these as easy (long radius) as possible.

In laying *suction pipe,* a *uniform grade* should be maintained, thereby avoiding air pockets or summits. Grade the suction pipe toward the supply, with a drop of not less than 6 *inches in each* 100 *feet.* It will be found economical to have grade given by a civil engineer.

The *suction pipe* and its connections *must be tight,* as a very small leak will supply the pump with air to its full capacity so that little or no water will be obtained, according to the size of the leak. Before covering the suction pipe it is recommended that it be tested with a pressure of not less than 25 or more than 50 pounds per square inch, to discover any leaks.

Wrought-iron pipe may be used for suction pipe of small sizes, but cast-iron flanged pipe is recommended for all sizes in which it can be obtained. When bell and spigot pipe is used it should be laid with the direction of the current from the bell end toward the spigot end.

All valves in suction and discharge pipe should be *gate valves.*

A *suction air chamber* is an advantage on long or high suctions, and is particularly recommended for single pumps, on all fire pumps, and any pumps which are to run at high speed, expecially for pumps of short stroke.

A *foot-valve*, under these conditions, insures a quick starting of the pump by maintaining the pipe full of water and free from air. When a foot-valve is used see that the area of its valve-seat openings is not less than the area of the pipe.

A *strainer* is always desirable but not necessary when water is clear and free from foreign matter that will clog the valves and passages of the pump. The area of the strainer openings should be at least four times the area of the pipe, to equalize the friction of water through the small openings, and because some of them are liable to become clogged. When strainers are used they must be frequently inspected and cleaned.

Extreme caution must be exercised while pipe is being laid and pump connected, to prevent foreign matter, such as sticks, waste, and rubbish from entering the pipe. Chips from threading pipe, sand, etc., will quickly cut the cylinder, piston, and valve of a pump, doing more damage than years of proper use, or perhaps, entirely disabling it.

A *priming pipe* connected to a supply above the pump or under pressure is a convenience for quick starting, and a necessity for a fire pump, and most large pumps of all classes.

Hot water cannot be raised to any considerable hight by suction. Thick liquids and hot water should always flow to the pump by gravitation.

Steam and exhaust pipes should be as straight as possible and of the full size called for by manufacturer's tables.
ample

In connecting the steam-pipe, proper allowance should be made for expansion. A gate throttle valve should be placed in the steam-pipe close to the pump. Means should be provided for draining this pipe before starting the pump.

A *heater* may be placed in the exhaust pipe to advantage.

To prevent freezing, drain the pump by opening all cocks and plugs provided for the purpose. In piping from these drips, valves should be placed close to the pump cylinders. The steam and water cylinder drips should never be connected into the same pipe unless a check valve is placed so as to close towards the water cylinder to keep it free from steam.

Erecting of a pump should be done by a thoroughly competent man.

Foundations suitable for the pump should always be provided.

All pipes should be properly supported so as to relieve the pump flanges from undue strains.

Keep the *steam cylinder* well oiled, especially just before stopping.

Keep the *stuffing-boxes* well and evenly filled with a good quality of packing. Don't screw them too tight.

Let the steam end alone if the pump begins to run badly, until fully satisfied that there is no obstruction in the water cylinder, water valves or pipes.

The pump should be located, if possible, in a light, dry, warm and clean place and have good care. Do not overlook the importance of this last suggestion.

Do not pull the pump apart to see what is inside as long as it does its work well.

XV

USEFUL INFORMATION

WATER

ONE cubic inch weighs .0361 pounds.

One pound = 27.7 cubic inches.

One cubic foot = 62.4245 pounds at 39 degrees Fahrenheit; 7.48 gallons U. S.; 6.2321 gallons imperial.

One gallon U. S. = 8.33111 pounds; 231 cubic inches .13368 cubic feet.

One imperial gallon = 10 pounds at 62 degrees Fahrenheit; 277.274 cubic inches; .16046 cubic feet.

One pound pressure = 2.31 feet in hight.

One foot in hight — .433 pounds pressure.

Petroleum weighs 6½ pounds per U. S. gallon, 42 gallons to the barrel.

To convert imperial gallons into U. S. gallons, multiply by the factor 1.2. To convert U. S. gallons into imperial gallons, multiply by the factor .8333.

A miner's inch is a measure for flow of water, and is the quantity of water that will flow in one minute through an opening 1 inch square in a plank 2 inches thick under a head of 62 inches to the center of the orifice. This is equivalent, approximately, to 1.53 cubic feet, or 11½ gallons per minute.

To find the diameter of pump plungers to pump a given quantity of water at 100 feet piston speed per minute, divide the number of gallons by 4, then extract the square root, and the result will be the diameter in inches of the plungers.

To find the number of gallons delivered per minute by a single double-acting pump at 100 feet piston speed per minute, square the diameters of the plungers, then multiply by 4.

To find the horse-power necessary to elevate water to a given hight, multiply the weight of the water elevated per minute by the hight in feet and divide the product by 33,000 (an allowance should be made for water friction and a further allowance for losses in the steam cylinder, say from 20 to 30 per cent.).

The mean pressure of the atmosphere is usually estimated at 14.7 pounds per square inch, so that with a perfect vacuum it will sustain a column of mercury 29.9 inches, or a column of water 33.9 feet high at sea level.

To determine the proportion between the steam and pump cylinder, multiply the given area of the pump cylinder by the resistance on the pump in pounds per square inch, and divide the product by the available pressure of steam in pounds per square inch. The product equals the area of the steam cylinder. To this must be added an extra area to overcome the friction, which is usually taken at 25 per cent.

The resistance of friction in the flow of water through pipes of uniform diameter is independent of the pressure and increases *directly* as the length and the square of the velocity of the flow, and *inversely* as the diameter of the pipe. With wooden pipes the friction is 1.75 times greater than in metallic. Doubling the diameter increases the capacity four times.

To determine the velocity in feet per minute necessary to discharge a given volume of water in a given time, multiply the number of cubic feet of water by 144 and divide the product by the area of the pipe in inches.

To determine the area of a required pipe, the volume and velocity of water being given, multiply the number of cubic feet of water by 144 and divide the product by the velocity in feet per minute.

XVI

USEFUL TABLES

Hights in Feet to Which Pumps will Elevate Water

Steam Pressure, 50 Pounds per Square Inch at the Pump
No Allowance made for Friction in Pipes, etc.

Diameter of Steam Cylinders	Diameter of Water Cylinders																
	2 Inch	2½ Inch	3 Inch	3½ Inch	4 Inch	5 Inch	6 Inch	7 Inch	8 Inch	9 Inch	10 Inch	10½ Inch	12 Inch	14 Inch	16 Inch	18 Inch	20 Inch
3½	230	147	102	75	58	37											
4	300	192	134	134	75	48	34										
5	469	300	209	153	117	75	52	38									
6	675	432	300	221	169	108	75	55	42	33							
7	920	588	408	300	230	147	102	75	57	45	37						
8	...	768	533	344	300	192	141	98	75	59	48	44					
9	...	972	675	496	380	243	169	124	95	75	61	55	42				
10	...	...	833	612	469	300	208	153	117	94	75	68	50	38			
12	...	...	...	881	675	432	300	220	169	133	108	97	75	55	42		
14	...	...	...	...	920	588	408	300	228	182	147	133	102	75	57	45	
16	...	...	...	...	...	768	564	392	300	236	192	174	141	98	75	59	48
18	...	...	...	...	...	972	650	490	379	300	243	220	162	122	95	75	61
20	...	...	...	...	...	...	833	600	469	370	300	272	208	150	117	92	75
22	...	...	...	...	...	...	1008	741	567	448	364	329	252	185	142	112	91
24	...	...	...	...	...	...	...	882	675	533	432	392	300	220	169	133	108
26	...	...	...	...	...	...	....	1034	788	626	508	460	356	258	197	156	127
28	...	...	...	...	...	...	....	....	919	726	588	533	407	300	230	181	147
30	...	...	...	...	...	...	....	....	1054	834	676	612	468	345	263	208	169
32	...	...	...	...	...	...	....	....	....	948	798	697	533	391	300	237	192
34	...	...	...	...	...	...	....	....	....	1070	868	786	603	442	339	268	217
36	...	...	...	...	...	...	....	....	....	....	972	881	675	495	380	300	243

The maximum limit of piston speed depends upon the head pumped against.

Friction Loss in Pounds Pressure per Square Inch

For each 100 Feet of Length in Different Size Clean Iron Pipes Discharging given Quantities of Water per Minute

Gallons discharged per Minute	½ Inch	¾ Inch	1 Inch	1¼ Inch	1½ Inch	2 Inch	2½ Inch	3 Inch	3½ Inch	4 Inch	5 Inch	6 Inch
	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds	Friction Loss in Pounds
5	24.6	3.3	.84	,31	.12	....	....	....	....	....	....	....
10	96.0	13.0	3.16	1.05	.47	.12	....	....	....	....	....	....
15	....	28.7	6.98	2.38	.97	....	....	....	....	....	....	....
20	....	50.4	12.3	4.07	1.66	.42	....	....	....	....	....	....
25	....	78.0	19.0	6.40	2.62	....	.21	.10	....	....	....	....
30	....	....	27.5	9.15	3.75	.91	....	....	....	....	....	....
35	....	....	37.0	12.04	5.05	....	....	....	....	....	....	....
40	....	....	48.0	16.1	6.52	1.60	....	....	....	....	....	....
45	....	....	....	20.2	8.15	....	....	....	....	....	....	....
50	....	....	....	24.9	10.0	2.44	.81	.35	.16	.09	.03	....
75	....	....	....	56.1	22.4	5.32	1.80	.74	.34	....	....	....
100	....	....	....	....	39.0	9.46	3.20	1.31	.60	.33	.12	.05
125	....	....	....	....	....	14.9	4.89	1.99	.90	....	....	....
150	....	....	....	....	....	21.2	7.0	2.85	1.32	.69	.25	.10
175	....	....	....	....	....	28.1	9.46	3.85	1.78	....	....	....
200	....	....	....	....	....	37.5	12.48	5.02	2.32	1.22	.42	.17
250	....	....	....	....	....	....	19.66	7.76	3.55	1.89	.65	.26
300	....	....	....	....	....	....	28.06	11.2	5.23	2.66	.93	.37
350	....	....	....	....	....	....	....	15.2	7.0	3.65	1.28	.50
400	....	....	....	....	....	....	....	19.5	9.0	4.73	1.68	.65
450	....	....	....	....	....	....	....	25.0	11.60	6.03	2.10	.81
500	....	....	....	....	....	....	....	30.8	14.26	7.41	2.70	.96

Weight and Capacity of Different Standard Gallons of Water

	Cubic Inches in a Gallon	Weight of a Gallon in pounds	Gallons in a Cubic Foot	
Imperial or English ...	277.274	10.00	6.232102	Weight of a cubic foot of water, English standard, 62.321 pounds Avoirdupois.
United States	231.	8.33111	7.480519	

Weight of Crude Petroleum, 6½ pounds per U. S. gallon,
Weight of Refined Petroleum, 6½ pounds per U. S. gallon, } 42 gallons to the barrel.

A "miner's inch" of water is approximately equal to a supply of 12 U. S. gallons per minute.

Pounds Pressure Lost by Friction

In each 100 Feet of 2½-inch Fire Hose, for given Discharges of Water per Minute

Diameter of Nozzle, Inches	PRESSURE AT HOSE NOZZLE									
	Head in pounds per sq. in.	20	30	40	50	60	70	80	90	100
	Head in feet	46.2	69.3	92.4	115.5	138.6	161.7	184.8	207.9	231.0
1	Gallons discharged...	110	134	155	173	189	205	219	232	245
	Rubber hose, pounds .	4.35	6.40	8.40	10.20	12.80	14.80	17.0	19.20	20.50
	Leather hose, pounds.	6.33	8.53	10.83	13.10	15.34	17.79	20.11	22.40	24.83
1⅛	Gallons discharged ..	139	170	196	219	240	259	277	294	310
	Rubber hose, pounds .	6.79	10.16	13.60	17.05	20.59	24.0	27.0	30.0	33.0
	Leather hose, pounds.	9.05	12.71	16.38	20.11	23.88	27.61	31.41	35.24	39.07
1¼	Gallons discharged ..	171	210	242	271	297	320	342	363	383
	Rubber hose, pounds .	10.28	15.64	20.85	25.46	29.50	39.0	43.81	49.42	55.0
	Leather hose, pounds.	12.84	19.0	24.07	30.11	35.94	41.57	47.36	53.25	59.20
1⅜	Gallons discharged ..	207	253	293	327	358	387	413	439	462
	Rubber hose, pounds .	15.0	22.96	29.40	40.50	48.20	55.70	64.70	72.0	79.26
	Leather hose, pounds.	18.81	26.39	35.01	43.38	52.0	60.40	68.59	76.73	84.87

HORIZONTAL AND VERTICAL DISTANCES REACHED BY JETS

Diameter of Nozzle, Inches	PRESSURE AT NOZZLE									
	Head in pounds, per sq. in.	20	30	40	50	60	70	80	90	100
	Head in feet	46.2	69.3	92.4	115.5	138.6	161.7	184.8	207.9	231.0
1	Gallons discharged	110	134	155	173	189	205	219	232	245
	Horizontal distance of jet	70	90	109	126	142	156	168	178	186
	Vertical distance of jet	43	62	79	94	108	121	131	140	148
$1\frac{1}{8}$	Gallons discharged	131	170	196	219	240	259	277	294	310
	Horizontal distance of jet	71	93	113	132	148	163	175	186	193
	Vertical distance of jet	43	63	81	97	112	125	137	148	157
$1\frac{1}{4}$	Gallons discharged	171	210	242	271	297	320	342	363	383
	Horizontal distance of jet	73	96	118	138	156	172	186	198	207
	Vertical distance of jet	43	63	82	99	115	129	142	154	164
$1\frac{3}{8}$	Gallons discharged	207	253	293	327	358	387	413	439	462
	Horizontal distance of jet	75	100	124	146	166	184	200	213	224
	Vertical distance of jet	44	65	85	102	118	133	146	158	169

Pressure of Water

Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch	Feet Head	Pressure per Square Inch
1	0.43	32	13.86	63	27.29	94	40.72	225	97.46	385	166.78
2	0.86	33	14.29	64	27.72	95	41.15	230	99.63	390	168.94
3	1.30	34	14.72	65	28.15	96	41.58	235	101.79	395	171.11
4	1.73	35	15.16	66	28.58	97	42.01	240	103.96	400	173.27
5	2.16	36	15.59	67	29.02	98	42.45	245	106.13	425	184.10
6	2.59	37	16.02	68	29.45	99	42.88	250	108.29	450	195.00
7	3.03	38	16.45	69	29.88	100	43.31	255	110.46	475	205.77
8	3.46	39	16.89	70	30.32	105	45.48	260	112.62	500	216.58
9	3.89	40	17.32	71	30.75	110	47.64	270	116.96	525	227.42
10	4.33	41	17.75	72	31.18	115	49.81	275	119.12	550	238.25
11	4.76	42	18.19	73	31.62	120	51.98	280	121.29	575	249.09
12	5.20	43	18.62	74	32.05	125	54.15	285	123.45	600	259.90
13	5.63	44	19.05	75	32.48	130	56.31	290	125.62	625	270.73
14	6.06	45	19.49	76	32.92	135	58.48	295	127.78	650	281.56
15	6.49	46	19.92	77	33.35	140	60.64	300	129.95	675	292.40
16	6.93	47	20.35	78	33.78	145	62.81	305	132.12	700	303.22
17	7.36	48	20.79	79	34.21	150	64.97	310	134.28	725	314.05
18	7.79	49	21.22	80	34.65	155	67.14	315	136.46	750	324.88
19	8.22	50	21.65	81	35.08	160	69.31	320	138.62	775	335.72
20	8.66	51	22.09	82	35.52	165	71.47	325	140.79	800	346.54
21	9.09	52	22.52	83	35.95	170	73.64	330	142.95	825	357.37
22	9.53	53	22.95	84	36.39	175	75.80	335	145.12	850	368.20
23	9.96	54	23.39	85	36.82	180	77.97	340	147.28	875	379.03
24	10.39	55	23.82	86	37.25	185	80.14	345	149.45	900	389.86
25	10.82	56	24.26	87	37.68	190	82.30	350	151.61	925	400.70
26	11.26	57	24.69	88	38.12	195	84.47	355	153.78	950	411.54
27	11.69	58	25.12	89	38.35	200	86.63	360	155.94	975	422.35
28	12.12	59	25.55	90	38.93	205	88.80	365	158.10	1000	433.18
29	12.55	60	25.99	91	39.42	210	90.96	370	160.27	1500	649.70
30	12.99	61	26.42	92	39.85	215	93.13	375	162.45	2000	866.30
31	13.42	62	26.85	93	40.28	220	95.30	380	164.61	3000	1299.50

Areas of Circles, Advancing by Eighths

	AREAS							
Diam.	0	1/8	1/4	3/8	1/2	5/8	3/4	7/8
0	.0	.0123	.0491	.1105	.1964	.3068	.4418	.6013
1	.7854	.9940	1.2272	1.4849	1.7671	2.0739	2.4053	2.7612
2	3.14	3.55	3.98	4.43	4.91	5.41	5.94	6.49
3	7.07	7.67	8.30	8.95	9.62	10.32	11.05	11.79
4	12.57	13.36	14.19	15.03	15.90	16.80	17 72	18.67
5	19.64	20.63	21.65	22.69	23.76	24.85	25.97	27.11
6	28.27	29.47	30.68	31.92	33.18	34.47	35.79	37.12
7	38.49	39.87	41.28	42.72	44.18	45.66	47.17	48.71
8	50.27	51.85	53.46	55.09	56.75	58.43	60.13	61.86
9	63.62	65.40	67.20	69.03	70.88	72.76	74.66	76.59
10	78.54	80,52	82.52	84.54	86.59	88.66	90.76	92.89
11	95.03	97.21	99.40	101.62	103.87	106.14	108.43	110.75
12	113.10	115.47	117.86	120.28	122.72	125.19	127.68	130.19
13	132.73	135.30	137.89	140.50	143.14	145.80	148.49	151.20
14	153.94	156.70	159.48	162.30	165.13	167.99	170.87	173.78
15	176.71	179.67	182.65	185.66	188.69	191.75	194.83	197.93
16	201.06	204.22	207.39	210.60	213.82	217.08	220.35	223.65
17	226.98	230.33	233.71	237.10	240.53	243.98	247.45	250.95
18	254.47	258.02	261.59	265.18	268.80	272.45	276.12	279.81
19	283.53	287.27	291.04	294.83	298.65	302.49	306.35	310.24
20	314.16	318.10	322.06	326.05	330.06	334.10	338.16	342.25
21	346.36	350.50	354.66	358.84	363.05	367.28	371.54	375.83
22	380.13	384.46	388.82	393.20	397.61	402.04	406.49	410.97
23	415.48	420.00	424.56	429.13	433.74	438.36	443.01	447.69
24	452.39	457.11	461.86	466.64	471.44	476.26	481.11	485.98
25	490.87	495.79	500.74	505.71	510.71	515.72	520.77	525.84
26	530.93	536.05	541.19	546.35	551.55	556.76	562.00	567.27
27	572.56	577.87	583.21	588.57	593.96	599.37	604.81	610.27
28	615.75	621.26	626.80	632.36	637.94	643.55	649.18	654.84
29	660.52	666.23	671.96	677.71	683.49	689.30	695.13	700.98
30	706.86	712.76	718.69	724.64	730.62	736.62	742.64	748.69
31	754.77	760.87	766.99	773.14	779.31	785.51	791.73	797.98
32	804.25	810.54	816.86	823.21	829.58	835.97	842.39	848.83
33	855.30	861.79	868.31	874.85	881.41	888.00	894.62	901.26
34	907.92	914.61	921.32	928.06	934.82	941.61	948.42	955.25
35	962.11	969.00	975.91	982.84	988.80	996.78	1003.8	1010.8

AREAS OF CIRCLES, ADVANCING BY EIGHTHS

Diam.	AREAS							
	0	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$
36	1017.9	1025.0	1032.1	1039.2	1046.3	1053.5	1060.7	1068.0
37	1075.2	1082.5	1089.8	1097.1	1104.5	1111.8	1119.2	1126.7
38	1134.1	1141.6	1149.1	1156.6	1164.2	1171.7	1179.3	1186.9
39	1194.6	1202.3	1210.0	1217.7	1225.4	1233.2	1241.0	1248.8
40	1256.6	1264.5	1272.4	1280.3	1288.2	1296.2	1304.2	1312.2
41	1320.3	1328.3	1336.4	1344.5	1352.7	1360.8	1369.0	1377.2
42	1385.4	1393.7	1402.0	1410.3	1418.6	1427.0	1435.4	1443.8
43	1452.2	1460.7	1469.1	1477.6	1486.2	1494.7	1503.3	1511.9
44	1520.5	1529.2	1537.9	1546.6	1555.3	1564.0	1572.8	1581.6
45	1590.4	1599.3	1608.2	1617.0	1626.0	1634.9	1643.9	1652.9
46	1661.9	1670.9	1680.0	1689.1	1698.2	1707.4	1716.5	1725.7
47	1734.9	1744.2	1753.5	1762.7	1772.1	1781.4	1790.8	1800.1
48	1809.6	1819.0.	1828.5	1837.9	1847.5	1857.0	1866.5	1876.1
49	1885.7	1895.4	1905.0	1914.7	1924.4	1934.2	1943.9	1953.7
50	1963.5	1973.3	1983.2	1993.1	2003.0	2012.9	2022.8	2032.8

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