THE POWER HANDBOOKS

KNOCKS AND KINKS

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CAUSES, DETECTION, AND CURE FOR MANY OF THE COMMONEST OF THESE TROUBLES OF THE ENGINE-MAN

PLAIN DIRECTIONS FOR PREVENTION AND REMEDY

COMPILED AND WRITTEN
BY
HUBERT E. COLLINS

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INTRODUCTION

THIS volume of The Power Handbooks is devoted to "Knocks and Kinks." The kinks, it is hoped, will serve to eradicate the knocks. For a long time Power has made a feature of the experience of practical men in locating knocks, covering the common and the unusual cases, and it has always devoted some space to little kinks which enable the engineer to master many of the vexatious troubles of his daily work.

These two classes of articles have been collected and arranged in a handbook of convenient size for desk or pocket, and with the other books of this series will, it is hoped, prove of value to the operating engineer.

Every engineer has been taxed to the limit of his capacity, at some time or other, to locate "knocks." Most engineers have had to call in help. Some "knocks" are only nerve-racking but the majority of them indicate operating defects and, as such, demand immediate attention or damage will result. The compiler acknowledges his indebtedness to various engineers who have contributed the articles to Power which form the bulk of this book.

HUBERT E. COLLINS.

NEW YORK, August, 1908.

I

CAUSES OF KNOCKS[1]

WHILE the word "knock" has many and varied applications, its meaning in this book is confined to that species of knocks with which every operator of a steam engine is familiar. In this class there are all degrees of knocks, from the small click to the nerve-racking (and sometimes engine-wrecking) pound.

When a knock develops in an engine it is usually evidence that something needs looking after; and, in general, knocks are, or should be, a feature for elimination. But to locate and remove all knocks in an engine is not always as simple a matter as might be supposed. One reason for this is that all parts of the engine are intimately connected, and a knock produced at one point is transmitted to all parts of the machine. Sometimes, due to the form or material of some particular part of the machine, the sound will appear to originate at this point, when in fact it may be produced at some distance from it. The principle is the same as that involved when a vibrating tuning-fork is held against a table.

Generally speaking, a knock in an engine is the result of lost motion between two parts. It therefore

[1] Contributed to Power by C. J. Larson.

follows that every bearing in the machine may be the occasion for a knock, provided sufficient looseness be afforded and that there is a reversal of forces acting. All bearings that have motion must of necessity have some clearance between the parts, otherwise there would be undue friction and heating. This clearance, when not excessive, is filled with oil which lubricates the parts and acts as a cushion between the metallic surfaces, reducing or removing the tendency to knock.

In nearly all bearings of a reciprocating engine there is a reversal of pressure at every stroke. The natural place to look for a knock, therefore, is in those bearings where pressures are greatest and where one would expect the most wear, viz., the main bearing, crank-pin, crosshead pin and slides. These knocks are usually easily located, while the engine is running, by flooding each bearing in turn with oil. When a bearing has comparatively little lubrication the knock will be sharp and metallic, but when flooded with oil the excess is squeezed out from between the bearing surfaces and cushions the blow, reducing the knock to a slapping sound. Ordinarily, therefore, simply keying up the bearing in question should remove the knock.

In line with the above, an engine which is quite noisy when supplied with oil in drops may often run beautifully when lubricated by means of a pressure or gravity system. The latter method, we are disposed to say, is the sensible way to lubricate an engine of any considerable size, at least. Many high-speed engines are lubricated by splashing the oil in the crank-case, and bearings are often run loose without knocking, for the reason that they are always flooded with oil.

The proper adjustment of the engine valves may have a great deal to do with quiet running. The object is to have the thrust at different bearings reversed as gradually as possible. Unless an engine has good steam distribution it is often useless to attempt to get it to run quietly. It does not necessarily follow that an engine with badly adjusted valves must knock, but the chances for smooth running are certainly much fewer. This is especially true in the higher-speed engines. The first step, therefore, toward removing knocks from an engine, if there is any doubt about the setting of valves, is to apply the indicator.

Causes of Some Knocks

Figure 1 shows a pair of indicator diagrams taken from a small Corliss engine having a very bad knock

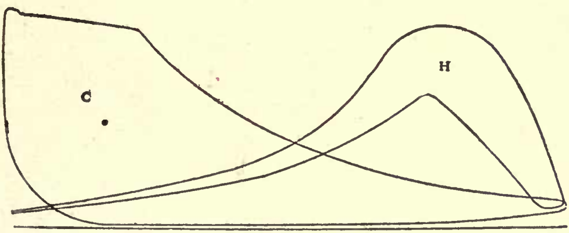
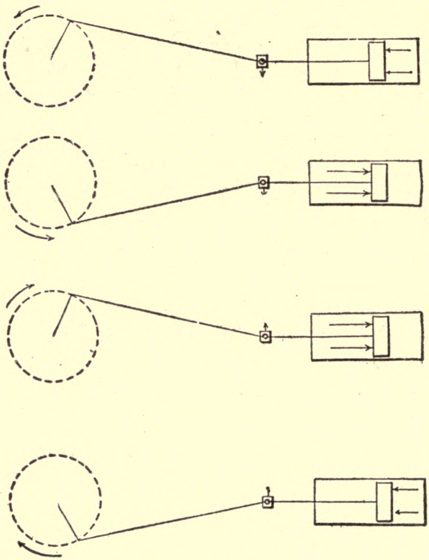


FIG. 1

in the head end. Many attempts at keying failed to improve matters, when the cause of knock was instantly located with the indicator. The engine had always run well until one night the engineer decided to examine the valves. In replacing them he accidentally got the head-end exhaust valve entered on the T head of the valve-stem upside down.

By referring to Figs. 2, 3, 4 and 5, it will be seen



FIGS. 2, 3, 4, 5

that when a horizontal engine runs "over," the thrust from the connecting-rod is downward, except at the instant the engine is passing centers, when the crosshead has nothing but its weight to keep it firmly against the slide. Should the compression be excessive, however, it can be seen that the crosshead might be lifted from the slide toward the end of each stroke. Normally, therefore, there should be no knock between the crosshead and the slide, even though the former has a chance to lift; but since from any variation in speed or pressure this might occur, it is not safe to run with excessive clearance between the top slide and crosshead. But when the engine runs "under," conditions are opposite. The crosshead is then forced against the top slide, except when at centers, and then the crosshead tends to drop onto the bottom slide from its own weight. It is necessary, therefore, to adjust the bottom crosshead shoes quite snugly when the engine runs "under," or there will be a knock at every center. This knock is not, however, as sharp as one produced at any of the other bearings referred to, since the pressure per unit surface of the crosshead shoes is comparatively small.

In vertical engines, direction of rotation and weight of crosshead have, of course, no effect as far as producing knocks between the latter and its slides is concerned. Vertical engines of the direct-connected type usually have the fly-wheel and the revolving member of the generator mounted on the shaft between the engine bearings, and the total weight of the rotating parts generally becomes so heavy that the shaft is not lifted from its bottom bearing by the upward pressure on the piston. It is therefore desirable to allow an unusual amount of clearance between the shaft and the top bearing, as this permits a freer distribution of the lubricating oil.

If, therefore, the pressure in the cylinder should from any cause be increased to a point sufficient to lift the shaft, the result will be a heavy pound when it returns to the bottom bearing. The above occurs in case water accumulates in the bottom end of the cylinder. Should the receiver pressure in a compound engine become excessive the shaft will lift; the same thing may also occur from excessive compression. Suppose the engine is running condensing, with the valves adjusted properly for that condition, then under non-condensing conditions the compression would be much too great and tend to lift the shaft and cause pounding.

If the weight on the bearings is not in excess of the upward force on the piston the bearing caps of vertical engines must, of course, be set up snugly. A good example of the latter case is the marine engine, which has no fly-wheel, and the weight on the bearings is that of the shaft only.

Knocks in the valve-gear can be located by flooding with oil, as described. The knock produced by an eccentric-strap which is too loose usually has a slapping sound unless the speed be high or lost motion great. The reason for this is that the surfaces between the eccentric and strap are large in proportion to the forces transmitted. In taking up an eccentric-strap, care needs to be exercised not to get it too tight, for while the pressure per unit of area is small, the sliding velocity is usually high. This is especially true in direct-connected engines, where the shaft is necessarily of large diameter.

In some single-valve engines the eccentric-rod connects direct to the valve-stem, making only one bearing at this point and consequently only one chance for knock. But very often it is necessary to transfer the motion of the eccentric by means of rocker-arms into line with the valve. This offset in motion is a necessary evil since it multiplies the number of bearings, and, as there must of necessity be some clearance in each, the aggregate may result in a number of knocks occurring simultaneously on the reversal of motion, which sounds like one heavy knock.

Large engines often have the carrier arms and brackets cast hollow, for the purpose of combining lightness with rigidity. This is excellent from an engineering point of view, but a large amount of surface is presented in these parts which greatly intensifies the sound of any existing knock.

Some years ago a large Corliss engine, which, while it ran well and gave good service, had what seemed a fearful pound about the middle of each stroke. Many efforts had been made to locate and remedy the trouble without success, and it was finally concluded that the cylinder was not in line with the slide and that the thump was caused by the piston striking the side of the cylinder. The piston was accordingly removed and a line passed through the engine, but it was found in perfect alinement.

It was a double-eccentric machine, and while operating condensing, the exhaust eccentric would be set somewhat in advance of the steam eccentric. It had been necessary to run non-condensing for some time due to lack of water, and the exhaust eccentric had been set back to get the proper steam distribution for the latter condition. This happened to leave the two eccentrics moving in unison.

There were two bearings for each carrier-arm stud, and one for each eccentric-rod and wrist-plate connection, making eight bearings in all. It had no doubt been observed that these various bearings were not particularly tight, but it had occurred to no one that there might be eight distinct knocks, all coming at the same instant, and all receiving the benefit of the resonance effect of the hollow arms and brackets. After these bearings had been carefully taken up and provision made for a better distribution of oil, the engine started off quietly, and every one concerned said, "Well, did you ever?"

Knocks in Shaft Governors

Nearly all high-speed and many of the so-called moderate-speed engines of the automatic cut-off class are provided with some form of shaft governor. The speed of the engine is controlled by shifting the position of the steam eccentric, either as to angle of advance or as to throw, and sometimes by a combination of both these movements. Knocks sometimes develop in these governors which are not easily located since the parts are revolving within a wheel and consequently difficult to study while in motion. There are several reasons for these knocks. Sometimes the bearings in the governor do not receive proper lubrication. These parts should be provided with some means of certain lubrication. Where an engine runs continuously for long periods the governor bearings soon become dry and bind; the oil which was applied before starting is thrown off by centrifugal force, and it is an uncertain method to try periodically to oil the governor with a squirt-can.

The fact that bearings are dry would often cause a knock of itself; but most shaft governors are very sensitive as to speed, and in consequence of this feature they often tend to move to extremes when any stickiness exists in the bearings of the governor arms. We have seen shaft governors which would allow the weights to rapidly alternate between the positions for no load and maximum load, striking the stops at both ends of their travel and producing knocks. This may occur so rapidly as to cause a surprisingly small variation in the speed of the engine.

Knocks will often be produced in the shaft governor on account of excessive strains upon the various parts. This condition prevails when the valve or valves move too heavily, as from lack of lubrication. When the boiler primes, the oil will be washed off the valve faces, causing undue friction, and if grit and particles of scale are carried over with the water, so much the worse. The result of the latter condition is most pronounced in an engine equipped with a snugly fitting balanced valve. The effect of the sticking valve in the case of a single-valve engine is to force the governor into the position for minimum cut-off, while the governor, responding to decrease in speed, struggles for the opposite. This condition, while tending to make the governor weights strike the stops, will also produce knocking in all the bearings through which the force is transmitted to the valve.

Sometimes a single-valve automatic engine may run quietly, with bearings and pins loose, at one steam pressure, but knock fiercely when the pressure is reduced. This looks like a paradox, but it is not when one analyzes the situation. In this type of engine the cut-off is shortened by decreasing the travel of the valve, while the lead remains practically constant. This is accomplished by moving the eccentric across the shaft in a direction which corresponds to a change of throw and angular advance. But as the cut-off is shortened the exhaust closure also comes earlier, and *vice versa*. As the steam pressure drops, the cut-off becomes later and consequently compression is reduced. Therefore, the real cause of knocking is lack of compression necessary to overcome the inertia of the reciprocating parts.

II

CAUSES OF KNOCKS[1]

A COMMON method of attaching a slide or piston-valve to its stem is by means of two nuts jammed together at either side of the valve. It is not an unusual occurrence to have these nuts work loose and allow clearance for the valve between the nuts and cause a knock at this point. This condition should be quickly detected by the action of the engine, as it means a reduced valve movement and irregular steam distribution. If the engine were running non-condensing, the above derangement would readily be noticeable by the uneven exhausts.

There is a possibility for knocks to exist in almost any form of engine valve. The type of valve which is least liable to produce a knock is the piston-valve, but even in this there may be the click of the packing rings. This may be caused by the rings fitting the grooves too loosely and rattling back and forth as the valve reverses its motion, or it may be due to the rings being slightly collapsed from a sudden increase in pressure in the port surrounding the rings. It will be observed that all the surface of the face of these rings, except that covered by the ribs on the port, is

[1] Contributed by C. J. Larson.

exposed to the direct pressure of the steam during the period that each ring is crossing the port opening.

Knocks Caused by Valves Lifting from Seats

In all forms of steam-valves free to lift from their seats a knock will result whenever the pressure becomes greater in the cylinder than in the steam-chest. This condition will arise from water in sufficient quantities to fill the clearance space, or from excessive compression due to faulty valve setting. The valve will lift from its seat as the piston approaches the end of the stroke, acting as a safety valve; then the instant the piston starts forward the pressure drops and the valve returns to its seat with a slam. It is usual to make valves so that they may lift a limited amount and act as reliefs, in case of emergency from water, but they should never be permitted to lift from steam pressure, ordinarily. When the compression is sufficient to force a regular slide valve from its seat, serious loss of steam results, as when the valve is not in contact with its seat, a direct communication between the steam-chest and the exhaust port is established. In engines with separate steam and exhaust valves, part of the steam filling the clearance space will be forced into the steam-chest.

If the steam-valves are set to open so early that full initial pressure is obtained before the piston has reached the end of the stroke, the result is much the same as if the compression were excessive. The resulting knock, however, is more in the nature of a clatter of the valve, for since the valve is already open it does not have to be forced from its seat, but the current of steam being forced back into the steam-chest until the piston reaches the end of the stroke will make the valve rattle on its seat.

It may be said here that excessive lead is often responsible for knocks in the various bearings, since it applies a heavy pressure suddenly against the piston at a time when it has considerable velocity.

Where independent exhaust valves are used, there may be clattering of these valves under certain conditions. If the engine operates non-condensing and the cut-off is very early, the steam will expand considerably below the atmospheric pressure. The result is that toward the end of the stroke, and before the exhaust valve has opened, the steam in the exhaust pipe will lift the valves, rush into the cylinder and cause a rattling of the valves. This will occur with any type of independent exhaust valve which is free to lift from its seat. With Corliss valves, it should not occur in a new engine, as the valves fit the ports quite snugly, but it may always be noticed in an old Corliss engine where the valves and seats are worn down even a small amount.

Knocks often occur from Corliss exhaust valves having end play. It would at first seem that there should be no occasion for the valves to move endwise, even if they have clearance in that direction. It is not safe to make the valves too snug-fitting endwise, especially in large engines, since the valve, being lighter than the walls of the chamber in which it turns, will expand faster, and if no clearance at the ends is allowed, an accident may result in starting the engine, unless all the parts have acquired an equal temperature.

These valves are round and fit the chamber quite snugly for some distance at each end. One end is slotted for the stem, with the result that there is more waste space for steam than at the back end. Now, when the exhaust valve opens, the steam in this waste space expands and drives the valve against the back

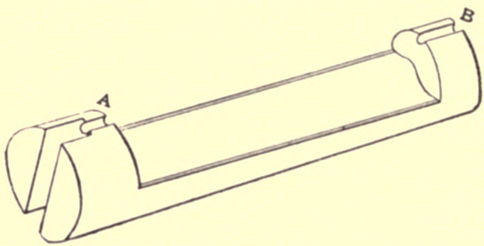


FIG. 6

bonnet, sometimes causing a very severe knock, even when the clearance is small. Then, when the valve has closed and steam is admitted, the small space at the back end will accumulate pressure faster than the larger space at the stem end, and the valve will be driven back. The end knock of an exhaust valve can be distinctly felt by holding any object firmly against the back bonnet.

The quickest way to correct such a knock is to cut passages for the steam in the ends of the valve large enough so that the pressure will always be the same at the ends of the valves as in the cylinder. Fig. 6 shows a Corliss exhaust valve, with channels cut at *A* and *B* for removing this knock.

Valves Chattering on Seats

Almost equally annoying as valves knocking is their chattering on their seats. This is caused by lack of lubrication, or too much spring in the various parts operating the valves, allowing them to move on their seats by jumps, as it were. Sometimes part of this chattering is caused by the valve motion having loose joints, giving the valve a jerky motion. In some cases it will be found that while ample oil is being supplied it does not find its way between the valve and seat. Grooving the face of the valve or seat will often greatly aid the distribution of the oil. In large engines, excellent results have been obtained by grooving the valve-seat and piping the oil into grooves under the valve, when no amount of oil sprayed into the steam would stop the grating of the valves. An inexperienced person might mistake this chattering of the valves for the piston "grunting for grease," but the former can readily be detected by the slight trembling of the valve connections.

Should the piston become loose on the rod, even a very small amount, a heavy knock will result, which usually gets worse rapidly. The most common practice is to have the piston held by a nut on the rod. This may not have been tightened sufficiently, allowing it to back off. It also happens that pistons are loosened by water in the cylinder.

A peculiar case of knock was recently found in the high-pressure piston of a vertical cross-compound engine of 2500 horse-power. The engine would pound fearfully on the bottom center, sometimes every revolution, then at times run quietly for some seconds, and at other times it would pound every other revolution for minutes at a time, reminding one of the exhaust from a "hit-and-miss" gas engine. The nut on the piston-rod had been tried several times, but it seemed

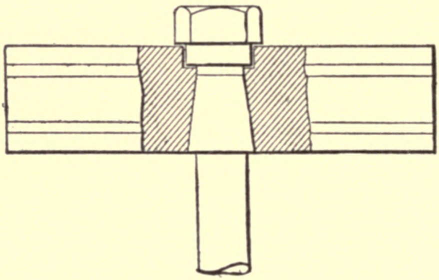


FIG. 7

tight. Fig. 7 shows the detail of the construction. It happened that there was insufficient thread on the piston-rod and that the bottom side of the nut did not actually bear on the piston. Since the nut was counterbored into the piston, one could not see when it came in contact, and its tightness on the thread was mistaken for its bearing on the piston.

It will be seen that the rod is tapered in the piston, and apparently on some down strokes the piston would be driven so firmly onto this taper that it was not removed when steam was admitted on the bottom center. Whenever the piston stuck there was no pound of course.

In built-up pistons, when a follower is secured with studs, there is always the possibility of a stud backing out and striking the cylinder-head. Cases have been known where through error in machine work the junk ring was not clamped firmly between the shoulder on the piston body and the follower. This will cause the junk ring to knock against these parts as the piston reverses its stroke.

If the packing ring travels over the counterbore in the cylinder to any extent, a part of its external surface is exposed to the initial pressure at the instant that the piston is at the end of the stroke. The result is that the ring is slightly collapsed, and as soon as the piston moves forward the ring expands against the cylinder walls with a sharp click. This action of the packing ring in the piston is identical with that described previously in connection with rings in piston valves. There will also be the same knock in pistons as in piston-valves if the rings fit the grooves loosely, except that since the velocities of pistons are so much higher, the knock resulting from a given clearance will be correspondingly greater.

A common method of attaching the piston-rod to the crosshead is by means of a thread and jam-nut. This arrangement affords an easy means for re-adjusting the clearance in the cylinder, but if the jam-nut should loosen, the rod may screw in or out of the crosshead, and allow the piston to knock one of the cylinder-heads. Any striking of the piston or the studs against the cylinder-head can be readily felt, by holding any object, as the end of a lead pencil, against the cylinder-head. Whenever any knock occurs within the cylinder it should be investigated immediately and its exact cause ascertained as, if the piston is striking the cylinder-head, it not only may wreck the engine but cause loss of life.

Engines "Out of Line" Cause Knocks

A great many knocks may exist from an engine being out of line. This term, however, is somewhat ambiguous. When the shaft is not at right angles to the center line of the cylinder the engine is said to be "out of line," but the same expression is used when the cylinder and slide are not in line, or when the path of the connecting-rod is not in line with the cylinder and slide. This latter condition would prevail if the shaft were moved endwise in the bearings.

Figure 8 shows a shaft out of right angles with the cylinder. Since the thrust from the connecting-rod does not bear squarely onto the crank-pin, it follows that on the forward stroke the crank brasses will bear sidewise against the face of the crank, while on the return stroke this side thrust will come on the head of the crank-pin. Therefore, if the brasses have any side clearance, a knock will result from the side slap of the rod. It also follows that if the shaft is not at right angles to the engine the connecting-rod will be out of line with the cylinder and slide. This amount is greatest at the centers, or where the pressures are reversed, consequently the evil that results is greatest.

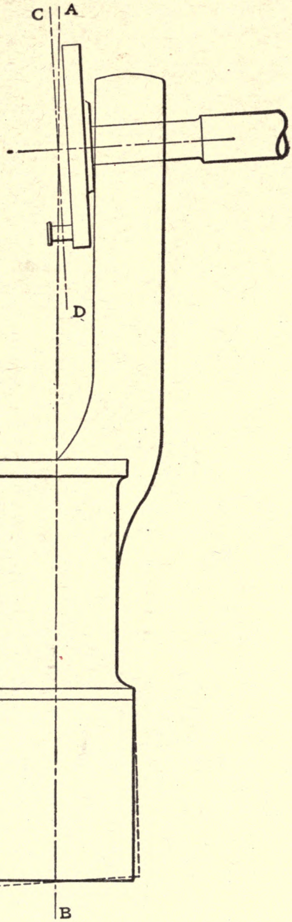


FIG. 8

The effect of the connecting-rod being out of line with the slide is a tendency to produce a knock from the crosshead being forced sidewise in opposite directions at each center, and also to cause side knocking of the connecting-rod in the crosshead. The reason for this will be understood by referring to Fig. 8, where *A B* represents the center line of the cylinder, and *C D* the path of the crank-pin, with the shaft out of line.

When the cylinder is not in line with the slide sidewise there is a tendency to knock at the crosshead on the head end of the stroke, similar to that just described above. Also if the piston is not snug-fitting in the cylinder bore, it will slam against the side of the cylinder and produce a knock. This is especially true in vertical engines, where the piston is unstable and responds to even a small force sidewise. The reason that there will be no knock at the head end in this case is that the cylinder is in line at that end. This is illustrated in Fig. 8, where the cylinder not in alinement is shown in dotted outline.

Another serious result in horizontal engines from the cylinder being out of line is excessive wear on the piston and cylinder. When the engine runs in this condition the piston has a slightly rocking motion, the effect of which is to wear the piston most on its edges and destroy its true cylindrical form, with the result that the piston is deprived of a large amount of surface for its support, and consequently the pressure per unit of surface becomes high and the wear excessive. This trouble may also result from the crosshead not being adjusted in the center of the slides.

III

CAUSES OF KNOCKS[1]

If the cylinder of a vertical engine be so designed that a large portion of the face of the piston is exposed to the steam port while the engine is on the center, and if the piston does not fit the cylinder snugly, a knock may result from the piston being suddenly forced against the opposite side of the cylinder from the impact of the inrushing steam. This trouble is often encountered with high-speed engines, and may sometimes be overcome by giving the engine more compression. The effect of this is to reduce the velocity of the steam in the port at the instant of valve opening. In horizontal engines where the steam ports are on the side of the cylinder this difficulty may also be met with, but it can manifestly not occur where the steam is admitted on the top side, as in a standard type of Corliss cylinder.

A condition which may exist in vertical engines and which is quite analogous with that of a horizontal engine having the shaft out of line, is that resulting from deflection of the shaft from the weight of the fly-wheel and generator. To one who has not had occasion to observe the same, it would appear that a

[1] Contributed to Power by C. J. Larson.

steel shaft two or three feet in diameter could not possibly spring an appreciable amount from any weight which the bearings would carry. But this is not the case. The amount of this shaft deflection is sufficient in all direct-connected units to be carefully allowed for in erecting the machine.

It can readily be seen that if the shaft sags between bearings, the path described by the center of the crank-pin will not be in a truly vertical plane, but will incline toward the fly-wheel side of the engine. Therefore, in order to have an engine of this description in perfect line as regards its shaft, the frames and cylinder should be set sufficiently out of plumb to make their center line coincide with the plane of the crank-pin travel. If this is not done there will be the tendency to knock, as previously described in connection with a horizontal engine having the shaft out of line.

If from any cause the crank-pin is not parallel with the shaft, the crank-pin brasses will knock sidewise. The point in the stroke at which the knock occurs will be determined by the direction in which the pin is inclined. It will be found impossible to key the crank brasses anywhere near snugness without heating, and this condition will also cause a side knock at the crosshead pin, if there is any clearance. The crosshead end of the connecting-rod will be thrown from side to side. The crank-pin can be tested for parallelism with the shaft by disconnecting the rod at the crosshead, slowly revolving the engine, and noting any side movement of the free end of the rod. Crank-pins are sometimes sprung from water in the cylinder.

Cases have been found where, due to mistake in workmanship, the crosshead pin has not been placed perpendicularly to the piston-rod. This condition will naturally result in a tendency to side-knock at the crosshead pin. To ascertain if the crosshead pin is perpendicular to the piston-rod, disconnect the connecting-rod at the crank and key the crosshead brasses tight, noting the position of the free end of the rod. Then disconnect the rod at the crosshead and connect it upside down (speaking of a horizontal engine), and again key tightly. If the free end of the rod retains its first position, the crosshead pin is true in this respect.

Knocks in the crank and crosshead pins of old engines may be the result of the pins being worn out of round, and consequently they cannot be keyed up properly. If this is the case they should be trued up or renewed.

Connecting-rod brasses have a habit of closing up and gripping the pin, on account of becoming hot. This will leave the brasses fitting the end of the rod or strap loosely and is often the source of knocks. Fig. 9 shows brasses which have been distorted from heating. The manner in which brasses tend to close up is as follows: When the surface of the brass next to the pin becomes hot suddenly, the brass tries to open as the result of expansion being greatest on the inside. The rigid strap or rod prevents any movement, and the metal is given a permanent set, then when the brass cools off the ends come together. Ten-inch brasses have been known to close up a quarter of an inch from getting hot suddenly. After the brasses have run hot it is usually necessary to relieve them on the sides where they hug the pin, and this leaves them more or less free to rock in the end of the connecting-rod, causing knocks. The brasses can be filed or machined square and true on the outside and heavy liners inserted to make them fit the rod tightly.

It should be remembered that as one keys up a connecting-rod, the center of the pin is being shifted slightly with respect to the rod. One brass remains

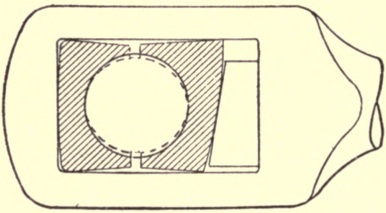


FIG. 9

stationary while the other is moved toward it by the wedge or key. When both wedges are placed between the two pins, keying up tends to make the rod longer between these parts. When one wedge is inside of the pin and another outside, the length would remain constant provided the wear was the same on both pins. But the greater wear always occurs at the crank-pin, due to the greater motion at this point. It will therefore be seen that keying up the connecting-rod will slightly change the clearance in the cylinder, and that ultimately the piston might knock the cylinder-head.

It is not usual to provide any means for taking up the wear of bottom bearings in vertical engines. Therefore as these bearings wear down the clearance in the bottom of the cylinder is reduced. Besides the possibility of the piston striking the head from these causes, it should be observed that changing the clearance volume in the cylinder has much the same effect as changing the lead and compression. Increasing the clearance will correspond to reducing lead and compression, and *vice versa*. It is therefore highly desirable to have the clearance in the cylinder equally divided at all times, since much change in clearance will tend to cause knocks from what will then correspond to improper valve adjustment.

To Determine Location of the Piston

Figure 10 shows a simple and reliable method of determining at any time the exact location of the piston in the cylinder. Make a clear mark all around the piston-rod at some point near the crosshead. When the crosshead is disconnected from the crank, pull the piston forward until it strikes the crank-end cylinder-head. Place a square even with the mark on the rod and transfer to the slide at *A*; then move the piston back until it strikes the back head, measuring the distance it moves. By subtracting the length of the stroke you have the total clearance in the cylinder. Measure this distance back from *A* and mark at *B*, then midway between these put a third mark *C*. When the engine is on the head center, a square placed at the mark on the rod should fall on the middle mark; if it does not, the exact amount which the clearance is out is shown by the discrepancy.

If excessive clearance is left between the cranks and bearings, allowing lateral movement of the shaft, a periodic surging is often set up, causing the cranks or

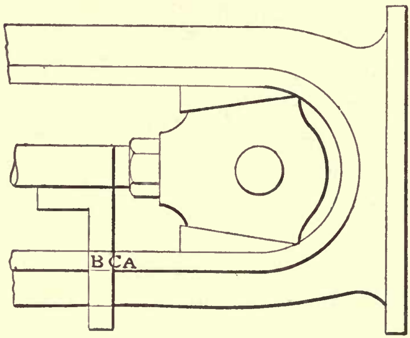


FIG. 10

collars to knock against the cheeks of the bearings. This will be instantly detected by the wobbly appearance of the rim of the fly-wheel.

When the fly-wheel is loose on the shaft, a knock will be produced at the key. This will usually occur just after the engine has passed the center. Many wheels are built in halves and the hub clamped onto the shaft by bolts. In wheels of any considerable size and weight it is very difficult to get these bolts tight enough to prevent slight movement between the wheel and shaft, and the surer way is to shrink the bolts in the hub a suitable amount. In built-up wheels a squeaking sound is often caused by slight movement in some of the joints. This may sometimes be removed by thoroughly tightening up all bolts, but even this may not overcome the trouble. There is absolutely no danger from this creaking noise, but it is very annoying, and it is invariably the subject of comment by all who come near the engine. When all other methods fail, it can usually be stopped by applying a little oil about the joints now and then. An erector's trick is to lightly coat all faces of parts that go together with graphite. This may not improve the job mechanically, but the wheel *won't squeak.*

IV

CYLINDER NOISES

In some engines there will be noticed a slight slapping or clicking noise at each end of the stroke. Usually

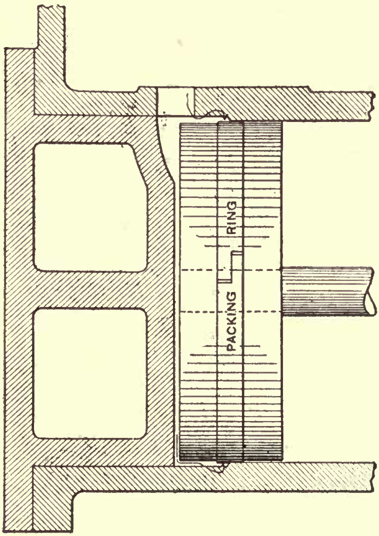


FIG. 11

this noise is not loud enough to indicate serious derangement, but it is very annoying, and in many cases mystifying, baffling attempts to locate it. An erector of analytical mind offers the following reasonable solution:

In such cases it will usually be found that the length of the counterbore, or its depth from the cylinder face, is such that the piston packing ring wipes over the edge of the counterbore a considerable distance, in some cases as much as half the width of the ring.

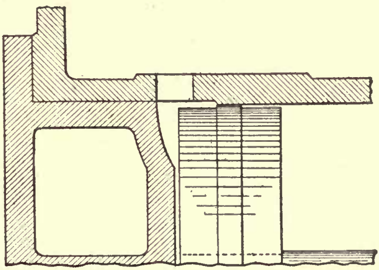


FIG. 12

This serves to prevent the ring from wearing a shoulder in the cylinder, but leads to the condition shown in Fig. 11. Steam is admitted before the end of the stroke, and when the piston reaches the dead center, a large portion of the ring surface projects over the counterbore and is exposed to the steam pressure, as shown by the arrows of Fig. 11. As the ring, whether a snap ring or a spring-adjusted one, is not held outward with any great force, it is instantly collapsed into its groove by this steam pressure, with a snap or click, producing the aforesaid noise. The obvious remedy is to modify the piston to permit the use of a narrower ring which shall wipe just flush with the edge of the counterbore, as in Fig. 12, thus exposing none, or very little, of its surface to the steam pressure. Even should a slight shoulder be worn, it is easily removed by a sharp scraper, with much less annoyance than a constant noise in the cylinder.

Another similar noise is sometimes caused by the imperfect fit of the packing ring in its groove. The ring, as originally made, may be narrower than the groove, or its edges may wear by long use until it is quite loose in the groove. Then at each end of the stroke, from the reversal of motion and from the steam pressure, the ring will jump or slip over to the other edge of its groove with a sharp "click," causing a most distressing noise. The cure is, of course, a new ring made wide enough to fit snugly into the groove.

V

READY DETECTION AND REMEDY

WEBSTER defines a knock as "a blow, a stroke," and a thump as "the sound made by the fall of a heavy body, as by the blow of a hammer." So, the operating engineer often hears the "thump" of some part of his engine, which tells him that some of the working parts are "knocking," and in due time he will find other evidence of the same trouble in heated parts as well. In event of an engine being already in operation, and, possibly, time in which to make inspection being very limited, it is desirable to be familiar with the very first indications of trouble, and then, if necessary, prove the evidence with tests.

Knocking, with the consequent heating of the pins or journals (or the trouble may be in the cylinder), is caused by loose adjustment, a loosened bolt or nut, in which case the remedy is easy of attainment; or some of the parts of the engine are out of alinement, which is a more serious matter. Knocks may occur in almost any of the working parts of the engine. The most prolific source, and the first to consider, is the crank-pin, and the troubles in the cylinder come second. After this the probabilities of trouble from knocking are about evenly divided among the other engine parts, not neglecting the arm of a fly-wheel, as will be shown by the citation of two instances in Chapter VII.

Knocking of the Crank-pin

Trouble with the crank-pin knocking may be caused by a variety of conditions and will be taken up in detail. First we will consider the indications of this trouble as they can be observed with all the connections made up. Assuming that the boxes are properly keyed or adjusted, and everything is in line, the appearance should be as shown in Fig. 13, which is a plan view of the crank-disk and adjacent shaft, with the crank-pin connections shown at each end of the stroke.

Referring to this illustration, first get the relative positions and names of the parts and lines in mind. These latter will be the same in the succeeding similar figures.

The line *A-B* is the center line of the cylinder, guides and crank-pin; *M-N* is the center line of the crank-shaft at right angles to *A-B; D* is a portion of the crank-shaft; *E* is the crank-disk, or in many cases a crank-arm only; *F F* is the stub end of the connecting-rod; *G G* is the crank-pin washer bolted on, or a solid part of the pin, as shown; *J J* are the crank-pin journal boxes; *K K* is the boss on the crank-disk or arm through which the pin projects; *L L* is the crank-pin itself, showing through the spaces at the end of the journal boxes. The crank-pin at the head-end center of its travel is indicated by *H*, and at the crank end by *C*. A few turns of the crank will allow the journal boxes to find their own center of location, and if the shaft is in proper alinement with the center line of the cylinder and the crank-pin is in proper alinement with the shaft, the appearance and position of the parts will be substantially as shown, with an even space all around

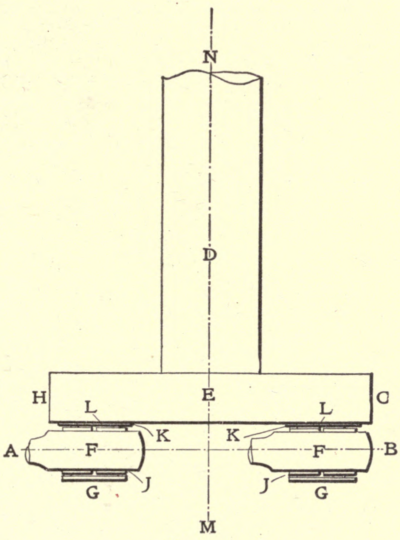


FIG. 13

the circumference of the crank-pin washer and the journal boxes, and the crank-pin boss, and the journal boxes, at all positions of the crank in its travel.

When a knock is located in the crank-pin, one of several conditions may exist. In the case of the crank-shaft being out of square with the center line of the cylinder, the condition may be as shown in Fig. 14.

The evidence of this condition may appear as follows: The line *A-B* being the center line of the engine, and *M-N* the true center line of the crank-shaft, we find the crank-shaft so located that its existing center is along the line *O-P*. This condition will act on the

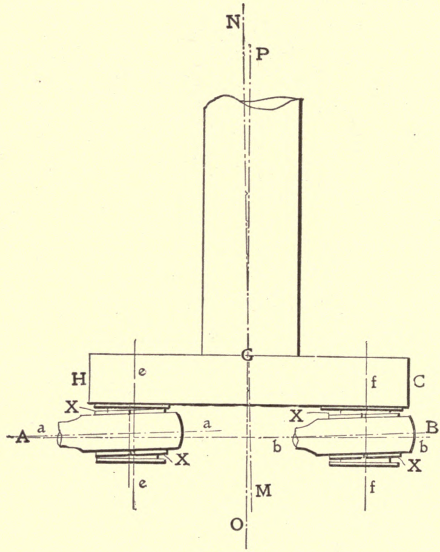


FIG. 14

crank-pin as follows: The shaft being on the center line *O-P*, it will pivot at the point *G* back of the crank-disk and throw the crank-pin out of line.

When the crank is at *H*, the center of the pin will be shifted and the journal boxes not having enough clearance on the sides, the center line of the connecting-rod will be so changed as to be in a direction indicated by the line *a-a*. When the crank is turned to the point *C*, the center line of the connecting-rod is changed to the other side of the line *A-B* in the direction indicated by the line *b-b*. Owing to the fact that in both positions shown in Fig. 14, *e-e* and *f-f* are at a greater angle from the line *A-B* than the lines *a-a* and *b-b*, the openings to be observed between the journal boxes and the washer at one side, and the boss at the other, will be at the points *XXXX* in both positions.

In Fig. 15, the lines *A-B* and *M-N* being the same as previously shown, we find the existing center of the shaft along the line *O-P* opposite its position in Fig. 14. When the crank is at *H*, the center line of the connecting-rod will be along the line *c-c*, and when at *C*, the same line will be *d-d*. The openings to be observed will be at the points *XXXX*.

With the conditions as shown in Figs. 14 and 15, the openings will be noticeable at the points indicated. There will be no noticeable opening when the crank is on the quarter positions. At the two quarters, the lines *a-a, b-b, c-c* and *d-d*, shown in Figs. 14 and 15, will be almost even with and parallel to the line *A-B*, although the pin center lines *e-e* and *f-f* in Fig. 14 will be at the same angle, so that when the positions are as at Fig. 14, and the crank is on either quarter, the openings *XXXX* will be in the same relative positions.

In Fig. 15, when the crank is on the quarters, the openings will be (for the same reason) on the corners shown in this figure. If time allows, a good way to verify the conditions as shown is to disconnect the connecting-rod from the crank-pin and swing it free over the pin to the positions *H* and *C*. When the pin is at *H*, Fig. 14, the crank-pin boxes would swing over the outside end of the pin away from the disk; and

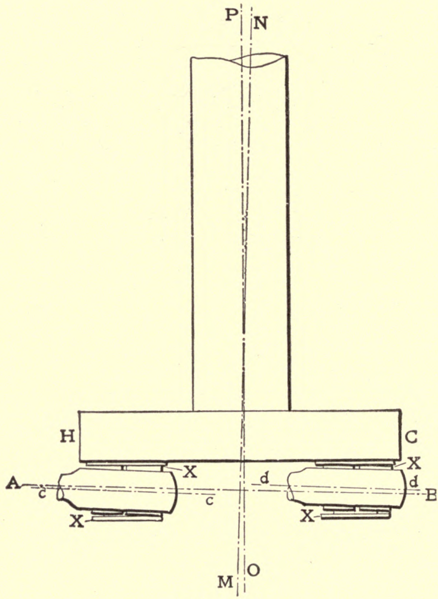


FIG. 15

when at *C*, it would swing in over the disk. In Fig. 15, the conditions would be reversed.

Figures 16 and 17 are elevation sketches of the same crank; *A-B* is the center line of the crank travel at right angles to the line *M-N*, the true center of the shaft, horizontal and level; *T-Q* being the position at the top quarter of the stroke and *B-Q* the position at the bottom. In Fig. 16, the line *O-P* is the existing center of the shaft where the outer end is lower than the crank end. With this condition of the shaft, the greatest openings between the journal box flanges and the adjacent surfaces, will be at the points *XX;* and in Fig. 17, with the line *O-P* showing the outer end of the shaft higher than the crank end, the greatest openings will be at *X X*. If the crank shaft is out of

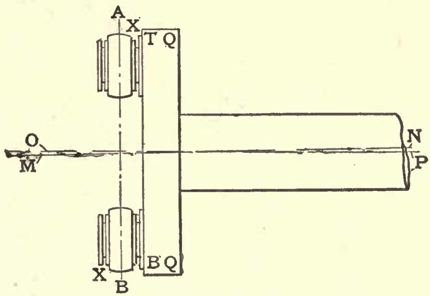


FIG. 16

level as in Figs. 16 and 17, the least amount of unequal space noticeable will be when the crank has reached the two centers.

For example, if in Fig. 14, the clearance spoken of was great enough so that the center line of the connecting-rod was not changed from the line *A-B*, then there would be a clearance all around between the edge of the crank-pin boxes and the crank-pin boss at *H*, but greatest at the point *X* on that side; and with the brass and cap touching at the point opposite, there would be an opening at the other point *X* as shown.

With the crank at the point *C* under the same conditions of clearance, the opening all around would be between the crank-pin cap and journal boxes, being greatest at the point *X;* and touching the crank-pin boss opposite the last point named, there would be an opening at the point *X* on that side. In Fig. 15, with enough clearance of journal brasses, the opening all around with the crank at *H* would be between the washer or cap and the boxes, and when the crank is

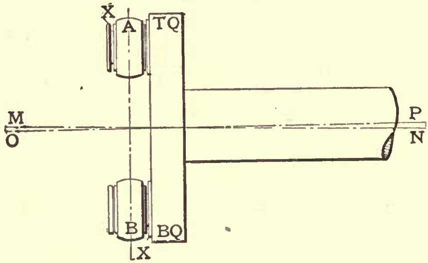


FIG. 17

at *C,* the openings would be between the boss and boxes, or exactly opposite to conditions in Fig. 14. Figs. 14 and 15 show the appearance of the openings when there is not enough clearance between the ends of the boxes, and the center line of the connecting-rod is changed from that of the cylinder line *A-B,* and this would apply as well to the appearance of the openings when the shaft was located as in Figs. 16 and 17.

Remedies for Knocking Crank-pin

When the trouble is such as depicted in Figs. 14 and 15, it is a matter of swinging the outer end of the shaft around so that the line *O-P* will coincide with the line *M-N*, as in Fig. 13. In order to do this, the outboard bearing must be moved in a horizontal line and parallel to the line *A-B*, in a direction towards the cylinder in Fig. 14 and away from it in Fig. 15.

Some engines have an outboard bearing similar to Fig. 18, where the bearing cap comes down on two separate half shells *b b*, adjustment being made by the bolts *c c*. These bolts are secured by lock-nuts. The

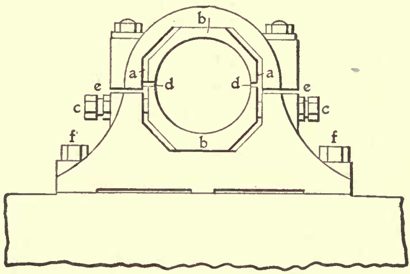


FIG. 18

shells have room in the frame for lateral movement, as shown by the spaces *a a* on each side.

To adjust this end of the shaft, first loosen up on the holding-down bolts of the cap, slacken off the lock-nuts on bolts *c c*, and slacken off the bolt *c* on the side toward the direction you wish to bring the shaft center. Then shove the shell over by setting up on the opposite bolt. After the desired position is secured, set up on the bolt *c* which you first slacked off, until the lower shell is fast between the adjusting bolts, and set up and secure the lock-nuts and the holding-down bolts.

On this style of bearing, the liner adjustment is made between the two shells at the edges and in the spaces *d*, leaving the spaces *e e* clear of liners.

Figure 18 is simply a representation of a type of bearing which has many variations in design, but all give opportunity for lateral adjustment. Some bearing blocks are without this adjustment, as shown in Fig. 19. In this case if the journal sets on a sub-base, it may be possible to chip out the bolt holes *f f* sufficiently to get the required adjustment. If not, a change in the position of the sub-base will be necessary. This may require considerable labor, for it may be necessary to cut away the joint between the sub-base and foundation and make a new joint, with the possible changing of the foundation bolts. Another thing to look out for, when the engine is direct-connected to a generator, is to see that the field or armature frame is set (after any changes in position of the shaft are made), so that the space between the stationary and revolving pieces is equal all around.

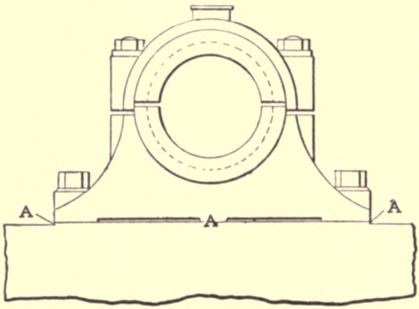


FIG. 19

Main bearings have been seen such as shown in Fig. 20, with quarter-boxes *c* and *d*, and a wedge adjustment *b* in front, but only the quarter-box *d* in the back or even no quarter-box at all, with a complete bearing from point *e* to point *f* on the circumference. On this style of bearing, when excessive wear has taken place and all the adjustment has been on one

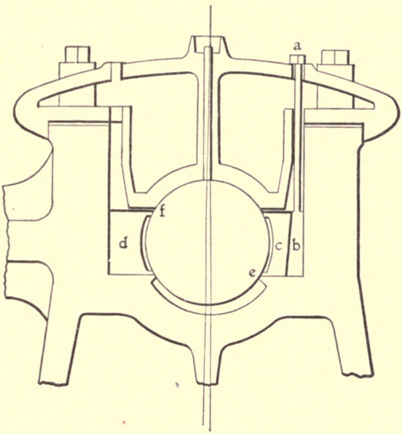


FIG. 20

side, a condition of affairs as shown in Fig. 14 may have been brought about. If there is a quarter-box *d* in the main bearing, a liner behind it will cure the trouble; if there is no quarter-box here, the bearing will need re-babbitting. When trouble is met with such as depicted in Figs. 16 and 17, it is a question of raising or lowering the outboard bearing so that the shaft will be level and the line *O-P* will conform to the line *M-N*.

If we have an outboard bearing on a sub-base as in Fig. 19, the matter of lining the bearing in a case like Fig. 16 is an easy one. Take thin steel wedges, say 2 inches wide, with a taper of not over $\frac{1}{4}$ inch to the foot, and drive between the bearing frame and sub-base (first loosening up on the holding-down bolts), until there is enough space to insert liners at *AAA*. Be careful to insert the same thickness of liner at each place. Before inserting a liner, examine it to see that all burs are removed on the edges and surfaces so that its entire surface will have a bearing. After a sufficient number have been placed to bring the shaft level, set up on the holding-down bolts in the sub-base. On very heavy engines, a hydraulic jack may be needed to lift the shaft in making these changes.

Where there is no opportunity to line up the outboard bearing, either the journal must be re-babbitted or the whole frame must be wedged up from the foundation and new grouting put in. When conditions are as shown in Fig. 17, the trouble is most likely in the main bearing. In this case the line *O-P* would practically be at the proper angle but the point of its divergence from the line *M-N* would be near the outboard bearing. If the bottom part of the main bearing is a shell sepasate from the frame, liners can be placed underneath to raise it. If there is no bottom shell, it will be a case of re-babbitting the bearing.

Crank-pin out of Line with Disk

Referring to Fig. 21 we have four different positions of the crank-pin in one revolution; at *H* the head end, *C* the crank end, *T-Q* or top quarter, and *B-Q*, the bottom quarter. In this case the shaft is at right

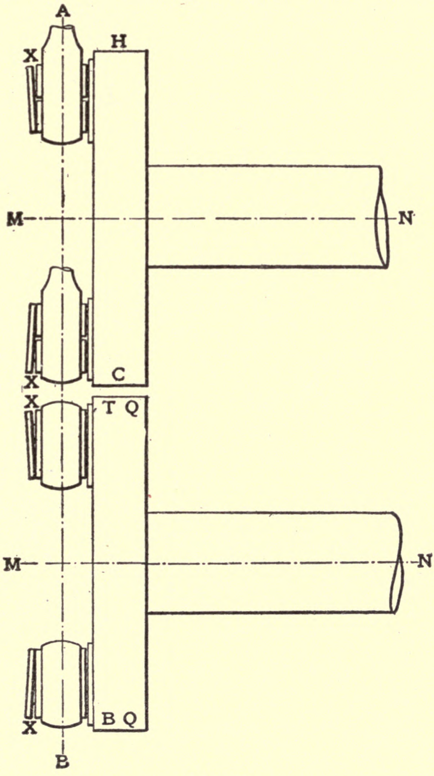


FIG. 21

angles with the line A-B, but the crank-pin is out of square with the shaft center line M-N; and in the illustration it will be noted that at all positions of the crank, the angle at which the crank-pin sets is inclined toward the line M-N. With an engine knocking from this condition of affairs the crank-pin journal can very likely keep its true center, but it will always bear against the outside cap or washer. In turning the engine around by hand the appearance will be as shown, with the exception of the bottom quarter, where the weight of the rod will shift the journal to the inside. If the pin were inclined at an angle opposite to the one shown, away from the line M-N, the positions above would be reversed. The greatest opening to be observed in this case are between the journal boxes and adjacent surfaces would be at $XXXX$.

This condition of crank-pin may be caused by being warped out of place in cooling after being shrunk in. In this event it would most likely be a matter of forcing the crank-arm or disk from the shaft and reboring the crank-pin eye. The work might be done with a boring-bar outfit, properly set. In any event a new pin would be required.

Loose Crank-pin

On most side-crank engines two ways of securing the crank-pin are used. Fig. 22 shows the pin either forced through the eye hydraulically, or shrunk in until it brings up on the shoulder *a* and is riveted over on the end *b*. In Fig. 23, the pin is forced through the eye as in Fig. 22 and is secured in place by a nut *a*.

Sometimes the pin loosens up in the eye and causes a knock. It can be found by close inspection at the

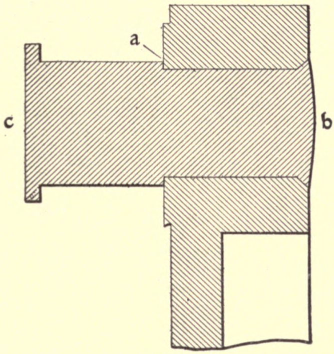


FIG. 22

back of the disk or by stripping the pin of its brasses. If this condition has long existed, the eye is not true and needs reboring before a new pin is inserted.

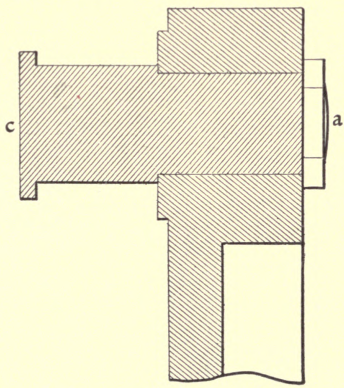


FIG. 23

Flat Crank-pin

More frequent than a loose pin, is a flat one, as shown in Fig. 24 at *a*. The reason the pin wears flat here is that the greatest pressure is just after it has left the

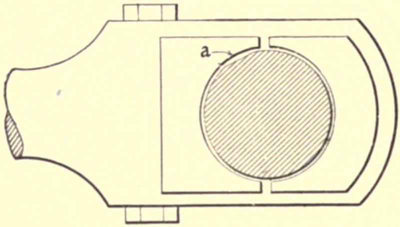


FIG. 24

centers and up to and arriving at the quarters. This does not allow proper adjustment of the boxes and causes knocking. A crank-pin 6 inches in diameter

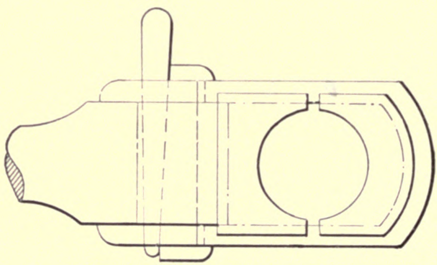


FIG. 25

has been known to wear flat $\frac{3}{8}$ of an inch. It is easily discernible when the pin is stripped and the only solution of the trouble is a new pin, or in an emergency, the filing of the pin round, by a mechanic who knows how to do it.

In Figs. 22 and 23 are shown crank-pins with the outside ends *c c* turned solid with the rest of the pin.

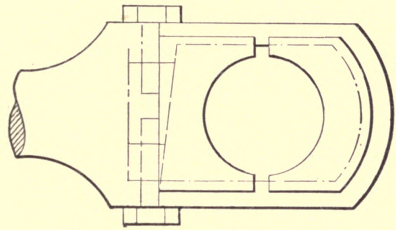


FIG. 26

This style of crank-pin is generally used with the strap, gib-and-key connection shown in Fig. 25. Crank-pins with a detachable cap at *c* are used with connections such as shown in Fig. 26.

Knocking at the Crosshead

The crank-pin being out of position as shown in Figs. 14, 15 and 21 and in a condition opposite to that

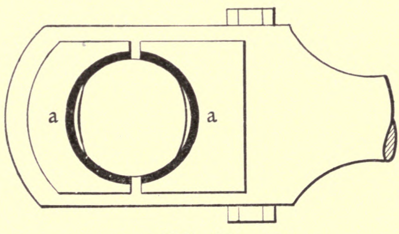


FIG. 27

spoken of in Fig. 21, will have an effect on the crosshead, causing it to knock sidewise, especially if it has flat or round shoes.

Trouble with the crosshead pin being out of line is not very frequent, but it sometimes wears flat on the

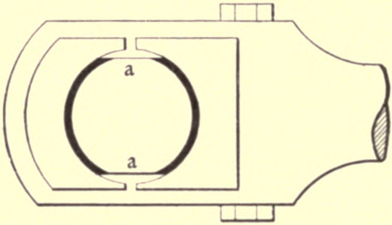


FIG. 28

two sides *a a*, Fig. 27. The pin should be taken out and turned true and the boxes turned to suit the new pin, then it should be flattened on the sides *a a*, as in Fig. 28. This will allow of considerable wear and adjustment without renewal of the pin.

Cylinder out of Line

Figure 29 is a sectional view of an engine cylinder and crosshead guides where the line *A-B* represents the center line; *C* is the crosshead, *D* the piston in the crank end of the cylinder and *E* the piston in the head end, *F* being the piston-rod. Owing to the fact that most engine cylinders are secured to the frame in some style similar to that shown, there is not much danger of the cylinder being out of line at the point where it joins the frame. The frame face and flange around it at *a* are bored and faced at the same setting with the guides. But on account of the human element in engine building and operation, there may be a setting

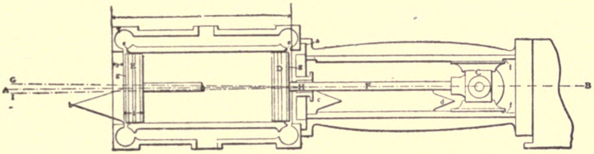


FIG. 29

of the cylinder which will let the head and rest too high along the line *G-H*, or too low in the direction of the line *I-H*, diverging either way from the line *A-B*.

Centering the Piston-rod

Sometimes the piston is low from wear in the cylinder. Again, the piston and cylinder may be all right and the crosshead be either high or low. An inspection of the piston-rod as it works through the stuffing-box gland while running, will tell if anything is wrong here. To find where the trouble lies, first take off the front cylinder-head, turn the engine until the piston is at *E*, and with a pair of calipers *b*, proceed to find out if the piston is central in the cylinder. To do this, place the divider point in the piston center with the caliper leg swinging around the counterbore. The latter should always be worked from, as the cylinder may be worn as well as the piston itself.

After centering the piston in the cylinder, turn the engine over so that the crosshead shoe will clear the head end of the guide, permitting a pair of inside calipers to be used, one leg resting on the wearing surface of the guide and the other point just "feeling" the bottom of the piston-rod at *c*. Now turn the engine until the piston is at *D* and the crosshead as shown at the crank end of the guide. Taking care that the calipers have not been changed, try the distance between the same guide and piston-rod at *d*. Take care to note if any shoulder is worn on the rod and if so try the calipers on the rod inside the shoulder away from the crosshead. The distance between the guide and rod at *d* should be the same as at *c*. Then with the calipers still set as at *c*, and the crosshead and piston remaining on the crank end of the stroke, again try the distance between the guide and rod at *c*. In measuring with a pair of calipers, a light pair should be selected, and the longer the legs the better. In using them, care should be taken to have the two points come into contact exactly opposite each other and that the "feeling" point is not forced into contact, but just "feels" the surface to which you are measuring. This is an art acquired only by practice.

After we have come the second time to point *c* on the rod and find the rod is lower than at point *d*, one of two things is wrong; either the head end of the cylinder is high or the crosshead is high, and if the rod is higher at *c* than at *d*, the head end of the cylinder is low or the crosshead is low.

There are so many designs of engine parts, that a detailed description of every move to make to change these conditions on even a few of the leading ones would be too long for the space at hand. But assuming that the engineer knows how to correct the conditions, it is suggested at this point that he change the hight of the crosshead while it rests at the crank end of the guides, until by frequent trial of the calipers at both points mentioned, *c* and *d*, he finds the rod exactly parallel with the guides. Then turn the engine over again until there is just sufficient room left at the head end of the guide to caliper the distance at *c*. Very careful calipering will now be necessary to detect a difference. If the rod is high now, the head end of the cylinder is high; if the rod is low, the head end of the cylinder is low.

If the piston is worn small on the bottom, as is often the case with horizontal engines, it must be raised. Some pistons are made with follower and bull rings, and in this event there is always some way of raising the center. If the piston is solid it can sometimes be turned around half way, raising the center. But in many instances neither one of these methods is available, and either a new piston must be substituted or the old one turned off and a new ring shrunk on. If the cylinder bore is worn badly it must be rebored and a new piston of the right size fitted.

All pistons should be so set that at each end of the stroke the first piston-ring will pass over into the counterbore as is shown at *e e*, and the crosshead shoes should go over each end of the guides as at *f f*. Sometimes they do not, and wear shoulders against which they bring up and cause knocking. A cylinder with no counterbore at all has been known and a shoulder at each end was inevitable. Another instance coming under observation was the case of a low-pressure piston on a vertical engine of large size being a loose fit, and the steam at each stroke on one end seemed to throw the piston over so forcibly as to result in a recess and shoulder in the cylinder, causing a heavy knock. These defects can be found by inspection.

Determining Clearance

While the front head is off, take the piston out and take the following measurements:

On the inside of the cylinder with a stick or rule long enough, take measurement 1, Fig. 29, which is the distance from the bottom of the cylinder at the back head to the front edge of the cylinder where the flange of the front head rests when in place; next take measurement 2, the thickness of the piston-head, then measurement 3, the amount the front head will extend into the cylinder when in place as indicated by the dotted line. Also take a fourth (4) measurement, the stroke of the engine.

Add together 2, 3 and 4 and subtract the sum from measurement 1; the difference will be the total clearance in the cylinder, one-half of which will equal the mechanical clearance *g g* for each end.

Now add this clearance to measurement 3, and replacing the piston set it up the distance of this last sum from the edge of the cylinder.

Example

Measurement $1 = 45\frac{3}{4}$ inches.
Measurement $2 = 5$ inches.
Measurement $3 = 4$ inches.
Measurement $4 = 36$ inches.
$45\frac{3}{4} - (5+4+36) = \frac{3}{4}$ inch total clearance.
$\frac{3}{4} \div 2 = \frac{3}{8}$ inch clearance in one end.
$\frac{3}{8} + 4 = 4\frac{3}{8}$ inches distance of piston-head from front edge of cylinder.

This is assuming that the crosshead and crank-pin are connected up and on the head-end center. The wear on the connecting-rod end connections also has a bearing on the clearance as the wear sets in. Depending on the amount of wear on the inside brasses, the distance between the crank- and crosshead-pins is gradually reduced.

With the style of rod connections shown in Fig. 24, the wear is all taken up toward the end and the tendency of wear is to lengthen the centers, thus reducing the piston clearance on the head end of the cylinder. When the clearance needs adjusting from this source, place liners under the inside brasses between them and the stub ends of the rod, in the event of a gib-and-key connection. With the wedge adjustment place liners under the outside brasses. The crank-pin brasses generally wear the most rapidly. If the piston-rod screws into the crosshead, secured with a lock-nut, the clearance can be adjusted here without reference to the connecting-rod adjustment unless the latter is brass-bound.

Where we have a horizontal engine of considerable size running over, the lower crosshead shoe is inclined to wear thin and bring the crosshead out of line. If this is not watched a very considerable knock may result. The reason is, that with this lost motion, when the engine passes the center, the crosshead is lifted and almost immediately thrown down again.

Most knocks in the main and outer bearings are caused by loose adjustment or the shaft being out of line as shown in Figs. 14, 15, 16 and 17.

Loose Fly-wheels

These very often give trouble, causing a knock hard to locate. The worst knock often occurs while the engine is running slowly without load and when the wheel turns over so that the key is at or near the point *c*, Fig. 33. (Chap. 7.)

VI

EFFECT OF INERTIA OF MOVING PARTS

USING compression to absorb the inertia of the moving parts is to some extent good practice, but going to the extreme of sacrificing economy to the smooth running of the engine should be avoided. E. F. Williams, a designing engineer, has this to say on the matter:

"The points of admission and release are generally manipulated by lap and lead on both sides of the valve, so as to secure the most power and best economy without particular regard to smoothness of running. As to the running effect of giving the valve more or less lead, that can be better determined by practical test than by theory.

"There are general principles, however, which are good guides to proper running conditions when understood.

"The inertia of the reciprocating parts is a leading factor in the operation of any reciprocating engine. The science of inertia is of more value to the designer of engines than to the operator, as where the engine is once designed and speeded, inertia becomes an inherent fixture entirely beyond the control of the operating engineer. It is of value, however, for the latter to understand the general rules for the distribution of inertia, so as to determine its force at the ends of the stroke where the steam pressure must change direction.

"Generally speaking, it should be borne in mind that while the steam pressures reverse at the ends of the stroke, the inertia effect reverses near mid-stroke, where the connecting-rod becomes tangent to the crank-circle."

It is well known that smoothness of action at the reversing points depends (other things being considered

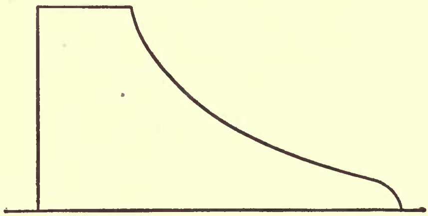


FIG. 30

equal), on how gradually or violently the pressures change. The most violent change would be brought about when conditions are as shown in Fig. 30; there being no compression and inertia being inconsiderable.

About the most favorable condition would be where the inertia about equaled the initial steam pressure, as at *a*, Fig. 31, or even as at *b*, in the same figure. The shaded diagram being the inertia, set over two engine diagrams joined on counter-pressure lines.

Less favorable conditions are established where the speed is sufficient to force quick reversals of pressure on the crank-pin, and not sufficient to generate an inertia effect equal to the initial steam pressure. High-speed engines, therefore, operating at pretty high pressures, require the most skilful management.

Take an ordinary example where the inertia runs up to say, one-half or three-quarters of the initial steam

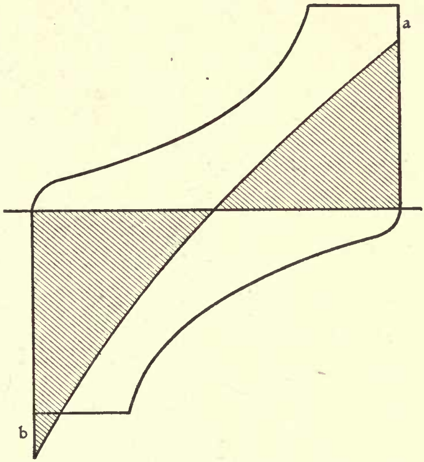


FIG. 31

pressure, as in Fig. 32. When the compression just about reaches and does not surpass the inertia as at *b*, it has very little effect on the reversing blow of the steam which is represented by the sudden rise of pressure *A* and *B* from the inertia line to the initial steam line.

If, on the other hand, the compression exceeds the inertia even by a little, as at *a*, it has a very quieting effect. It is important in this connection for the engineer to know how great the inertia effect is at each end of the stroke. In the first place the actual weights of piston, piston-rod, crosshead and connecting-rod must be known. This information can generally be obtained of the manufacturer; if not, the only way is to take the parts out and weigh them. A close approximation may then be made as to the mean of the

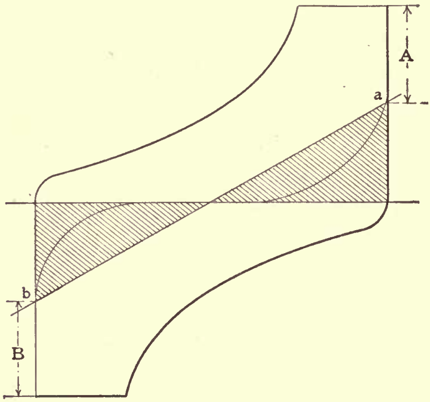


FIG. 32

inertia at the two ends of the stroke, equaling the centrifugal force.

Assume the reciprocating parts of an engine 16 by 24 inches in size to weigh 704 pounds, and the engine speeded to 200 revolutions per minute. The area of the piston = 201 inches; $704 \div 201 = 3\frac{1}{2}$ pounds, the weight of reciprocating parts per square inch of piston area.

The centrifugal force of a body weighing 704 pounds revolving in a circle 24 inches in diameter at the rate of 200 revolutions per minute is

$$\frac{200^2 \times 704 \times 2}{5870} = 9600$$

pounds, nearly, or

$$\frac{200^2 \times 2}{5870} = 13.58$$

times the weight, nearly.

As the weight is in this case $3\frac{1}{2}$ pounds per square inch of piston area, the mean inertia effect for the two ends of the stroke would be $3.5 \times 13.58 = 47.03$ pounds per square inch on the piston. Now, if the pressure by compression is carried up to, or a little above, this at the end of the stroke, the quieting effect will be good on the moving parts; and if this can be done without distortion of the steam diagram at the cost of economy, the result will be most desirable.

The inertia is greater at the back end than at the front, owing to the angularity of the connecting-rod.

This latter is greater or less, depending on the proportionate lengths of rod and stroke.

The effect of lead in a general sense is that when the admission takes place (when the clearance is small) at or near the end of the stroke, the effect is more sudden than if admitted when the piston is some distance back from the end of the stroke, and the effect is more gradual when the piston is receding than when it is advancing toward the cylinder-head. It often occurs from this cause that lead occasions thumping, while an absence of lead is more favorable to smooth reversals.

The points given here on this phase are only a few of those to be considered for a full understanding of the subject, but if understood are a guide for further changes if necessary.

VII

SOME CURIOUS KNOCKS

Early in Chapter V a case was referred to of trouble occurring in a fly-wheel spoke. Some years ago while indicating the engines in a street-railway power house, in the vicinity of New York City, and while slowing down one of the engines a hard pound was heard which was so loud that there was some debate as to the advisability of starting up again. But as everything in sight was as it should be, the engine was started; at the first revolution the pound began and continued once every revolution, increasing in volume as the engine was speeded up until a good speed was reached, when the noise disappeared. It caused much worry until it was finally located in one spoke of the wheel. The wheel was 22 feet in diameter and built up with hollow spokes fastened with bolts to the hub, and at the other end to a segment of the wheel rim somewhat after the style shown in Fig. 33. On taking off the segment a piece of 3-inch extra-heavy pipe *b*, 3 or 4 feet long was found inside the spoke. As the wheel turned slowly the pipe would run down the arm at each revolution and ram the rim; for some reason it came back to the hub easily, until sufficient speed had been attained for centrifugal force to hold it permanently out against the rim.

Some one had evidently been ramming sand out of the spoke, and jammed this piece of pipe in there and

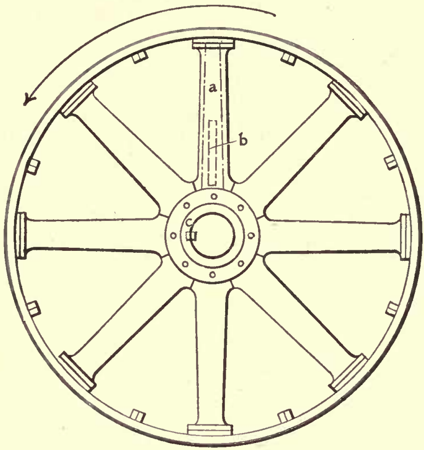


FIG. 33

left it; or possibly the pipe was inserted for a core in the foundry. The engine had run some ten years without loosening the pipe.

An amusing case of offending fly-wheel occurred in a large power plant several years ago. When a new engine was started up there was a strange rumbling sound in the vicinity of the wheel, which was sufficient to make any one apprehensive. A thorough examination revealed nothing, and the engineer representing the engine builder laid the trouble to the generator. After several attempts at starting, it was discovered that when the engine got above a certain speed the noise ceased. It was a segmental wheel, linked together at the rim. In order to make a "stocky" looking wheel, the builders cored out the rim, making a hole about five inches square all around the rim. After the engine had run for some weeks, making the noise when starting and stopping, and had caused many days and nights of brain racking, a conscience-stricken laborer confessed that he had accidentally

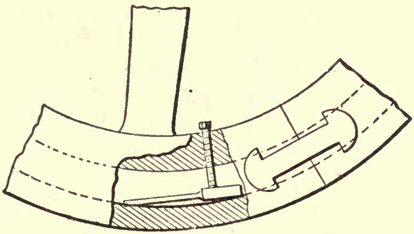


FIG. 34

dropped a large wrench into the cavity in the rim before the last segment was put in place. To remove a segment would have been expensive, so the method of stopping the noise shown in Fig. 34 was used. A hole was drilled and tapped from the inner side of the rim into the cavity and a long stud provided. By turning the wheel the wrench was got directly under the hole and the stud was screwed down, clamping the wrench firmly. This wrench has now traveled farther than any other member of its tribe.

VIII

RIGGING UP TO TURN AND REFIT LARGE PISTONS — A CRANK-PIN TURNING DEVICE[1]

This not only is a large job done in a satisfactory manner in a shop having small tools only, but also it shows how an efficient special machine was built from material rescued from our old friend and standby, the scrap heap.

The plant where the work was done is a large power station with eight compound engines having cylinders 42 and 86 inches by 60 inches stroke; steam pressure, 180 pounds; revolutions per minute, 75. These engines were built to develop 4500 horse-power, but are run anywhere up to 6000 horse-power. Six engines are in commission at a time while a standby engine is kept just turning over so that at a moment's notice, in case of accident, it can be started at full speed. The remaining engine is undergoing any necessary overhauling.

The pistons of the low-pressure cylinders are 86 inches diameter by 15 inches deep. Two inches from each face there is a packing-ring groove *B*, Fig. 36, $1\frac{1}{4}$ inches wide. The pistons cost when new $664, and

[1] Contributed to Power by Dixie.

when put in are about 0.032 smaller in diameter than the cylinders. After a year and a half they are found to have worn approximately $\frac{3}{8}$ inch, and if they could not be repaired would be a total loss.

It was decided to try cutting grooves in one of the pistons and lining it with babbitt. This was done and gave such highly satisfactory results that as they become worn and can be spared the other pistons receive the same treatment.

The largest lathe in the repair shop is a 24-inch Putnam with a bed about 10 feet long. It was, of course, out of the question to turn up the 86-inch pistons on this lathe, but it came in for the work on the piston-rods later on. Out in the scrap heap was a bent piston-rod — not very much bent, but just enough to put it out of business for the work for which it was intended.

This was brought in and a couple of journals turned on it about six feet apart and true with the taper seat for the piston. Two bearings were fitted to these journals. Masonry piers were built 18 x 24 inches and high enough to suit the job. To these 10 x 5-inch I-beams were secured by bolts built into the masonry. On top of the 10 x 5-inch I-beams, 12 x 5-inch I-beams were bolted, and to these the two bearings were secured, all of which is shown in the half-tone, Fig. 35. The slide at the left, upon which the tool slide is mounted, is equipped with a carriage having screw feed, which serves for longitudinal feed for the tool. The tool slide is taken from the 24-inch lathe previously referred to. The pistons are mounted on the end of the piston-rod which forms the spindle of the machine, and six dovetailed grooves 1¼ inches wide by ¼ inch deep are turned in them, as shown at *A*, Fig. 36.

This job took eight hours, which is pretty quick time. After grooving, a wooden form lined with asbestos board and taking in about ⅙ of the circumference is clamped to the piston and the grooves are

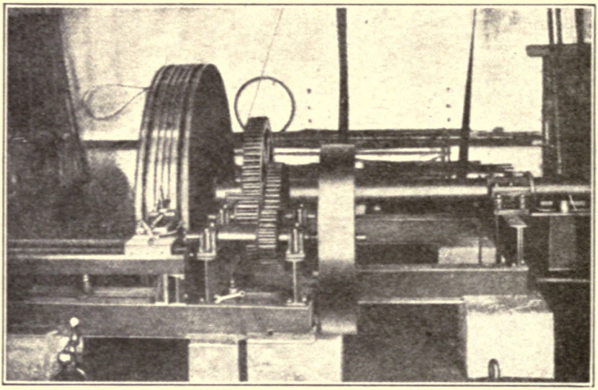


FIG. 35

poured with babbitt. The six segments of babbitt are not allowed to abut, there being a space of about ½ to ¾ inch between their ends. After all the segments are filled with babbitt, the babbitt is pened into the groove, the pening at the same time that it compresses the metal spreads it so that the segments creep toward each other, making a perfectly tight joint at their ends. After the segments are all pened solid they are turned to size, the whole job taking but three days.

The first cylinder fitted with one of these doctored pistons has now a fine glazed surface on it, and is the only one that has.

The crosshead shoes of these engines are also babbitt

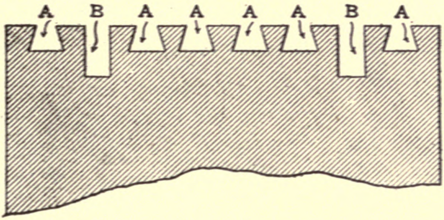


FIG. 36

lined and are turned to size in the same fixture, but between the bearings. The rig shown in Fig. 37 shows

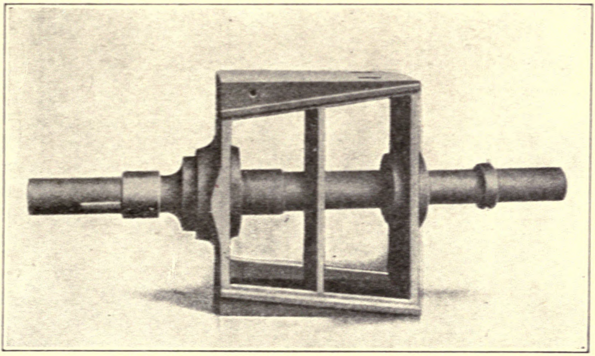


FIG. 37

a spare crosshead secured to a shaft with journals on it which fit the bearings on the turning rig. The large gear on the spindle of the turning rig is split so that it·

can be easily shifted from the one job to the other. The tool slide is also portable, of course.

A Crank-pin Turning Rig

Figure 38 shows a crank-pin turning device. In this rig *A* is a rotating sleeve journaled in the casting *B*,

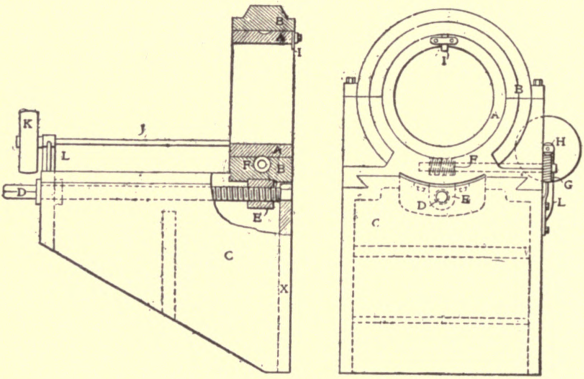


FIG. 38

which slides on the angle casting *C*, being actuated by the screw *D*, working in the nut *E*. A worm *F* meshes with worm teeth cut in the center of the periphery of *A*. The worm *F* is driven through the worm gear *G*, the worm *H*, and the shaft *J* by the pulley *K*. The sleeve *A* carries the turning tool *I*. The sleeve *A* is not shown in halves in the illustration, but can be so made. It is, of course, understood that no attempt has been made to make the drawing to scale.

The face *X* of the angle casting *C* is set true with the crank web, and the sleeve *A* concentric with the crank-pin. The tool *I* is set to depth of cut and the machine is started. The feed is by means of a star wheel on the end of the screw *D*.

Turning Long Piston-rods on a Short Lathe

The spare piston-rods for these engines are about two feet longer than the longest lathe in the shop. The threads on either end for the nuts and the taper for

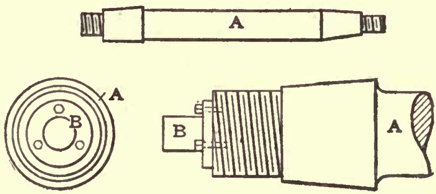


FIG. 39

the pistons have been left a "little full." When it came to fitting a new rod, these oversizes were noticed, and it was necessary to devise some means for doing the job of turning them to size. This is shown at Fig. 39. *A* is the piston-rod, *B* is a small cast-iron piece drilled for three cap screws. The ends of the rods are drilled and tapped so that the piece *B* can be secured to it. The piece *B* is caught in the chuck and the rod is supported in a steady rest having wooden jaws. Even if the piece *B* is not central with the rod, the chuck jaws can be easily set so that the rod runs true. This made it possible to reduce the threads and the tapers, instead of re-threading the nuts, which would, of course, prevent their being interchangeable, as the threaded portions on the spare rods were not all exactly alike.

A Piston-nut Wrench

Another useful tool is shown at Fig. 40. It is used for tightening the piston nuts when in place in the

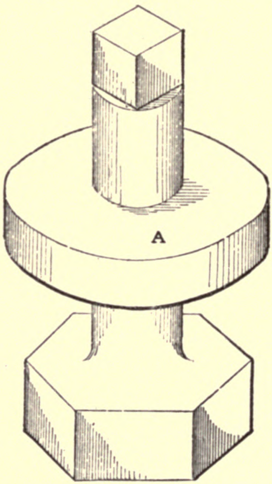


FIG. 40

engine. The flange *A* fits, with a little play, the manhole in the cylinder cover. The piston is brought to top center, the wrench is put in place, and the flange *A* steadies it so that there is no danger of its being displaced while the wrench bar is turning it.

IX

REPAIRING A BADLY BROKEN CYLINDER

A 30 x 72 Corliss engine which had been in service twenty-six years at Atlantic Mills, Providence, parted

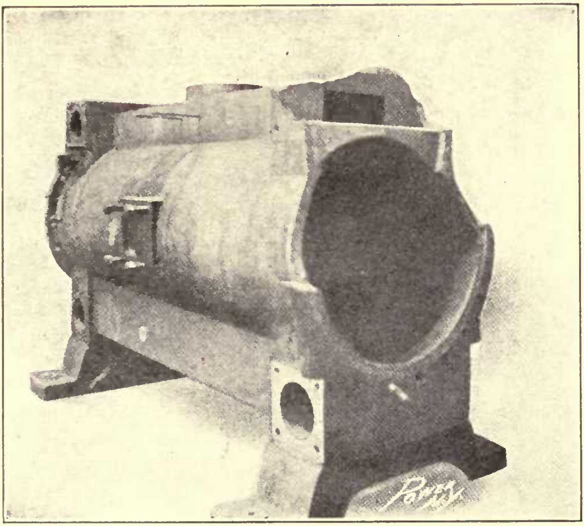
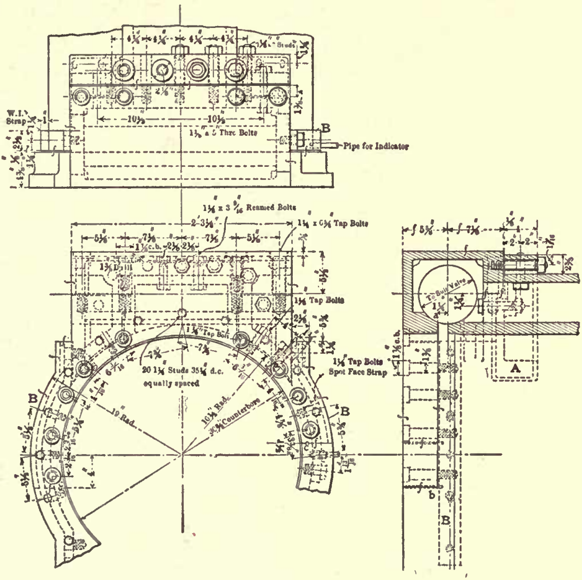


FIG. 41

and let the piston knock the back end of the cylinder into the condition shown by the photograph reproduced herewith (Fig. 41). To make a new cylinder would require a month and this delay in the department run by the wrecked engine would seriously embarrass the whole plant. It was concluded that the broken cylinder could be repaired, and that the engine could be put in running order in about two weeks. It was sent to the shop and fourteen days following the accident it was put to work carrying the full load, and has run ever since without developing a leak.

The break, as will be seen by the photograph, followed roughly the outside line of the port, or at least did not extend inside of that line, so that Mr. Giles decided that it would be feasible to plane the cylinder off along the line *a b* in the drawing, Fig. 42, retaining the full port width. A reinforcing piece *A* was secured to the cylinder with its face nearly in line with the back edge of the valve chamber. As the cylinder wall was not of sufficient thickness to bolt to endwise, wrought-iron straps, one inch in thickness and 22 inches wide, were secured to the cylinder by $1\frac{1}{8}$-inch tap-bolts. The steam-chest was planed off upon the lines indicated by the drawing, and the breaks in the flange and continguous cylinder wall squared. A pattern was made for a piece to fit the surfaces thus created, and the casting bolted to the cylinder and reinforcing pieces as shown, those holding longitudinally being put in first and the surfaces forced into contact at right angles thereto under heavy pressure, aided by blows with a battering-ram while being bolted up. The joint was made tight by running a soft copper wire around inside of the bolts, which were heated to as high a temperature as practicable before being set up, so that in cooling they would draw the surfaces tightly together. This wire terminated at *c* and *c*, where $\frac{3}{8}$-inch holes were drilled, which, after the wire had been cut off, were plugged with brass



REPAIRING A BADLY BROKEN CYLINDER.

FIG. 42

plugs chilled before driving, so that their expansion would close the longitudinal gap. It is the worst case we have ever seen, and the owners of the engine are warm in their expression of appreciation of the ingenuity, enterprise and thoroughness which thus helped them out of a bad situation.

X

REMOVING A TIGHT PISTON-ROD FROM CROSSHEAD

THE piston-rod of an 18 x 24-inch slide valve engine having become scored and worn so that it was impossible to keep packing in the stuffing-box, it was decided to put in a new rod and substitute a set of metallic packing for the old fiber article.

The rod was $3\frac{1}{2}$ inches in diameter and was turned down to 3 inches where it was fitted in the crosshead and secured with a key. But that rod did not want to be removed; in fact it positively refused to budge. It seemed to feel as though the man who put it there twenty years ago had meant it to stay, and stay it did. Backing-out keys were of no avail, and only served to disfigure the crosshead, and gas jets used to expand the crosshead had no apparent effect.

Chain tongs were also brought into play, but while they would very likely have twisted the rod off it would not come out. One of the boys remarked that he "guessed nothing short of a dose of water in the cylinder would fetch the old thing out," and the right idea was revealed.

The engine was placed on the forward or crank center and the slide valve moved so as to cover the back steam port and leave the forward port wide open. A $\frac{1}{2}$-inch pipe was then run from the discharge of a small feed pump and connected to the forward drip-cock of the cylinder. The steam-chest drips were closed and the piston-rod packed with hemp.

The pump was then started, and when the gage on the pump registered 80 pounds the rod walked quietly out of the crosshead. As an 18-inch piston has an area of about 258.5 square inches, the aggregate force exerted on that rod was over 10 tons. The writer therefore desires to register a vote in favor of the water cure for cases of stubbornness.

Core Sand in Cylinder

When starting a new engine the engineer should satisfy himself that everything is in first-class running order, and not trust to the assurance of the erecting men or builders. It is not inferred that the builders are not to be relied upon, but a great many things that are very small in themselves are sometimes overlooked, and at some subsequent time give rise to serious trouble and delay. As an example of this, an engine which had recently been installed and had been run only a few times with light loads. It had a crosshead of the hollow box type, the bottom guides being oiled by the oil passing down through the crosshead from the top guides through holes drilled for that purpose. An emergency arising, the engine was started up and a heavy load thrown on, causing the engine, which had not been very closely adjusted, to pound heavily.

After running a short time the bottom guides began to smoke, and before the engine could be stopped a shower of sparks was flying from the guides.

Investigation showed that the sand remaining from the core had not been entirely removed, and when the engine commenced to pound it had dislodged the sand,

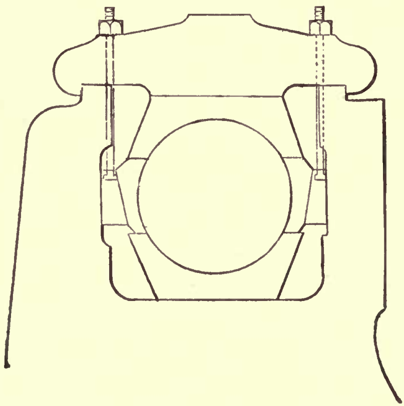


FIG. 43

which naturally found its way out with the oil on to the guide. It required a lot of time and hard work to get the guides and crosshead in proper order, which would have been avoided by a careful examination before starting the engine.

Adjusting Quarter-Boxes

One of the troubles of engineers handling engines having the quarter-boxes adjusted by means of wedges, as shown in Fig. 43, is that of securing and maintaining accurate adjustment of the wedges. The following scheme may prove of value to some of the boys: Make four brass rings or collars, as shown in Fig. 44, to fit

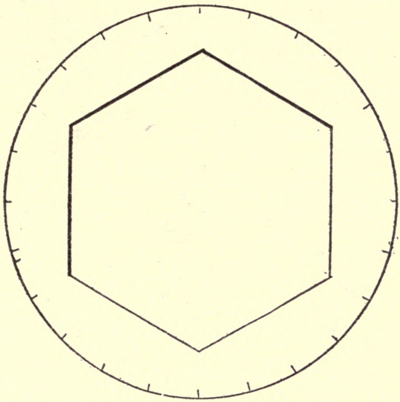


FIG. 44

over the hexagon nuts on the adjusting screws, laying off the edges in twenty-fourths, marking the spaces plainly.

An arrow should be cut in the cap of the pillow-block as a registering point. Place the collars on the nuts and lower the wedges until the bolts are free, then draw up the wedges carefully, being sure to give each nut an equal number of turns, until the proper adjustment is very nearly secured. You may now adjust the bearing by twenty-fourths of a turn of the nuts, and once having obtained the required adjustment it will be easy to take up the boxes quickly and accurately.

XI

SOME MARINE PRACTICE[1]

THERE are lots of tricks which are practised in the engine-rooms of steamships, that are entirely unknown to the average stationary engineer. The following is a sample.

The method used to take up lost motion is different from what is usually done ashore. Marine engines of the propeller type are usually bolt connected, as shown in Fig. 45. The brasses are held together by a bolt on each side, doweled so that it can't turn around. The nuts have a round shank fitting into an enlarged hole, with a set-screw engaging it, as shown in the figure. Liners are placed between the brasses, mostly made of thin brass and paper. The brasses are fitted with a little room at the sides, so that they can be moved back and forth a little, endwise of the pin. (We are dealing now with crank-pin boxes.)

A rough and ready method of finding how slack the box is consists in prying it back and forth with a pivot-bar inserted between the brasses and the crank webs. Most of the bearings on the engine can be tried in this manner, with the exception of the crank-shaft. This endwise motion also insures a better running engine, as

[1] Contributed to Power by Eugene L. Griggs.

it keeps it from binding. The shaft tends to work ahead a little, through the wearing of the thrust bearing, and if everything was fitted snug there would be trouble right off.

To take up the slack on a crank-pin box, it is placed on the top center, as the pin has greatest diameter at this point, if it has worn any, and it is also easier to

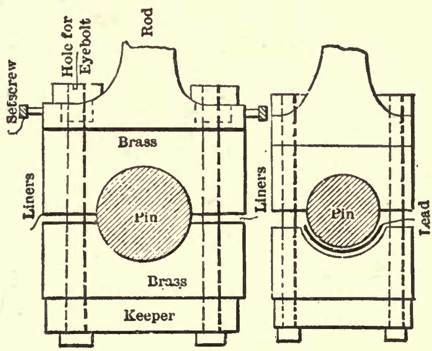


FIG. 45 FIG. 46

get at. The bottom brass is then lowered down about an inch by partially unscrewing the nuts on top. A thin liner is then taken out from each side, and the bolts set up solid again. It is just the same as though the whole bottom end of the connecting-rod were forged in one piece. The nuts on top are marked in some way, so that they can be set up a little farther than they were before, also they can be put back just as they were originally, if necessary. There must always be strain enough on them to hold the liners in place. The nuts are moved with a heavy fork wrench and sledge.

Leads are often taken off, which will tell exactly how the boxes fit, also how much slack there is. Fuse wire is used, usually about No. 19, English wire gage. This wire is cut in lengths a little less than half the circumference of the pin. They are placed in position as shown in Fig. 46. Then the bolts are set up to the marks on the nuts, squeezing the wire out flat to the thickness of a piece of thick writing paper; the nuts next slacked up and the leads removed. They are then tried in a wire gage, and should be from 28 to 32 thick, English standard, for a crank-pin box on a good sized engine. If found to be all right the nuts are screwed up to the original marks.

In taking down the boxes (stripping the pin, as it is called), eyebolts are screwed into small holes tapped into the bolts, tackles hooked on, and the bottom box lowered down into the crank pit. Then a block of wood is adjusted under the crosshead to take the weight, after which the connecting-rod is swung to one side and the top box removed.

The rod is generally forked at the top with a bearing on each side. In taking leads the top boxes are removed, the wires inserted and the boxes placed back again. The principle is the same as the bottom end.

The crank-shaft journals (binders, as they are generally termed) are treated like the wrist-pin boxes. It is always advisable to take leads on the binders, it being impossible to pry the whole shaft back and forth.

All propeller engines of any size are piped up so that water can be played on the principal bearings. For the crank-pins there is a pipe with holes in the bottom, running each side of the pin at right angles to the shaft. By opening a cock a sheet of water is directed downwards on both sides.

The main journals, besides being water-jacketed, have a pipe suspended over them, with a goose-neck joint, allowing it to be moved in any direction. The eccentrics are treated likewise. Now if any bearing starts to heat, it is easily kept under control, because salt water is so cheap.

The guides have water circulating around or through them, as the case may be — in fact most engines won't run well without it.

It is astonishing what a small stream of water will keep anything cool which would otherwise run hot. Salt water is free on a steamer, and a small pump will jerk it overboard again without much trouble.

XII

RE-BABBITTING LARGE ENGINE BOXES

A LARGE engine box is composed of four pieces and a wedge, and some of the larger ones have two wedges. Before starting the job, one should see that all the necessary tools and appliances are at hand and one should also note the position of the eccentric and see if the shaft and bottom box are over the line. In all engines there is a "line of square" on the bed plate under the main bearing and a similar line on the bottom box, the latter of which should be directly over the former. The time to make them so, if they are not, is before dismantling the box. After you have taken out all but the bottom piece, with a good spirit level examine the shaft for line and note what is necessary if anything to true it; then slip the eccentric and fly-wheel, slack off the nuts on the outboard bearing and take the weight of the shaft off the bottom piece with a good jack. The shaft may be blocked up then and the bottom piece drawn out. Run the babbitt out of the bottom piece and weigh it and place the piece on a good solid bench in a good light, and where it may be gotten at from any side. Use an arbor $\frac{1}{16}$ of an inch in diameter smaller than the shaft and set the liners on each end of the box so that you can get $\frac{1}{16}$ of an inch more babbitt than before. When the ends and sides are secure heat the babbitt enough to char a pine stick and then pour it. The pouring should be continuous for each piece, that is, all at one time. When cold enough the arbor may be removed. With a heavy hammer go all over the new babbitt and hammer it hard to make it firm; then dress it on the edges some, after which the sides, top and edges may be hammered. The boxes with the wedge or wedges may then be clamped together, but be sure that the outside dimensions of the boxes and wedges are a little less than the inside measurements in the bed plate. Caliper the shaft, which will give you the size to which the babbitt as now assembled may be bored out. The pieces taken off by the tool may be caught and saved by spreading some canvas on the floor under the lathe. The oil grooves may then be cut with a diamond pointed cold chisel, after which the edges of the grooving should be all gone over with a file to make sure that everything is smooth. The bottom box may then be put back in the bed and after oiling it well the shaft may be put down on it. At this point the shaft should be trued if it is necessary, for if the bottom box is properly placed the rest will come naturally into the correct position. Put in the wedges, if there are two, and if only one put in the back side piece first, and when all are in give them a good oiling. A box that was re-babbitted in the manner described fourteen years ago is giving good service still. It has given no trouble during this time, and promises to last for twenty-five years more.

XIII

KEYING UP CRANK-PINS

Crank-pins and bearings heat from one or all of the following causes: First, poor lubrication; second, boxes not being properly adjusted; third, out of line; fourth, grit or other foreign substance becoming lodged in the bearings; fifth, crank-pin and bearings not sufficiently large to withstand the strain imposed upon them by the action of the reciprocating parts. Assuming that the pin and box are properly proportioned and the rod in line, let us begin by keying up the crank-pin.

Place the crank-pin on the upper quarter and revolve the fly-wheel in the opposite direction to that in which it turns in operation for about one-eighth of a revolution. Slack the set-screw and you will find that a light blow from a soft hammer will be sufficient to take up the lost motion or slackness in the boxes. In some cases the key can be pushed down by hand a considerable distance without the use of the hammer, whereas, if we keyed up the crank-pin without moving the engine in the opposite direction, a sharp blow would be required to move the key. During the operation of driving the key if one hand is placed beneath the key, one can readily learn the weight of blow that must be given in order to move it. Where boxes have been newly fitted we may not be able to drive the key to its proper position at the first keying. Perhaps an engineer of large experience might be able to drive the key the required amount at the first trial, but it is safer to drive it gradually until we arrive at the required position. Even experienced engineers usually drive the key a little at a time; after two or three keyings the position is found where the pin will run without pounding or heating.

Some engineers drive the key in the following way: A line is drawn across the gib and key about half an inch above and parallel with the strap, as shown in Fig. 47 at *A B*. Now by driving the key we can readily see how far the line upon the key has moved from the line upon the gib. A lead pencil is better to draw the line with than a steel scriber, because we can rub out the lines at will, whereas, if we use a steel scriber, the lines drawn from time to time will become confusing and likely to lead to trouble. Usually the key is driven about $\frac{1}{32}$ to $\frac{1}{16}$ of an inch at each keying. When driving the key in a large engine it is best to jar or rock the fly-wheel from one-sixteenth to one-eighth of a revolution in opposite directions. This enables us to take up the lost motion very readily. During the operation of rocking the wheel the finger of the person who is driving the key can be kept on the space, between the boxes, as shown at *D*, Fig. 47, which will enable him to detect any looseness and he will continue to drive the key until all, or nearly all, of the lost motion is taken up.

Another method of driving the key is to drive it down hard, scribe a line across the gib and key and then drive it back about $\frac{3}{32}$ of an inch and make the set-screw tight. This method is a poor one, because one is liable to bind the boxes on the pin.

Another method often successfully employed with light connecting-rods is to disconnect the connecting-rod at the crosshead and let an experienced man rock it up and down while another strikes the key. With a little experience the correct position of the key can

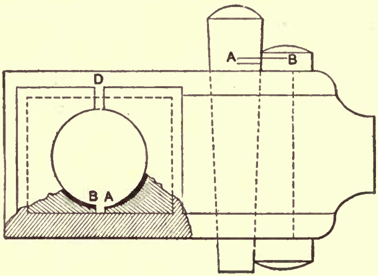


FIG. 47

be quickly found in this way. After the correct position is found it is desirable to draw a line across the gib and key for reference.

Another method is to drive the key partially down; then, if steam is up, rotate the engine slowly, keeping your finger on the space *D*, as before mentioned, and as the crank-pin reaches the centers tap the key with the soft hammer until the lost motion is taken up. The finger being kept upon the space between the boxes enables one to determine when the correct position of the key is being approached. It is clear that the finger cannot be kept on the boxes of a center crank engine, especially if the space between the disks is narrow. If the engine runs over, the half box farthest from the cylinder should be well chamfered and have proper oil grooves cut in it. If it runs under, then the half box nearest the cylinder should be well chamfered and have proper oil grooves. In general both boxes are chamfered and have oil grooves cut in them. The square edge should not be allowed, as it acts as a

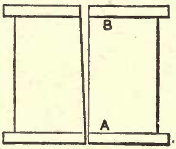


FIG. 48

wiper. The square edge and the edge that is chamfered are shown in section in Fig. 47 at *A* and *B*, respectively.

If the boxes are not fitted square, they are apt to bind the crank-pin on one side, producing pounding and heating. Fig. 48 illustrates this. It will be seen that the boxes are nearly closed at *A*, while a considerable amount of opening is shown at *B*. It will be readily understood that as the boxes wear they will meet at the point *A*, permitting pounding and heating. Engineers of limited experience need not be alarmed at a certain amount of heat in their crank-pin boxes. Crank pins of high-speed engines give good results, even though they heat sufficiently to make it uncomfortable to hold the hand against them. Crank-pins of high-speed engines usually heat quicker than those of slow or medium-speed engines. Suppose we have two crank-pins whose diameters are equal, it can be readily seen that the pin on the higher-speed engine will traverse a greater amount of surface in a given time than the pin on the slower speed engine, hence the greater liability to heat. It will be noticed throughout that the sense of feeling plays an important part in determining the position of the key, but the sense of hearing is no less important.

No matter how well the bearings are fitted, if they

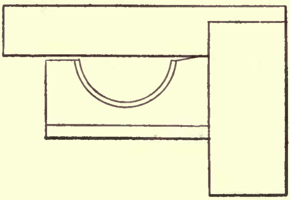


FIG. 49

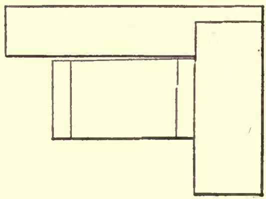


FIG. 50

are not properly lubricated they will heat. Where glass body oil cups are used, engineers or oilers are liable to believe that the cups are working properly because they see the oil lowering in the glass body. Nevertheless we know that if a thread of waste, a hair or other similar material becomes fastened in the point of ejection, it is liable to lead the drop of oil in an entirely opposite direction to that which was intended. Some oil is "lumpy" and oil that has been used and filtered over may contain water. Also, if a draft of cold air is allowed to blow across the oil cups, it will change the consistency of the oil, making it thicker and less liable to flow.

All these things should be taken into consideration in order to prevent heating and pounding. The square or surface plate should be used in fitting up boxes. If we use the square as shown in Figs. 49 and 50, we can readily discover whether the boxes are square or otherwise. When a square is not available, the calipers may be used. Generally a surface plate can be improvised by making use of some portion of the engine bed or steam-chest. If this cannot be done, a heavy level board may be used.

XIV

TESTING FOR A LOOSE CRANK-PIN[1]

A CERTAIN engine pounded when the crank-pin passed the centers, and several tests were applied in efforts to locate the cause without success. The engineer was convinced that the crank-pin was loose, but could not readily prove that his idea of the matter was correct. The pin was riveted in place, and where it was beaded on the inside of the disk crank there was no escape of oil or any other evidence to show that the parts did not remain in perfect contact.

The crank was placed on the inside center, the crank-pin boxes were drawn together firmly by the wedge provided for this purpose, and the crank was then turned a short distance by a lever applied to the fly-wheel; but this test failed to turn the pin, although it was carefully marked to determine how much it moved.

The boxes were again adjusted to working conditions, the crank placed on the inside center, and steam quickly admitted to each end of the cylinder alternately. Although there did not appear to be any unnecessary lost motion while the engine was running, this test showed much more than was expected, making it impossible to tell whether the pin was loose or not.

[1] Contributed to Power by W. H. Wakeman.

The main-bearing quarter-boxes were then clamped firmly to the shaft by screws provided for this purpose, and the test again applied. Lost motion was less than before, but still too great for a satisfactory test of the pin. Furthermore, the crank, although containing enough cast iron to make it apparently very strong, would spring badly every time steam was applied to the piston, and there was enough lost motion in the crank-pin boxes to prevent a satisfactory test. This was taken up as before, after which a square was held on the face of the crank and against the end of the connecting-rod. When steam was again applied to each end of the cylinder, a slight movement of the end of the connecting-rod plainly indicated that the pin was loose.

Of course the crank would spring every time pressure was brought to bear on it in either direction, but this movement was not confounded with that of the connecting-rod, as only the latter indicates the condition of the pin.

A new pin was roughed out and then turned down nearly to the correct size on a working day. On Sunday the old pin was taken out, which was not a difficult job, as it proved to be loose after the riveted head was cut off, and it had been loose long enough to wear the hole oblong, making it necessary to rebore it in order to secure a perfect fit.

Investigation showed that the old pin had been put in about five years previous, without reboring the hole, although the pin that came with the engine was loose enough to be dangerous. If the job had been thoroughly done at that time, nothing further would have been needed; but in addition to omitting the reboring of the old and worn hole, the new pin was simply driven into a cold crank with a sledge hammer, and the end of it beaded or riveted, which, further experience taught, is not sufficient to make a permanent job. The crank should be heated and shrunk onto the pin, especially where hydraulic apparatus is not available for forcing the pin into place.

XV

TWO NARROW ESCAPES[1]

THERE are illustrated and explained in the mechanical papers from month to month so many engine wrecks in which lives are lost and much valuable property destroyed, that it appears as if an engine is almost as dangerous as a boiler at the present time.

These illustrations are object lessons which should lead to precautions that will prevent them elsewhere, but they do not always answer this purpose. Two cases are presented herewith which show narrow escapes from serious accidents, and as other engines now in service may be found in the same condition, if a remedy is promptly applied in every such case it will save future trouble and expense.

A certain engine pounded badly on the head end occasionally, but never on the crank end. This is one of the worst kinds of pounds to locate, as it may not be possible to find it when heard, and when a search is fairly begun it disappears as if by magic.

On making an examination of the external parts, nothing was found that served to locate the pound. The cylinder-head was removed and on putting a socket wrench on one of the follower bolts to remove it, the

[1] Contributed to Power by W. H. Wakeman.

whole piston and rod turned together. This showed at once that the piston-rod was loose in the crosshead, but when the piston was turned back to the position in which it was found, there was no evidence to show where trouble was to be found, because the check-nut was apparently in its place, although not duly fastened.

This is what took place in daily practice. When turned to the position in which it was found there was

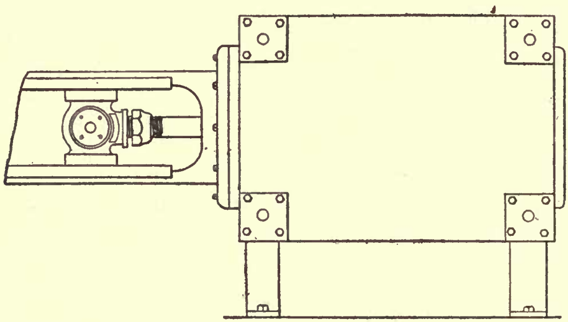


FIG. 51

little or no noise, but occasionally the rod turned out of the crosshead, as shown in Fig. 51, until piston and cylinder-head came together, causing a heavy pound on the head end at each revolution. Had this turned much farther it must have caused a bad wreck, but it would turn in again, stopping all noise for several hours, after which the trouble would be repeated, and this had continued for several weeks. Of course it did not take long to put the piston in place and tighten the check-nut with a suitable wrench, but it was a narrow escape from serious trouble.

Figure 52 illustrates another case in which there was a heavy pound once in each revolution, but here it was on the crank end. This was a larger engine than previously mentioned, as it developed more than 1000 horse-power; therefore, every time it was shut down and its internal parts examined it cost several dollars.

The connecting-rod is a tapering fit in the piston

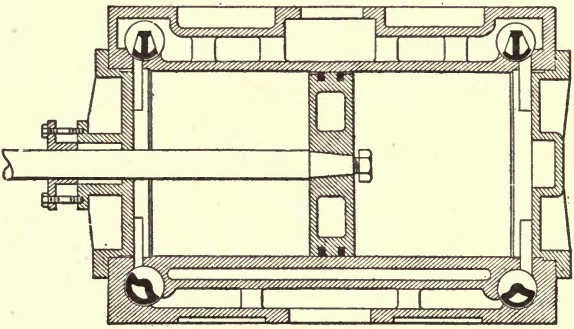


FIG. 52

which is held in place by a large nut. This nut became loosened, hence when steam was admitted to the crank end of the cylinder it forced the piston off until the nut prevented further movement at this point. This jarred the whole engine and caused it to vibrate badly, but when the head end was reached and a charge of steam admitted to this end of the cylinder, it forced the piston on again, as shown, but on account of the tapering rod this was a comparatively easy movement, making little noise.

Of course this was a very dangerous condition of affairs, but the engine was run several days after efforts had been made to locate the cause of trouble, until an expert was summoned, discovered the loose nut, and tightened it in about five minutes.

XVI

SOME PRACTICAL KINKS[1]

Every engineer who has to cut and fit his packing should make a substitute piston-rod. This can be made of wood and of different diameters, corresponding to the sizes of rods found in the plant. Suppose there are four different diameters of rods, then the substitute

FIG. 53

rod will have the appearance shown in Fig. 53. A piece of soft pine wood may be employed for the purpose. If a lathe is convenient, place the piece of wood in it and turn it down to correspond with the diameters of the engine or pump piston-rods, as the case may be. If the distance from shoulder to shoulder be made six inches, and there are four different diameters, the piece of wood will be twenty-four inches long.

This tool will be found to be very handy; instead of waiting for the engine to be stopped in order to obtain the length of the piece of packing needed, the size may be taken from the wooden rod and the packing cut at leisure without danger of one's being burned or scalded.

[1] Contributed to Power by William Kavanagh.

Besides, if the wooden rod is accurately made the packing can be fitted to a finer nicety than could be done by wrapping the packing around the actual, hot piston-rod, trying to locate the required length. Those especially who have charge of compound engines, where the cylinders are superimposed on each other, or even where the cylinders are in tandem, will find this substitute rod a time-saver as well as an accurate tool.

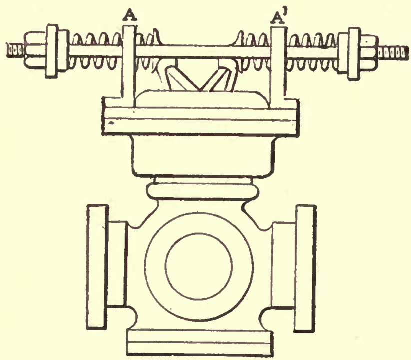


FIG. 54

At a certain plant they carry two hundred pounds pressure on the boilers and employ reducing valves. Two of the compound engines are supplied with steam at 150 pounds, while 110 pounds pressure is delivered to the Corliss engines. It was found recently that one of the compound engines was receiving full boiler pressure, the reducing valve being inoperative. Investigation disclosed that the rods supporting the springs were corroded, and that all the valve needed was a little oil in the guides through which the rods passed. The reducing valve is shown in Fig. 54. The corroded parts were at A A^1.

There is a common exhaust main for all of the engines, and included in the line is the elbow at M, Fig. 55, which had a bad habit of breaking; it had to be renewed three times. Different ideas were expressed with respect to the cause of the breaks. Some said it was water-hammer, others said the castings were bad and so on. The cure was easily affected by inserting an expansion joint, as shown at P.

Heine boilers are used and each boiler is fitted with a pop safety valve. Through some accident or because of poor management the safety valve on boiler No. 1 used to blow continuously. No amount of tension on the spring would stop the blowing, so it was determined to take the valve off and ascertain the difficulty. A substitute valve was bolted in position so the boiler could be used in the meantime. Inspection of the old valve showed that the seat was badly cut; more than a dozen grooves had been eaten into it by the action of the steam. The disk was also badly grooved.

One of the engine-room assistants was detailed to grind out the grooves, the method employed being as follows: A piece of pine wood about half an inch thick was attached to the back of the valve disk by means of two ¼-inch tap-bolts, as shown in Fig. 56. On top of this piece of wood was fastened a ½-inch pipe flange into which was screwed a piece of ½-inch pipe of sufficient length to reach about two feet above the bottom of the valve. Diametrically across the valve body was bolted a section of pine board in which had been drilled a hole to coincide with the valve center. The ½-inch pipe extended through this hole, the sides of which

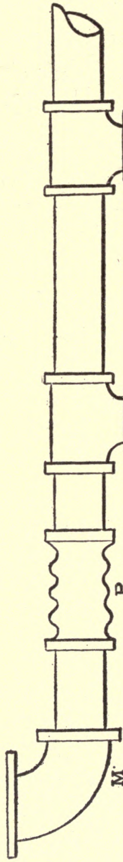


FIG. 55

acted as a guide for the pipe. A carpenter's brace was fitted to the ½-inch pipe and by turning the brace to either the right- or left-hand, the valve disk was made to assume a corresponding motion.

Ground glass and machine-oil composed the cutting agent. By raising the valve disk the seat could be painted with the ground glass and oil. Then the disk was lowered and its rotation effected by the means

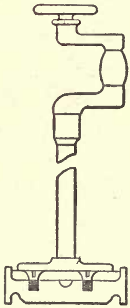


FIG. 56

described. The grinding process was kept up until all of the grooves were worn away. The ground glass was then dispensed with and oil was used alone in the finishing process. It took considerable time but the job was so satisfactory that it paid. If a lathe had been at hand some time could have been saved, although in any case the disk would have to be ground to the seat. The oil-finishing process develops a highly polished surface that, in all probability, cannot be surpassed by any other method.

An accident occurred to the stuffing-box of one of our high-speed engines, the stud-bolts being broken off close to the cylinder-head. Owing to the fact that there is a diaphragm in front of the stuffing-box, the stud-bolts could not be drilled out. The stuffing-box is cast in the plug, which is fitted into the cylinder-head, as shown in Fig. 58, and it was withdrawn as follows:

A heavy piece of timber, for a brace, was bolted across the cylinder-head, having a hole central with the stuffing-box center, as shown in Fig. 57. A $\frac{3}{4}$-inch iron rod of sufficient length to reach from the stuffing-

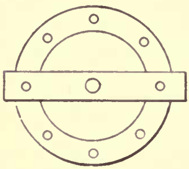


FIG. 57

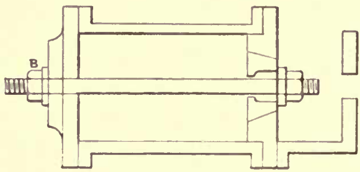


FIG. 58

box to the wooden brace was threaded on both ends and nuts and washers attached. By screwing up the nut at *B*, Fig. 58, the plug was easily drawn into the cylinder. The old stud-bolts were then drilled out and new ones inserted. It will be noticed that the taper or plug looks toward the crank end, so that the steam always exerts a pressure to maintain the plug in position.

The engineers who erected this plant made a serious mistake with one of the Corliss engines. The holding-down bolts were more than six inches too short and to remedy the mistake sockets were screwed on one end of the bolts, into which were screwed short bolts to make up the deficiency. Fig. 59 shows one of the original holding-down bolts with socket and short bolt attached. Later the sockets on the holding-down bolts of the outboard bearing stripped, and it was at first

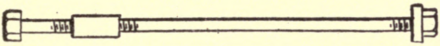


FIG. 59

thought that in order to renew those sockets the fly-wheel, shaft and bearing would have to be removed.

The foundation of the outboard bearing is built in line and close to the buttressed portion of the engine-room wall, as shown in Fig. 60. At *A* is a plan view

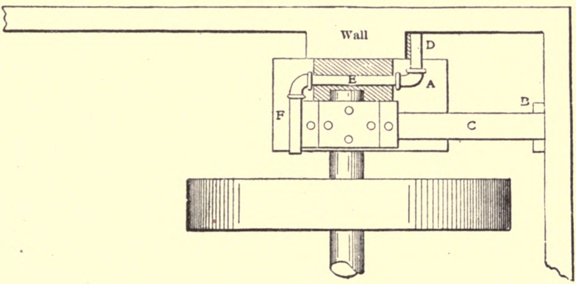


FIG. 60

of the bearing, shaft and fly-wheel. The fly-wheel is fourteen feet in diameter and, of course, the removal of this machinery would entail considerable expense, not to mention loss of time. A section of heavy plank *B* was secured to the wall by means of expansion bolts. A piece of planking three inches thick and twelve inches wide was adjusted from the wall to the bearing *C*. This plank was spiked to *B*, and was sufficiently long to prevent movement of the bearing in that direction.

This part of the problem was easy, but to prevent movement of the bearing in the opposite direction was where the difficulty lay; many ideas were tried out before the plan adopted was evolved. Three 2-inch nipples were cut to fit, as shown at *D E F*, the long nipple *E* being right-and-left. The elbow at *F* was made left because the pipe tongs could be employed to screw up the nipple *E* in the direction of the bearing. A heavy strain was brought to bear upon the nipples *D* and *F* by screwing up the nipple *E*. After this was done the nipple *E* was wedged in place by means of blocks of wood in order to prevent vibration when the engine was running. The engine is now running satisfactorily and to all appearance this bearing will need no further strengthening.

XVII

HOW A NOISY PISTON VALVE WAS CURED[1]

In a certain plant there are three high-speed engines of a prominent make, which are belted to three Mather dynamos and are used for lighting and power purposes. During the summer season two of the engines are sufficient for all requirements, but during the winter all the engines are required. The engines and dynamos are designated as Nos. 1, 2 and 3. No. 1 was built about ten years ago, and was installed in the plant some time before Nos. 2 and 3. Owing to this, No. 1 is worn in the cylinder, valve chamber and bearings to a greater degree than the others. At the time the writer took charge of this plant he noticed that Nos. 1 and 3 were run almost constantly. Upon inquiring why No. 2 was not run, he was informed that she was belted to an "explosive dynamo," and for this reason she was only run during the time the other engines required to be cleaned or keyed. How No. 2 was cured of her explosive propensities appeared in a previous issue of Power, in "Some Things Worth Knowing." The noise or "grunting" of No. 1 attracted the attention of the writer the moment he entered the engine room. Upon asking

[1] Contributed to Power by William Kavanagh.

why the engine was allowed to run without oil — this being his first impression — he was informed that "All the men in this country could not do anything with her to stop the noise. She made that noise since the day she was first started," and, furthermore, "the local expert had the indicator on the engine to try and discover the difficulty." The expert suggested that a new valve and valve chamber be sent for, but the fact that the builder of this engine was over 400 miles away, and also that the engine would have to be shut down, precluded any possibility of making the change. The owner of the engine was satisfied to have all of the local machinists and experts try and stop the "grunting," but they never succeeded. The writer, after becoming acquainted with the plant, determined to see if he could do anything to lessen the noise. The first thing he did was to order three hand oil pumps, one for each engine. No 1 received special attention. Oil was pumped into the valve chamber regularly, but there was no decrease in the noise. The cylinder oil was changed, and in one case 6 pints of cylinder oil were pumped into the valve chamber without causing the slightest effect on the noise. From this it was quite apparent that something must be done besides pumping oil into the cylinder. When an opportunity occurred, the bonnet of the valve chamber was removed and the valve inspected. At first sight there was no apparent reason why the valve should make any noise. The valve was highly polished and showed plenty of lubrication. Fig. 61 is a longitudinal elevation of the valve and valve chamber. The valve is admitting steam to the cylinder through the ports B and is exhausting at the end D through the passage E. The guide K is bolted fast to the rod that maintains the cylindrical parts of the valve in position. Fig. 62 shows how the guide K was constructed. It will be noticed that the leg or sliding part of the guide shown at P is

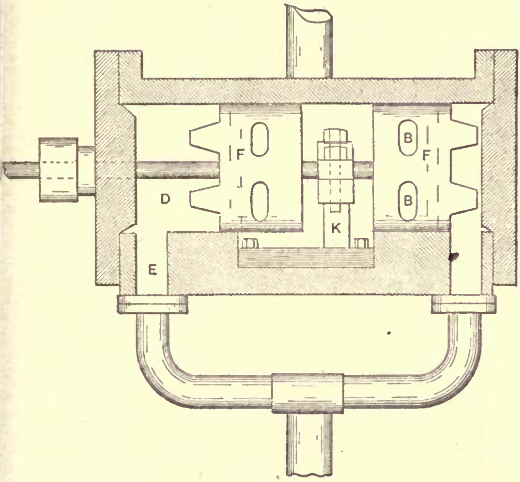


FIG. 61

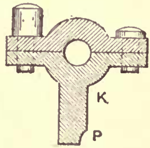


FIG. 62

FIG. 63

FIG. 64

worn away about $\frac{3}{16}$ inch. From the appearance of the guide and guide-way, which is shown in Figs. 63 and 64, it is evident that they were placed in the valve chamber by the builder, and it would be, or might be, wrong to meddle with it; so it was determined to try some other means before attempting to remove or meddle with it. To remove the valve from the chamber necessitated the removal of K. Before the valve was removed the "lead" was noticed. The lead on the head end was $\frac{1}{8}$ inch, while on the crank end the lead was hardly noticeable. After the valve was removed it was inspected thoroughly and calipered to locate the worn side. No fault being found, the valve was replaced. The engine being on the inboard center, the valve was given $\frac{3}{16}$ lead, then the engine was turned over to the outboard or crank center to test for the lead. Now it was immediately noticed that the engine could not be turned over on to the crank center without breaking the valve rod, because the "horns" or guides shown at *D*, Fig. 61, struck the stuffing-box head of the valve chamber. This was indeed a discovery, so the valve had to be replaced as found, as the engine was needed to meet the evening load.

Now we see we have discovered at least two things: First, that the guide *K* is worn all at one side; second, that there is a structural mistake in the valve or valve chamber. Before the bonnet was replaced the valve and chamber were smeared with cylinder oil and plumbago and the engine started for the night. The next day the valve was removed and $\frac{3}{8}$ inch sawed off from the horns, as shown at *D*, which permitted the valve to be properly set. While the valve was out of the chamber staggered grooves were filed, as shown at *F*, around the cylindrical parts of the valve. It was not considered expedient to file complete circles. After the grooves were filed the valve was replaced with proper lead on both centers, the engine was started, and the noise was considerably reduced, together with a saving of 300 pounds of fuel in twelve hours, the load being the same. Now that the grunting did not vanish altogether, the writer could not reconcile himself to the guide *K*. He reasoned in this way against the guide, *that a piston valve should be free to rotate if necessary*, and that the guide *K* prohibited the valve

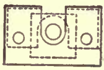


FIG. 65

FIG. 66

from finding its own center or riding position; so he resolved to dispense with the guide altogether. In order to do this it was necessary to make a yoke to maintain the valve stem in position. A piece of heavy sheet copper, which was the only available metal, was used. The heavy lines in Fig. 65 show the copper

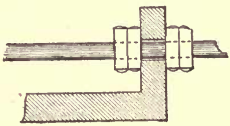


FIG. 67

yoke; the dotted lines show the builders'. It is self-evident that the copper yoke must be put on in the reverse direction to that of the builders' in order to maintain the valve-stem in position should the valve rotate or require to be rotated by hand. Referring to Fig. 66 we see the valve-stem is fitted with a square nut on the end, which nut is shown behind the copper yoke in dotted lines. Now it can be seen that the square corners of the nut must be preserved if we desire to be able to rotate the valve by hand. If we look at Fig. 67 we will see that the hole in the cross-head that operates the valve and stem is enlarged, and the lock-nuts, as shown, permit the stem to be rotated. Now, since the square nut prohibits the valve from rotating it is evident that rotation takes place in the crosshead between the lock-nuts, as indicated. Everything being ready, the valve was put in position without the guide *K*, and all the grunting vanished.

XVIII

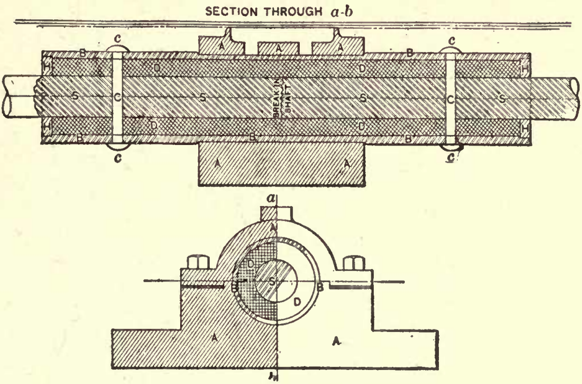
EMERGENCY REPAIRS AND RUSH JOBS

In time of trouble, the man who can take hold and come out with the nearest to a permanent repair is the one who will win out in the end every time. In all, or nearly all, break-down jobs there are usually several ways of making repairs, and all may be very good, but the man who has a large plant waiting for him to do something is not usually going deeply into the various ways of performing a piece of work, but will usually satisfy himself that the way he has outlined will do the work, and then go on and finish it without departure from his original method.

One of the worst things to contend with is a broken shaft. This piece of shafting was used as a jack shaft, and carried on one end an 18 x 60-inch pulley with an 18-inch double belt, and on the other a 24 x 72-inch pulley which ran a gang edger. This shaft was $3\frac{15}{16}$ inches diameter, about 14 feet long, and was carried in three bearings, a center and two end ones. The bearings were all on bridge-trees and keyed in by wedges that were driven in from the side next to the draft, instead of the rear side. One evening it was noticed that the machine on this shaft slowed down and stopped, and very naturally it was supposed the belt had come off. A hasty examination showed that the shaft was broken in two in the center bearing and the collars on each end had prevented its coming out.

A piece of 8-inch pipe was placed in a lathe and bored out for about 1 inch to the depth of $\frac{1}{16}$-inch cut. Both ends were turned the same and then a piece of old dead plate out of a furnace door about 1 inch in thickness was bored out to shaft size; then the outside turned off to the size of the bored-out pipe end. Two of these were made. Then the middle bridge-tree was taken out of the way and the shaft center was blocked up until it was level; a top side line run on it and it was made perfectly straight. Before this was done, however, one of the collars had been slipped over the shaft (one on either side of the break) and then the pipe was put on over shaft also. Then the shaft was lined as explained before, the pipe divided on either side of the break, the collars drove into the ends of the pipe and then ran the whole full of babbitt metal. When the pipe was in the lathe a place was turned in the center long enough for a bearing, and when the pipe was run full of metal the center bridge-tree had only to be let down far enough to take in the tail bearing of an old engine we had, that had been through a fire, babbitt the same, and the job was practically done. After the babbitt had cooled sufficiently, two holes were drilled through the pipe and shaft and two 1-inch machine-steel pins were driven in and was ready to start. The job lasted until we got a new shaft from St. Louis, and when the new one came we did not put it in right away. In Figs. 68 and 69 a sectional view of the repair after it was finished is given. It wasn't very pretty, but it did the work, and that was all we wanted it to do, and it seemed strong enough to have lasted much longer than we used it.

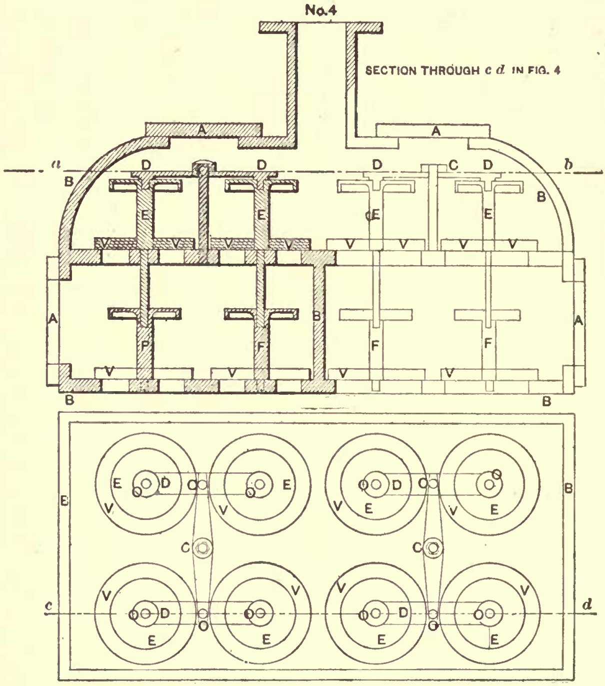
A, center bearing box; *B*, piece of 8-inch pipe; *D*, babbitt lining between pipe and shaft (head removed



FIGS. 68 and 69

in lower view); *S*, shaft; *C*, pins that go through shaft; *H*, cast-iron head, light driving fit in end of pipe.

Trouble was had with a large compound circulating pump on account of it losing so many valves and valve studs. The studs were very light on the lower end, would break off on the slightest provocation and then both valve and stud would go. On account of the way the top of the water end was made it was impossible to get to the studs to drill them out when they were broken off without taking off the whole top of the pump, and that was too big a job to try to do in one day. As we were almost compelled to run condensing or shift part of our load to another station, the breaking



FIGS. 70 and 71

of these studs became a serious thing. In correcting the trouble the arrangement for holding the valves in that is shown in Figs. 70 and 71 was designed. All the old valve studs were removed, a drill run through all the threaded holes into which they had formerly been screwed. This hole was drilled entirely through the upper valve plate, and as the suction valves were directly underneath the discharge, we drilled on into the suction valve plate deep enough to remove all traces of threads, but did not go through the suction plate. Then all holes were reamed and the end of all studs were made a snug fit where they

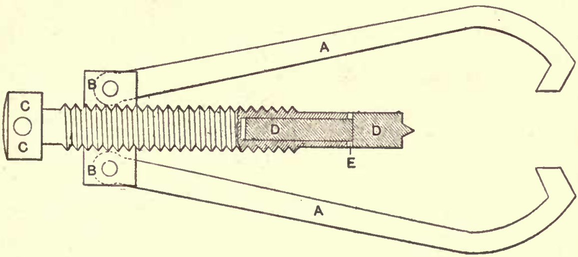


FIG. 72

went through the valve plate. A joint was made on the plates with the shoulder shown on the studs.

A A A A = Bonnets, both discharge and suction.

B B B B = Cast-iron body of water end of pump.

C C C C = One-inch bolt threaded into the discharge plate and holding down crossbars across the top of all four valves.

D D D D = Bridges extending from one valve stud to another.

O O O O = Pins that go through bridges and into top of studs.

E E E E = Studs that hold soft rubber valves in place.

V V V V = Valves, 16 in number and all held in place by two 1-inch studs.

Figure 72 shows a very useful device that is for the purpose of pulling the cranks off of Corliss engine valve stems. The sketch explains itself. *A A* are hooks to catch on inside of bonnet, *D D* is a center, of which various lengths can be used in order to bring the jack within range of different size engines.

Engine Repairs

Here are a couple of safety appliances which we put on our Allis cross-compound Corliss engine:

Have read of an engine being wrecked by the breaking of the bolt which held the crosshead shoe in place. On the out stroke the bottom shoe slipped from under the crosshead and on the return stroke it was caught between the crosshead and the cylinder, causing a wrecked engine.

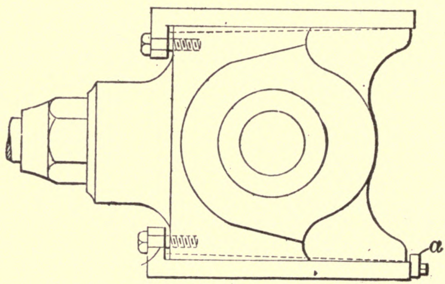


FIG. 73

To guard against this happening to our engine, we drilled and tapped three holes in the end of each shoe and bolted a piece of ½-inch iron across the end, as shown at *a*, Fig 73, so that if the bolt *b* should break, the shoe could not slip from under the crosshead and cause any damage, but would give us warning to shut down.

On the crosshead end of the connecting-rod, which is of the solid-end style, as shown in Fig. 74, there is some danger of the upper bolt breaking and letting the

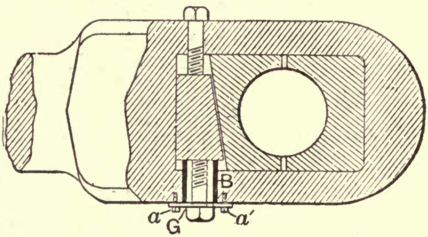


FIG. 74

wedge drop down, which would cause the shutting down of the engine possibly at just the time when it is most needed.

As a precaution against this, we drilled two holes through the washer *c* and drilled and tapped two holes in the bottom of the rod to receive bolts *a* and *a′*. Then after getting the wedge nicely adjusted we took a careful measurement of the space between the bottom of the wedge and the bottom of the rod, and had a piece of wrought-iron tube cut the proper length, so that when it was put in as at *B*, Fig. 74, it would be a snug fit between the washer and the wedge. If the upper bolt should break, the washer and piece of tube would keep the wedge up in place.

When having to draw the wedge up farther to make up for the wearing of the brasses and pin, a liner can be placed between the washer and the piece of tube.

On a Rice & Sargent engine with rod made like Fig. 75, the bolt *A* broke above the wedge and

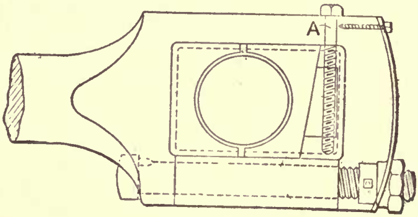


FIG. 75

allowed the wedge to fall, causing a shut-down when the engine was very much needed.

As a quick repair job a piece of $1\frac{1}{4}$-inch rod was threaded to fit the wedge and was screwed through the wedge with a Stilson wrench until it brought up against the bottom of the rod; then a few more turns and the wedge was brought up tight enough and the engine was ready for a start.

When ordering a new bolt, they had one made long enough so that the end of it would just touch the bottom of the rod, leaving no chance for the wedge to get away again.

A most ingenious and effective repair job is that shown in Fig. 76. The break occurred to an old-fashioned box bed type of engine, a crack developing under the main bearing at *A* and gradually extending downward until a clean break was made in both sides of the bed.

Had the cap of this bearing been made and fitted as

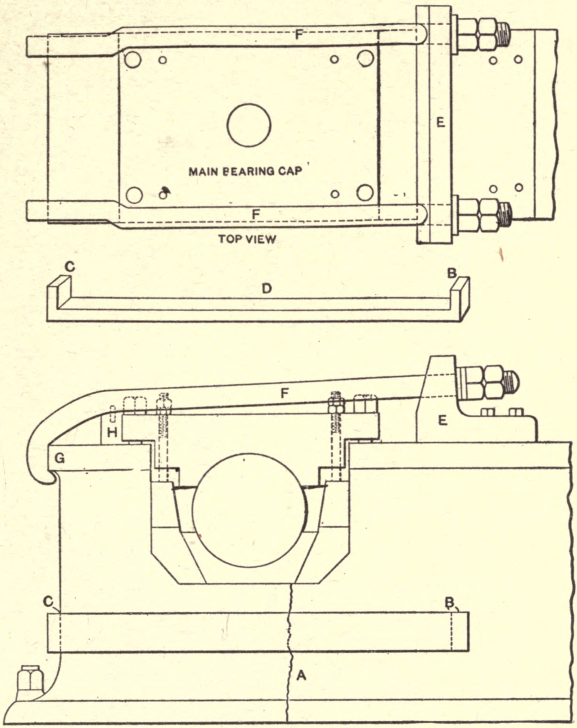


FIG. 76

shown in Fig. 77, it is hardly probable that the crack would ever have appeared, as the lips of the cap fitting over the lugs on the bed would have held and strengthened the bed casting.

As it was imperative that the plant should continue in operation with the least possible delay, and it would have taken several weeks at least to install a new engine, it was decided to attempt temporary repairs.

A rectangular hole $4 \times 1\frac{5}{16}$ inches was made in each side of the bed at *B* and the corners of the bed squared up at *C*. Two bars of $4 \times 1\frac{1}{4}$-inch iron were then made with the ends bent as shown at *D*, the distance

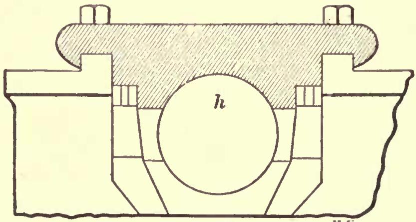


FIG. 77

between angles being less than the distance between *B* and *C* by the amount of opening at the break.

The bars were then heated to a red heat, care being taken not to heat them at the angles, and they were then shoved into place, as shown in Fig. 76, and allowed to cool off, the shrinkage of the bars drawing the sections of the bed together so that the crack was hardly noticeable.

A cast-iron bracket *E* was then made and bolted to the top of the engine bed, as shown in Fig. 76, the flange or lug of which extended out over the sides of the bed, as shown in the top view, holes being cored in the casting to receive the stay-rods *F F*, which were made of 2½-inch square iron, one end being bent and made in the form of a hook to fit over the heading at the top of the bed at *G*.

The bracket ends of the rods were turned and threaded to receive nuts as shown, and an offset put in the rods to enable them to clear the bolts in the cap. The block *H* was placed under the rods to prevent their bearing on the cap, and was secured by a dowel pin in the rod.

When finished and the nuts properly tightened the engine was probably stronger and more serviceable than before the break occurred, and is still in that condition.

XIX

TEMPORARY REPAIR TO BROKEN SHAFT

THE enclosed sketch, Fig. 78, shows how a quick repair was made to a broken shaft. The shaft was used for driving two printing machines and an ink

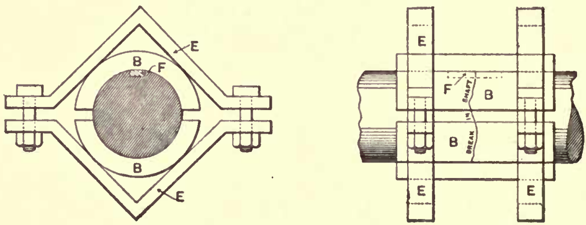


FIG. 78

mill. It was owing to the ink mill becoming locked that the shaft had broken. The broken shaft was uncoupled, so that the rest of the works could run. The broken ends were then propped up from the floor, so that a keyway could be chipped as shown by the dotted lines *F*, about 1½ inches long, ¼ inch deep and ½ inch wide, in both halves, thus making a keyway 3 x ½ x ¼. An old key was then filed to fit. When in place it was filed to the same level as the shaft, as shown. A cast-iron bush *B* was then taken in halves, from a very broad pulley. Then two pair of strong driving clamps *E E*, such as are used by turners for driving large shafts, etc., when turning, were procured. With these were clamped the bushes over the broken part, thus binding the whole together. For a quick repair this will be hard to beat, as the shaft was standing only 1½ hours.

XX

HANDLING MACHINERY WITHOUT MARRING IT

COPPER hammers are largely used for driving keys and other work about the machinery. When new a copper hammer is soft but hardens rapidly with use, so that after short service it will bruise work nearly as much as steel. The only remedy is to take out the handle and anneal it every time it is used.

A better plan is to use a babbitt or lead hammer. If made of lead simply, they get out of shape quickly. To prevent this take a piece of copper pipe, iron pipe size, and drill a hole in the center for handle and fill it with lead. This will keep its shape.

A better plan still is to use hardwood blocks on end. These can be used to put against the part to be driven and hit with a hammer, or larger blocks can be used by striking on end with simply the blocks themselves. Blocks 4 to 5 inches square and 2½ to 3 feet long are handy on large engines for driving the stub end of connecting-rods back and forth when keying up.

Altogether too many people handle finished machinery with steel bars. The ordinary man will put a bar against a polished part or against a bearing and bruise it beyond repair. I have known a piece of 8-inch shaft with four bearings on it, all nicely skidded for handling that had long indentations made on every bearing by bars being placed against them instead of against the skids.

Where machinery is to be moved it pays to have wooden levers made as in the sketch. Fig. 79. These

WOODEN LEVER.

FIG. 79

should be made from maple, the square part 4 x 6 inches and 9 to 10 feet long. After a short experience one man can do more with them than two or three can with a bar. They will do a great deal of the work usually done with a jack and will have it all done before a jack could be gotten into position.

When using wooden blocks and levers they should be kept clean, but it is always good practice to strike them on end on some solid substance before using to jar off any loose dirt that may have become attached to them.

XXI

TO FIND DEAD CENTER

On all engines when preparing to set the valves, the first thing to look after is to find and adjust all lost motion in valve gear, then proceed to place the engine on the "dead center."

Owing to the fact that the crosshead will remain

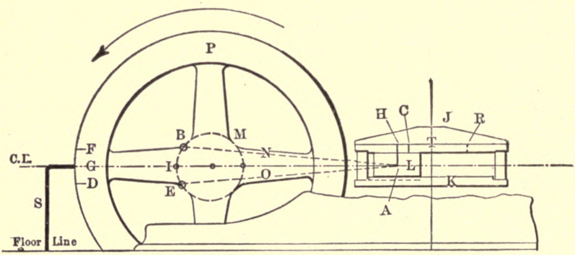


FIG. 80

stationary a short time before the crank-pin passes the center and remain so until the crank has passed the center a short distance, it is best to take greater care than to simply note the point where the crosshead rests at end of the stroke, in order to find "dead center." The process illustrated in Fig. 80 is an accurate process.

In the figure *J* and *K* represent two crosshead guides, *L* the crosshead, *M* the path of crank travel, *N* and *O* the center lines of connecting rod at different positions, and *P* the balance wheel.

To start with, make a tram *S* out of any material convenient, preferably round steel. Turn engine around till crank is at point *B*, at any point where the crosshead is still moving with the other parts, near the end of the stroke. Then mark *A* on the crosshead running the mark up close to nearest or most convenient guide bar. (Where crosshead is down in frame of engine, use a straight edge across the top of the holding-down bars.) Then mark *C* on guide bar opposite mark *A* on the crosshead. Now with tram *S* (one point on given mark on the floor opposite fly-wheel and on a line with one edge of the rim) scribe mark *D* on the face or side of the rim, making the arc come to edge in either case.

Next, turn engine so that the crank passes the center and the mark *A* on the crosshead again comes to mark *C* on the guide bar.

Then take tram, again resting it on the same point on the floor and scribe arc to mark *F* on same edge of rim of wheel. The crank will then be at *E*.

Now place prick punch marks as near edge as possible at points *D* and *F* and with a pair of dividers, bisect the distance from *D* to *F* and make mark *G*.

Turn the engine back so that, with the tram resting at same point on the floor, the other point of tram will touch *G*.

The crank will then be on "dead center" at *I* and the mark *A* on the crosshead will be opposite mark *H* on guide bar, which you will now mark as the point where crosshead reaches end of stroke.

Turn the engine to the opposite end of stroke and repeat the foregoing moves and you will then have both "dead centers" found and marked.

Now, with a marking chisel, go over all scribe marks and make them permanent and where the center punch marks are on rim of wheel, the floor and frame, it is a wise precaution to place marks with chisel around the

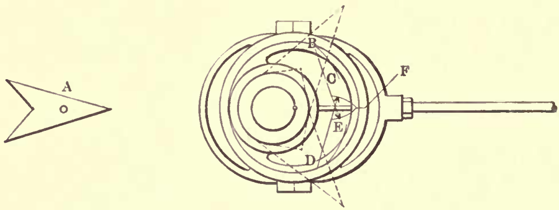


FIG. 81

center punch marks thus $\boxed{\cdot}$. It is also a good means of finding your marks in future.

NOTE: In some cases there is no fly or balance wheel or it is not convenient to tram to them for marking. If so, the rim of the crank disk or even the surface of the shaft will do. The further you get from the shaft center, the better however.

The next move is to get the eccentric "dead center."

Where the eccentrics are fastened by set-screws, friction keys or keys easily withdrawn, loosen up on one or the other of them as the case may be and turn the eccentric around the shaft while finding the centers. Where there is a fixed eccentric such as governor eccentrics, the engine itself must be turned around to find the points we are after.

Make a tram *A*, Fig. 81, out of a board or sheet steel and place at point indicated a nail or some pointed piece of iron or steel, just far enough out so that you can scribe arcs *B*, *C* and *D*, *E*, bringing the arcs down to the edge of the eccentric at points *B* and *D*.

Be careful in using the tram, to have the end on

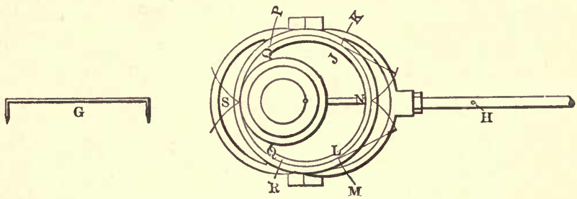


FIG. 82

shaft or boss of eccentric the same distance away from eccentric in both instances.

Take a pair of dividers and from points *B* and *D*, scribe arcs so that they will exactly intersect on the eccentric edge at *F*. This will be the center line of eccentric.

Make another tram *G*, Fig. 82, make mark *H* with center punch on eccentric rod and with one leg of tram on point *H* scribe arcs, *J*, *K*, and *L*, *M*, on the eccentric-strap coming to points *J* and *L* on the edge of strap.

With the dividers scribe from points *J* and *L* arcs exactly intersecting each other at point *N*.

With marking chisel, make permanent marks at point *F* on the eccentric and point *N* on the eccentric-strap. Bring the eccentric around so that both points correspond and you have the eccentric on one "dead center."

From points *J* and *L* scribe arcs *O, P* and *Q, R*, ending at points *O* and *Q* on edge of strap, from points *O* and *Q*, scribe arcs intersecting at point *S* on the edge of strap. With chisel, make mark on strap at *S*. Bring eccentric around so that point *F* corresponds with point *S* and you have the eccentric on the opposite end of travel.

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