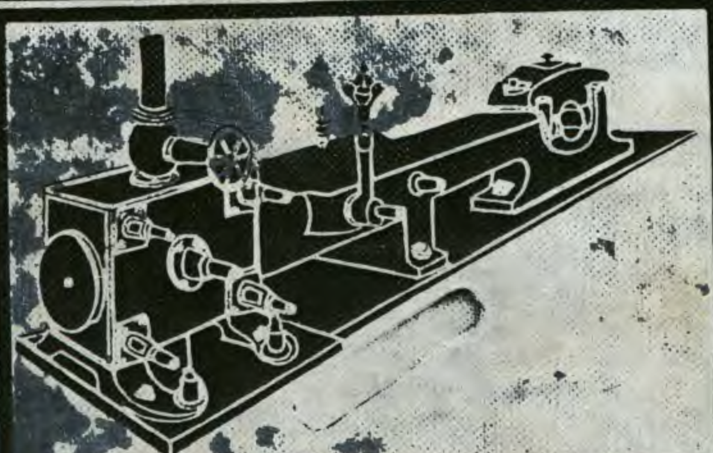
STEAM ENGINE TROUBLES

HAMKENS



STEAM ENGINE TROUBLES

A PRACTICAL TREATISE

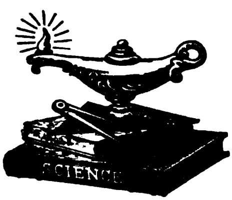
FOR THE ENGINEER, TELLING HOW TO LOCATE AND REMEDY TROUBLES WITH A STEAM ENGINE

CYLINDERS, VALVES, PISTONS, FRAMES, PILLOW BLOCKS AND OTHER BEARINGS, CONNECTING RODS, WRISTPLATES, DASH-POTS, REACHRODS, VALVE GEARS, GOVERNORS, PIPING, THROTTLE AND EMERGENCY VALVES, SAFETY STOPS, FLYWHEELS, OILERS, ETC., ARE ALL TREATED AND IF ANY TROUBLE WITH THESE PARTS ARE FOUND, THE BOOK GIVES THE REASONS AND TELLS HOW TO REMEDY THEM

BY

H. HAMKENS

MECHANICAL ENGINEER



THOROUGHLY ILLUSTRATED WITH DETAILED ENGRAVINGS

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PREFACE

DURING a number of years the writer has been in the closest touch with steam engines, small and large, as designer and superintendent of erection, and has always felt that the troubles which an engineer encounters form a large and not unimportant part of his professional life. With the great number of different designs of engines in use it keeps engineers and power plant owners guessing why things go wrong with certain parts of engines, and being without the necessary theoretical knowledge, they are at a perfect loss to know how they can be prevented or remedied. It seems to be the hardest thing in the world to get any real inside information in case of trouble, for the reason that nobody wants to fix the blame and nobody wants to take it, and be made responsible. Another thing, it is very often a thankless job for the engineer to point out defects of an engine which may in future become a source of trouble.

The development and perfection of the steam engine is in a large part due to designers who made a close study of the existing designs with the idea of remedying defects and making improvements, in order to limit breakdowns and repairs, and in this way making engines more reliable, easier to handle and increasing their economical performance. But the constant improving and changing has added so great a number of differently arranged parts that it is almost impossible for the practical man, whose time is limited and who has not the opportunity to come in personal contact with the new features which have been introduced, to keep informed.

These considerations led the author to write a series of articles on "Steam Engine Troubles" which were published in *Power* a short while ago. The interest which was manifested in these articles led him to revise and extend the same, which are now issued in book form.

The author acknowledges with thanks many suggestions which were made by Charles H. Bromley, associate editor of *Power*.

January, 1919.

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INTRODUCTION

For the operating engineer or the prospective purchaser it may be of interest to know which design and construction of different parts of an engine may best be selected and which should be avoided. In other words, it would be a good thing if engine troubles in general and particular could be listed, so as to give interested parties a chance to forestall costly and dangerous accidents.

In the following pages the troubles which the principal parts of steam engines are subject to are described, good design is contrasted with bad, the most suitable material for certain parts and the most approved construction of the same is pointed out.

The advantages and disadvantages of many designs are discussed, the troubles they are heir to and ways to overcome or minimize them. The directions given for correcting existing evils are easily understood, by following them breakdowns and costly accidents can be eliminated.

The principal considerations in the building of foundations, setting of templets and lining up an engine are given with a complete account of how to erect the engine in the language of the man on the job.

There are directions for lubrication, valve setting and testing, also how to locate trouble with an indicator, and what adjustments are to be made on new engines and maintained on old ones. Speed regulations, so essential to good service, are treated.

Simple rules for calculating safe pressures of main-bearings, stresses in flywheels, strength of cylinder walls, etc., are given which enable an engineer to find out whether the engine he runs is constructed on sound principles or not.

H. HAMKENS.

January, 1919.

STEAM ENGINE TROUBLES

CHAPTER I

CYLINDERS

Engine cylinders must be of correct size. Stresses in cylinder walls. Radiation losses. Bolts and nuts. Slide-valve cylinder. Square-corner Corliss cylinder. Lagging. Built-up cylinders. Steam jackets. Sleeves for cylinders. Material for cylinder castings. Insufficient drainage. Location of valves. Inaccessibility of cylinders with valves in heads. Ground joints. Regular joints.

THE power of a steam engine is, to a large extent, determined by the size of its cylinder, and its usefulness depends on the construction and material of the same. Many engine troubles are due to the size of the cylinder which is used, and to the power developed; the cylinder may be too large or too small, and the engine may not carry a sufficient load or be overloaded, either one is liable to be a source of trouble. Overloading will put too heavy a strain on the moving parts, while too light a load will be a heavy drain on the coalpile, altogether out of proportion to the work done. Since most of the engines in use are manufactured, that is, made to standard designs and from existing patterns, their power and economical range is dependent on certain predetermined factors, principally steam pressure and speed. Twenty years ago the majority of stationary engines were designed for 80 to 100 lbs. boiler pressure and a piston speed not to exceed 600 ft. per minute. The cylinder walls, size of bolts, thickness of flanges, piston rods and other working parts were designed for the prevailing boiler pressure with a factor of safety more or less determined by practice; tne dimensions of many engines were fixed by a rule of thumb, which answered very well under the existing conditions.

As long as boilers and engines are well matched and of the proper size this method may give fairly good results, but as soon as a change is made either by increasing the steam pressure or speeding up the engine in order to get more power the whole arrangement is thrown out of balance and trouble may be expected. Engine cylinders of small bore, 12 ins. or less, have, as a rule, a much larger factor of safety for the cylinder walls and other parts than the larger sizes, for the simple reason that on small castings time is saved in molding and casting by making the castings a little heavier than is absolutely necessary for safety's sake. The time saved more than compensates for the cost of the extra metal used. Medium and larger size cylinders, however, are made mostly without any excess of metal and they are, therefore, more limited in regard to pressure. If the steam pressure is raised to any considerable extent it is well to do some investigating and a little figuring; not only should the strength of the cylinder walls be ascertained, but also that of the steam chest, bolts and other parts, which are connected with the cylinder.

It is a good rule to let the combined stresses in the cylinder walls in no part exceed 1500 lbs. per square inch, whether due to steam pressure or other forces. The principal stresses due to steam pressure in the cylinder barrel are tensional, tending to disrupt the metal in two directions; one of which is longitudinal and the other circumferential. The latter is practically twice as great as the former, as a simple calculation will show. To get the correct idea of this we will take the two views of a cylinder, shown in Fig. 1, and assume that the inside diameter of the same is 1 in., that the cylinder walls are 1 in. thick and the steam pressure 100 lbs. per square inch; then the pressure on each half of the circumference for 1 in. in length will be 100 lbs. and the tension on each side will be

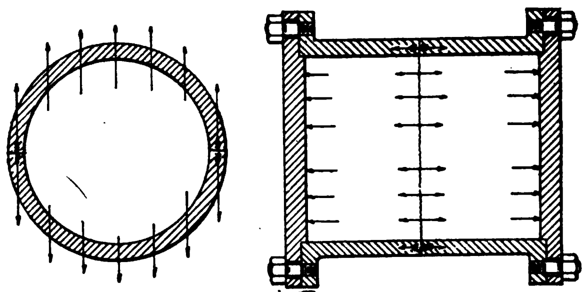


FIG. 1.—Stresses in cylinder walls.

50 lbs. per square inch. If both ends are closed the end pressure in both directions will be $\frac{0.785 \times 100}{3.14} = 25$ lbs. per square inch. The combined stresses will be $50+25 = 75$ lbs. per square inch. The cylinder would, therefore, be safe for $\frac{1500 \times 100}{75} = 2000$ lbs. per square inch. If the diameter was 10 ins. the safe pressure would be $\frac{2000}{10} = 200$ lbs., etc. Without any complications and great refinements a simple calculation of this kind, if applied to a cylinder, will indicate the danger point

which may be reached in reboring, or if the steam pressure is to be increased, it will also show the factor of safety allowed for the cylinder walls.

The price is very often the governing factor in choosing one engine in preference to another, and a slide valve engine may be selected or a high speed automatic engine, although it is a well-known fact that their steam consumption is very much greater than that of a Corliss engine. The lowest bidder is, as a rule, successful in securing the order. The consequence of this is that man-

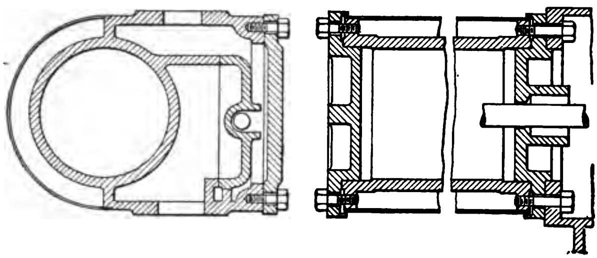


FIG. 2.—Slide valve cylinder with unprotected steam chest.

ufacturers endeavor to reduce the cost of their engines as much as possible, save time and labor in order to produce a machine, which apparently is just as good as a more expensive one, but will be found in the course of time to be of a very inferior kind and a source of no end of trouble and expense.

To illustrate this the cylinder of a slide valve engine, shown in Fig. 2, which represents a design much in use, will serve the purpose to point out two of the most troublesome features connected with engines. One of the bad features about it is that the whole steam chest and cover are not covered with insulating material or lagging, in consequence of which there is a constant loss of steam through radiation. Assuming that 1 lb. of steam is condensed per square foot of uncovered surface per hour and taking the uncovered surface to be 10 sq. ft. there will be 10 lbs. of steam condensed per hour; for an ordinary boiler this would be equivalent to a loss of 2 lbs. of coal per hour or about 3 tons per year. The steam condensed in this way is not only a total loss, but it very seriously interferes with the lubrication of the valve and

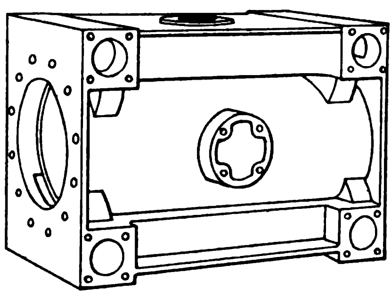


FIG. 3.—Corliss cylinder with exposed surfaces.

piston and increases the friction and the wear of the moving parts.

Another bad feature of the construction is that the connections between the cylinder and frame, back head and steam chest cover are made by means of tap bolts, where studs and nuts should be used. If the different parts are disconnected a number of times the threaded holes in the cast iron will wear out and there will be trouble in making the joints tight.

In larger engine cylinders of the Corliss type the radiation may be just as bad as in the common slide valve cylinder. Fig. 3 shows the old style square corner Corliss cylinder with a large exposed surface; not only are the ends unprotected from radiation, but the whole top is uncovered and so is the bottom. Besides, the exhaust chest is cast directly on to the cylinder barrel, which increases the radiation considerably. This construction became so troublesome, that in more recent designs the exhaust chamber is cast separated from the cylinder barrel with a space between them. To add to the gen-

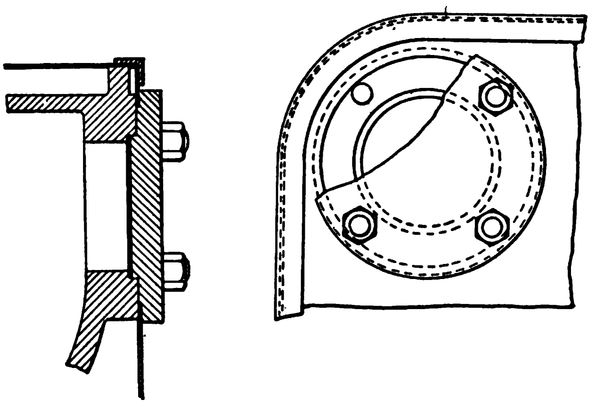


FIG. 4.—Bonnet flanges designed to hold the lagging in place.

eral wastefulness in some cylinders, side ribs are cast on the top and bottom plates for supporting the lagging. These ribbed cylinders are nothing but radiators, they condense a lot of steam and make the engine room uncomfortably hot on warm days. The lagging used to be made of wood, which, after a few years, would begin to rot or be charred from the heat of the steam.

In order to give the best service all the heated parts of a cylinder must be covered with non-conducting material, either asbestos or magnesia, and this again should be protected by cast iron or steel plates, so arranged that they can be removed readily without disturbing any of the other parts, as, for instance, on Corliss engines the bonnets or the valve gear. The common practice on Corliss cylinders is to hold part of the lagging in place by letting the bonnet flanges overlap, as shown in Fig. 4. If, for some reason or other, the side lagging plates have to be removed the whole valve gear must be dismantled, which is no small job and throws the engine out of com-

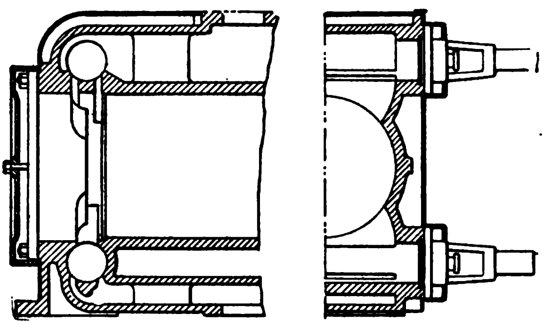


FIG. 5.—Corliss cylinder with protection for heated surfaces.

mission for a considerable time. The object of the construction is, of course, to reduce the cost without regard to any future trouble which it may cause. The most approved construction of the lagging for a Corliss cylinder is shown in Fig. 5, in which all the parts can be removed without disturbing the bonnets, valve gear, or cylinder head; all the heated parts connected with the cylinder as, for instance, the bonnets and heads are protected by means of polished covers, which serve the double purpose of preventing radiation and covering the joints of the lagging plates.

Very peculiar constructions of cylinders which give a great deal of trouble are found in many New England mills

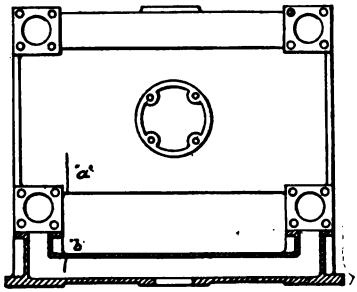


FIG. 6.—H.P. Cylinder with separate exhaust chamber.

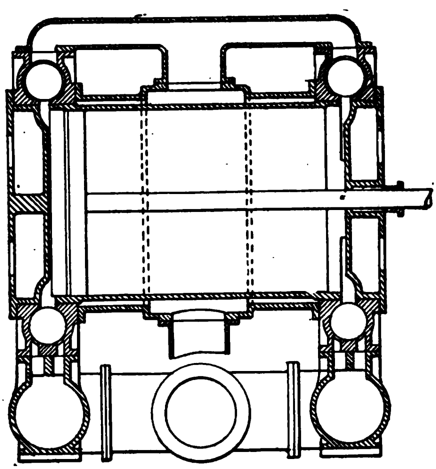


FIG. 7.—Low-pressure cylinder made in sections.

principally on tandem or cross compound engines of the larger sizes. These special designs are shown in Figs.

6 and 7, which represent the construction of a high- or low-pressure cylinder, respectively. In the high-pressure cylinder the exhaust chamber forms a separate casting which is bolted on below and acts as a pedestal for the cylinder. The difference of expansion between the cylinder barrel and the exhaust chamber is considerable, the joints are, therefore, apt to leak or the castings will crack at either *a* or *b*. The low-pressure cylinder in Fig. 7 is made in several pieces, it consists of an outside casing with the valve chambers attached, a sleeve forming the bore pressed in and two exhaust chambers with a connecting tee placed underneath, supporting the cylinder. Trouble is sure to result from the number of different castings, all bolted together; the joints are very hard to keep tight, but the most troublesome part is that the cylinder is insufficiently supported, which results in a rocking motion, loosening of the joints, leaks and finally rupture of some of the parts. Cylinders of this type found favor with some manufacturers, who had not the facilities in their foundries to cast the whole structure in one piece, and with others from the mistaken idea that shrinkage strains could be eliminated by casting them in several pieces.

The steam jacketing of low-pressure cylinders in compound engines was quite a fad for a while and the sleeve construction was favored by many engineers, until later it was shown by actual tests that the supposed saving in steam was not borne out by the facts, at least not in this design. As will be noted on the drawing the low-pressure cylinder is not jacketed with live steam but with steam from the receiver, the heating effect is, therefore, much reduced, the condensation first in the enlarged exhaust chamber of the high-pressure cylinder and next in the jacket of the low-pressure cylinder more than offset the slight gain in steam actually consumed in the cylinders. Except in pumping engines steam jackets have almost become obsolete.

If the sleeve of a jacketed cylinder is not put in the right way it may be the cause of considerable loss of steam and of a very annoying pound in the engine. Fig. 8 shows a method of inserting a sleeve as practiced by some engine builders, which must be condemned as imprac-

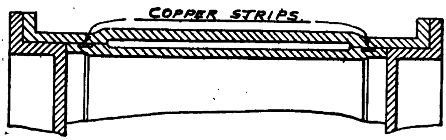


FIG. 8.—Impracticable sleeve construction.

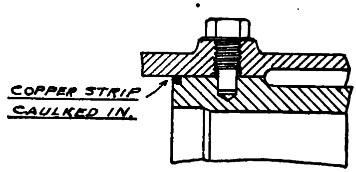


FIG. 9.—Correct sleeve construction.

ticable. A wide strip of copper is pounded into a groove at each end of the sleeve, it is turned the same diameter as the outside of the sleeve with a certain allowance for forcing in. The supposition is that the copper ring will expand more than the cast iron when heated and form a tight joint. The supposition is erroneous and the method is wrong, the copper might as well be left out. The correct way to make the joint is shown in Fig. 9, which consists in turning a groove with a slight taper in each end of the sleeve and after the latter is drawn in tight in inserting a $\frac{3}{8}$-in. square copper strip into the groove and caulking it. If a leak should develop in the joint all that will be necessary to stop it will be to expand the strip of copper a trifle more by hammering. Fig. 9 also shows a method to secure the sleeve at each end and keep it from moving lengthwise in the cylinder; this is effected by drilling a hole through the outside of the cylinder into the enlarged part of the sleeve and after tapping it part of the way a special tap bolt is screwed in. There should be two of these tap bolts at each end of the cylinder.

The material of which steam cylinders are made is of great importance as far as the life of an engine is concerned; in many cases the iron is too soft and the wear excessive, making repeated reboring necessary. The iron must be close grained and hard, but it should also have a tensile strength of between 25,000 and 30,000 lbs. per square inch. Remelted pig iron with a good percentage of selected scrap melted in a cupola seems to give the best results. On no account should a cylinder be made of air furnace iron which, as a rule, is not suitable for this purpose, since it is mostly very soft and porous, although its tensile strength may be higher than required. Some foundries have used a certain percentage of steel in their cylinder castings, but the result has not been very satisfactory, since it is almost impossible to make that kind of castings homogeneous, they are liable to have blow holes and soft spots. Attempts have been made to chill the bore of cylinders by lining the core with bricks or some other hard material; they have not been crowned with success. It is advisable to steer clear of such experimental affairs which are, in most cases, nothing but trouble makers for the engineer and power user. The best thing to do is to stick to methods which have proved to be successful in the course of time.

A good deal of trouble with cylinders is due to insufficient drainage. There is always some water in the cylinder of a running steam engine, which should be expelled as soon as it forms; all pockets in which water can accumulate must be avoided. This requirement is strangely overlooked by many engine builders who produce designs that almost seem to invite trouble in this respect. One of the worst designs is shown in Fig. 10. As

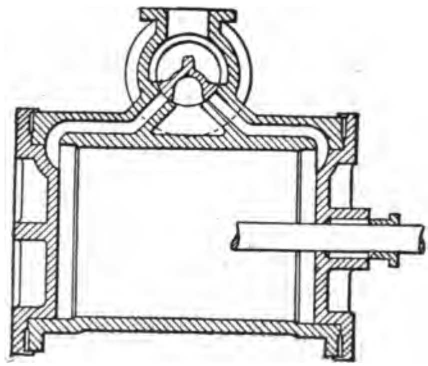


FIG. 10.—Rocking valve cylinder with insufficient drainage.

will be noted the steam chest is placed on the top of the horizontal cylinder, steam is distributed by means of a rocking valve. However, it is quite immaterial what the shape of the valve is, the main trouble is that its location is radically wrong. It is located without regard to drainage of the cylinder, and any water which is carried over through the steam pipe or is condensed in the cylinder must either be drained off through special valves at the bottom, or forced out with the exhaust in an upward direction, contrary to the laws of nature. Unless cylinders of this style are provided with very large snifting valves, one at each end, they are liable to be fractured by water. Similar trouble may be expected from any design in which

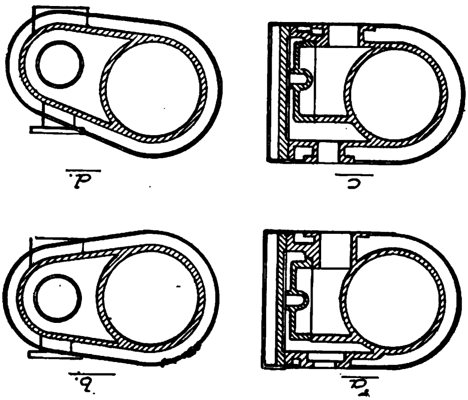


FIG. 11.—Diagrammatic views of cylinders for drainage.

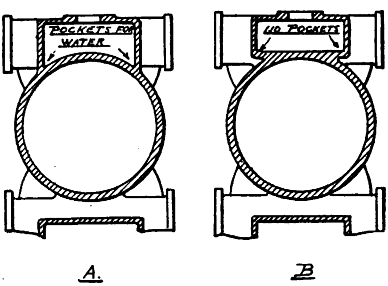


FIG. 12.—Cross-sectional views of steam chests.

the valve is located on the side of the cylinder in such a position that the water cannot be drained automatically. In Fig. 11 are shown four diagrammatic views of cylinders, in which *a* and *b* show the wrong location for a slide valve and piston valve, respectively, while *c* and *d* give the correct position of the same as to drainage.

In most Corliss engine cylinders the drainage is perfect at least as far as the cylinder proper is concerned, but in some designs water will collect in the steam chest, which may cause trouble if it is thrown over in a body by an abnormal rush of steam. Fig. 12 shows two cross-sectional views of Corliss cylinders of which the design

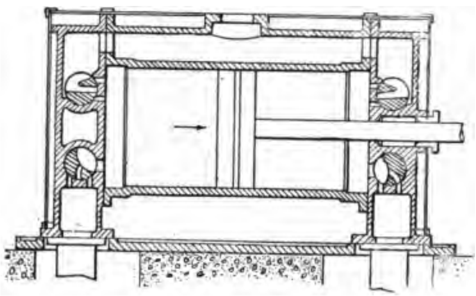


FIG. 13.—Corliss cylinder with valves in heads.

shown in *B* is to be preferred, since it is self draining, while the pockets shown in *A* will fill with water.

In order to reduce the clearance space in cylinders the valves are sometimes placed in the heads, this we find in many engines of the Corliss type, for which a claim of high economy is made. If the exhaust ports are located, as shown in Fig. 13, there is very little chance for water that accumulates in the bottom of the cylinder to drain off; it will have to be forced out by the piston. A very troublesome feature of this construction is that the only way to get at the inside of the cylinder and at the piston and packing rings is to remove the back head including the valves, bonnets and everything connected with them; the valve gear must be disconnected and very likely the end of the cylinder must be propped up. This is even in large plants with plenty of help and lifting facilities quite a job for the engineer to undertake, the consequence is that cylinders of this design are not examined as often as would be desirable. It is advisable to steer clear of such troublesome designs even if the economy is a shade better than with the ordinary kind.

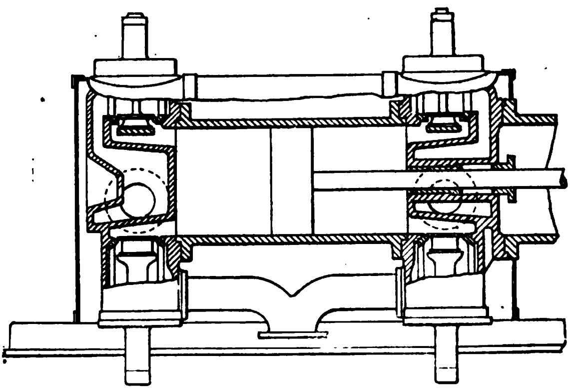


FIG. 14.—Complicated cylinder design with poppet valves.

The acme of complicity is shown in Fig. 14, representing a cylinder construction for high pressure and superheated steam. This design was originated several years ago in Europe and has been copied by engine builders in the United States. The poppet valves which are used are located in the cylinder heads, part of the valve gear is also attached to the latter. The cylinder proper consists of a simple cylindrical casting of symmetrical shape which is free to expand and contract. The inside of this cylinder is even less accessible than the one described previously. After removing the lagging, a part of the non-conducting material around the rear flange of the cylinder barrel must be knocked off in order to get at the nuts, steam and exhaust flanges must be disconnected, part of the valve gear taken off and the back head with the valves moved back on the foundation plates. If there is no spare engine to take the load, this work will throw the plant out of commission for the best part of a day, before everything is in running order again.

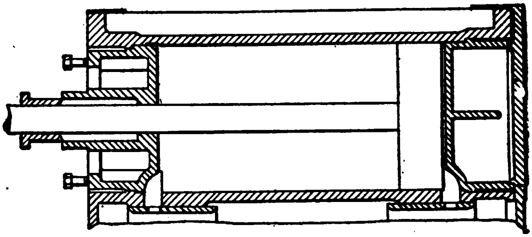


FIG. 15.—Cylinder and heads with ground joints.

The joints between the cylinder and heads will give trouble by getting leaky, which may be due to several causes. On some engines the joints are ground very carefully, obviating the use of gaskets. These ground surfaces are very apt to be damaged in handling of the heads, which, of course, will make them leak; if the leak is not stopped immediately the out-rushing steam will very likely damage the surface to such an extent that the surfaces have to be reground, which is a tedious and expensive job since often all the studs have to be removed and later put in again.

Fig. 15 shows a design in which the difficulty about regrinding has been successfully overcome, the heads are made without the customary flange, they are both inserted from the back; the front head is drawn onto its seat by bolts while the back head is held in place by a plate bolted to the rear end of the cylinder. The ground surfaces are very narrow, if a leak should develop a copper ring made of wire could be used for quick repair. The construction is rather expensive and has, therefore, not found much favor with engine builders.

The design commonly used is shown in Fig. 16, in this the head is made a good fit a short distance into the cylinder, the depth varying from $\frac{1}{2}$ to $1\frac{1}{4}$ in. according to the size of the cylinder; the balance of the head is relieved

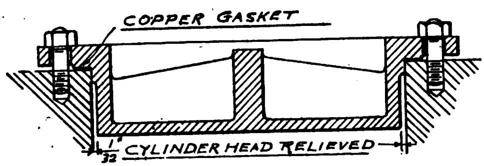


FIG. 16.—Regular joint between cylinder and head.

about $\frac{1}{32}$ in. all round, in order to make it enter freely. It is important to make the space as small as possible to prevent continuous condensation in the same. The inner surface of the head should be highly polished, also for the sake of avoiding condensation. The best material for the joint is a plain ring of soft copper not over $\frac{1}{16}$ in. thick and from $\frac{3}{4}$ to $1\frac{1}{4}$ in. wide according to size. Formerly corrugated copper rings were much used, but they are easily damaged, in which condition they will not make a tight joint. Attempts have been made to make gaskets of a combination of corrugated copper and asbestos, they may be used for cylinders of small bore but not for the medium and larger sizes.

CHAPTER II

VALVES

Common slide valve. Partly balanced valves. Balancing rings. Influence of the angularity of the connecting rod. Equalization of cut-off. Valves for automatic engines. Prof. Sweet's valve. Piston valves. Adjustable valve seats. Adjustable piston-valve ring. Corliss valves. Port boring rig. Single-ported Corliss valves. Double-ported valves. Corliss valves which wear leaky. Steam valves must lift. Water in cylinder. Porter-Allen valves. Multiported valves. Gridiron valves. Effect of high-pressure steam and superheat. Poppet valves.

NOTHING is of greater importance on an engine than the valve or valves, on the operation of which depends, to a large extent, the success of the machine. The ingenuity of designers and engine builders has been taxed to the limit to make valves which will remain tight, are easily moved and give long service. The common slide valve has been used more than any other valve; it is, however, gradually being abandoned for the reason that it is so extraordinarily wasteful in the use of steam and subject to many other evils. Its simplest form is shown in Fig. 17; the principal trouble with it is due to the steam pressure acting on its large surface, causing excessive wear of valve and seat. Insufficient lubrication or hard spots in the valve or its seat are very often responsible for uneven wear and leaks. Leaky valves must be refitted and if the wear is uneven a slight cut should be taken off valve and seat and both of them scraped to a perfect fit. One of the things to look out for on valves of the older type is that they do not wear shoulders; this part was formerly often overlooked and the result was perhaps a broken valve rod or a tremendous waste of steam. Excessive

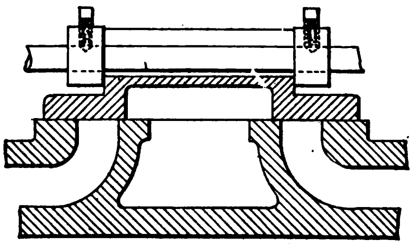


FIG. 17.—Common slide valve.

friction may make the eccentric run hot, in order to avoid this the valve must be balanced either partly or entirely.

A partly balanced slide valve is shown in Fig. 18, in which the partial balancing is effected by fitting a cup-

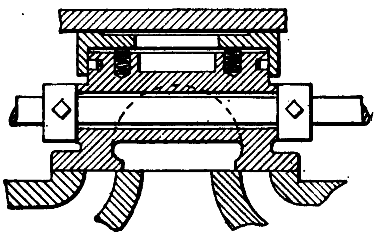


FIG. 18.—Partly balanced slide valve.

shaped ring over a cylindrical extension on the back of the valve. Several small spiral springs of German silver wire serve the purpose to hold the ring out against the face of the steam-chest cover to compensate for any wear which may take place. A packing ring keeps the steam from blowing through. There are several little things which do not work out exactly as contemplated, since there is no motion to the balancing ring except in sliding over the steam-chest cover, some gummy oil or a little dirt is very liable to get between the ring and the part of the valve it fits on, causing it to stick, rust may also play its part. This kind of valve, therefore, should be often examined and cleaned. One of the conditions of tightness is that the face of the steam-chest cover is perfectly parallel with the valve seat, which requires some

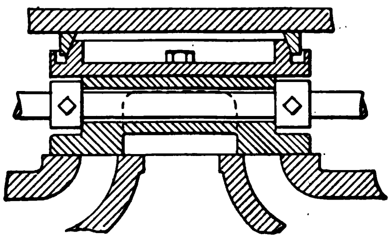


FIG. 19.—Conical balancing ring for slide valve.

very careful machining and fitting, more perhaps than the ordinary machine shop will give it.

An improvement in the balancing ring is shown in Fig. 19. In this design no cylindrical extension on the back of the valve is required, there are simply two flat surfaces provided to which the balance ring is bolted. The latter has a conical extension on the outer side over which a ring is fitted; the ring is split to allow it to expand when put in place. A small keeper prevents steam from leaking into the inner space. When the steam-chest cover is put on a slight pressure is exerted on the ring which has the tendency to force it against the finished surface of the cover and compensates for any wear. The ring will also adjust itself automatically if the surfaces of valve seat and cover are not exactly parallel.

The angularity of the connecting and eccentric rods makes valve motions irregular; slide valves made in one piece are more influenced by this than other valves which are divided into two or more parts, each of which can be adjusted independently of the others. On a common slide valve the four cutting edges are rigidly connected, a change of position of one changes also the positions

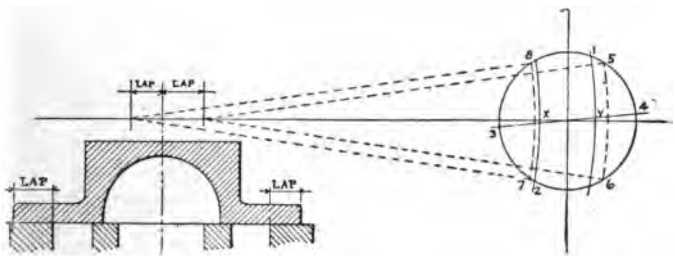


FIG. 20.—Equalization of cutoff.

of the other three, which is important in adjusting the valve for equal cut-off, etc. If the eccentric rod is long its angularity is often disregarded, but that of the connecting rod must be taken into account; its effect is to carry the piston ahead of its proper positions on the forward stroke and to make it lag behind on the return stroke. This is demonstrated in Fig. 20, in which the circle represents the path of the eccentric and may also be taken as the crank circle on a reduced scale. If the cut-off is to take place at $\frac{7}{10}$ of the stroke at both ends, represented by points X and Y in the diagram, with a ratio of the connecting rod to the stroke of 3 : 1, the relative positions of the crank pin will be at 1 and 2 for the forward and return strokes, at which the valve must close. The arc through which the valve opens on the forward stroke is 3–1, and on the return stroke 4–3: 3–4 representing the lead positions of the crank pin. The arc 5–6, therefore, represents the opening of the valve on the out stroke, and 7–8 on the back stroke. Transferring these positions to the motion of the valve we find that the lap on one end must be greater than on the other in order to give equal cut-off.

When the automatic engine made its appearance some

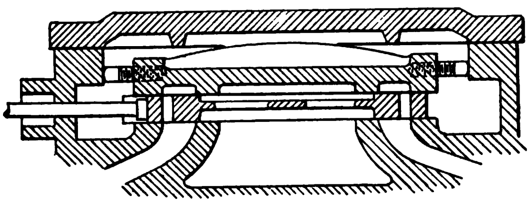


FIG. 21.—Prof. Sweet's balanced valve.

thirty or forty years ago many engineers tried their hands at designing balanced valves, only a few of these designs have survived, the greater number of them have been abandoned on account of their tremendous waste of steam. This latter fact has given the automatic engine a bad standing amongst power users. There seems to be only one really perfect balanced slide valve for automatic engines in existence, which was originated by Prof. Sweet. The main features of this valve are a flat rectangular plate with ports cut in and a self-adjusting pressure plate which fits over the valve. Fig. 21 shows a modification of this valve which has been adopted by a number of engine builders. The only troublesome feature about the valve seems to be that it requires the utmost care in fitting. All of the surfaces of the valve, seat and pressure plate have to be perfectly true, the valve itself must be of uniform thickness. This perfection which is necessary to make the operation of the valve successful can only be attained by successive planing, grinding and hand scraping. A good feature of the valve is that it will lift off its seat and relieve the cylinder of a moderate quantity of water.

Another form of valve which has found many advo-

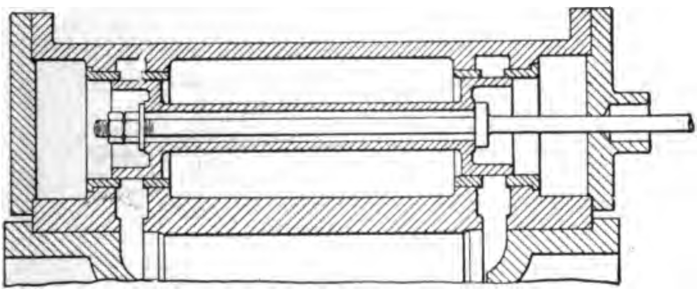


FIG. 22.—Plain piston valve.

cates amongst engine builders and users is the piston valve; it has, however, in the course of time gained a bad reputation. Fig. 22 shows a piston valve for a high-speed engine that has been used very extensively. Ordinarily the valve is made without packing rings with a very close fit in the seats by careful grinding. Since there are no means provided to take up any wear the valve is bound to wear out of round on horizontal engines. Another bad feature which it has in common with all piston valves is that it cannot lift off its seat. In this respect all piston-valve engines are alike, there is always danger of having a wreck, they cannot relieve themselves of any large quantity of water even if they are provided with relief valves. It seems to be impossible to construct piston valves which will give ample relief in case of water in the cylinder like a slide valve; designers of engines acknowledge the fact, and no attempts are made to improve conditions in this respect. Cylinder relief valves are used instead, and means must be provided to prevent water from entering in order to make the engine safe.

To make a piston valve steam tight, however, has been

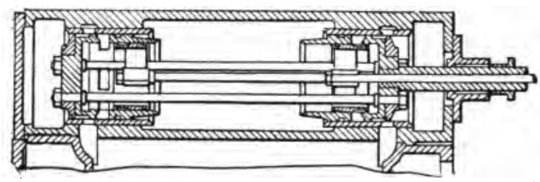


FIG. 23.—Piston valve with packing rings.

tried in a number of designs, and as must be admitted with partial success. A very complicated arrangement to attain this result is shown in Fig. 23; it is used in a large number of engines and consists of a main valve and a cut-off valve, the latter working inside of the other. Both of them are provided with packing rings to make them steam tight. The main valve works in bushings which can be renewed in case of wear. There are so many rods, nuts, rings, etc., used in connection with this valve, that it will be advisable to call in an expert, if anything goes wrong with the valve; very few engineers are competent to make any adjustments or repairs on the same.

The combined weight of the valve and all its parts is considerable, the work to move it at the speed which the engine runs, on which it is used including friction is quite an item in the coal bill.

A valve of similar kind but without the multiplicity of parts is shown in Fig. 24, the claim for originality of the same rests in the way the valve seat is made adjustable to prevent leakage. The valves are simple cylindrical castings with ports in their circumference, they work in sleeves which are split and can be contracted by means of adjusting screws. With a little care it is a simple matter

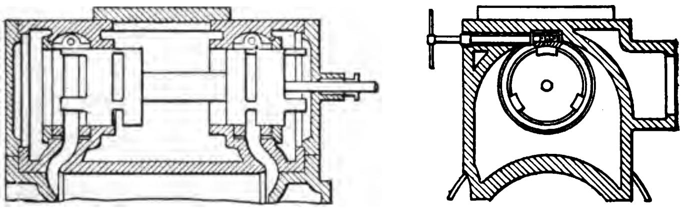


FIG. 24.—Piston valve with adjustable seat.

to make any adjustment. However, some caution must be used, if the ring is contracted even a trifle too much it will pinch the valve and cause either excessive wear or a break of some of the valve gear parts. Since the adjusting screw is accessible from the outside the construction is not fool-proof and cannot be recommended on that account.

A piston valve which probably gives the least amount of trouble of any piston valve ever designed is shown in Fig. 25. It consists of an outer ring and a taper plug fitted inside of it with two end plates for holding them together. The ring is split and can be expanded by forcing in the plug, which increases the diameter of the valve and permits the taking up of any wear of the valve and its seat. The adjustment can only be done when the engine is shut down by taking off the bonnets and slacking the screws which are tapped into the center cone, the nuts on the valve stem are next tightened and the valve tried by hand. In this way the adjustment can be made to a nicety. The outer ring is provided with keepers to prevent steam from blowing through. The ends of the ring

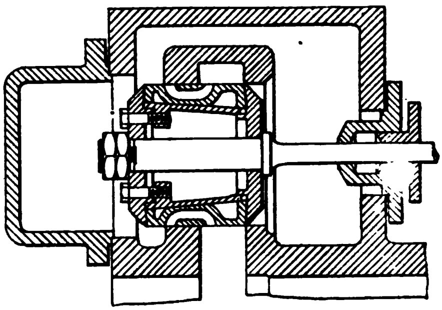


FIG. 25.—Piston valve with adjustable ring.

have conical surfaces over which the end plates are fitted, this prevents the ring from being expanded by steam that might leak through into the inside. Steam and exhaust valves are separate, whereby the condensation due to different temperatures in the valves is reduced to a minimum. This is a great advantage over other designs in which steam is admitted and exhausted through the same valve.

The valves which have found the greatest favor amongst engineers and power users were first introduced by George H. Corliss, who conceived the idea of distributing the steam in the cylinder through rotary plugs placed at the bottom and top of each end of the cylinder, the plugs or valves for admitting the steam were located above and those for the exhaust below the bore. In this way he reduced initial condensation and provided the most effective drainage of the cylinder which has ever been devised. His efforts were crowned with singular success, and the engines he designed became famous for their great economy in the use of steam, efficiency and

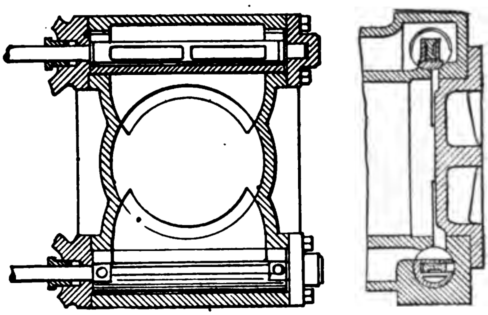


FIG. 26.—Early design of Corliss valves.

reliability. The location of the valves on the early designs of Corliss engine cylinders is shown in Fig. 26; no expense was saved in their construction, the valve stems were made of bronze running the full length of the ports with a bearing in the back bonnet. The valves were carefully fitted over the stems, and spiral springs as a rule were used to hold them on their seats. This precaution was much needed on the exhaust valves which seated upwards, the steam valves would lift in case of water or too much compression in the cylinder. The trouble with the bronze valve stems is that they wear rapidly in the stuffing boxes; to have them replaced by new ones is quite an expense. For that reason Corliss engines of recent design are provided with tee head valve stems made of steel.

The valves of old Corliss engines and also of the cheaper grades of new engines are single ported. On account of their long travel they are subject to considerable wear and are liable to become leaky. They will sometimes wear shoulders, which are hard to remove, at least on the seats, with ordinary tools. To avoid trouble and loss of steam on that account it would be a good

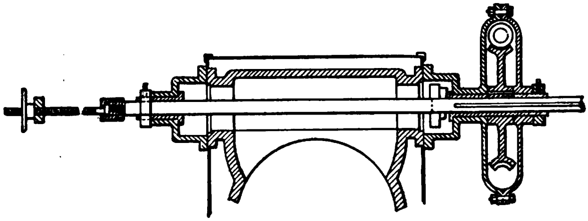


FIG. 27.—Port boring rig for Corliss valves.

thing, if a port boring rig was provided, which could be used to take a slight cut through the port hole if the seat shows any wear. A simple device of this kind is shown in Fig. 27; it consists of two brackets, which are bolted to the porthole flanges, and a boring bar with one or two cutters. One of the brackets has a gear case attached. The boring bar can be made of a piece of cold-rolled steel or shafting with a feed screw connected at one end. The drive can be made of a worm and wheel or a combination of spur gears. Many an engineer who is handy with tools could fit up the rig himself without much expense and be prepared for an emergency. For small and medium-size cylinders the boring rig can be driven by hand, for larger ones a motor can be used.

Reboring the porthole necessitates also a returning and refitting of the valve; for that purpose it must be set out of center in a lathe and the side which bears on the seat must be turned to fit the new diameter of the porthole. If there are no facilities for turning the valve it can be fitted by filing it to a template.

The steam and exhaust valves of a Corliss engine require much attention. In buying and installing an engine it is important to be sure that the valves and portholes are perfectly round and parallel. If they are not, it will take a long time to get the valves to seat properly, with that smooth oil-soaked surface which is desired to make them run easily and with the least amount of friction. Some manufacturers grind the valves on special machines with emery or carborundum wheels, but this is a rather dangerous thing to do. Unless the valves are of very close-grained iron and are carefully cleaned after the grinding process, some particles of emery may remain in the pores of the cast iron and quickly grind out the valve seat.

Emery, carborundum and all sharp-cutting materials are bad things to use on the moving parts of an engine. A good file or a scraper is best. The most satisfactory way to make the valves tight is to bore the portholes round and straight and then draw file the valves to a good fit. Valves fitted in that way, while perhaps not tight at first, will soon come to a lasting and perfect bearing. After a few days' run the new valves should be taken out and examined and the high places removed with a smooth file or a scraper.

Single-ported Corliss valves are seldom used on new engines, as they require a long travel and give slow steam admission and release, which is accompanied by wire drawing and loss of economy. Double-ported valves and cylinders are essential for modern Corliss engines. The steam and exhaust passages into the cylinder should be as short, and the flow of the steam and exhaust as direct, as possible without complicated and tortuous bends. Fig. 28 shows a good construction of steam valve. As indicated by the arrows, the change of direction is easy and

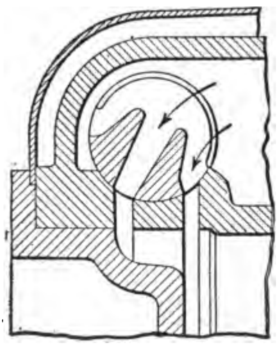


FIG. 28.

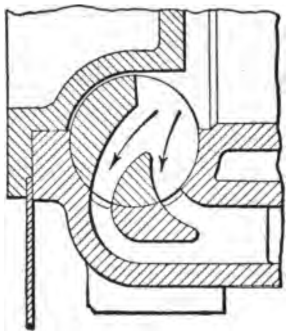


FIG. 29.

Good construction of steam and exhaust valves.

gradual. The exhaust valve shown in Fig. 29 has the same characteristic features; besides, it is located high up in the counterbore of the cylinder, which is an important factor in steam consumption, since the clearance is reduced considerably.

The arrangements of steam and exhaust valves shown in Figs. 30 and 31 are undesirable, though they are found in many engines. In Fig. 30 the steam has to pass over the top of the valve, and it enters the cylinder after changing its direction twice, which results in a loss of pressure. Fig. 31 shows the exhaust valve located below

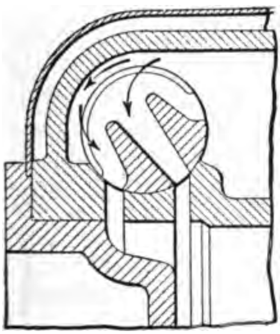


FIG. 30.

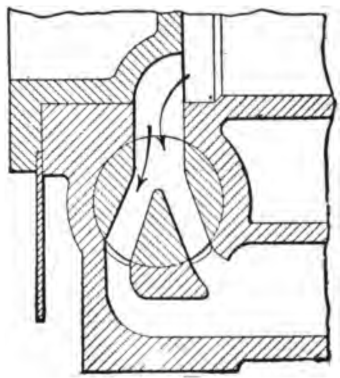


FIG. 31.

Undesirable construction of valves.

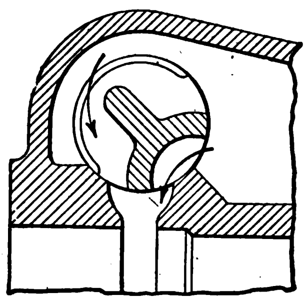


FIG. 32.—Double-ported steam valve.

the counterbore of the cylinder, with long passages and increased clearance. No high economy can be expected from designs like these.

Another undesirable steam valve is shown in Fig. 32. It is a so-called double-ported valve with a single steam port in the cylinder. This kind has a large clearance and the steam pressure acting on the valves when closed is much increased, consequently they open hard and the wear is excessive. Besides, the valve gear parts are likely to give trouble on account of the heavy strain put on them in opening the valve. Trouble may also be expected from the valve in wearing a shoulder on the seat, which, after

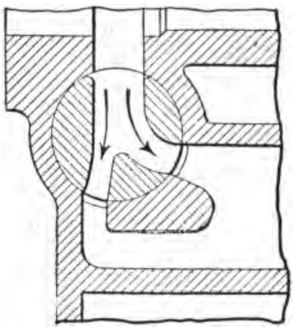


FIG. 33.

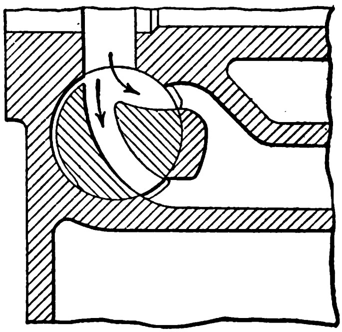


FIG. 34.

Exhaust valves which will wear leaky.

a while, will need reboring of ports and, probably new valves.

Some exhaust valves are designed, as shown in Figs. 33 and 34. Neither one of them can be recommended, since they will eventually leak and give trouble, especially the one shown in Fig. 34. The port above the horizontal center line cannot be made tight, unless the valve is forced upwards by means of springs, which are very objectionable. The seat of a Corliss valve should always be downward and not at an angle to the vertical line. Gibs and springs to hold the valves on to their seats must be avoided.

Steam valves should be made so they will lift off their seats and give relief in case the compression should run too high through a disarrangement of the valve gear or slipping of the eccentric, or if a considerable amount of water should get into the cylinder. What engineers call "a dose of water" is a dangerous thing and must not only be guarded against, but taken care of in the most approved way. The number of cracked cylinders and blown-out heads, caused by neglect to provide relief valves, is legion. Therefore a relief valve of ample size should be located at each end of the cylinder in as low a position as possible. The relief valves and the lifting steam valves are the only "cure" that we know of to counteract the destructive effect that will result from water in any considerable quantity being left in the cylinder after the exhaust valves are closed.

To prevent water from reaching the cylinder at all, it is of the greatest importance that the steam piping should be well covered, either with asbestos or 85 per cent magnesia, in order to reduce the condensation to a minimum, and also to have it well drained. There should be no possibility of water accumulating gradually in pockets and then discharging into the cylinder all at once. Even steam separators may become dangerous and a source of trouble, if the water is allowed to collect in them. Water cannot be compressed to any appreciable extent, and an engine is not designed like a pump. The piston speed is much too high and the area of exhaust valves is not large enough to discharge a great quantity of water at anything like the usual piston speeds. The velocity in well-designed pumps rarely exceeds 5 ft. per second, or 300 ft. per minute, which is about one-half of the ordinary piston speed of steam engines. The valve area of pumps is equal to or even larger than the piston area, while the exhaust ports of a steam cylinder are only about 15 per cent of the piston area, so it can be easily understood why an engine is wrecked if any considerable amount of water is to be discharged from the cylinder, even with the exhaust valves wide open. The relief valves can take care of only a small quantity of water at the end of the stroke, when the exhaust valve is closed and the piston travel reduced, and so can the steam valves by lifting off their seat.

The original idea of Corliss to separate the action of the valves for steam and exhaust and make them perfectly independent of each other appealed to many engine designers, who tried their hands at making improvements. Most of them have given up the unequal struggle, but a number of the engines they built are still in existence. A design which attracted a great deal of attention and found many advocates is shown in Figs. 35 and 36, illustrating the Porter-Allen valves. Instead of placing them above and below the bore of the cylinder these valves are located on the sides, the steam valves on one side and the exhaust valves on the other in vertical position. Each valve is provided with a pressure plate holding it on its seat; the pressure plates for the steam valves are adjustable, while those for the exhaust valves are bolted rigidly to the cylinder. It can easily be seen that a removal of any of the valves is a very troublesome and tedious affair, a lot of nuts have to be unscrewed and large joints to be broken, the large gaskets for the joints are easily damaged, and if so, expensive to replace. It stands to reason that an engineer will not be very anxious to take off the covers and examine the valves, especially on the larger sizes. The steam valves will lift slightly off

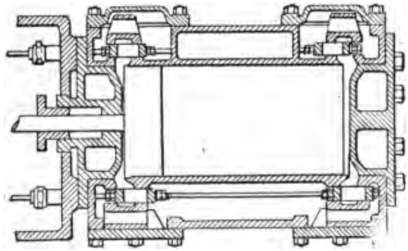


FIG. 35.—Horizontal section through Porter-Allen valve.

their seats, not sufficient, however, to relieve the cylinder of a large quantity of water. Engines with this type of valves have been used to a considerable extent in rolling

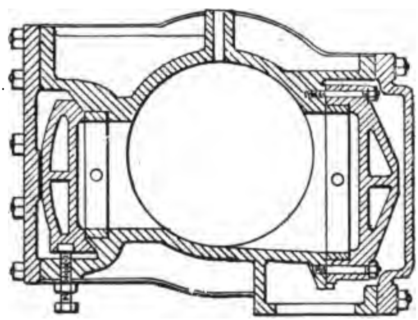


FIG. 36.—Vertical section through Porter-Allen valve.

mills for large powers and very fluctuating loads, the valves are naturally of large proportions and great weight; unequal expansion between valves and seats is often very troublesome, especially in starting up, which cannot safely be done until the engine is thoroughly warmed up.

In order to reduce the travel of valves, multiported valves have been designed, the most noted of which are the so-called gridiron valves with flat surfaces. Instead of having one or two cutting edges, they have a number of them, sometimes as high as a dozen or more. These valves require thorough lubrication and careful handling; they are easily broken since they are necessarily of frail construction and subject to shrinkage strains. Fig. 37

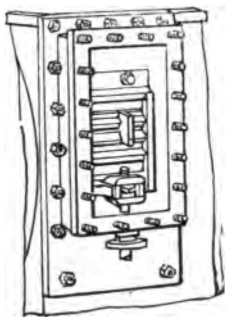


FIG. 37.—Vertical gridiron steam valve.

shows a gridiron steam valve placed on the side of the cylinder; the motion is vertical, therefore, the full weight of the valve, including friction, has to be overcome by the valve gear every time the valve is opened. When released the pressure on the valve stem closes the valve, assisted by its own weight. The steam chests, there are two of them, one for each valve, are separate castings, bolted to the cylinder. This is done for two reasons, first to make the seats of special hard iron and second, which is more important, to make them removable for refacing in case they become cut or damaged. Valves placed vertically on a steam cylinder are hard to lubricate, the lubricant will not stay on the surfaces but runs off.

A more elaborate form of gridiron valve construction is shown in Fig. 38. The steam and exhaust valves are placed in horizontal position, similar to Corliss engine practice, above and below at the ends of the cylinder.

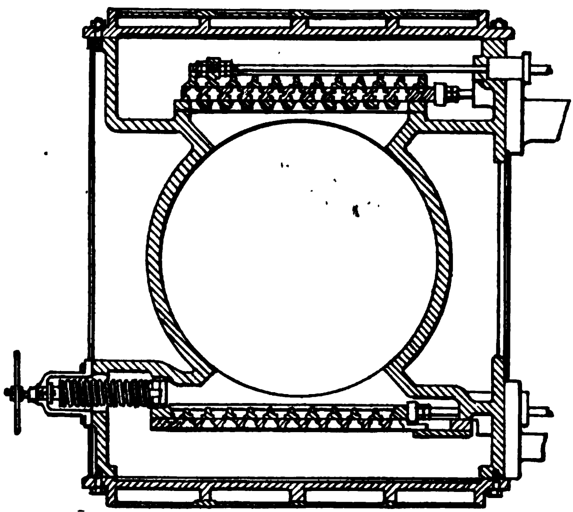


FIG. 38.—Horizontal gridiron valve construction.

The valves and seats are flat castings with a lot of slots cast in; they can all be refaced or replaced if necessary. The whole design is well thought out and executed, but, of course, if the least bit of hard foreign matter gets in between the valves and seats they are easily ruined beyond repair, making it necessary to have spare parts on hand. It is evident that with so many flat surfaces and cutting edges working together, the most excellent workmanship and the most careful treatment is required. To complicate matters on this design, there is an extra cut-off valve for the steam, which slides on the top of the regular steam valve. The valves are not balanced but are exposed to the full steam pressure, the steam valves always and the exhaust valves at intervals, with their large seating

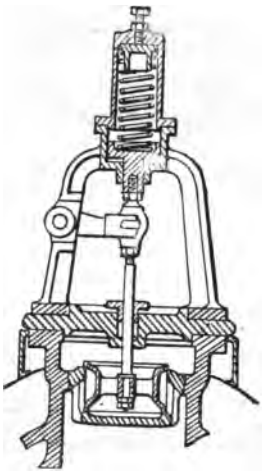


FIG. 39.—Double beat poppet valves.

areas they require, therefore, extremely abundant lubrication, otherwise they will pull hard and even stick.

While Corliss valves are identified with American steam-engine practice and were brought to their highest perfection in the United States, poppet valves have been developed in Europe, mostly Germany and Switzerland. It is only in recent years that valves of the latter design have been taken up by American engine builders, and the more successful of them are strictly of European origin.

High pressures and superheated steam puts valves with sliding surfaces, such as slide valves, piston, Corliss and gridiron valves out of business, for all of which a steam pressure of 150 lbs. per square inch and 50° of superheat is about the limit. The trouble with the higher pressures in connection with sliding surfaces is that the latter work very hard and require an excessive amount of

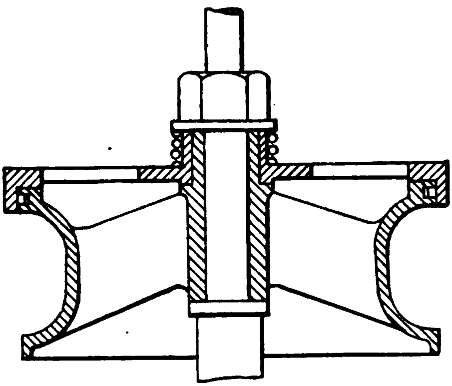


FIG. 40.—Poppet valve in two parts.

lubrication. Balanced valves are hard to keep tight under high-pressure steam. Superheat makes lubrication more difficult, besides it has a tendency to distort the castings on account of unequal expansion.

Poppet valves are usually of the double beat type, as shown in Fig. 39, for two reasons, first to reduce the lift and second to make them as nearly balanced as possible. A disadvantage of these valves is that the clearance is very large if they are located on the cylinder similarly to

Corliss valves. Some engine designers place them in the heads, this reduces the clearance, but makes trouble about getting at the inside of the cylinder. Another disadvantage connected with them is that the direction of the flow of the steam is changed several times, which results in a loss in pressure. They are not the ideal valves by any means, but for high pressure superheated steam we have nothing as yet which does the work better; they are subject to a great many evils and require close watching.

The principal troubles seem to be leakage and hammering.

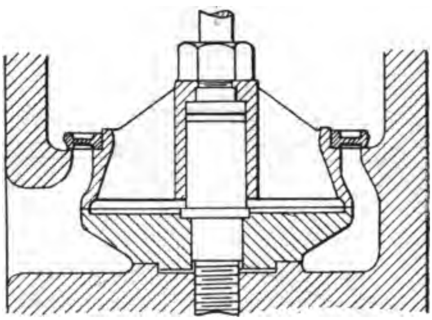


FIG. 41.—Poppet valve with flexible ring.

A change in the temperature of the steam will affect the valve sooner than the seat, it will expand more and commence to leak. To prevent this the steam valve is sometimes made in two parts, as shown in Fig. 40, in which an upper ring is fitted over the valve body and made to yield slightly in seating. Another way that gives good results is shown in Fig. 41, it consists in shrinking on a flexible steel ring which forms the upper seat; the ring will deflect under pressure and allow both lips of the valve to seat tight.

Hammering of poppet valves is generally prevented by cushioning on slow speed engines and by suitable cam

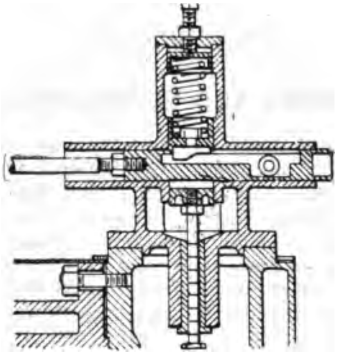


FIG. 42.—Cam motion for poppet valve.

motions on higher speeds. A very effective motion of the latter kind is shown in Fig. 42, which makes the valve operate noiselessly at 250 revolutions and over.

CHAPTER III

PIPING AND SEPARATORS

Principle of natural drainage for piping. Supporting of pipe lines. Blowing out of pipe lines. Combination of separator and throttle. Ordinary steampiping is not safe for superheat. Pipe joints. Welded joint. Lap joint. Screwed and shrunk joints. Pipe-joint clamp for stopping leaks. Steam separators. Stratton separator. Separators with baffle plates and corrugations. Direct flow. Prof. Sweet's and vertical separators. Receiver separator for high pressure and superheat.

Many accidents happen to the steam piping and throttle valves, which are caused by faulty design, construction or workmanship. The steam piping connecting boilers and engine must have ample provision for expansion and contraction; bends and elbows must be made to a long radius. Long and complicated steam pipes give poor economy. Natural drainage of the pipes is important; there should be no chance for water to collect in pockets. The principle of natural drainage is shown in Fig. 43. *A* shows valves and pipes arranged correctly. Water condensed in the pipe will either run back to the boiler or toward the engine. If it collects over the engine, it should be discharged through the drain pipe, which is usually opened to clear the pipe before the engine is started, or it will run into the cylinder and out through the exhaust when the cylinder is warmed up before starting. While the engine is running and taking steam, no water can accumulate in the vertical pipe.

View *B* shows the valve in an undesirable position near the boilers; there is a chance for water to collect

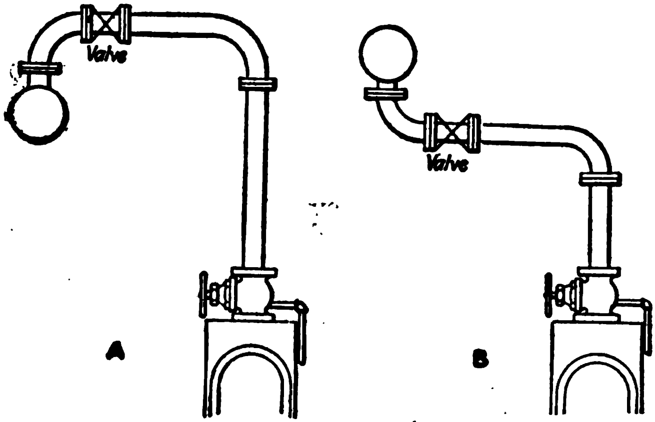


FIG. 43.—" A " shows a correct and " B " an incorrect steam main.

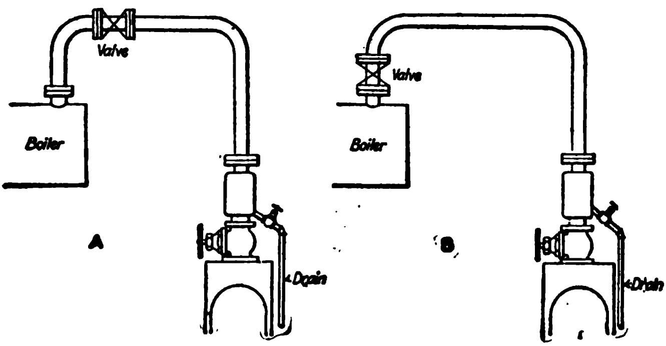


FIG. 44.—No water can collect in the layout " A ", if the drip valve is open; but " B " will collect condensate.

over the valve when closed, and on opening this would be thrown toward the engine, causing water-hammer.

If the engine is connected to a steam main an arrangement similar to *A*, Fig. 44, should be used. In this no water can collect except over the engine, whence it will be drained, as stated before. In *B*, Fig. 44, water will collect in the pocket between the main and the valve and may cause trouble.

To allow for expansion, loops or bends in the pipe should be placed in a horizontal position with a drop toward the engine. If more than one boiler furnishes steam, each should be provided with an automatic non-return stop valve that will close when the pressure in the main exceeds that in the boiler. A butterfly valve or an automatic stop valve should be placed in the steam pipe to be instantly closed from various parts of the engine room, by hand or by an automatic safety stop on the engine, if the normal speed is exceeded.

The steam pipes must be well supported by pipe hangers or brackets to relieve them of all strains due to their weight. For ordinary steam pressures flanged joints should be used with the flanges screwed on. All horizontal pipes should be pitched about 1 in. in 10 ft. for good drainage. If a pipe line has been newly erected or if any changes have been made in the old piping a careful test should be made by gradually increasing the steam pressure; the full pressure must not be turned on at once. A new pipe line must be blown out before it is connected to the engine or the separator. It is sometimes a revelation to note the things that have been deposited in the pipes; waste, wrenches, files, tobacco, bolts, gaskets are only a few of them. If any of such rubbish is allowed to get into the engine it will cause serious damage, perhaps a wreck. A separator located over the engine will act as a safeguard and prevent any foreign matter to reach the cylinder, but it must be cleaned out before the engine is put into regular service.

The combination of steam separator and angle throttle valve, shown in Fig. 45, is not to be recommended. In this arrangement there is a bending strain in the neck of the valve owing to the weight of the separator and steam piping; water hammer may also give trouble, and breaks in the neck are not infrequent. The separator and steam

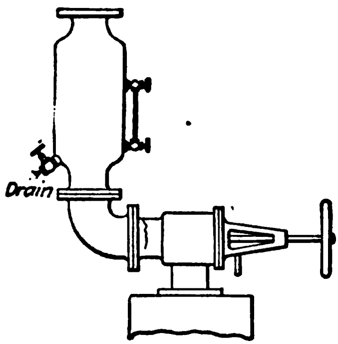


FIG. 45.

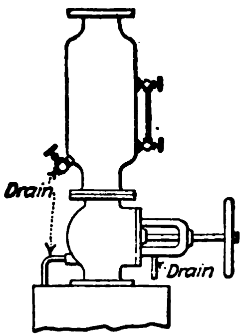


FIG. 46.

The right and wrong way to connect separator and throttle.

pipe should be placed directly over the globe valve, as in Fig. 46. Drains should be provided as shown.

The tendency in recent years is to raise the steam pressure and superheat the steam, if a change is made in a plant in regard to either one of these conditions alterations in the piping and stop valves are also in order. No ordinary steam piping put up years ago will be safe for superheat; cast-steel fittings must take the place of the former cast-iron ones and the old gaskets of paper, asbestos, rubber of other soft material should be replaced with some made of copper. Cast-iron pipes will not do for high pressure or superheat, new pipes of wrought iron or steel must be substituted with cast-steel or forged-steel flanges put on in the most approved manner.

The best connection between pipes and flanges is a weld which makes the joint perfectly solid as shown in Fig. 47. The process can only be carried out by the use of special machinery, it makes the joint perfectly homogeneous and eliminates the possibility of a leak between pipe and flange. The next best joint is made by lapping the

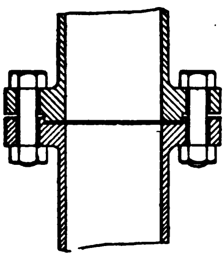


FIG. 47.—Welded flange joint.

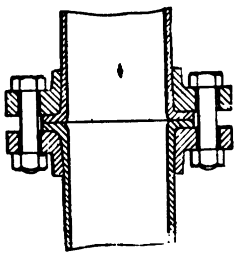


FIG. 48.—Lap joint for pipes.

ends of the pipes over, as shown in Fig. 48. The ends must be perfectly true, they are drawn together by means of wrought-steel flanges which conform exactly with the inner surface of the lap. The flanges must have a good fit on the pipe and lap and be made loose enough to swivel if a change in the position of the bolt holes should be desirable. This feature is a great convenience in erecting not shared by the welded joint.

The regular practice heretofore has been either to screw or shrink flanges on to a pipe, as shown in Figs. 49 and 50. The objection to the former is that the threads weaken the pipe and to the other that a shrink fit depends entirely on the care with which the job is done. For shrinking on a flange an allowance must be made for the expansion of the metal in heating and the subsequent contraction; the allowance varies between 2 and 3 thousandths of an inch, according to the size of the pipe, per inch of diameter. The pipe must be made perfectly round where the flange is to go either by turning or pressing, the flange is brought to a red heat in a fire, then placed over the pipe and allowed to cool; the end of the pipe is hammered over into a chamfer on the flange. If

FIG. 49.—Screwed joint.

FIG. 50.—Shrunk joint.

the job is done right the flange will hold, if not it is sure to come loose in time and leak.

A leaky pipe joint is very troublesome, but there are ways to stop a leak if it is not too bad and taken care of in time. The first thing an engineer will do is to try to stop it by calking, which may make it worse after a while: the next thing in order will be a clamp around the pipe near the flange and some packing driven in between. Fig. 51 shows a pipe joint clamp manufactured for this purpose which may overcome the trouble, it consists of a packing ring and two iron split rings; the packing ring is placed next to the leaky joint, the smaller iron ring follows and the large ring is slipped over the other two, which are forced against the joint by means of the set screws. The device is extremely simple and easily applied.

Every engine, large or small, should be provided with a steam separator in the pipe line near the engine, to catch water and dirt carried along with the steam. The

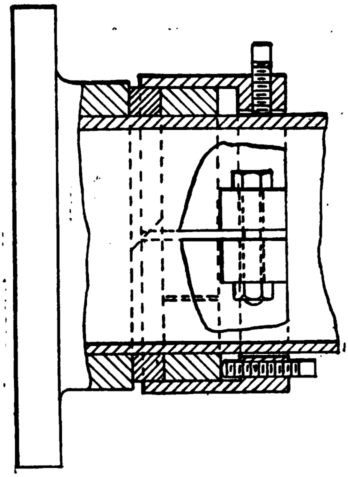


FIG. 51.—Pipe joint clamp.

best place for a separator seems to be right over the throttle valve on top of the cylinder, but it may also be located in the horizontal pipe leading to the engine. The main consideration is to make it large enough; a separator of insufficient capacity is worse than useless, it may even be dangerous if water collected in it is forced over into the engine by the rush of the steam. There are many steam separators on the market, most of which act on the principle of having any entrained water separated from the steam by a change of direction and velocity. One of the principal requirements is that all parts are accessible without disconnecting the steam pipe. In this respect the Stratton Separator, shown in Fig. 52, seems to fill the bill. It can only be applied to a hori-

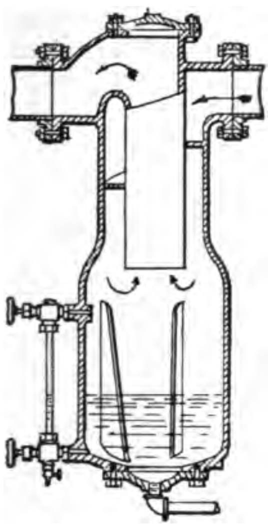


FIG. 52.—Stratton separator.

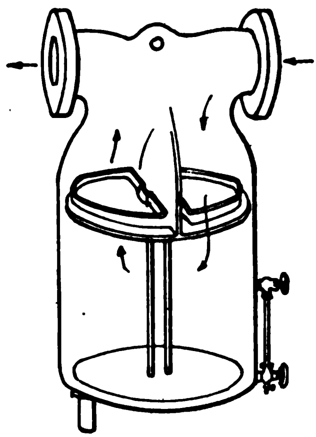


FIG. 53.—Separator with short collecting chamber.

zontal pipe; there are covers on the top and bottom, so that the whole inside can be inspected and cleaned. The steam is made to whirl around the inner pipe in a downward direction, on which course the water is supposed to be liberated by centrifugal force and collected in the bottom of the enlarged part. Dry steam flows upward through the inner pipe to the engine.

Another design of separator to be placed in a horizontal pipe is shown in Fig. 53. The steam plunges downward on one side into the enlarged lower part and rises on the other side, water collects at the bottom. Troughs are provided in the upper part of the large chamber to intercept any water which is caught on the sides. This separator is made purposely short, the object of which is to make the moisture in the steam impinge on the surface of the water in the bottom, where it is caught and retained.

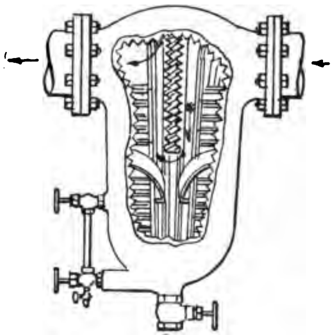


Fig. 54.—Separator with baffle and sides corrugated all over.

There are no loose parts on the apparatus and the facilities of examining the inside and cleaning are limited.

In some separators the direction of the steam is changed by baffle plates as shown in Fig. 54, which shows the arrangement for a horizontal pipe. The surfaces touched by the steam are corrugated acting on the principle of a washboard. The entering steam strikes the corrugated surfaces which retain the moisture and lead the water toward the bottom, partly separated by two plates from the main chamber. This separator seems to be very effective, the corrugations evidently add to its efficiency. The inside is not accessible except through a small hole at the bottom.

The baffle-plate feature is used in many other designs, one of which is illustrated in Fig. 55. It is made in two

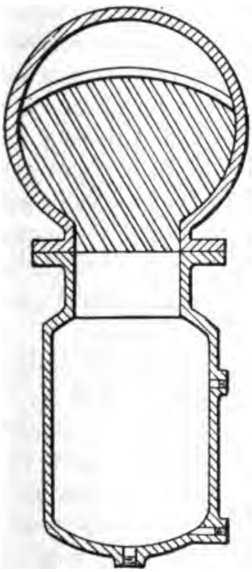


FIG. 55.—Separator in which only the baffle plate is corrugated.

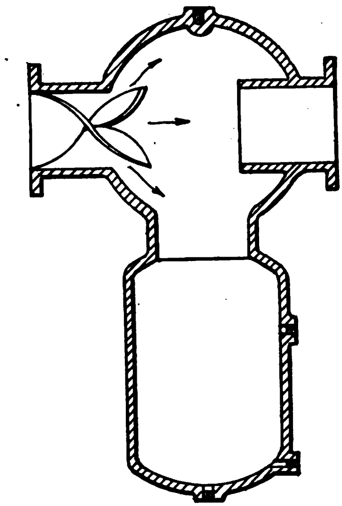


FIG. 56.—Direct flow separator.

parts, the lower part in which the water collects is a separate casting, bolted to the baffle-plate chamber; this construction makes the inside accessible without disconnecting the entire separator from the steam pipe. The corrugations in the baffle plate run at an angle to the rising current of steam which is supposed to prevent the steam from carrying any of the moisture caught by the plate over the top of the same. No other part of the separator is corrugated.

While some designers make a great point of the value of the baffle plate and corrugation, others claim that the baffle forms an obstruction to the free flow of the steam and causes a loss in pressure, and that corrugations are liable to retain the moisture long enough to be picked up again by the steam. Fig. 56 illustrates a separator in which neither baffle nor corrugations are used, nor is the direction of the flow of the steam changed to a great extent. The steam enters a large chamber through a screw-shaped nozzle creating a rotary motion which throws the entrained water against the walls of the chamber by centrifugal force. Dry steam flows straight into the nozzle on the opposite side, which projects partly into the inside of the chamber.

A separator that has found great favor with designers and users of high-speed engines is shown in Fig. 57. It was originated by Prof. Sweet, who, as the designer of one of the most efficient high-speed engines, knew probably more about the necessity of using dry steam in that type of engine than any other engineer. In some respects it resembles the Stratton Separator, but there is no centrifugal action. The steam enters on one side, takes a downward course until it reaches the mouth of the smaller inside pipe, whence it flows upward and out on the opposite side. The distinguishing feature of the separator consists in the perforated cylindrical plate which is placed close to the wall of the large chamber. Each perforation has a lip pointing downward, on which the moisture of the steam is deposited, dripping into a conical perforated plate and down into a reservoir. The moisture which collects on the inner vertical pipe is also led down into the water receptacle. The separator can be taken apart without disturbing the connection with the steam pipe. The large vertical chamber is protected from radiation by a steel jacket and insulating material.

A vertical separator to be placed directly above the throttle valve and, therefore, in the most effective position is shown in Fig. 58. The entering steam strikes a conical

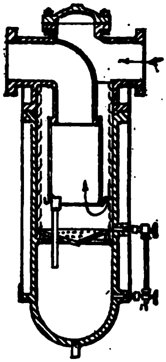


FIG. 57.—Prof. Sweet's separator.

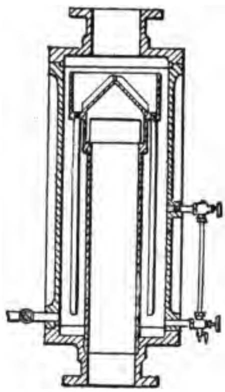


FIG. 58.—Vertical separator.

baffle plate provided with ribs on which part of the moisture is deposited to be discharged through small pipes into the well at the bottom. It is forced upward again by a vertical extension on the rim of the baffle and flows down into the large chamber and finally through side openings into the inner pipe and to the engine. Changing the direction of the flow of the steam repeatedly has the tendency to separate the entrained water very effectively. The outside of the separator body is covered with non-conducting material and a steel jacket. The inside of the separator is not accessible except by disconnecting the steam pipe and taking off the top cover.

For high pressure superheated steam and severe service neither one of the afore illustrated and described designs of separators will answer. They lack strength and

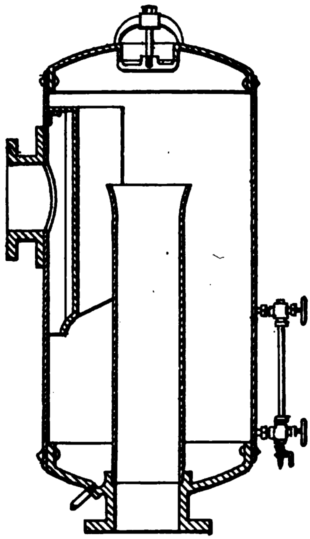


FIG. 59.—Receiver separator.

capacity, unless they are made of cast steel and unusually large. Fig. 59 illustrates a receiver separator which is designed for severe conditions. It is built like a boiler of sheet steel with cast-steel bottom and nozzle riveted to the shell and its inside is accessible through a regular manhole on the top. Steam enters on the side striking a baffle plate which deflects it on both sides, there is an opening in the baffle for the water to run out and collect in the bottom of the receiver. The steam freed from water enters a vertical pipe which reaches well up into the receiver. On account of the large capacity of this separator which should be at least twice the volume of the piston displacement, there is very little danger from flooding the engine with water; it also acts as an equalizer of the steam supply for greatly varying loads and is especially adapted for electric power stations.

The purpose of the gauge glass on a separator is generally taken to be to show how much water is collected in the same, so as to give the engineer some idea, when to blow it out. It should rather be in its place, though, to show that no water is collecting but that every particle of it is immediately drained off into an efficient trap. Of course there must be a valve in the drain pipe near the separator for emergencies, but this valve should be open for free egress of the moisture. The importance of keeping the water from collecting in the separator cannot be emphasized enough, although some manufacturers claim their apparatus should have water in the bottom against which the current of the steam and particles of moisture in the same impinge.

CHAPTER IV

THROTTLE AND EMERGENCY VALVES

Two ways of closing a valve. Throttle with pilot valve. Angle valve. Valve seat with taper thread. Quick-closing valves. Prof. Sweet's throttle. Large valve construction for rolling mills. Balanced throttle valves are not reliable. Emergency valve. Butterfly valve. Throttle closed by means of a sprocket chain. Throttle closed by steam pressure. Emergency valve placed over the throttle. Arrangement for closing the throttle automatically and from certain places in the engine room. Speed limit device attached to the crosshead.

The throttle valve of an engine should be of very rugged construction and quick closing. Slow-closing gate valves can, therefore, not be used, although they would offer the least resistance to the steam on account of the passage being perfectly straight without any bends as in the customary globe or angle valves.

There are two ways of making the valves close, against the steam pressure or with the same, both ways are shown in Fig. 60. Valves closing with the pressure are to be preferred, for the reason that they will stay closed if anything happens to the spindle, screw or yoke, while valves constructed on the other principle have the full steam pressure always acting on these parts when closed, and will open of their own accord if any of the parts mentioned give way. On the other hand valves having the full pressure on top open hard and must, therefore, be provided with a pilot valve or a bye-pass.

Fig. 61 shows a valve with a pilot valve, by opening the

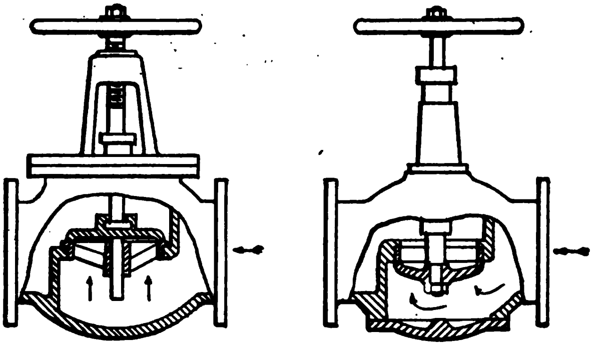


FIG. 60.—Two ways of closing a throttle.

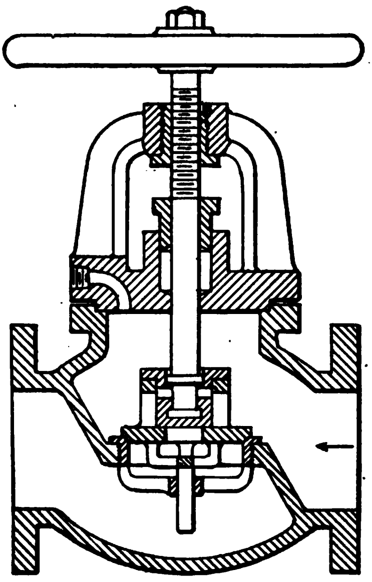


FIG. 61.—Throttle with pilot valve.

latter the pressure can be equalized on both sides of the main valve, taking the strain off the spindle and other parts in opening. The small valve is also used for warming up the cylinder before the engine is started. However, the engine should not be started with the pilot valve, otherwise the pressure on the main valve will not be sufficiently reduced to open it; at least on the larger sizes, since the engine would use considerable more steam than can pass through the small opening of the valve and prevent the pressure from becoming equalized.

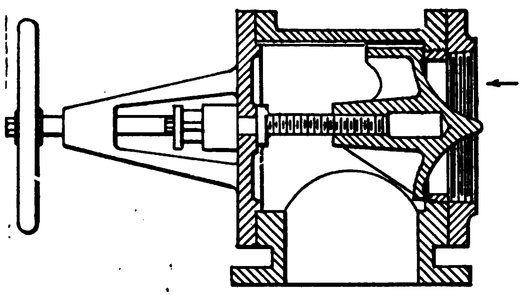


FIG. 62.—Angle throttle valve.

Throttle valves closing against the pressure do not need a pilot valve or bye-pass; with them all that is necessary to warm up the engine is to just break the valve and let through a small amount of steam. In that respect that type of valve is very convenient. Fig. 62 shows an angle valve of this type, which has some good features. The angle construction in this case is not objectionable, since the elbow connecting valve and piping can be made of steel strong enough to eliminate all danger of it cracking. The valve disc is of unusually substantial construction with easy curves at the end which offers a minimum of resistance to the steam; it is provided with a tongue running in a groove on the valve body, which keeps it from turning and makes it seat always in the same place. The screw thread on the spindle is inside of the valve and works in the hub of the valve disc; it is removed from outside influences and cannot be damaged, which is often the case with an outside thread. The valve stem can be repacked with the valve in any position, a collar on the spindle making a tight joint on the bonnet; in fact there is hardly any necessity for a stuffingbox and packing. This is different from other designs which have the troublesome feature that the stem can only be repacked when the valve is either wide open or entirely closed, and on which the packing is always exposed to the full steam pressure making it wear rapidly on account of the combined turning and sliding motion of the spindle. There is one thing, however, on this valve which will not please the engineer, there is nothing to show whether it is open or closed or how much it may be open, since the spindle does not move lengthwise and there is no thread or any other mark which would indicate the position of the valve.

The valve seat in Fig. 62 is screwed into place with a taper thread, it has no shoulders, can easily be removed and without much expense be repaired or renewed. Some valves are provided with a straight thread and screw up against a shoulder; they are liable to come loose in time and give trouble, unless the threads are a perfect fit in the seat or they are secured in some other way.

If anything goes wrong with an engine every second counts and it is of the greatest importance to have some way of shutting off the steam immediately. For this reason quick-closing throttle valves have been designed. Prof. Sweet took the lead in this respect with the quick-closing valve shown in Fig. 63. This throttle, which he has been using on his high-speed engines, is altogether different from the regular run of valves. In the first place the passage through the valve is straight, there are no turns for the steam to take, consequently there is no resistance and no loss in pressure; another advantage is that it is self-draining, which the regular globe-pattern throttle valve is not. But the principal advantage of the

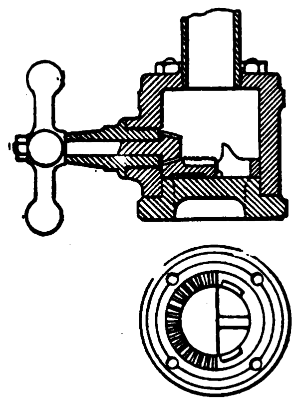


FIG. 63.—Prof. Sweet's throttle.

valve is that it takes only about two turns of the hand wheel to close or open it. The valve seat is always protected by the valve itself, it is never exposed to the action of the steam, neither is it affected by rust; it will, therefore, remain tight almost indefinitely.

A similar valve construction for large rolling-mill engines is shown in Fig. 64; it has three openings instead of one as Prof. Sweet's, and on account of its size is provided with a second valve on top of the main valve, which

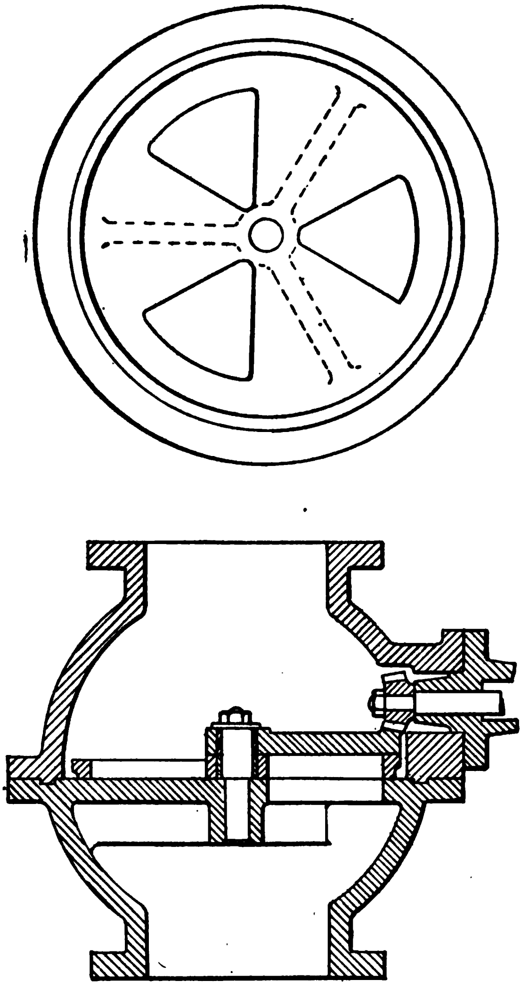


FIG. 64.—Quick-closing throttle for rolling mill engines.

acts as a pilot valve to equalize the pressure on both sides of the opening. This valve is either full opened or closed by one turn of the hand wheel.

Balanced throttle valves of the double beat and also of the piston type have been tried again and again but they have never been a great success, since they cannot be made tight permanently. If either one of these two kinds of valves is used on an engine, as they are sometimes on reversing engines for rolling mills or hoists, there should be another valve in the line of the regular mush-

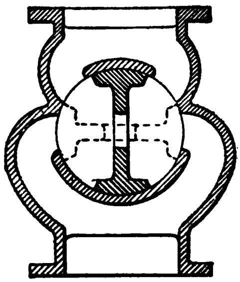


FIG. 65.—Emergency valve.

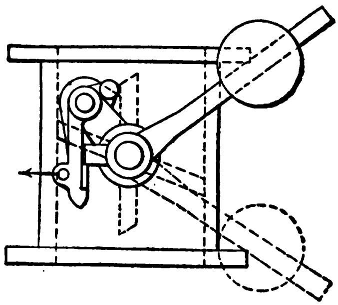


FIG. 66.—Butterfly valve.

room type, which is closed if the engine is to be shut down for any length of time, in order to keep steam from leaking through the main valve and giving the engine a sudden start.

An engineer cannot always be at the throttle or near it, he has to attend to other duties about the engine room. There should, therefore, be some means provided for instant action if trouble comes which makes it necessary to slow down the engine or for an immediate shutting off of the steam. For this purpose emergency valves or overspeed prevention devices have been designed, which either operate automatically or are under the control of the attendant at different places on the premises.

An emergency valve which is usually placed directly over the throttle is shown in Fig. 65, it can be operated either by hand or by a special overspeed device. If a latch at the end of the valve stem is tripped the valve is

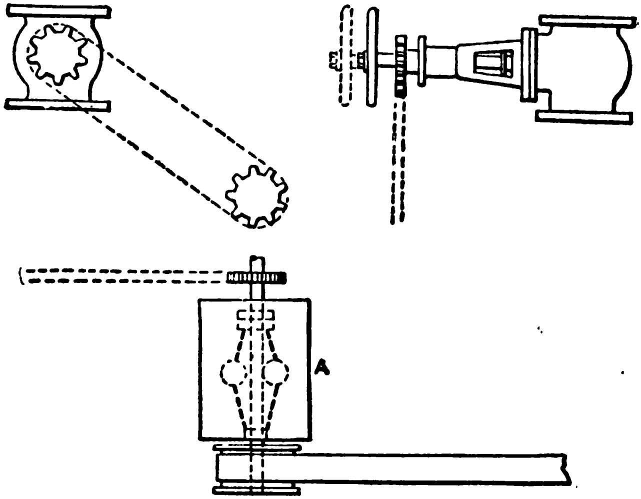


FIG. 67.—Stop valve operated by flyball governor.

moved by a weight to the position shown in dotted lines shutting off the steam.

Instead of the emergency valve shown, a butterfly valve provided with a weight and tripping device may be used, as shown in Fig. 66. Neither of these is steam-tight, but when closed they will stop the engine. Care must be taken not to have the stuffingbox too tight, or the weight may not be sufficient to close the valve.

Fig. 67 shows an arrangement in which a sprocket chain is made to turn a wheel on the valve stem and close the throttle as soon as the speed limit *A*, which is a small governor, throws in a clutch. The power to do this work is derived either from the crankshaft by means of a belt, or from a heavy weight which has to be wound up by hand. This device is liable to get out of order, especially if the valve stem works tight. To overcome the trouble with tight valve stems is the object of some designs. Fig. 68 shows one in which the steam pressure is utilized to

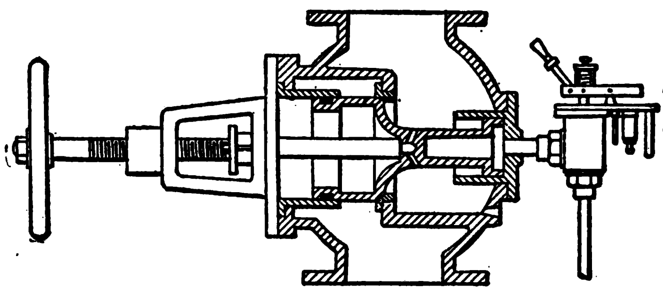


FIG. 68.—Steam operated trip on throttle valve.

close the throttle quickly in case of emergency. The throttle-valve stem is not rigidly connected to the valve. As soon as it is screwed back the steam forces the valve open; it enters also through the small holes and fills the space above the valve. Steam also passes the small piston, which is slightly less in diameter than the bore of the little cylinder that is connected to an auxiliary valve operated by a trip-lever. As soon as this lever trips the auxiliary valve, the steam behind the small piston escapes, throwing the main valve out of balance, closing it. The trip-lever may be operated by an auxiliary governor set for a certain speed, by an electro-magnet or by hand. On an old engine a valve, as shown in Fig. 69, may be placed right over the throttle valve. It has the same tripping device as the one in Fig. 68 and is operated on the same principle. It is opened by a small handwheel which works a pinion and a rack, the latter being cut into the valve stem.

Complications and intricate devices on these valves should be avoided; everything must be handy for the

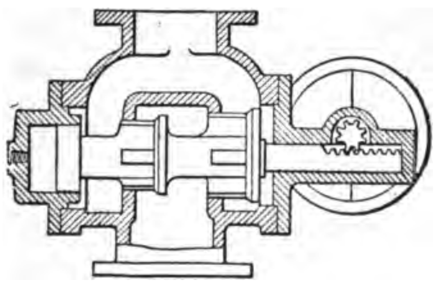


FIG. 69.—Another type of quick-closing valve.

engineer and arranged so that he can quickly shut off the steam in case of emergency. The throttle-valve handwheel of an engine is located in the most dangerous position if the engine runs away. It takes a man of iron nerve to step into the pass of the flywheel to shut off the steam by turning the handwheel of a slow-closing throttle. Means should therefore be provided for shutting off the steam from both sides of the cylinder and from certain places inside and outside of the engine room; and besides, there should be an automatic device that will stop the engine if the speed exceeds five or ten revolutions above the normal. A complete arrangement which will accomplish this is shown in Fig. 70, consisting of a steam, electric and hand-operated system. The methods of opera-

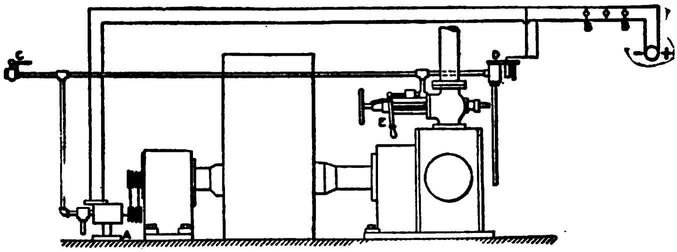


FIG. 70.—Diagrammatic layout of emergency engine stops.

tion are as follows: The speed-limit device *A*, which consists of a small governor inclosed in a cast-iron dustproof case, and driven by chain, ropes or belt from the end of

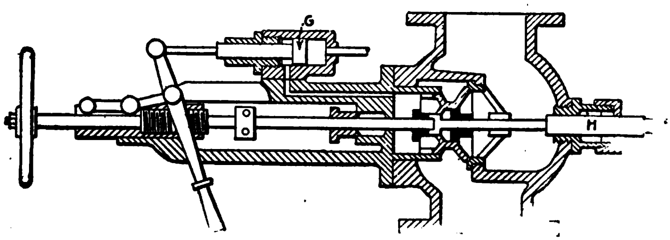


FIG. 71.—Section of steam operated valve.

the mainshaft, opens a small relief valve and makes electrical contact as soon as a certain speed is exceeded. Steam is exhausted from one side of the small piston *G*, in Fig. 71, which shows the throttle valve in detail; this snaps the toggle links and trips the valve, which is closed by steam acting on the plunger *H*. The same result can be obtained by closing any one of the switches *B*, located in convenient places in the engine room, or by opening hand relief valve *C*, also by tripping the valve *D* by hand or throwing over hand lever *E*. The throttle can also be closed by the handwheel, independent of the tripping devices.

With this or a similar system the engineer should

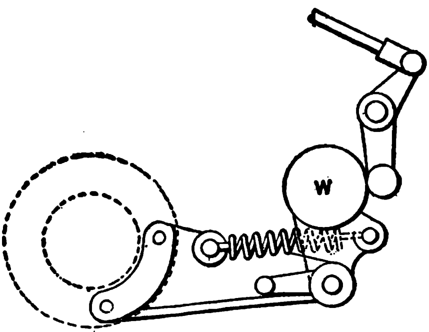


FIG. 72.—Emergency stop connected to crosshead.

make it a practice to test the apparatus when stopping the engine, for if anything is allowed to get out of order the whole thing may become useless. There is such a thing as overdoing it, and unless a man has the time and skill to attend to all these devices, the simpler methods described are to be preferred.

Fig. 72 shows a simple speed-limit device which may be connected to the crosshead. The weight *W* is held in position by a spring. When the engine exceeds a certain speed, the inertia of the weight will make it fly ahead at the end of the stroke and strike a lever connected to the governor, tripping a latch and releasing a spring, which will force the governor to its highest position and prevent steam entering the cylinder. The trip-rod of this device can be connected to a self-closing butterfly valve or some of the other safety devices.

CHAPTER V

PISTONS

Good and bad piston-rod construction. Straight fit with key. Sharp corners are dangerous. Taper fit. Cold rolled steel is unfit for piston rods. Piston-rod construction for rolling mills. Threaded piston rod. Limit of stresses in rods. Method of securing the rod to the piston. Follower bolts must be of suitable material. Different designs of follower and bolts. Junk rings. Packing rings.

It seems that engine accidents run in series. Aside from flywheel accidents, which probably cost more lives and money than all other engine trouble combined, many accidents are due to faulty piston-rod construction. Years ago any old connection between crosshead and piston rod was considered good enough, and the cheapest and most dangerous was the most common. A troublesome connection between the piston rod and crosshead is shown in Fig. 73. The rod was slightly reduced in diameter at the crosshead, with a straight fit in the latter. A hole for the key was cut into the rod, with the corners made square. The trouble came as soon as the slight shoulder had hammered into the crosshead or been reduced on the rod, creating a pound. The key had to be taken up, more pounding and keying followed and, finally, rupture through the sharp corners of the hole, as shown. The construction is bad and should be avoided.

Fig. 74 shows a piston rod with a taper fit in the crosshead and the hole for the key half-round at the ends. This is a slight improvement over the first-described straight rod. If the diameter of the rod is not increased, the only advantage is that it will not hammer into the crosshead, the taper being about $\frac{7}{8}$ in. per foot; but the disadvantage is that the taper reduces the diameter of the rod at the farthest end of the keyhole so much that there is not sufficient metal left to resist the alternating forces of the steam pressure acting on the piston and rod, the metal crystallizes and a break results. If the rod is made sufficiently large in the crosshead, with

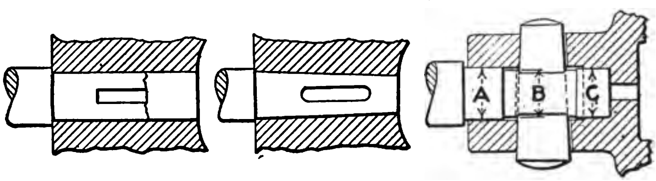


FIG. 73. FIG. 74. FIG. 75.
Piston-rod fastenings in crosshead.

a corresponding increase in the key, the design may be satisfactory.

It is well to investigate the kind of steel in a piston rod. Cold-rolled is not fit to be used, but open-hearth steel of 0.30 to 0.35 carbon can be recommended. If, for reasons of quick repairs or easy dismantling, as in rolling-mill work, a keyed piston rod is desirable, it should be extra large in diameter and the reduction at the crosshead should be made in three offsets. The diameter A, Fig. 75, should be $\frac{1}{8}$ in. smaller than the rod, B, $\frac{1}{16}$ in. less than A, and C about $\frac{1}{16}$ in. less than B, and A and C should be easy driving fits, while B should be at least $\frac{1}{32}$ in. loose in the bore. The end of the piston rod should fit against the bottom of the bore C, a small hole leading into the center core of the crosshead. The key must be located in part B, the key and holes to be half-round at the sides.

A disadvantage of the keyed piston rod is that the clearance in the cylinder cannot be adjusted. Where this is essential, the piston rod should be screwed into the crosshead and provided with a large check nut. The outside diameter of the thread should be about the same

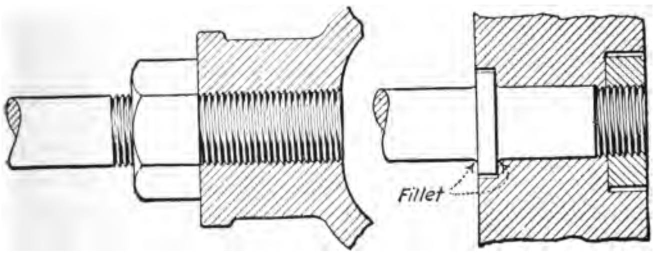


FIG. 76. FIG. 77.

Threaded piston-rod fastenings.

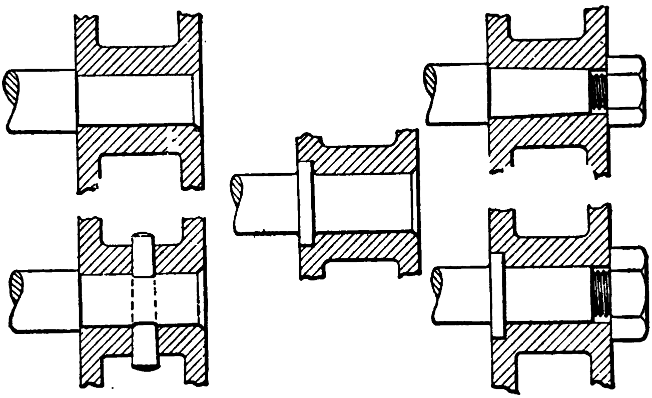
as the diameter of the rod and the length of the thread in the crosshead from $1\frac{3}{4}$ to 2 times the diameter of the rod. Care must be taken to make the thread a good fit. Fig. 76 shows the connection of a threaded piston rod and crosshead. The number of threads per inch may be made as follows: For $2\frac{1}{4}$- to $2\frac{1}{2}$-in. diameter rod, 8 threads per inch; for $2\frac{3}{4}$- to 3-in. diameter rod, 7 threads per inch for $3\frac{1}{4}$- to 4-in. diameter rod, 6 threads per inch; for $4\frac{1}{4}$- to 5-in. diameter rod, 5 threads per inch; for $5\frac{1}{4}$- to 6-in. diameter rod, 4 threads per inch; for $6\frac{1}{4}$- to 8-in. diameter rod, 3 threads per inch. The threaded piston rod is to be recommended if economy is desired, for it enables the builder to reduce the clearance in the cylinder to a minimum and to adjust it to be equal at both ends of the stroke.

The stress of piston rods in tension and pressure, whether keyed or screwed, should not be more than 6000 lbs. per square inch in the weakest section, and the shearing stress of keys and rods must not exceed 5000 lbs. per square inch. A piston rod occasionally breaks in the piston, but this does not happen as often as in the crosshead. The reason for this is that most of the rods are pressed into the piston spider, where they are held very tightly and are not subject to hammer action. as at the crosshead end.

Fig. 77 shows a good way of securing the piston rod to the piston. The rod has a large collar against which the piston spider is pressed. Liberal fillets should be turned on both sides of the collar. The end of the rod is threaded and has a steel nut screwed on, which is let into the piston. This makes the face even without any projections. The nut is set up tight and secured by a lockscrew.

Other ways of holding the piston on the rod, frequently used by engine builders, are shown in Figs. 78 to 82. Fig. 78 shows the rod slightly reduced in diameter, pressed into the piston and the end riveted. Fig. 79 shows the same construction with an additional cotter pin driven through; this weakens the rod and is open to the same objections as the key and hole at the crosshead end. If the rod is pressed in solid the cotter is of no use, but if it is not a perfect press fit the hole weakens it so much that there is danger of a fracture. A number of rods have failed at this point and caused much damage. Fig.

80 is an improvement on Fig. 78. The collar on the rod is an excellent thing, but, of course, there is the uncertainty of the press fit. Figs. 81 and 82 are used frequently. Fig. 81 shows a taper rod with a thread and nut at the end, and Fig. 82 has a straight fit with a collar at one end and thread and nut at the other. Both of these types will hold if the rods are pressed in tight, but the disadvantage is that the large nut projects and causes a



FIGS. 78, 79, 80, 81 and 82.
Piston rod fastenings in piston.

continuous loss of steam, for the reason that a cavity for the nut must be provided in the back cylinder head, which increases the clearance space and also the surface on which steam will condense.

Troublesome parts of a piston are the follower bolts, which deserve thorough consideration. For these bolts, or studs, the best grade of Norway iron or its equal should be used; it should be carefully tested for tensile strength and elongation. Mild steel or, worse, high-carbon steel should not be used.

A much-approved follower-bolt construction is shown in Fig. 83. The bolts are made to go through the piston, one end is screwed tight into the piston spider against a collar on the stud, and the end is riveted over to hold the bolt. The shank has play in the hole to give the bolt a chance to bend if not perfectly square with the follower. A nut holds the follower onto the piston. This nut is important, for if one comes off there is usually some damage done, either to the piston, the back head or to the exhaust valve and its gear. Some engineers prick-punch the nuts and studs to keep the nuts from backing off; others put in split pins—dangerous things if not properly secured in the holes. If the nuts are not a good fit on the bolts, so that they have to be screwed on all the way with a socket wrench, they cause the engineer a lot of worry. The bolts and nuts should be given the most rigid inspection and replaced if not perfect.

Follower bolts of different designs are shown in Figs. 84 to 86. Figs. 84 and 85 are similar, the only difference being that the studs in Fig. 84 are provided with a collar and, therefore, cannot back out or become loose as long as the follower is in place. The only thing that can come off in the design shown in Fig. 84 is the nut. Fig. 85 shows a stud screwed tightly into the piston, with a nut to hold the follower. The disadvantage of this is that the stud may come out, since there is nothing except the tight fit in the tapped hole to hold it in place. In Fig. 86 a tapbolt with a long thread is used—not such a bad construction in itself if the bolt is made a close fit in the tapped hole and if the thread is perfect, but often this is not the case and then there is trouble. Besides, every time the follower is taken off bolt and hole are exposed to wear, and if this is done often the bolts will get loose and give trouble.

Much thought has been given to improvements on follower bolts, though it seems a simple thing to design some device that will hold a bolt securely in its place. But there always appears to be some drawback. This

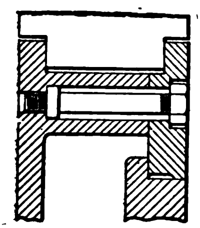


FIG. 83.

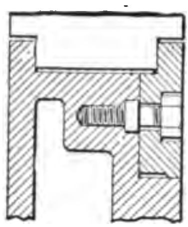


FIG. 84.

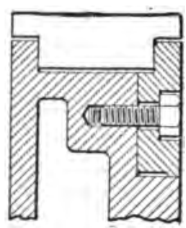


FIG. 86.

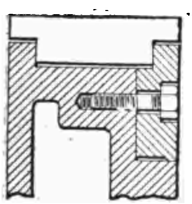


FIG. 85.

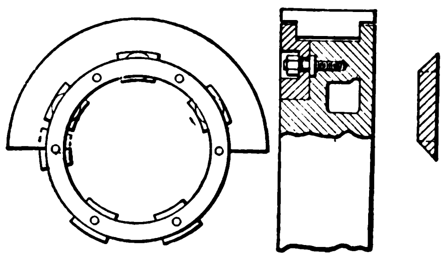


FIG. 87.

Piston rod follower fastenings

is why some engineers insist on a piston without junk ring, follower or bolts. If the piston is made long and lubrication is ample, it may give good results for a while, but eventually it will wear out of center and, not having any provision for adjustment, will give trouble; that is, of course, on horizontal engines. Pistons on vertical engines are not subject to wear and do not need any central adjustment.

An excellent but expensive method of securing follower bolts and nuts is shown in Fig. 87. It consists of a steel ring which is shown in cross-section. A dovetailed groove to correspond with the ring is turned into the face of the follower, the edges of the ring are made square in a number of places, and the groove in the follower is chipped out in the same number of spots, to register with the projecting parts of the ring. This will allow the ring to be inserted in the groove. It can then be turned around into such a position that the narrow parts will be over the follower bolts. A number of countersunk machine screws hold the ring in place. The upper part of the elevation in Fig. 87 shows the ring in position in the groove of the follower, while the lower part shows certain parts cut away at the edges. A number of engines on which the follower bolts gave considerable trouble were provided with rings as described.

Figs. 88 to 91 show different designs of junk rings for pistons. Fig. 88 shows the most approved way of arranging the junk ring, which is made in two parts—one turned to hold the packing ring and the other to fit into a recess in the first. The follower and piston hold the two parts of the junk ring firmly together. Adjustment is made by means of set screws of tough bronze, with check nuts. The face of the two rings is lined with red metal strips hammered into dovetailed grooves and turned flush with the outside of the rings. This construction allows the removal and inspection of the packing ring by simply taking off the follower and the outer half of the junk ring; neither the adjusting screws nor the inner half of the junk ring have to be disturbed. Fig. 89 shows a modification of Fig. 88. The junk ring is made in one piece. If the packing ring has to be taken out and examined, the

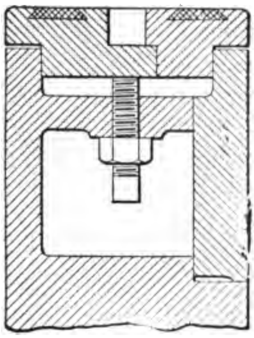


FIG. 88.

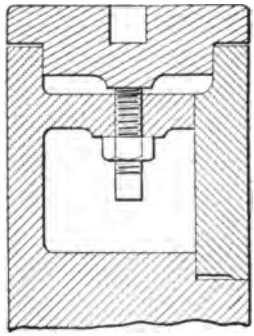


FIG. 89.

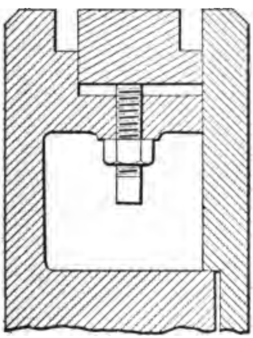


FIG. 90.

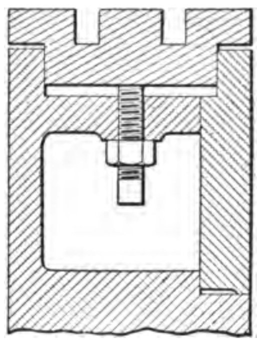


FIG. 91.

Forms of junk rings.

adjusting screws must be loosened and the whole junk ring removed from the cylinder. In Fig. 90 the junk ring covers only part of the piston, so that the latter is partly supported on its own rim and also on the follower. As the piston wears, the junk ring will have to be adjusted, and from that time on the weight of the piston will be supported by the much-reduced face of the junk ring. In this arrangement there are two packing rings used. The disadvantage is that if the second packing ring has to be examined or renewed the junk ring has to be removed. The construction is old and cannot be recommended. Fig. 91 shows a solid junk ring with two packing rings. All the objections to the design of Fig. 89 hold good in this case. There are many opinions about the design of the packing ring or rings, and there is a variety of them in use. It would seem that two rings in a piston would be better and more economical in the use of steam than one, but practice shows this is not true. If the time taken to turn two grooves and make two rings is spent on one groove and one ring making a perfect fit with about $\frac{2}{1000}$ in. clearance on the sides, one ring will be the more satisfactory. On horizontal engines the junk ring bears on the bottom half and it may be assumed that this part will be steam-tight, if properly lubricated. The only leak, then, that can occur is on the upper half; therefore, if the packing ring is made to fit well on the sides of the groove, so that no steam can get under the ring, all that is required to make it bear against the upper half of the cylinder is a slight pressure upward, and it appears that a snap ring will best perform this function, with the joint in the ring at the bottom. The ring may have the inner and outer surfaces turned concentric or eccentric and should be fitted with a certain amount of spring to make it exert a pressure of 2 to 3 lbs. against the cylinder walls. The fit all around must be perfect, which means that it should be turned to the exact size of the cylinder after the joint has been made.

A simple keeper, as shown in Fig. 92, should be used. The joint in the ring is made by letting the two ends overlap. A strip of bronze is riveted to one end of the ring. If this strip is well fitted no steam will escape

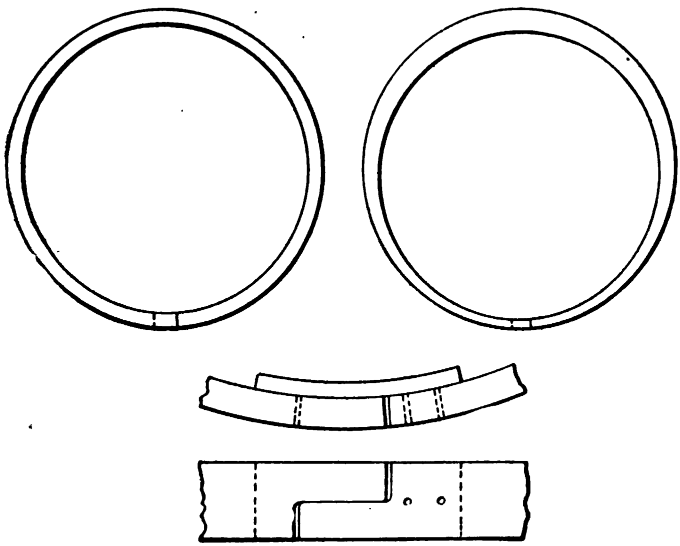


FIG. 92.—A simple keeper.

or get back of the ring and force it against the cylinder wall. Snap rings of this kind may be used with good results up to 30 in. diameter of cylinder. For larger sizes the ring may be made in two or more parts, using bronze keepers, as shown in Fig. 93, which in this form have a small projection at the back, over which a spiral spring slips. The object of this is to hold the parts of the sectional ring against the cylinder wall. Figs. 94 and 95 show the packing ring held out by steel springs.

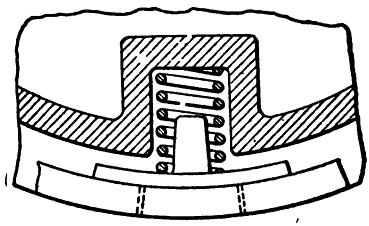


FIG. 93.

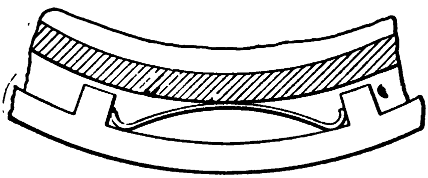


FIG. 94.

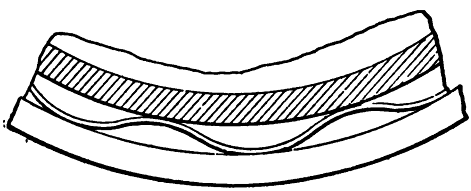


FIG. 95.
Piston ring springs.

This is not considered good practice, as the tempered springs will break sometimes and the pieces work their way out and cut the cylinder.

CHAPTER VI

FRAMES

Guides must be in line with the cylinder. Bored guides have advantages over other types. Girder frames only for light work. Pillow blocks must be strong and well supported. Heavy duty frame for hard service should be in one piece. Two-piece frames are apt to give trouble. Unsupported guides will deflect. Frames must be well supported on the foundation. Oil should drain into a reservoir and be filtered. Oil shields and guards.

Bored guides are the standard. Fig. 96 shows this type. The trouble with the V and the flat guides, shown in Figs. 97 and 98, respectively, is that it is almost impossible to get them in line with the bore of the cylinder, which

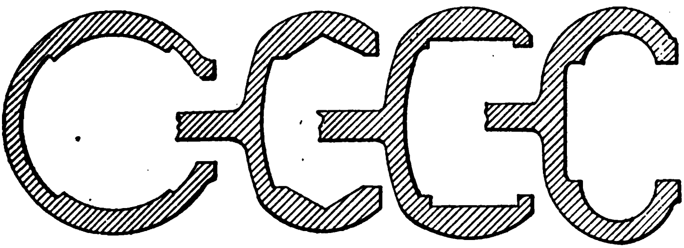


Fig. 96. Fig. 97. Fig. 98. Fig. 99.

Various types of engine guides.

is essential in a well-running engine. The only rational way to attain this is to run a boring bar through and do the facing of the end for the cylinder head and the boring of the guides at the same setting; this will insure perfect alignment, a thing which cannot be accomplished with the other kind of guides.

An advantage of the bored guides is that the crosshead can accommodate itself to the connecting-rod and crank-pin. If for some reason the crankshaft should get out of line, perhaps through unequal wear of the main and outer bearings or a settling of the foundation, the bored guides will permit the crosshead to roll sideways without throwing any extra strain on the pins or the connecting-rod, while the V or the flat guides will make the crosshead shoes bind on the sides and run hot. Another good feature is that with bored guides the lubrication seems to be more effective than with the other two styles. The oil has a tendency to spread evenly over the lower guide when struck by the cylindrically shaped crosshead shoe. On the V guides it will run toward the center groove where the crosshead shoe cannot touch it, and on the flat guides it will spread over one side only and leave the other dry unless the guide is level. Very few crossheads used for the V or the flat guides can be removed without taking off at least one of the shoes; frequently both of them have to come off, and the engineer has his hands full when he undertakes to do the job. With the old-style shoes that lap over both ends of the crosshead and have a wedge for adjustment inside, it is even necessary to remove the connecting-rod. All this trouble is avoided with the bored guides, where the crosshead can be rolled around and taken out through the opening in the side. A very antiquated style is shown in Fig. 99. The bearing surface is milled in half-moon shape to a small radius. These are even worse than the V-shaped or the flat style, since it is almost impossible to make any correction by filing or scraping if they are not parallel and central.

The object of the frame of an engine is evidently to counteract the forces transmitted from the cylinder through the piston, piston rod, crosshead, connecting-rod and crank to the shaft and main bearing, and also to prevent the engine from moving on the foundation. With this in view we can readily see that the girder frame should be used only for light work. The tendency is to overload an engine, either from a mistaken standpoint of steam economy or from a desire to increase the output without going to the expense of adding to the equipment. To do this on a girder frame is a rather hazardous undertaking

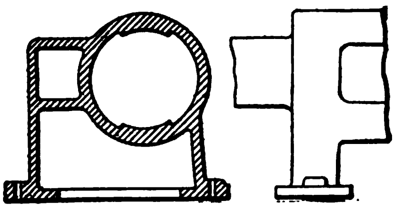


FIG. 100.—Upper and lower guides rigidly connected.

and may result in disaster, unless the frame is built on certain approved lines. As explained previously, the girder frame should have bored guides, and it would be well if the girder part could be of box section instead of the conventional I-beam section. The slide pedestal should be cast onto the frame, not bolted on, and the upper and lower guides should be rigidly connected, as shown in Fig. 100. If the end of the guide barrel has no support on the foundation, as, for instance, in the original girder frames made by George H. Corliss, there is quite a noticeable deflection every time the crank turns over its upper or lower center. Of course the bolting on of a support is only a convenience for engine builders, to make the frames do for right- or left-hand engines. Another convenience which manufacturers introduced years ago, and which is still a source of trouble to the engineer, is the bolted-on main pillow block. Bolts will stretch and nuts will get loose in spite of all precautions, which, of course, cannot happen in this particular instance if the frame and main-pillow block are cast together.

The design of a pillow block needs some serious attention. If made, as shown in Fig. 101, there will be a cer-

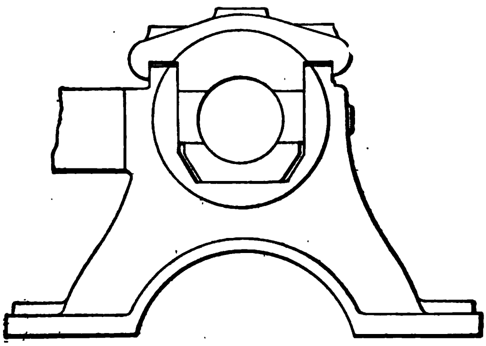


FIG. 101.—Improper pillow block design.

tain amount of deflection in the center, which may result in fracture. Some of the most valuable metal has been cut away, supposedly under the impression that a graceful arch would best answer the purpose of sustaining the weight of the shaft, flywheel and crank. An architectural design rarely meets the requirements of engine practice, where the forces constantly change in magnitude as well as in direction. A dead load is altogether different from the strain caused by a revolving flywheel and the reciprocating parts of an engine, with all the attendant forces due to the pressure on the piston. Beautiful lines and curves are of little value to the engineer—strength and rigidity are what he expects of an engine frame. The late Prof. Sweet's "straight line," backed by a lot of solid metal, is vastly superior to elegant curves and fanciful shapes on engine work.

Fig. 102 shows a good design of main pillow block. The

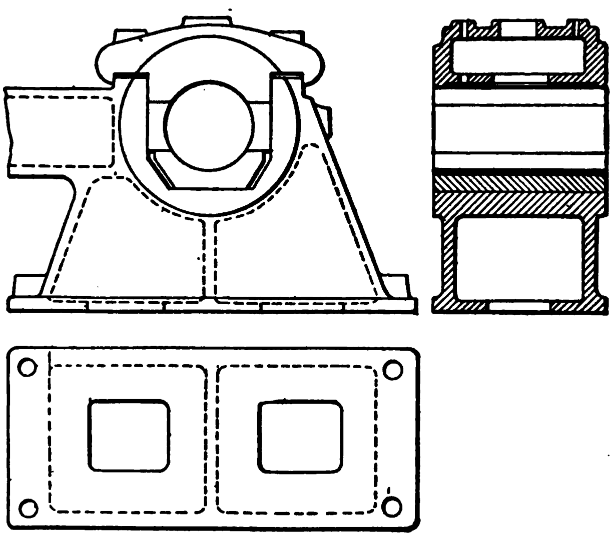


FIG. 102.—Good pillow block design.

base has full bearing on the foundation, and the shaft is supported in the most natural and direct way. There are just sufficient openings in the base for removing the cores, otherwise it is one continuous surface. This is a substantial construction, and if designed in the proper proportions it will give satisfaction. There is no attempt made to save expense by sacrificing strength, as in the pillow blocks shown in cross-section in Figs. 103 and 104.

Fig. 103 illustrates a case of insufficient support at the base line and is contrary to all sound principles. There are no return flanges provided, consequently it is impossible to use grouting on this construction. The object of this design is to save a few dollars by letting the inside of the pattern leave its own core. Fig. 104 is another design from which trouble may be expected. It is made of rib

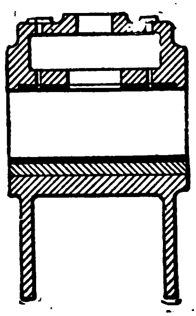


FIG. 103.—Insufficient support of pillow block at base line.

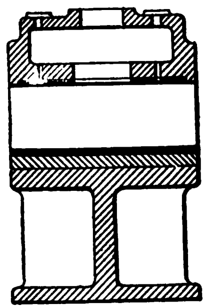


FIG. 104.—Ribsection shown is weaker than box section.

section with the idea of saving core work, the base is broad and long, but rib sections do not compare in strength with box sections; they seem to have a natural tendency to develop cracks, probably owing to shrinkage strains.

For heavy, continuous service an engine should be provided with a so-called heavy-duty frame as shown in Fig. 105, which is made in one casting with a wide continuous bearing on the foundation. Guide barrel and main pedestal are in one piece. In this one-piece design it is possible to counteract all strains in the most direct way and there are no sharp breaks in the lines, or offsets with sharp corners where cracks could develop. The argument that a frame of this kind is subject to dangerous shrinkage strains cannot be maintained, for the casting can contract freely in the mold in all directions. Another advantage is that it requires little machining and fitting and is easy to line up in erecting if it is once fitted right in the shop, which is not always the case with the two-piece frame shown in Fig. 106. On the latter design the frame and

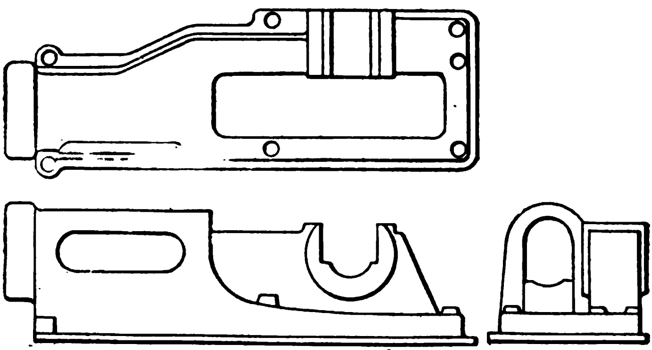


FIG. 105.—One piece heavy-duty frame.

guide castings are machined separately and bolted together. In shipping, the two pieces are taken apart again and the chances are that there is more or less trouble in lining up when the engine is erected, as many an erecting engineer has found out to his sorrow. The joint between the two pieces is the weak spot; tap-bolts are mostly used to hold the two parts together and on some designs, even links are shrunk in, which is, of course, open to severe criticism.

Guides that are not supported on the foundation, as shown in Fig. 107 are not fit for heavy work. There is always a certain amount of deflection in them, and often the lower crosshead shoes give trouble on account of this—

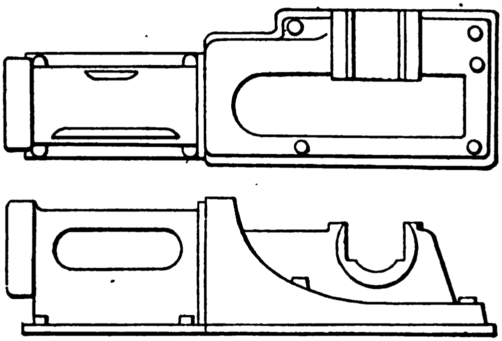


FIG. 106.—Two-piece frame.

a puzzle to the engineer, who wonders what can be the matter with the crosshead that makes the lower shoe and guide run hot, and why they require such a lot of oil.

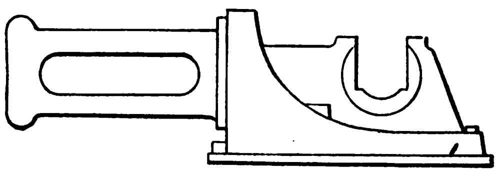


FIG. 107.—Insufficient support for guides.

Even if a foot is bolted on under the guide, as shown in Fig. 108, the design is only an excuse for a heavy-duty frame and does not compare in strength and rigidity with the one shown in Fig. 105. For extremely heavy work the bottom of the frame should be a continuous plate with enough openings for removing the cores, but for less strenuous work it will be sufficient if a broad flange is run around the outside, with a rib turned up to form an

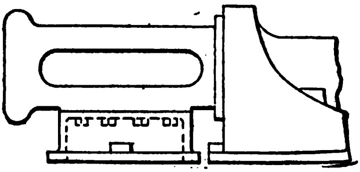


FIG. 108.—Pedestal under guide.

oil channel, as in Fig. 109. This is far superior to the method shown in Fig. 110, where the edge at the bottom is reinforced by a narrow strip, which does not afford a good bearing on the foundation and will not keep oil from running on the floor if any gets on the frame. An oil

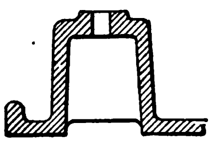


FIG. 109.—Broad flange forming oil channel.

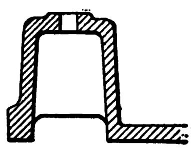


FIG. 110.—A narrow strip does not afford a good bearing on the foundation.

channel all around the frame is one of the things that a modern engine should be provided with. If it is not, some oil is bound to get on the floor and, worse, on the foundation, where it will destroy the grouting and weaken the concrete.

Oil used for lubrication should be free from grit and dirt; therefore, it is best to filter all of it, even the new

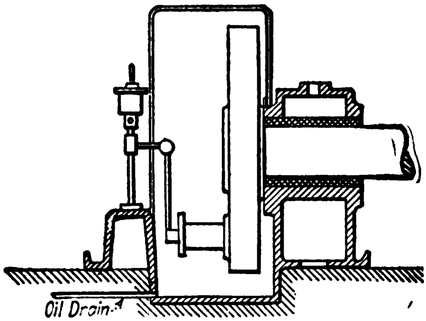


FIG. 111.—Crankpin section.

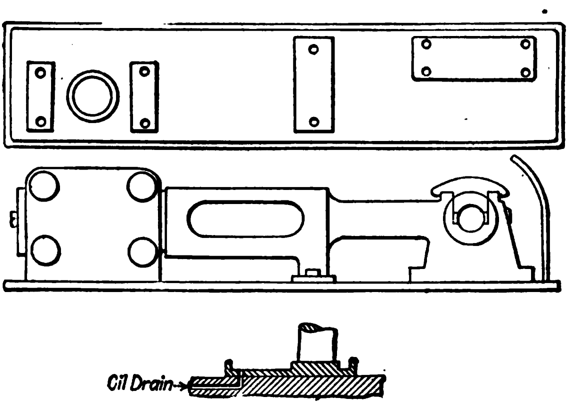


FIG. 112.—Cast-iron sole-plate under frame.

supply. Filtered oil is often better than the original article, since the volatile parts have been eliminated in the filtering process. One of the advantages of the heavy-duty frame is that it has a bottom cast in the crank pit and oil used on the guides, crosshead, crankpins and main bearing can be made to drain into this pit and from there to a tank or direct to a filter. Fig. 111 shows a cross-section through the crank pit at the main bearing. With the girder frame matters are different. It is open on all sides and has no provision for catching oil. The common practice is to collect in tin pans the oil that is thrown off. It is almost impossible to keep an engine room clean with these contrivances, which will begin to leak in a short time or are likely to be upset, making the floor slippery and nasty. A much better way is to provide a continuous

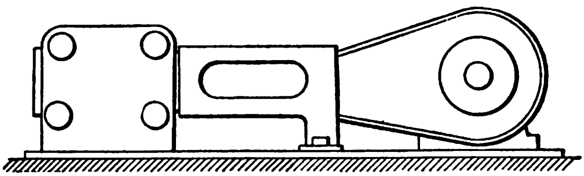


FIG. 113.—Oil guard for high speed engine.

cast-iron sole plate under the whole girder frame, as shown in Fig. 112. The plate may be made in one piece or in two bolted together, according to the size of the engine, with a separate plate for the outer bearing. The outer edge of the plate should have a bead running all around it, and if there are any holes, as, for instance, under the cylinder for the exhaust pipe, they should also have beads around them to prevent oil from running through. The plate should slope from all sides toward the crank or some other convenient point, and the oil should be drained off. A simple oil shield for the crank, as in Fig. 112, is generally sufficient for a slow-running engine, but if the speed is over 100 r.p.m. or the piston speed over 600 ft. per minute, the connecting-rod and crank should be inclosed by an oil guard, as in Fig. 113. The combination of the continuous sole plate and this kind of oil guard can be recommended; it will pay for itself in a short time.

CHAPTER VII

BEARINGS

Main-bearings in four parts. Quarter boxes must be adjustable. Lower box removable. Wedge adjustment. Chain oilers. Water must not be used on a hot bearing. Outer bearing must be well lubricated. Pressure on bearings to be within certain limits. Diagram for calculating the pressure on bearings. Undesirable features of bearings on old designs.

OF main-bearing designs there is a great variety, but the one using two quarter-boxes, an upper and lower shell and a cap which fits over the jaws of the pillow block,

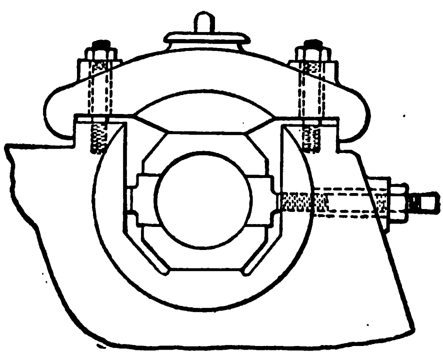


FIG. 114.—Common main bearing.

seems to be the accepted standard. Fig. 114 shows an arrangement that we find in many engines. Its disadvantage is that it has screw adjustment for the outer quarter-box only; if the opposite quarter-box wears, shims will have to be used on that side. The jaws should be made wide enough to let the bottom box be rolled out by lifting the shaft. This is a very important feature which is not carried out in some engines. If the lower box has to be removed on the eccentric side, there is usually more or less trouble.

For ordinary factories a main bearing like Fig. 114 will be satisfactory, but where the work is heavy and continuous, as in rolling mills and for large direct-connected

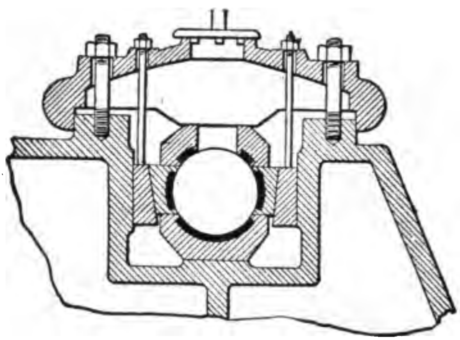


FIG. 115.—Main bearing adapted for heavy service.

engines, the one shown in Fig. 115 will give better results. In this design both quarter-boxes are adjusted by means of wedges. The wedges should always have full bearing on the quarter-box, to prevent the latter from tilting and binding on the shaft. Many main bearings run hot and give trouble on account of the wedges being too short, as shown in Fig. 116, in which case the shaft in bearing against the upper corner will throw the lower edge of the quarter-box inward—in other words it will make the quarter-box act as a lever.

Fig. 117 illustrates a bearing that is well adapted for direct-connected engines; it has about all the refinements that a main bearing needs. The quarter-boxes are taken up by wedges. The lower box is made cylindrical, and it can even be made with a spherical seat if desired. Cylindrical and spherical shells can be removed by simply relieving them of the weight of the shaft; there is no need to take the generator apart, which must be done if a flat-bottomed lower box has to be taken out. The bearing is provided with chain oilers, which will carry a good supply from the reservoir over the shaft, assuming, of course, that the chains are running. It is not wise to depend on chain oilers entirely; if used in connection with a gravity oiling system, they are a convenience to the engineer. The oil in the reservoir must be renewed from time to time and all sediment removed. If this is neglected, a bearing may run hot in spite of the chain oilers.

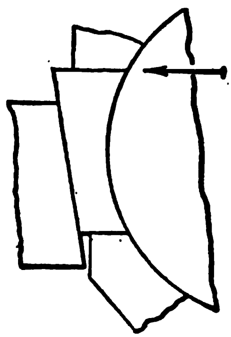


FIG. 116.—Wedges are too short for the quarter box.

If a bearing does run hot, one of the worst things that can be done is to turn water on it; water always contains more or less grit and dirt unless it is distilled, and it will spoil a bearing. If grit of any kind once gets into a bearing, it will be found hard to remove. A hot bearing should be treated with plenty of high-grade oil, cylinder oil will help some, but, of course, if it gets into a bearing, it is difficult to get it out again. There is

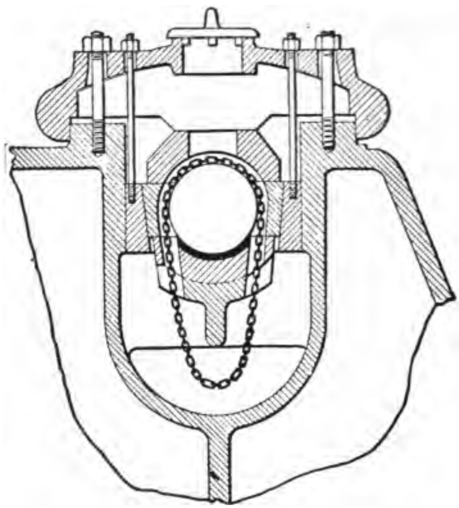


FIG. 117.—Main bearing for high-speed engines.

no cure for a hot bearing except to find and remove the cause.

The outer bearing often gives more trouble than the main bearing, because the shaft turns always in the same position, while in the main bearing it is moved forward and backward, owing to the push and pull of the connecting-rod. In fact, the pressures vary constantly in accord with the forces acting on the crankpin; therefore, the outer bearing must be provided with ample means for lubrication and should be watched very closely on a new engine. It is best to use a construction similar to the main bearing. Quarter-boxes are not absolutely necessary, but a removable-bottom box is most desirable. Fig. 118 shows such an arrangement. It also shows a chain-oiler, which is of more use on an outer bearing than on a main bearing. The cap must be provided with a large

FIG. 118.—Good type of outer bearing.

opening, enabling the engineer to examine shaft and oiler without difficulty. A copious supply of oil is to be recommended.

Fig. 119 shows an outer bearing made in four parts. The quarter-boxes can be adjusted by means of set-screws—a convenient way, since there is hardly any side strain in this case. There are two oil shields bolted to the bearing—one on the inner and the other on the outer side, which serve the double purpose of holding the boxes in place and collecting the oil that runs off. The outer shield has a hole in the center to feel the shaft. A grooved ring is provided on the shaft inside of the inner shield; the ring is shrunk on, its function being to keep oil from creeping along the shaft. If necessary a wiper may be attached to it.

If a bearing is large enough and well lubricated it should run for years without being rebabbitted. To attain this the pressure must not exceed 100 lbs. per square

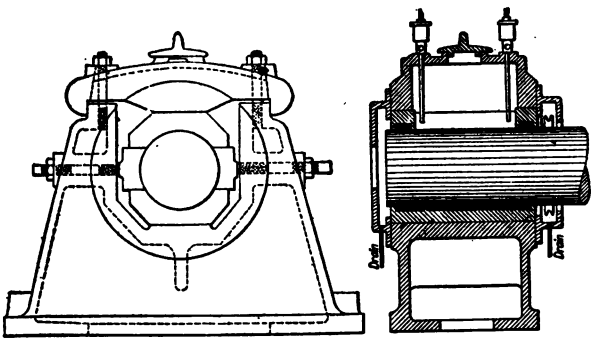


FIG. 119.—An outboard bearing of excellent design.

inch of projected area. Any pressure over 130 lbs. per square inch is dangerous and liable to heat the bearing, even with a good supply of oil. On the main bearing it is not so important to take the steam pressure on the piston into account, as the weight of the crank, shaft, flywheel, armature, etc., because the dead weight of these parts is always acting in the same direction, while the pressure, due to the force of the steam, acts alternately in opposite directions, giving the oil a chance to work its way between quarter-boxes and shaft. The lower shells of main and outer bearing are the parts exposed to a continuous grind that will give trouble if overloaded. The old rule, to make the diameter of the crankshaft equal to one-half the diameter of the piston and its length equal to the same, gave good results for ordinary belt or rope drives, but for rolling mills, electric light and power plants and other installations where very heavy flywheels are required, this rule is deficient and the size of the bearing must be carefully considered. To determine approximately the pressures on main and outer bearings, it is well to lay down a simple diagram, as shown in Fig. 120, and to ascertain the reactions A and B. Without going into any refinements the following calculation will give a pretty good idea of the pressures one has to deal with on a single-cylinder engine.

Let

F = Weight of flywheel;
G = Weight of armature;
S = Weight of shaft;
C = Weight of crank;
A = Reaction of main bearing;
B = Reaction of outer bearing;

Then

$$A=\frac{F\times b}{e}+\frac{G\times d}{e}+\tfrac{1}{2}S+C$$

$$B=\frac{F\times a}{e}+\frac{G\times c}{e}+\tfrac{1}{2}S.$$

The lower-case letters in the formula are taken from Fig. 120. These formulas, of course, are not quite correct, but they will answer for all practical purposes. To get the pressure per square inch *A* and *B* must be divided by the projected area of the respective bearings.

Some of the older engines are provided with main bearings which are entirely unfit for present conditions. Fig. 121 shows one in which the cap does not hook over the jaws; the full steam pressure carried on the outstroke from the piston through the intervening parts to the shaft and bearing has to be sustained by the end of the frame without any support from the cap. The bending strain on the outer jaw is, therefore, considerable, due to the leverage on which the pressure acts. If the wedges

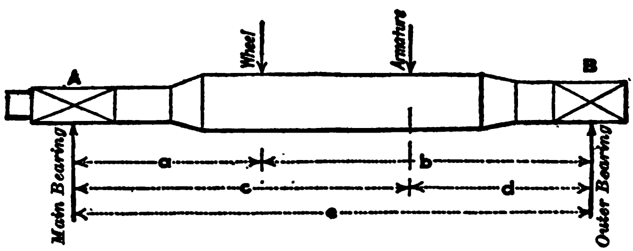


FIG. 120.—Diagram to assist in calculating pressures on bearings.

do not bear properly on the quarter-box and frame, but only on the upper part, the pillow block is liable to spring and may develop a crack or break off. A small amount of metal in the form of lips on both sides of the pillow-block cap with a tight grip over the jaws would be a great improvement. Another inferior feature of the design are the slender wedges, two on each side, which hold the quarter-boxes in place, if properly designed they should extend over the full length of the bearing. There is no loose bottom box, the babbitt is simply poured into a recess in the frame; if the bearing has to be rebabbitted the shaft with everything on it has to be jacked up sufficiently to remove the old lining, pour the new metal and

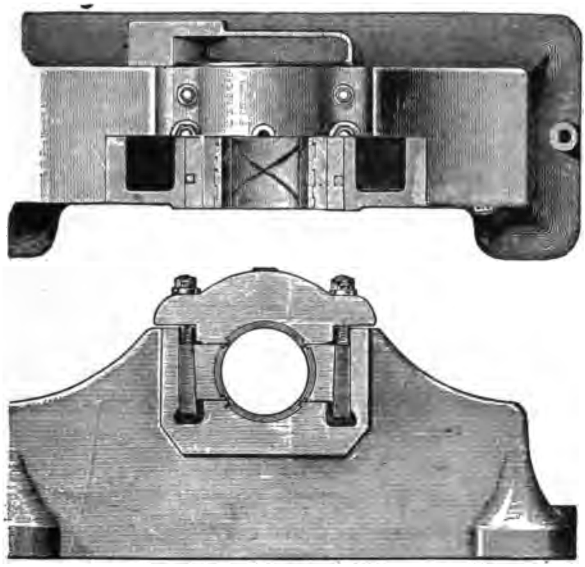


FIG. 121.—Undesirable mainshaft bearing.

scrape the bearing. On large engines this is a hard and tedious job.

CHAPTER VIII

CONNECTING-RODS

Gib and key ends. Strap end rods. Inside adjustment increases distance between centers. Wedge adjustment. Marine end rods are not a success on stationary engines. Hatchet and forked rods have their drawbacks. Solid end rod for continuous service.

Two of the older designs of connecting-rod ends are shown in Figs. 122 and 123. Rod ends like Fig. 122, with gibs and key, were used first. They answered the purpose for slow speeds, say 60 or 65 r.p.m.; with increased speeds the gibs gave trouble and were abandoned for through bolts like Fig. 123. Many rods of this kind are still in use. The trouble with them is that the keys will hammer into the boxes and have to be taken up frequently. These strap-end rods were very convenient when forgings were made of wrought iron welded together from "selected scrap"; it was easier in those days to make the rods in several pieces than in one. With the advent of open-hearth steel the built-up rod became obsolete and the solid-end rod, forged in one piece from a billet, made its appearance. The strap rod, shown in Fig. 124, is disappearing; no modern engine builder would make one except for a special occasion. On rods of this kind both keys are on the inside. As the boxes wear and the keys are taken up, the distance between centers gradually increases and the consequence is that the clearance in the cylinder begins to vary. On old engines this has to be watched closely, otherwise there may be trouble with the piston striking the back head. To keep the distance between centers about the same, shims are put between the boxes and the straps, an expedient which can not be recommended.

On engines where the crosshead pin cannot be removed, or which have a center crank, the connecting-rod must

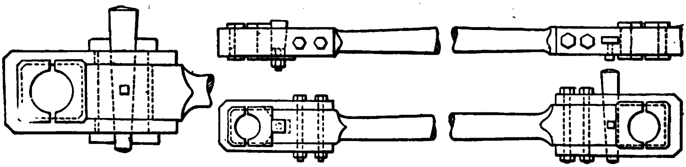


FIG. 122. FIG. 124.

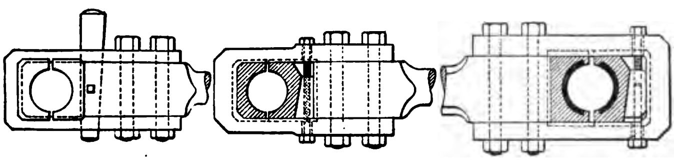


FIG. 123. FIG. 125. FIG. 126.

FIG. 122.—Connecting-rod end with gib and key.
FIG. 123.—Connecting-rod end with key and bolts.
FIG. 124.—Common strap end connecting-rod.
FIGS. 125, 126.—Strap connecting-rod ends with wedges and bolts.

be made so that it can be taken apart, and under such conditions the rod ends shown in Figs. 125 and 126 can be used to advantage, the former for the crosshead end and the latter for the crank end. The adjustment is made by means of wedges held in place by tapbolts. The wedge at the crank end is placed on the outside and the one at the crosshead end on the inside, so that any adjustment for wear will leave the distance between the centers of boxes practically constant. The marine end shown in Fig. 127 is sometimes used on the crankpin, but with indifferent success; it is very hard to adjust, especially in the larger sizes. Here again we have to do with shims, which are placed between the cap and the rod, and as the boxes wear some of them are taken out and the nuts are drawn up tight again—certainly the crudest way of adjustment that can be conceived.

There is also the hatchet rod, shown in Fig. 128, a sub-

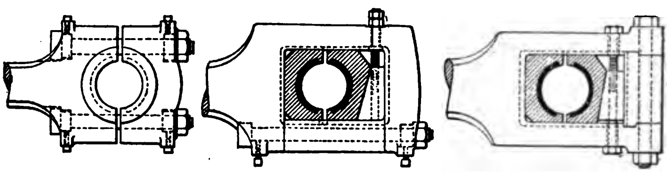


FIG. 127. FIG. 128. FIG. 129.

FIG. 127.—Marine end connecting-rod.
FIG. 128.—Hatchet rod end.
FIG. 129.—Slotted rod end.

stitute for the marine rod. The upper side is solid, but the lower is cut out so that the rod can be lifted off the crankpin by removing the large bolt and the block that fills the gap. Fig. 129 represents a forked rod which is cut through at the extreme end. The opening is closed by means of a steel block which is recessed into the two sides of the rod. A large bolt holds the ends and block together. The adjustment of these two rods is effected by wedges located near the outside of the rods. Neither the hatchet nor the forked rod finds much favor. They are cumbersome, require the highest class of workmanship and, if the blocks and bolts are not a very good fit, will give a good deal of trouble.

In an up-to-date Corliss engine the connecting-rod designs just described should not be accepted; the solid-end rod is the only one which can be recommended. There are, of course, many different designs of the latter, for each of which a number of claims of superiority are advanced. Fig. 130 shows a solid-end rod provided with boxes which, it is claimed, cannot pinch the pins. They are not split in the center in the customary way, but are slotted out on one side, into which the adjustable

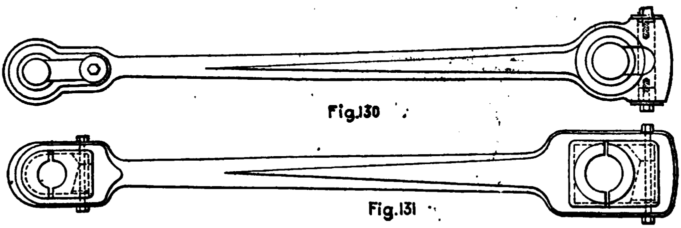


FIG. 130.—Connecting-rod with cylindrical boxes.
FIG. 131.—Ideal connecting-rod for Corliss engines.

part of the box fits. On the crosshead end the adjusting wedge is located parallel to the crosshead pin; the object of this is to make the wedge accessible outside of the crosshead, where it can be given close attention. The advantages of the rod seem to be that the boxes are cylindrical and can, therefore, be fitted easily and that most of the operations on the rod are performed by turning, boring and milling. The disadvantages lie in the long slot at the crosshead end, which is apt to spring, and in small surface cf the adjusting wedge at the crank end.

The rod shown in Fig. 131 represents what might be called the ideal one for a Corliss engine; it fills all the requirements and, on account of its simplicity, strength and reliability, has been almost universally adopted as the standard. There are no frills about this kind of rod; everything seems to have been logically developed. The rod is forged in one piece, turned in the lathe, the sides planed, and the ends slotted out and squared to receive the boxes and wedges. The surfaces that come in contact

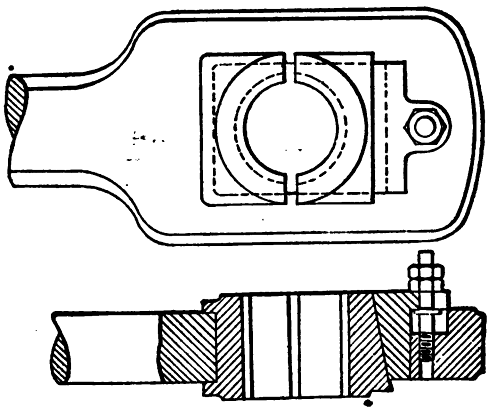


FIG. 132.—Wedge bearing full on box.

between the boxes and wedges are very large and the pressure per square inch on these surfaces is consequently much reduced. The best material for the wedges is forged steel. Cast iron is not suitable for this purpose; it is likely to break through the threaded hole and give trouble. The tapbolts must have a good fit in the wedge, but they should be loose in the rods, with at least $\frac{1}{16}$-in. play or, in the larger sizes, $\frac{1}{8}$-in. If they are made to fit tightly, there will be trouble, as they will break in a short time. Norway iron or its equivalent is the best material for the tapbolts; mild steel is likely to crystallize and break. A small fillet should be left between the shank and the head of the bolts, and sharp corners must be avoided.

It has been said that the space above the wedge for adjustment is objectionable and that the box and wedge should bear over their entire surface. In Fig. 132 an attempt is made to overcome this difficulty. The wedge is inserted from the front, it bears over the full surface of the outer box and is held in place and adjusted by means of a screw-bolt. A set-screw from underneath helps to secure the wedge; this is placed in a horizontal position instead of vertical, as in Fig. 131. This small set-screw, which is used as a check in addition to the large bolt, shows that the confidence of the makers of this kind of rod is not quite supreme as to the staying qualities of the wedge. The projecting bolt and nuts, in front, where the engineer has to feel the rod, are objectionable.

CHAPTER IX

HOOK RODS

Troublesome hook-rod designs. Some rods are liable to be thrown out of gear. Hook rods of antiquated design are lifted by hand. Reynolds' hook rod. Finders and clamps. Telescopic rods.

THERE are many different designs of hook rods—in fact, every Corliss-engine builder seems to have one of his own. Very few are entirely satisfactory; none is

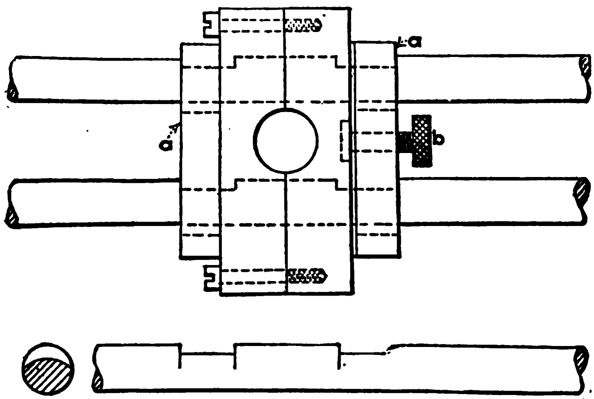


FIG. 133.—An inconvenient hook rod.

perfect. It is discouraging to note that after more than fifty years of Corliss-engine designing and building in this country, human ingenuity has been unable to overcome all the difficulties which the problem of producing a good hook rod presents. The most troublesome hook rod and its disengaging device are shown in Fig. 133. Instead of being in one rod, as most of the others, this one has two rods running parallel, both notched in two places, as shown in the detail. Two bars *aa* drop into the notches and are forced against the shoulders of the rod by tightening the set screw *b*, which has a knurled head. To disengage the rod from the hook block, screw *b* is loosened and the bars *aa* are lifted out of the notches.

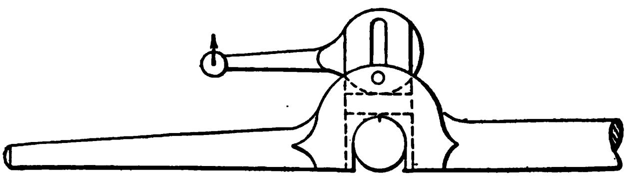


FIG. 134.—Hook rod with lifting lever.

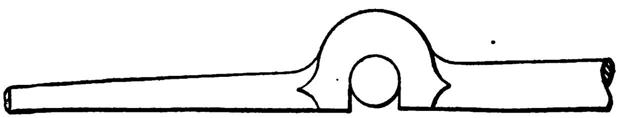


FIG. 135.—Common hook rod.

In throwing the rods into gear again, the bars must be moved close up to the block and secured in the notches—the most awkward thing imaginable and dangerous to the fingers of the attendant. Anyone who has to handle this kind of device every day and does not get his fingers nipped once in a while is lucky.

Figs. 134 and 135 represent two antiquated designs of hook rods. The difference between the two is, that the one shown in Fig. 135 has to be lifted out by hand and must be held by the operator or by slipping a ring on a chain over the end, while the one shown in Fig. 134 is lifted by moving the lever over in the direction of the arrow, thus throwing a small gib in position, which will make the rod slide over the hook pin. No provision is made to hold these rods on the pin except by their own weight; they are liable to be thrown out of gear and give trouble, if the valve gear begins to work a little harder than usual or if the load increases suddenly. To remedy this a dovetailed slide is arranged on the under

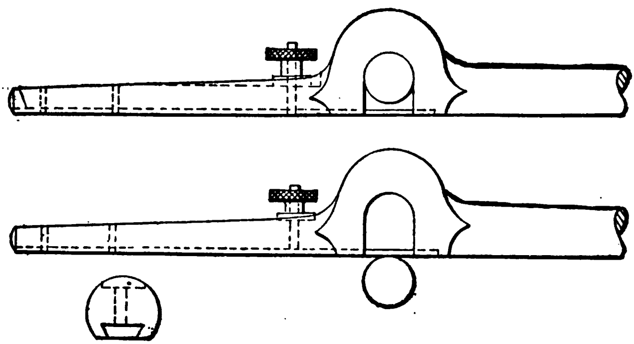


FIG. 136.—Hook rod with dovetailed slide.

side of the rod shown in Fig. 136. The slide answers the double purpose of keeping the rod in place and letting it slide forward and backward over the pin when lifted out. The rod has to be lifted by hand.

To avoid the lifting of the hook rod has been the aim of many designers, most prominent among them the late Edwin Reynolds, of Milwaukee, to whom the device shown in Fig. 137 is credited. The object of the different designs is to let the rod slide through a block engaging and disengaging it at will in a certain place. In Mr.

Reynolds' device a casting A is bolted to the wristplate and bored out to receive the block B, in which is located the pin C, having a handle D fastened to the outer end. The hook rod E runs through holes in A and B, holding them together. The pin C has a shoulder on which a spiral spring rests, forcing the taper end of C into a corresponding hole in rod E. To disengage the rod and block, D is turned around until a small pin strikes a stop, when C

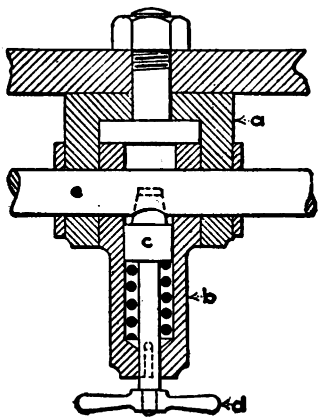


FIG. 137.—Reynolds' hook rod.

can be pulled back, the neck of B having a slot, through which the small pin passes. When the pin gets on the outside, D is turned about one-quarter of a turn, which locks C. To engage the rod again, D is turned back so that the pin can slip through the slot, when C will be made to find the taper hole in E by moving the wristplate forward and backward. To lock the mechanism the handle D is turned ahead till C is tight in the rod E. It was soon demonstrated that this device would do for light work only; on heavy work the force applied to the taper pin was not sufficient to hold the hook rod tight. To remedy the trouble the arrangement shown in Fig. 138 was adopted, in which pin *C* acts only as a finder, while the screw *F* secures the rod. Two operations are required to throw the rod in and out of gear, and it is rather confusing to the engineer to remember which of the two handles he has to turn first.

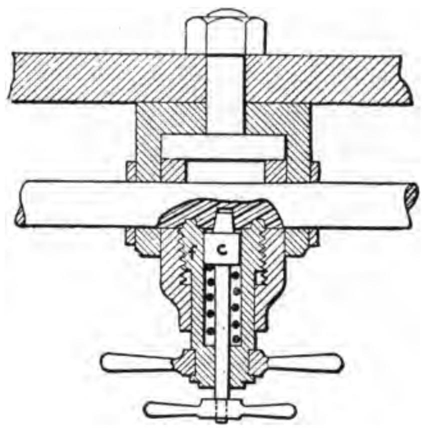


FIG. 138.—Locking device for heavy engines.

Fig. 139 shows a modification of the last-described disengaging device. The number of parts has been reduced; it has the finder and the screw to tighten the rod, but there is only one handle and one-half turn in one direction or the other will release or fasten the hook rod. This device fills all requirements except that of taking up the wear in the wristplate. However, it will last for years with proper attention and adequate lubrication.

The two devices shown in Figs. 140 and 141 use a finder to locate the rod in the required position, but to secure

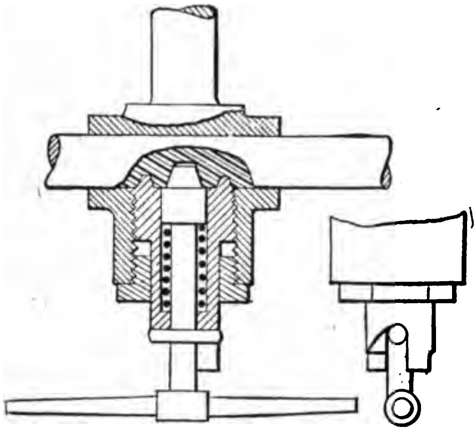


FIG. 139.—Locking device with one handle.

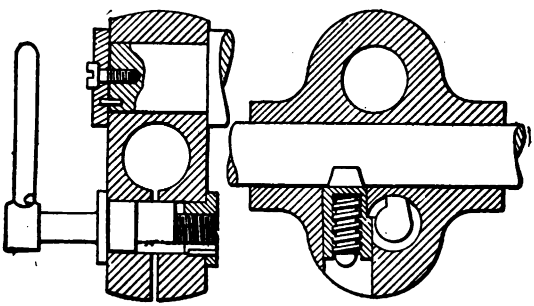


FIG. 140.—Hook rod with clamping device and finder.

it they draw the hook block, which is split half way through for that purpose, together by means of a handle and screw and clamp the rod. In Fig. 140 the performance is ingeniously carried out by simply turning the lever one way or the other, which, at the same time, works the finder and loosens or clamps the rod. In Fig. 141 finder and clamping device are separate, each working independently of the other. Therefore the engineer always has to remember which to manipulate first Devices of this kind that depend on clamping the rod are likely to give trouble in the larger sizes, where much force must be applied to the lever to get an efficient grip on

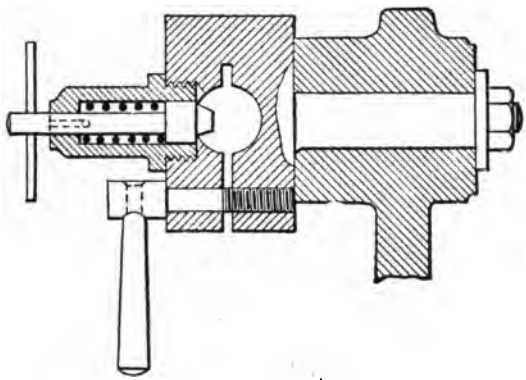


FIG. 141.—The clamping device and finder are separately operated.

the rod. To insure a good hold, a piece of pipe or even a hammer is often used to force the lever over and tighten the rod. Neither of the devices is immune from trouble; there are cases on record where the rod slipped in the block and part of the valve gear was broken.

Some hook rods are made on the telescopic principle; that is, the rods are made to slide in a sleeve provided with a disengaging device similar to those shown in Figs. 138 and 139, or they are made on the same order as shown in Fig. 142, in which a toothed clutch is used. In this device care must be taken to have the rod and sleeve engage in the right place. If the clutch catches on the

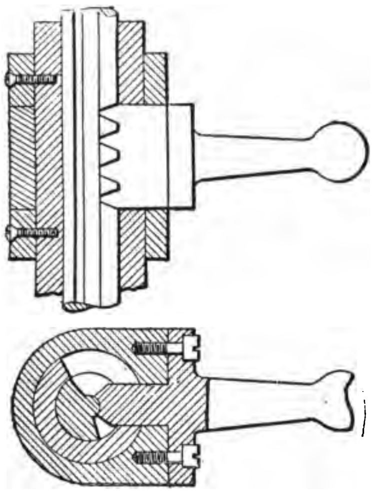


FIG. 142.—The toothed clutch.

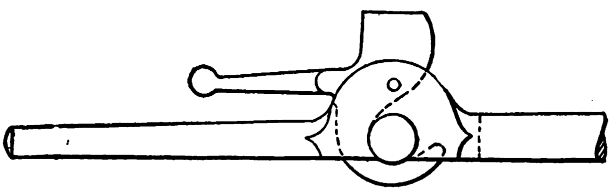


FIG. 143.—A simple and safe engaging device.

wrong tooth, the result is very likely to be a twisted valve stem or broken bonnet.

There is another device, shown in Fig. 143, that deserves mention. It is of similar style to the rod, Fig. 134, but much improved inasmuch as it will not only lift the rod out in turning the handle to the right and let it slide over the hook pin, but it will also keep the rod from jumping off the pin when thrown in gear. There are many other devices, from taper pins to hardened steel latches, but little faith can be placed in them.

CHAPTER X

DASHPOTS

Vaccum dashpot. Objections to the old open dashpot. Inverted pot. Ball and socket joint should be adjustable. Leathers in dashpots are objectionable. Single plunger pots are not satisfactory for varying loads. Differential plunger. Varying loads require frequent adjustment on dashpots. Steam dashpots are not a success. Spring dashpot.

An engineer can often perceive by the sound of the dashpots when something goes wrong with the engine. There is no such thing as a noiseless or soundless dashpot, although some builders may be found who claim that they have them.

An old-time style, still in use on many engines, is shown in Fig. 144. It represents the original vacuum dashpot, the object of which is to create a vacuum in the lower chamber by having the valve gear lift the plunger, and to let the atmospheric pressure force it down again as soon as cut-off takes place. If there is no leak, the force to push the plunger down will be constant for any lift, whether the cut-off is short or long; the pressure will be equal to the area of the plunger in square inches multiplied by the atmospheric pressure. Air in the lower chamber will weaken the action of the pot. To keep it out the plunger must fit well, it should have a number of grooves cut in which, when filled with oil, will act as packing. A small snifting valve A should be provided to relieve the pot of air that may pass below the plunger; the little ball valve, shown in detail, which may be made of a piece of hexagon steel or bronze, will answer the purpose. No oil should be allowed to accumulate in the bottom of the pot; it must be discharged either through the valve *A* or through a pet-cock *C* in

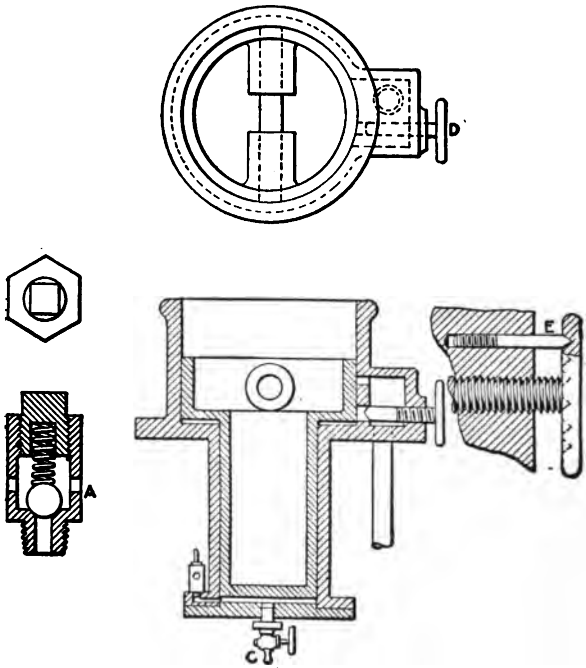


FIG. 144.—An old style dashpot.

the cover. The upper chamber and the large part of the plunger act as a cushioning device for the dashpot in the following manner: Air is drawn in by the ascending plunger through a slotted opening in the side of the pot and on the downward motion is discharged freely until the lower edge of the plunger passes the slot; from that moment cushioning takes place, the confined air being compressed and forced out through a hole near the bottom of the upper chamber, which can be opened or closed as required by the screw *D*. The little check *E*, consisting of a round pin inserted in a hole in the air chamber on the side of the pot and forced by a spiral spring into a countersunk hole drilled into the inner face of the hand-wheel, keeps the screw from turning on its own account. The principle of this pot is all right, but its action leaves much to be desired. The pot is open at the top, and being placed near the floor, it is almost impossible to keep out grit and dirt, which will wear out the pot and impair its action. The pin in the upper part of the plunger for the dashpot rod is a detriment; if the pot is not in line with the valve gear, the dashpot rod stub end will tend to force the plunger over and reduce the quick action desired for good steam distribution and a quiet-running engine. The old rule was to make the diameter of the large part of the plunger twice the size of the small part, which makes it heavy and sluggish in action.

The troubles with the old open dashpot have been overcome in the design shown in Fig. 145. The ratio between the diameters of the plunger has been reduced, and the plunger has been extended to the top of the cushion chamber, preventing dirt dropping into it; the dashpot rod is held in place by a ball joint, eliminating friction due to bad alignment. The air for the cushion chamber is drawn from an annular space around the vacuum pot which, having a number of small holes drilled in the outer shell, acts as a silencer. The cushioning valve is operated by a handle, a slight turn of which will change the effect to suit. The snifting valve and drain-cock are in the bottom of the pot. This kind of dashpot will give good results if it is kept in first-class order.

Care should be taken to remove every particle of core sand from the inside of the plunger and the annular space of the pot; if this is neglected, there may be trouble

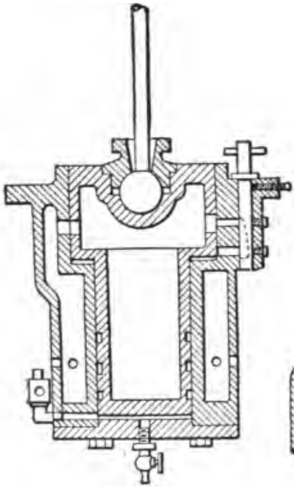


FIG. 145.—Improved dashpot.

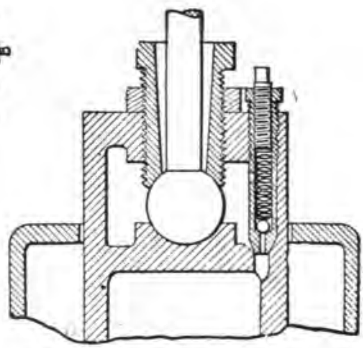


FIG. 146.—Note the snifting valve.

if the vibrations in running shake it loose so that it can work its way between the working surfaces of the plunger and the pot.

Fig. 147 represents what is called an inverted dashpot, in which the plunger is stationary and the pot moves. A removable cap makes the cushion chamber accessible. This kind of pot is practically dustproof. A little snifting valve is located on top of the stationary plunger—a bad place for it. Anything that obstructs this valve puts the pot out of service, and the whole apparatus has to be taken apart to remove the obstruction, which means a shutdown. Some kerosene squirted in through

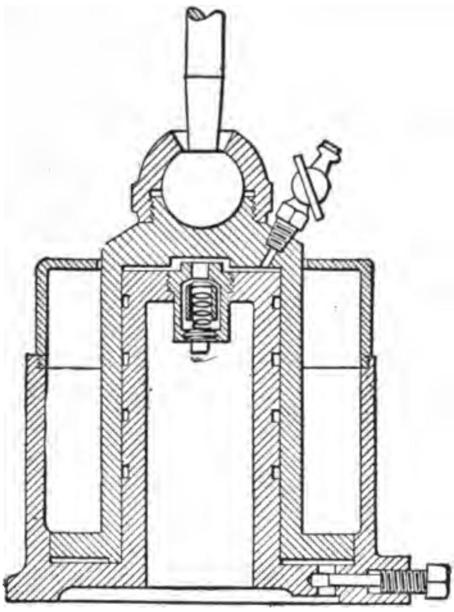


FIG. 147.—The inverted dashpot.

the pet-cock will sometimes wash it out. This dashpot seems to be constantly out of order; the plug gets loose and changes its position, making the action of the pot uncertain. There is no adjustment on the ball joint for the dashpot rod; if the ball wears and becomes loose, the nut must be taken off and reduced at the bottom by filing. These little imperfections on a thing that is otherwise all right are annoying.

Fig. 146 shows a way to overcome the trouble with the snifting valve, pet-cock and adjustment of the ball and socket. The top of the pot is enlarged and a thread cut inside a hole in it instead of on the outside, as in Fig. 147. A screw with a hole for the dashpot rod is screwed in. A check nut will hold the screw tight in place, and any wear or play in the ball joint can be taken up without difficulty. The enlarged head serves also to receive a combination snifting and air valve, which controls an opening to let in air, oil or kerosene, as required. Air made to enter and leave the pot through this valve passes through a chamber that surrounds the ball, thus preventing noise which it would make in passing through a pet-cock.

The inverted dashpot has an advantage over the first two pots described, inasmuch as it has a large outer casing that forms a natural reservoir for the air used in cushioning. No new supply from the outside is needed; the same air is used over and over again. As shown in Fig. 147, the pot has a turned flange at the bottom which performs the cushioning part by dropping into a bored recess of the casing. The sides of the casing slant toward the recess into which the flange is fitted, so that the cushioning takes place gradually. The screw at the bottom, which regulates the cushioning, is troublesome. It has a square head for adjustment by a wrench. This is unhandy, since the wrench is frequently mislaid. A knurled handwheel with a simple check would save a lot of trouble. Air and oil are discharged from the cushion chamber, below the floor through a small hole in the bottom.

Another inverted pot is shown in Fig. 148. It has no ready adjustment for the ball-and-socket joint, the cap being held on by tapbolts. The cushion chamber is not recessed; air is drawn in through little ball check valves in the bottom which are liable to get out of order by dirt or gummy oil getting into them. It is evident that a partial vacuum will be created in this chamber every time

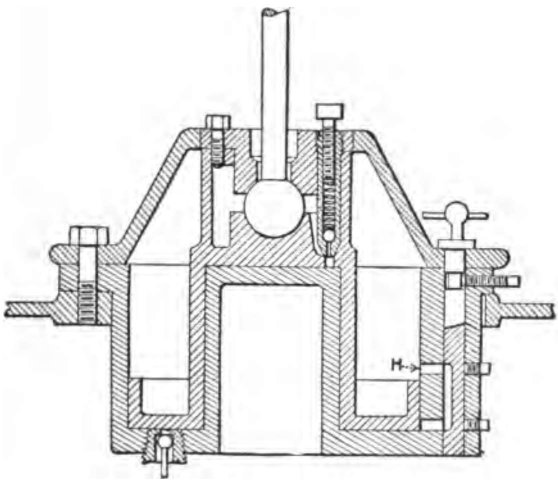


FIG. 148.—More elaborate inverted dashpot.

the pot is lifted, which would not be necessary if the regular vacuum chamber were made large enough to do the work. The amount of cushioning is regulated in a quick and effective way. On one side of the cushion chamber a hole is drilled vertically in an extension into which is fitted a straight plug having a groove milled on one side; this groove connects the two holes *H*, and by turning the plug the opening through the holes and, consequently, the cushioning can be closely regulated. This pot is about as nearly soundless as one can be made. There are no leathers required to counteract the shock of the descending plunger or pot, as on the other kinds. This is a great advantage, since the leathers wear out and have to be renewed.

The dashpots so far described are expensive and require

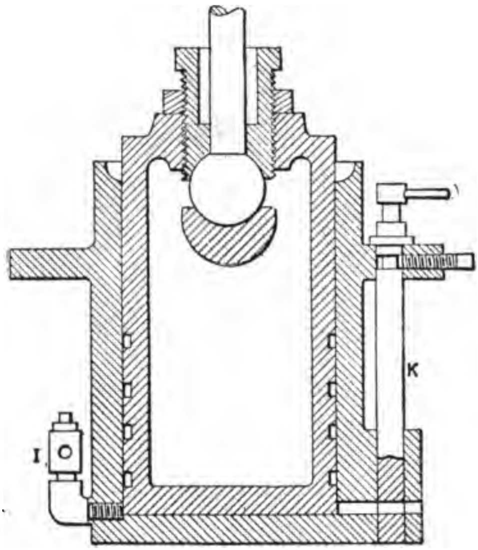


FIG. 149.—Simple plunger dashpot.

skill in manufacturing, on account of the exact fits which must be made on the two concentric parts of the pot and the plunger. If one of the parts is out of true, the device cannot work satisfactorily. To avoid some of the difficult fits, there are dashpots made of one plug and vacuum chamber without the extra cushioning part. Fig. 149 illustrates one of this kind. The purpose of the arrangement is that the lifting of the plunger will create a vacuum sufficient under all conditions to close the steam valve, and that in coming down the plunger will compress the air that has entered the pot, forming an effective cushion, the air to be discharged when the plunger is near the bottom. To accomplish this result, the diameter of the plunger must be comparatively large, since full atmospheric pressure cannot become effective as in the two-chamber pots. Some manufacturers persist in using single-plunger dashpots and obtain good results with them for steady loads, but for varying loads they are not satisfactory since they have to be adjusted every time the load changes considerably. If the snifting valve *I*, in Fig. 149, and the air-inlet valve *K* are set for a heavy load, the plunger will stick near the bottom when the load gets light, and if adjusted for a light load, it will hammer as soon as a heavy load is thrown on. Compressing air in the lower part of the pot with a blow has quite a heating effect. This can be seen if oil is poured around the plunger; as soon as it gets to the bottom of the pot, it will evaporate and be discharged in the form of a white vapor or smoke. Leather cannot be used in this pot to deaden the blow of the plunger, for it would soon be converted into charcoal.

A dashpot that made quite a sensation some years ago is shown in Fig. 150. It is provided with a differential plunger which in its upward movement compresses air above the larger part and forms a partial vacuum below. Air is forced through the holes drilled into an extension on the side of the pot, making a passage between the upper and the lower chambers, stopped by a taper plug. Most of the air will pass from the upper into the lower chamber in the upward motion, and on the down stroke will be forced back again, so that a continuous exchange of air takes place between the two chambers. On the down motion a partial vacuum will also be formed in the upper chamber and it is claimed this will have a retarding effect on the plunger. This pot is open to the same objections as the one with the single plunger and

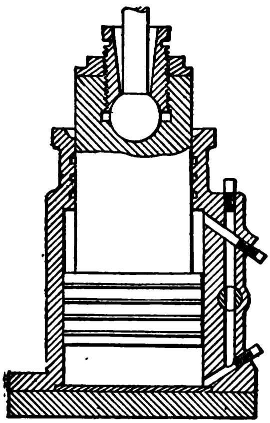


FIG. 150.—Differential plunger dashpot.

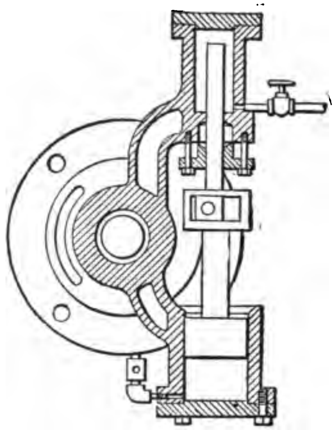


FIG. 151.—Steam dashpot.

chamber—it is deficient on varying loads and needs frequent adjustment.

Perhaps the most trying conditions for dashpots are those that prevail when a direct-connected engine is run on the friction load with, say, 150-lbs. pressure at the throttle and the brushes thrown off the commutator. Under these circumstances the steam valve will barely open, and the lift of the dashpot plunger is so small that it requires a high vacuum to force it down again after the valve gear has been tripped. Few dashpots will work under these conditions. It is unpleasant to see Corliss engines run on very light loads with full-pressure steam and the throttle wide open. It looks almost as if the valve gear were trying to pound itself to pieces. The steam pressure should be regulated to some extent according to the load, and every engineer who has any regard for his engine will attend to this. One of the requirements of a perfect-running valve gear is that the dashpots must always come down to the lowest position. For a varying load this is hard to attain without repeated adjustment, but there are times when the attendant cannot watch them, with the result that the valve gear gets noisy and troublesome. To overcome this, steam dashpots have been tried. The principle of such a pot is shown in Fig. 151. It consists of an upper steam cylinder and a cushion pot below, with the necessary piston for each. A small steam pipe with a stop valve is connected to the steam cylinder. The steam arm is attached to the block which slides in the slotted hole in the middle of the piston rod. The action is simple, and if everything is in working order it is positive. The valve gear pushes the steam piston up into the cylinder and the pressure of the steam forces it down again, as soon as cut-off occurs, till the cushion piston touches the bottom. There can be no doubt about the effectiveness of this device, but one has to contend with steam piping, stuffing-boxes, wear of steam piston, troublesome connection to the steam arm and also with varying steam pressures. Although in theory steam dashpots are good, in practice they have failed. They are hard on all the other parts of the valve gear.

Another pot is the spring dashpot, shown in Fig. 152. The weak point of this is that it relies on

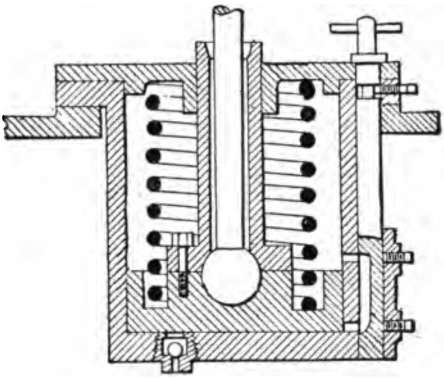


FIG. 152.—Spring-operated dashpot.

the action of the spiral spring to force the plunger down.

CHAPTER XI

GOVERNORS

Flyball governor for slow speed. Stepbearing. Weighted governor. Spring governors. Wearing parts of governor running in oil. Governor with one tension spring and roller bearings. Enclosed governor with oil pump. Inertia governor. Combination of centrifugal and inertia governor. Oil pots. Method of testing governor springs. Many accidents are caused by small pin. Safety lever. Antiquated collar on governor column. Change of speed. Governor drives. Fastenings of levers, gears, and pulleys. Equalizing cut-off on Corliss engines. Friction in cross-shafts. Unequal distribution of load in cylinders of compound engines. Adjustable levers. Emergency governor.

THE governor of an engine may be regarded as a permanent watchman, overlooking the performance of the apparatus with an observant eye. If more power is required, it drops, apparently on its own account, and lets the engine take more steam; and, as the load decreases, it rises and reduces the amount of steam to suit. We owe this device to the genius of James Watt. The old pendulum, or flyball, governor still holds its own. The principle on which its action is based is so simple and effective that it is still widely used. A few changes have been made to adapt it to various conditions. Engines designed by Watt ran slowly and so did the governors. As engine speeds increased, the governors were made to run faster and their size diminished.

Fig. 153 shows the customary form of flyball governor in use on many engines. The speed of these governors ranges from 50 to 60 r.p.m. Flyballs are from 6 to 9 in. diameter. The lift of the governor sleeve is from 3 to 4

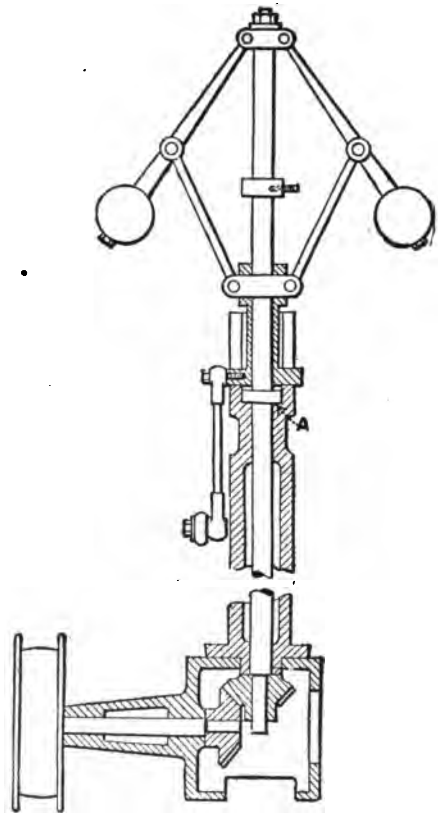


FIG. 153.—Common flyball governor.

in., with an additional drop of $1\frac{1}{4}$ in. to the safety stop. As the regulation of this governor is not close, it should be used only on engines for sawmills, stoneyards, machine shops, flour mills and factories with belt or rope transmission, where the friction load is a large percentage of the total and where close regulation is not essential. One of the troublesome features of the design is the collar *A* which supports the weight of the governor. If not always well oiled it will wear into its seat and change the height

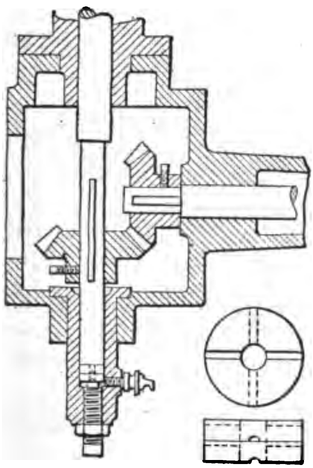


FIG. 154.—Step bearing for flyball governor.

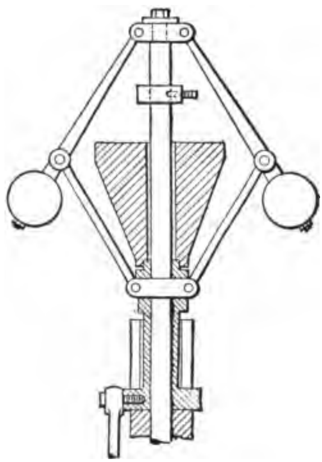


FIG. 155.—Weighted flyball governor.

of the governor and the cut-off of the valve gear. It may also cause the gears to mesh too deep and break the teeth. A ball bearing is sometimes placed under the collar, but this does not improve matters if the lubrication is insufficient.

To avoid the trouble with a collar, it is advisable to extend the spindle below the gear box and use a step bearing, as shown in Fig. 154. The end of the spindle should be hardened and ground. Two plates are to be placed under the spindle, the upper one of phosphor bronze and the lower of steel; each should have a hole in the center and oil grooves on both sides. The steel plate should rest on the end of a large set-screw, which will permit the governor to be adjusted for height and also to take up wear in the step. The set-screw should be secured by a check nut. The step bearing can be kept well oiled, since oil from the governor spindle and sleeve will drain into the bottom of the gear box; there is no danger of its running dry. It may be cleaned from time to time by pouring kerosene into it and draining it through a pipe-plug or small pet-cock near the bottom.

For engines that require closer regulation, run at a higher speed, and especially for compounds, a weighted governor like that shown in Fig. 155 is much used. By increasing weight and speed the governor is made more powerful and less liable to be affected by friction. The weight must not revolve with the spindle, it should simply rest on the sleeve. The hole should be at least $\frac{1}{8}$ in. larger than the spindle to prevent contact causing friction. Everything about the sleeve and the joints must be free and well-oiled, otherwise the governor will stick in some places and make the engine race.

Governors that are actuated by centrifugal force and weights are slow in action; to attain greater sensitiveness and closer regulation, spring governors are used on many modern engines. Fig. 156 shows an example, with a compression spring in the top of the governor, adjustable by two nuts on the sleeve. The balls are made either in one piece with the lever or separate, as shown in the detail. If separate, they may be adjusted for height and accurate balance. Spring governors are simple in their conception, but they should not be experimented with except by competent mechanics. One case is known where an engineer

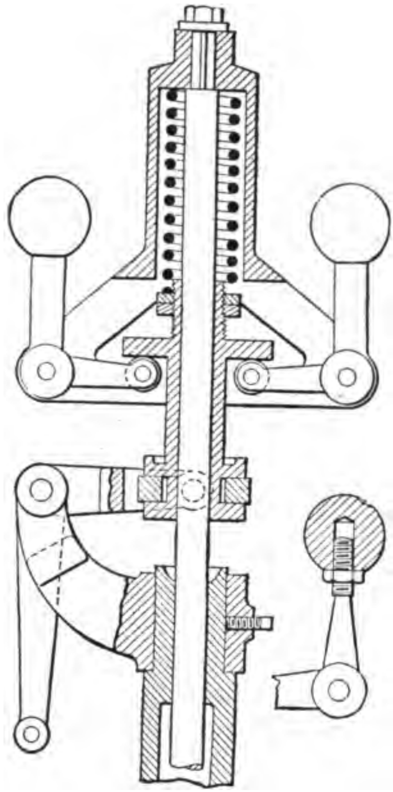


FIG. 156.—Flyball spring-loaded governor.

cut a couple of inches off the spring to make the engine run faster. He did not get exactly what he had expected and telegraphed the builders that their governor was no good and that they should immediately send a man to fix it.

The rollers at one end of the arms which actuate the sleeve have a tendency to wear flat on the circumference and grooves in the lugs of the sleeve, either of which will materially affect the reliability of the governor. The pins

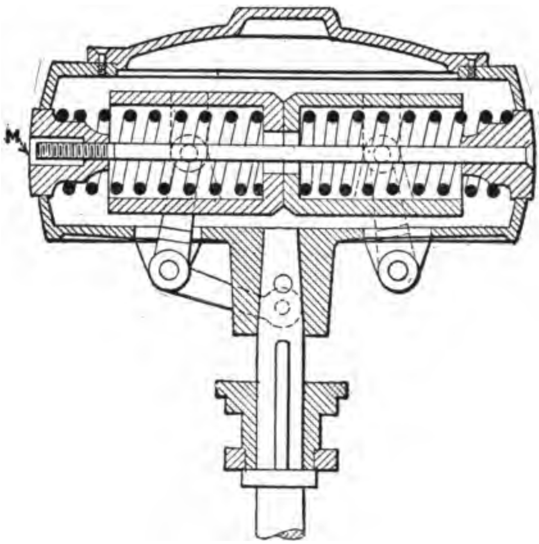


FIG. 157.—Enclosed spring governor.

in the arms are also subject to considerable wear on account of the heavy pressure exerted by the spring and the incessant oscillations of the balls. To obviate this defect, the governor shown in Fig. 157 has been designed. Instead of balls this has flat semicircular weights, each with a large hole in the center to receive a compression spring. The weights are supported by the arm by means of pins located in the same plane as their center of gravity. The centrifugal forces of the weights are directly opposed to the resisting forces of the springs, consequently there is no pressure due to either of these forces acting on the arms, and little wear on the pins. The springs can be adjusted, and the speed of the governor and engine varied

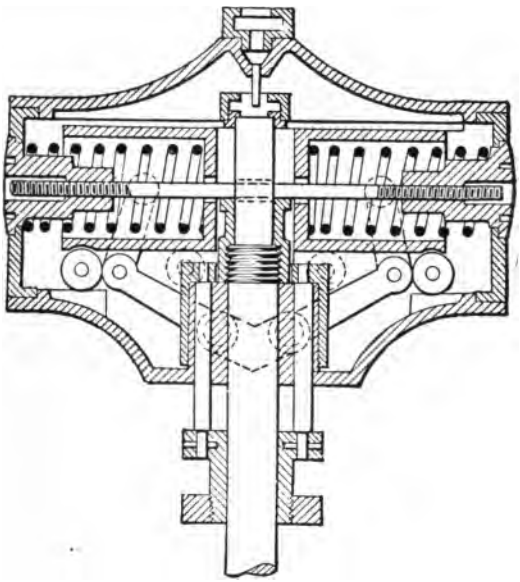


FIG. 158.—Enclosed spring governor with oil bath.

between certain limits by tightening or loosening the nut *M*, increasing or decreasing the pressure on the springs. On account of the high speed at which these governors run, it is almost impossible to keep the joints lubricated; a new supply of oil can be put on only when the governor is at rest, and this is soon thrown off when it is up to speed; there is no way of oiling the joints as long as it is in motion.

This deficiency has been overcome in the design shown in Fig. 158. The weights and springs are arranged as shown in Fig. 157, but the arms act on a sleeve connected to the collar by two studs passing, closely fitted, through holes in the hub of the casing, which extends well up into the inside and keeps the oil from running out. The weights are supported by rollers which run on planed surfaces of the casing. All parts in the lower half of the casing work in a bath of oil. The oiling of the upper pins, rollers and slides is effected through the cap on top while the governor is in motion. This governor is dust-proof and it cannot throw off any oil except that which works its way out through the two holes around the pins in the hub. For compactness and sensitiveness these inclosed-spring governors are about all that can be desired. The two springs are a disadvantage; unless they are evenly matched the governor is liable to be out of balance. Compression springs have a tendency to buckle if the ends are not truly square. It is advisable to pay attention to this before inserting the springs.

For testing the springs platform scales can be arranged inexpensively in the following manner: Two iron bars 2 in. wide by 1 in. thick, each about 4 in. longer than the base of the scales, are placed as shown in Fig. 159. They should have holes drilled in the end for two $\frac{5}{8}$-in. bolts with long threads and two nuts at each end to clamp the bars. The upper bar must be provided in the center with a long screw about $\frac{7}{8}$ in. diameter. The spring to be tested is placed on the platform, resting on a casting with a projection for the spring to fit over, as shown in detail. A similar casting is placed on top of the spring except that this one has a countersunk hole drilled in the upper side into which the pointed end of the set-screw fits. With everything in place, the scales are balanced without putting any pressure on the spring; the reading on the beam represents the weight of the bars, rods, screws, etc., and the spring. If the length of the spring was measured when free, the deflection of the same caused by the weight of the bars, etc., can be noted. The set-screw should now be turned one revolution at a time and the pressure after

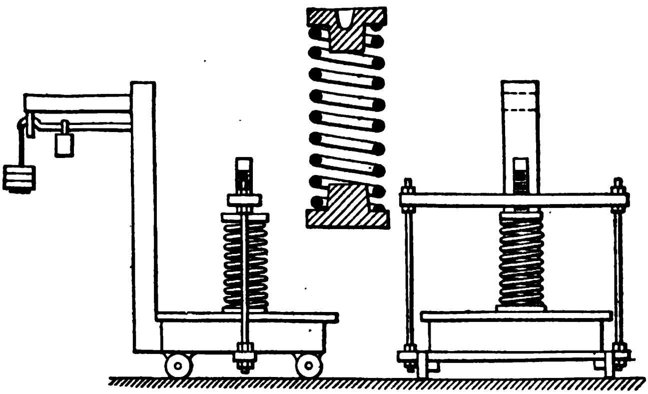


FIG. 159.—Testing springs on platform scale.

each turn recorded. For a $\frac{7}{8}$-in. screw it will take nine turns to compress the spring 1 in.; the difference between the first reading and the one after nine turns is called the stiffness of the spring, meaning, of course, that it takes that many pounds to compress it 1 in. The test must be continued for the full working range of the spring, after which the readings for each turn of the screw can be compared. For a good spring the increase in pressure for each turn must be uniform.

To obviate trouble arising from the use of two compression springs, the design shown in Fig. 160, using one tension spring, has been introduced. The speed can be adjusted by changing the tension of the spring. The

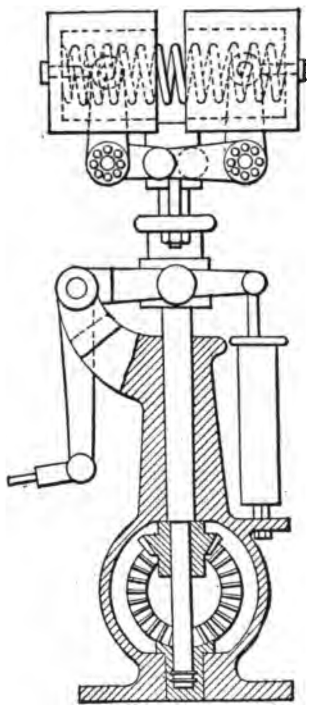


FIG. 160.—Tension spring governor.

arrangement of the weights is about the same as in Fig. 157, but there is no outside casing; everything is in plain view and can be watched. Roller bearings are provided for the pins in the weights as well as in the arms, reducing the friction to a minimum. The roller bearings retain the oil for a long time, so there is not much need for a fresh supply, and hardly any oil can be thrown. The

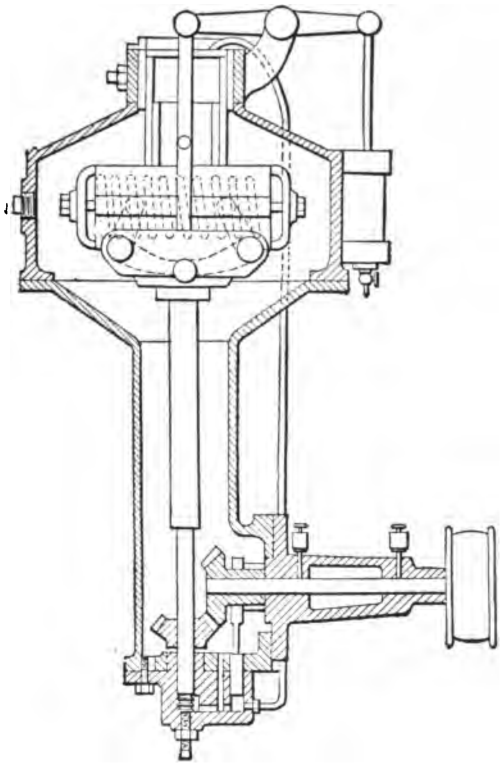


FIG. 161.—Enclosed tension spring governor.

governor spindle is supported in a step bearing without any adjustment for wear, and the gear on the spindle is located above its mate, which is subject to the same objection as raised in regard to Fig. 153.

Another centrifugal-spring governor with one tension spring is illustrated in Fig. 161. In this design all the rotating parts are inclosed in a stationary casing, making it impossible for oil to be thrown on the engine and floor or on the generator, if the engine is direct-connected, The lubrication is effected by a small pump located at the lowest point near the step bearing. When the governor is running and the pump is in working order, an abundance of oil will be forced to the top of the casing and from there over the working parts. This way of

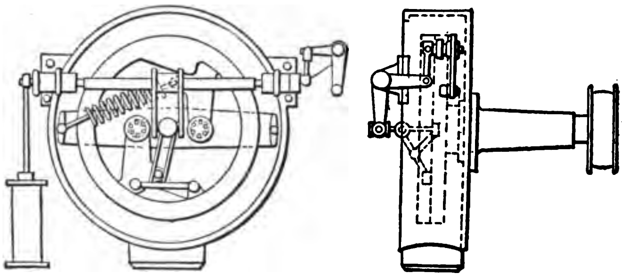


FIG. 162.—Inertia governor.

oiling is of doubtful value, for, if the pump gets out of order, the governor will have to run without oil until somebody discovers the trouble. It would be far better to take the oil supply from a continuous gravity oiling system with a stream of oil running into the top of the governor.

Inertia influences the moving parts of a governor to a great extent. Its action in the flyball and centrifugal spring types described is detrimental, as it either unduly retards or accelerates the motion of the balls and weights. In an appropriate design, however, it can be used to advantage. Fig. 162 shows an inertia governor that is in use on a number of engines, mostly direct-connected, noteworthy for close regulation on varying loads. In this design no gears are used for rotating the governor weights, also no vertical spindle nor step bearing. The shaft on which the pulley is located also supports the governor weights; there are only two bearings on the shaft to be looked after. The weights are annular, pivoted on a double arm keyed to the shaft; a spiral tension spring holds them in position, and a link forms a connection between the two. As the governor begins to rotate, the centrifugal force tends to move the weights away from the center, thus actuating an arm which in turn rotates a shaft and thereby controls the governor rods and cut-off levers by a series of intermediate levers and links. The many levers, bell-cranks, pins, links, ball-and-knuckle joints on this governor are its weak points. The revolving parts are protected by a substantial case, which also serves as an oil shield. This type closely resembles shaft governors on high-speed engines and, like them, has the pivots for the weights, which are liable to give trouble on account of friction and excessive wear, even if provided with roller bearings.

The construction shown in Fig. 163 represents a combination of a centrifugal and an inertia governor, the weights *A* forming the centrifugal and the wheel the inertia part. A tension spring located on a plane with the center of gravity of the weights counteracts the centrifugal and inertia forces. The wheel *C* is supported on a collar of the governor spindle on which it can rotate. The vertical pins *D* connect the wheel with the weights *A* by brackets which slide up and down over the pins, according to the outward or inward motion of the weights.

The action of the governor is as follows: With the engine running at normal speed the weights, wheel and spring will be in equilibrium, but when the engine slows down or speeds up, the governor spindle will follow suit. The weights will remain for an instant in their position, but the wheel, being free to move on the spindle, will

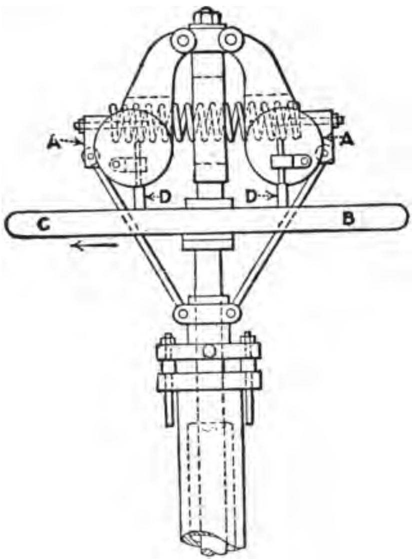
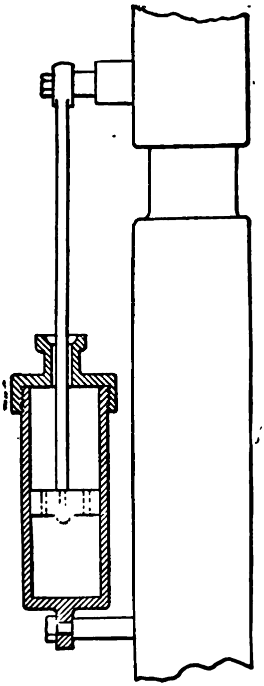


FIG. 163.—Combination centrifugal and inertia governor.

continue at the former speed, and force the governor weights inward, thus admitting more steam to the cylinder when the engine slows down and lags behind, with the opposite result, when it speeds up. The arrangement is ingenious and should give good results if everything is in working order, but here again one has to deal with a multiplicity of parts. The pins of the inertia wheel are liable to stick in the little brackets on the weights, or the wheel itself may bind on the spindle; in either event much trouble will follow. Complications on a governor should be avoided. All the parts of this



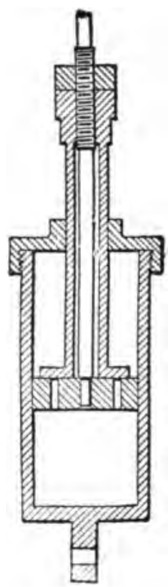


FIG. 164.—Non-adjustable oilpot. FIG. 165.—Adjustable oilpot.

one are exposed to view, but there is no provision made to prevent oil from being thrown off. If the load on an engine fluctuates considerably, the governor is liable to jump. Owing to the inertia of its parts it has a tendency to overtravel its true position on an instantaneous change of load. This may become troublesome unless an oil pot be provided to prevent violent motion.

A simple oil pot in use on many engines is shown in Fig. 164. To let the oil pass through, a number of holes are drilled in the piston, some of which may be plugged, according to the resistance required. The device is simple, but it has no outside adjustment. Fig. 165 shows the same style of pot which may be adjusted from the outside. The holes in the piston can be closed or opened to any degree desired by lowering or raising the sleeve. This makes a good oil pot.

Oil pots like that shown in Fig. 166, in which the flow of the oil is regulated by a valve in a pipe connection on the outside, are likely to give trouble on account of air which may accumulate at the top. If there is any air in the pot, it will be alternately compressed and expanded and make the governor unsteady. There should be a little valve at the top to let out air and take in oil. A stuffingbox is needed for this pot, which, of course, increases the friction. The oil pot shown in Fig. 167 has some good features. As it is made of brass it is not subject to rust like pots made of cast iron. The piston has a groove turned on the outside with four holes at right angles to each other drilled toward the center. The piston rod is made of a piece of brass tubing riveted into the piston with two holes drilled through, one above the other, close to the piston. A small piston valve inside of the tube controls the passage of oil through these holes and is adjusted by a screw and check nut on the top of the tube. The object of the groove and the four holes in the piston is to cause the oil to circulate around the circumference and make it tight. The kind of oil to

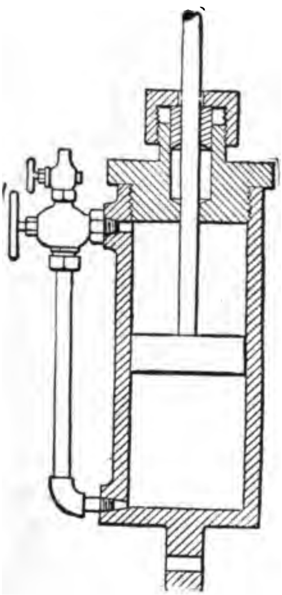


FIG. 166.—Oilpot regulated by valve.

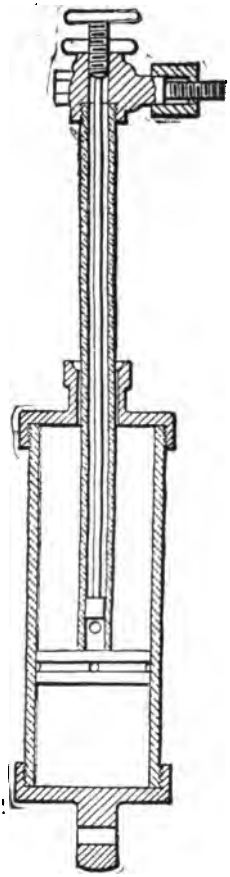


FIG. 167.—Adjustable oilpot with relief ports.

use in an oil pot must be learned by experience. It depends of course on the locality and on the season. That used in winter must be frostproof. Glycerine has been substituted in some cases, in others kerosene or a mixture of kerosene and machine oil. It depends on the prevailing conditions. All such devices must be watched to avoid gumming and sticking.

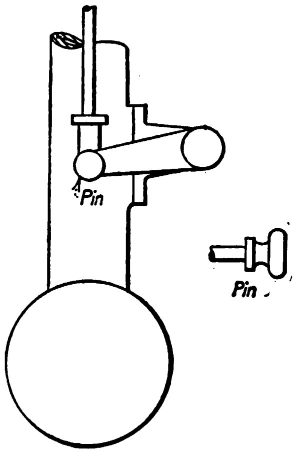


FIG. 168.—Small pin which has caused many accidents.

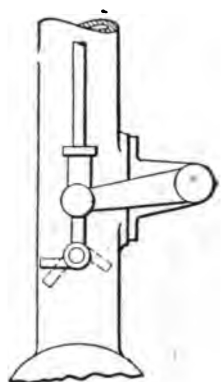


FIG. 169.—Safety lever.

Some accidents to Corliss engines have been caused by a small pin used on the governor, shown in Fig. 168. In starting the engine the governor is raised sufficiently to insert the pin in a hole in the governor column. With the governor in this position the engine will take steam full stroke. As the speed increases the lever will be lifted off the pin and the engine will begin to cut off. That is the time the pin should be taken out of the hole, but some engineers leave it in, and if anything should happen to the governor belt or gears to stop the governor, the lever will drop down on the pin, the engine take steam full stroke and will probably run away, since the pin prevents the governor from throwing in the safety cams. If, instead of the pin, a little lever is substituted, as shown in

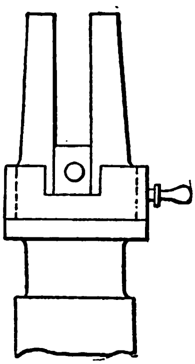


FIG. 170.—Unsafe collar for governor.

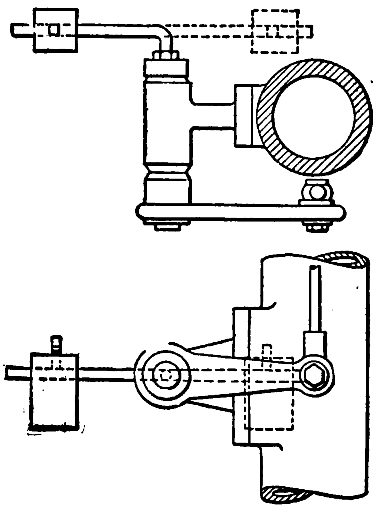


FIG. 171.—Small weight for change of speed.

Fig. 169, it will automatically drop into the position shown in dotted lines as soon as the governor rises, and the danger will be avoided.

A still more antiquated design than the little pin is shown in Fig. 170. It consists of a collar located on the governor column, having a slot into which the governor sleeve may drop provided it is in the right position.

If the slot is not in the right place the governor cannot drop sufficiently to make the safety cams operative. Wherever that ominous little pin or the collar is still in use on an engine, the engineer and proprietor had better be cautious. No insurance company will tolerate either one of

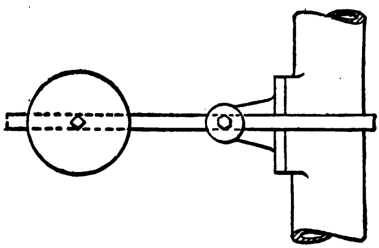


FIG. 172.—Too heavy weight makes governor sluggish.

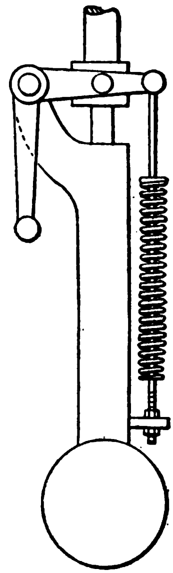


FIG. 173.—Spring used instead of weight.

these contrivances. Some engine builders are using the collar with a spring inside to throw it automatically into the proper position as soon as the governor rises. This, of course, eliminates the danger if everything is in good working order.

It is often desirable to change the speed of an engine, and if the change only amounts to two or three revolutions a minute, a lever and weight can be added, as shown in Fig. 171. If arranged as shown in full lines, the speed of the engine will be decreased, and if as in dotted lines, it will be increased. A large weight on a long lever, as shown in Fig. 172, making a considerable change of speed possible will render a governor sluggish and its action uncertain on account of the inertia of the parts. A spiral spring will give better results; it must be located in a central position to the governor, as indicated in Fig. 173. Attaching a heavy weight or a spring to the side of the sleeve or to the governor lever will increase the friction and make its action unreliable. If the speed of an engine is to be permanently changed, say ten or more revolutions per minute, it will be advisable to change the governor pulley, making it larger in diameter for higher speeds and smaller for slower. A change in the ratio of the gears will give the same result. Change of speed by means of frictional drives using conical pulleys or friction disks should be used only in extreme cases where the variation in speed amounts to 40 to 50 per cent.

There are more belt drives in use on governors than all other drives combined, in spite of the fact that every designer and engineer knows the short-comings of a belt. It will stretch, break, slip, run off, run crooked, get soaked with oil, etc.; still, we find it in use on almost every engine. The reasons for this are that it is cheap, noiseless, easily put on and repaired; it is always in plain view and can be watched and replaced if it shows signs of weakness. At the same time it is a constant source of worry.

Gear drives came into use some years ago, but most of them have been discarded. They were expensive, required a lot of lubrication, the bearings ran hot, the babbitt would wear out and the teeth of the driving gear on the main shaft would break.

On some large engines, where the distance between the main shaft and the governor is considerable, ropes are used, generally three of them. They are, of course, never of the same length and have not the same tension—one rope will do most of the driving while the others do hardly any work. If a rope gets too slack, it must be respliced, which requires considerable skill. If carelessly done every time a splice runs over the sheave there will be a jerk on the governor, causing an irregular speed.

Silent-chain drives are in use on some engines that require close regulation. They must be carefully covered to prevent dirt getting into the chain; if anything gets out of order the engine may be thrown out of service for quite a while. It is advisable to have spare parts of the chain and extra sets of chain wheels on hand, to be prepared for an emergency.

The trouble with some belt drives for governors is that the belt is of inferior quality, of insufficient width, or the governor pulley is too small. On many engines the governor belt runs directly on the main shaft, but for high-speed governors a pulley is used, which must be well secured. If it ever becomes loose and slips, there may be just enough friction between shaft and pulley to keep the governor revolving, preventing it from throwing in the safety-cams, but admitting more steam than is required, making the engine speed up and perhaps run away. A double leather belt should be used, not less than 3 in. wide. It must be endless, even and tight, without metal joints or lacing, and it must be kept free from oil.

Considerable trouble is experienced with governor levers, gears and pulleys coming loose; all of these parts must be secured in the most approved manner. It is best to use two ways of fastening, so that if one should fail the other will insure safety. A poor way to hold a lever on a shaft is by means of a taper pin; next to that is the use of only one set-screw. Fig. 174 shows a good

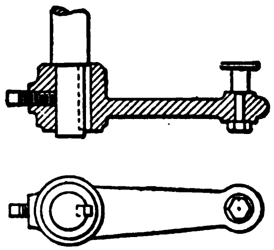


FIG. 174.—Good fastening for governor levers.

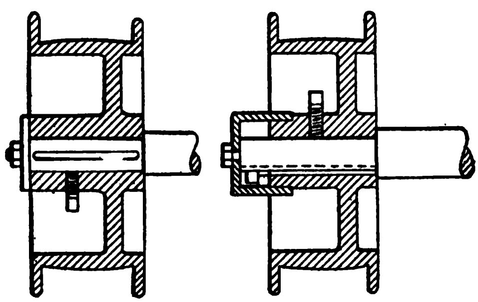


FIG. 175.—Method of fastening governor pulleys.

way to fasten governor levers, by a feather and a cupped case-hardened set-screw. The same method is used in Fig. 175 to hold the governor pulley securely on its shaft. As an extra precaution, a washer with stud and nut keeps the pulley from working off. A driven key may be used in the pulley hub, in which case a cap should be provided, held in place by a stud and nut to prevent the key getting loose and to protect the attendant from being caught by the key. Fig. 176 shows an approved way of holding the

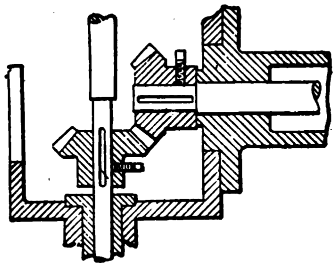


FIG. 176.—Gear fastenings on shaft and spindle.

gears on the shaft and spindle. Feathers and set-screw together are also employed, the driven gear being placed below, which is the safest position for it.

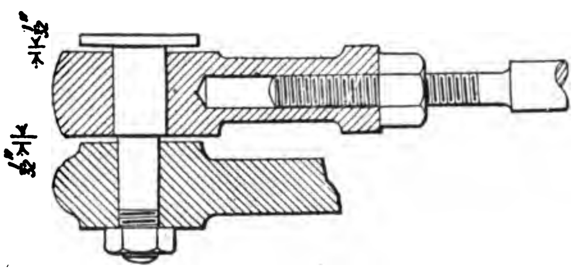


FIG. 177.—Stub ends of rods should never fit tightly.

The stub ends of governor rods should have about $\frac{1}{32}$ in. play on each side, as shown in Fig. 177; under no circumstances should they fit tight between the lever and the head of the pin. The lift of a flyball governor is usually limited by a collar on the spindle, as shown in Fig. 178. This collar must be placed high enough so that the governor cams will trip the valve gear and prevent steam entering the cylinder with the sleeve or weight touching the collar. A set-screw let deep into the spindle and screwed up tight should secure the collar; if it slips down and keeps the governor from rising to the highest

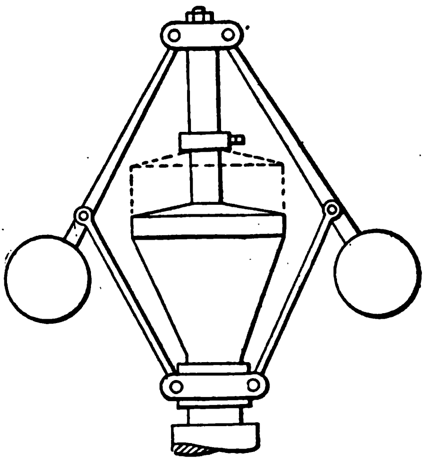


FIG. 178.—The lift of a flyball governor is limited by a collar.

position, the chances are that the engine will run away on a light load.

All the parts which connect governor and valve gear must be as light as is consistent with safety; therefore, of the arrangements shown in Fig. 179, the one marked *A* is the better. The double lever in *B* makes the upper rod in some engines long and heavy. It may begin to vibrate and interfere with the cut-off on the head end. If the rod is so long that it will sag even if made of a piece of pipe, it must be cut in two and supported by a carrier, thus increasing the friction.

Especial care should be used to locate the pins in the cam levers in such positions that cut-off at both ends of the

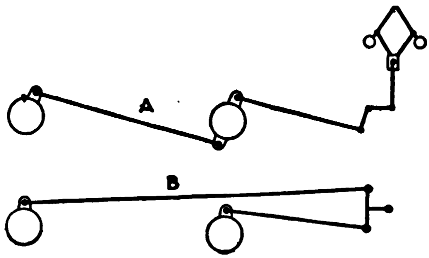


FIG. 179.—Arrangement of governor rods.

cylinder is equal for any load, or at least as near so as the angularity of connecting-rod and eccentric rod will allow. For equal cut-off the crank pin travels through a greater arc on the crank end than on the head end. In

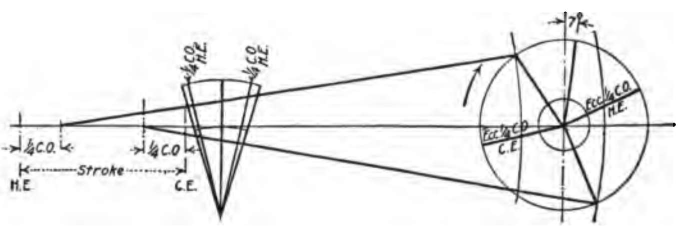


FIG. 180.—Travel of crankpin for ¼ cutoff.

Fig. 180 the arcs are shown for one-quarter cut-off. The difference in travel should be compensated in the valve gear and setting of the governor-cams. Using the ordinary rocker arm of a single-eccentric engine and an angular advance of about $7°$ for the eccentric, the positions of the rocker arm show that here the conditions are reversed in relation to the piston travel. In other words, the rocker arm moves slowly with the piston moving fast at the head end and it moves fast with the piston moving slowly at the crank end; provided, of course, that the motion of the crankpin is uniform. The cam on the head end should, therefore, be made to travel through a greater arc for a given cut-off than the one on the crank end.

The usual way to lay out valve gears is to assume that the angles through which the crank and eccentric travel are alike for equal cut-off, and it keeps an engineer guessing why he has to change the governor rods almost every time he indicates the engine, in order to equalize the load at both ends of the cylinder. Unless the angularity of the connecting-rod and eccentric rod is taken into account, the ordinary Corliss valve gear will give equal cut-off for only one position of the governor, that being the one for which it is adjusted by blocking. For all other positions the cut-off at both ends of the cylinder will be unequal. If an engineer will go through all the phases of indicating his engine, from the latest cut-off to the earliest by lifting the governor one-quarter inch at a time and blocking it, he will get an instructive set of diagrams. Trying to equalize the cut-off on an engine without blocking the governor is, of course, useless unless the load is steady.

On cross-compound engines trouble with the regulation may be caused by friction in the bearings of the cross-shaft. Even if this shaft be made of a piece of pipe, there will be a certain amount of deflection. The bearings should, therefore, have spherical seats so they can adjust themselves. Fig. 181 shows a simple arrangement that may be used to advantage on both sides of a cross-compound engine. The spherical part of the bearing gives it a chance to accommodate itself to defects of the cross-shaft due to deflection or bad alignment.

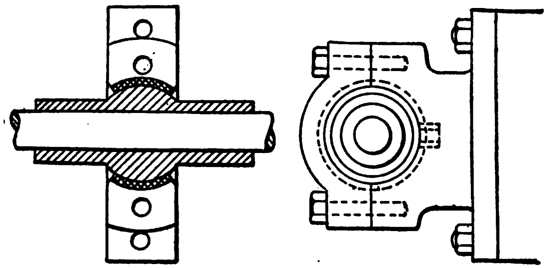


FIG. 181.—Spherical bearing for governor cross-shaft.

Unequal distribution of the load on the two cylinders of compound engines is often very troublesome. It is generally due to faulty construction of the governor

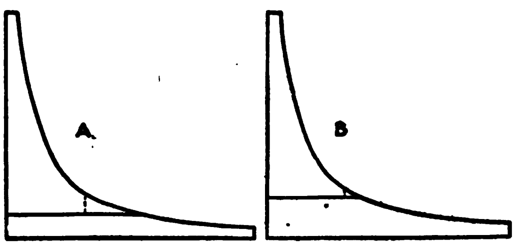


FIG. 182.—Combined cards of a compound engine.

levers. Referring to the two diagrams in Fig. 182, showing the combined cards of a compound engine, *A* indicates that the high-pressure cylinder carries a greater load than the low-pressure. To equalize the burden the cut-off in the low-pressure cylinder must be shortened, which will raise the receiver pressure and distribute the load equally, as shown in diagram *B*. The simplest way to accomplish this is to lengthen the governor lever on the low-pressure side.

The levers shown in Fig. 183 will answer the purpose. One of them has a number of holes drilled in the end; the pin can be changed from one hole to another to raise or lower the receiver pressure. A more accurate adjust-

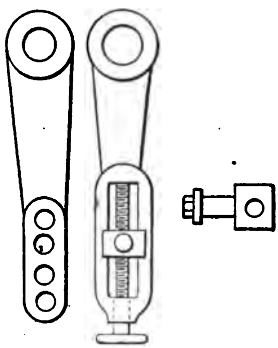


FIG. 183.—Slotted governor lever.

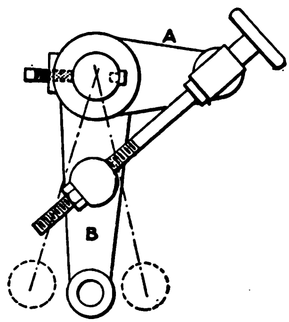


FIG. 184.—Adjusting device for governor lever.

ment can be made with the other lever, which has a slotted end provided with a screw and handwheel. The position of the pin on this one can be changed gradually without disconnecting the rod. If a double lever be used, the adjustment must be on both ends. The same result is sometimes obtained by moving the low-pressure governor lever ahead or back without changing its length, which can be done by an arrangement shown in Fig. 184. There are two levers used in this case. *A* is secured on the shaft, while *B* is loose and adjusted for position by means of a screw. For the greatest possible refinement this device is sometimes furnished with a slotted end on the loose lever.

It must always be understood that a governor will regulate the speed of an engine only if everything is in working order, and it is the engineer's business to keep it so; but even the most untiring care and vigilance cannot prevent the breaking of a stub end or rod, slipping of

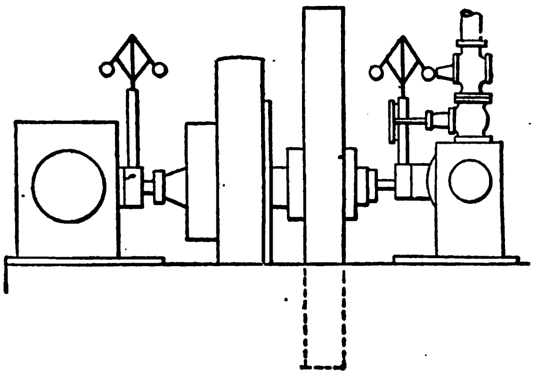


FIG. 185.—Cross-compound engine with two governors.

the belt, breaking of levers and pins and other accidents, which will endanger life and buildings. It seems, therefore, that one governor alone should not be relied on, but that the engine should be provided with a second device which would act and prevent it from running away if the regular governor should fail.

Fig. 185 shows a cross-compound engine with two governors—the one on the low-pressure side to regulate the cut-off of the engine and the one on the high-pressure side to operate an emergency valve should the engine exceed its normal speed by from five to ten revolutions. The second governor will pull a latch on the stem of the emergency valve, releasing a weight which will close the valve and shut off steam from the engine.

CHAPTER XII

RELEASING GEARS

Old crab claw. Reynold's radial gear. Other radial gears. Overhung pins and hookblocks. Straight springs. Trip motions with rollers and cams. Gravity trip motion. Valve-gear catch plates.

The old crab claw, Fig. 186, is disappearing. It has done good service on slow-speed engines running with a steady load and using low-steam pressures. The trouble with it is that the catch plates and pins are subject to excessive wear, which makes it expensive to keep in repair.

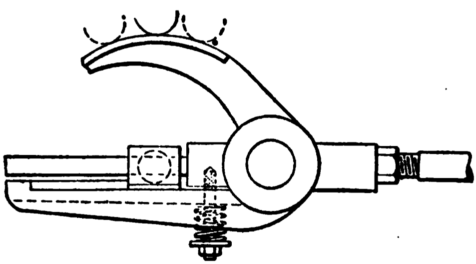


FIG. 186.—Crab claw.

On account of its weight the crab claw is liable to miss hooking on. The leverage on the curved arm changes for different cut-offs, as indicated in dotted lines in the sketch, throwing a variable strain on the governor.

The radial valve gear, Fig. 187, originated by Edwin Reynolds, has some decided advantages over the crab-claw gear, although it, too, is becoming obsolete. The hook used to be made of bronze and was liable to break; the hole for the pin wore out of round, making the valve gear noisy and unreliable. The hook should, therefore, be made of forged steel, and the hole should have a bronze bushing that can be replaced when worn. The curved spring is troublesome, for it is almost impossible to make two springs of this kind alike in shape and temper; some of them will last a lifetime, others break after a short while. If they are too stiff, they will throw a greater strain on the

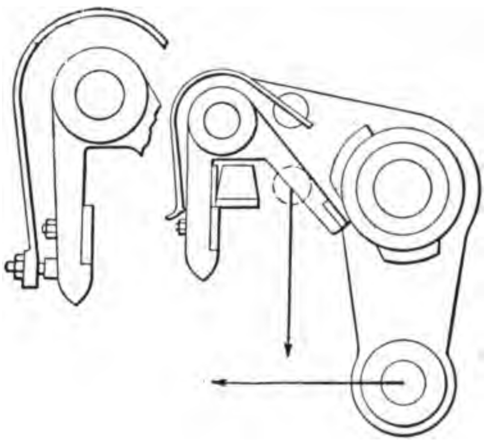


FIG. 187.—Reynolds' radial gear.

governor, and if too weak, the catch plates will slip and wear off rapidly until they will not pick up. It is common practice to push a piece of leather under the spring and exert more pressure on the hook and to change the catch plates as soon as the engine can be shut down. A spring adjustable by a set-screw at the extreme end which bears against the hook, as shown in detail, is being used. This may or may not be an improvement; it depends on the pressure put on the hook. If the engineer does not use good judgment, but tries to make sure of hooking up even if the catch plates are worn a little and exerts all the pressure the spring will stand, the shock on the governor to release the hook will be excessive and the strain on the valve gear much increased. As the toe plate on the trip lever of the hooks wears, the engine will take more steam for the same position of the governor and its speed will increase. This should be watched for and the plates be changed to another edge when the wear becomes notice-

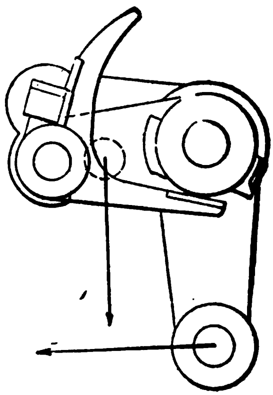


FIG. 188.—Hook located on the steam arm.

able. A piece of leather on the hook is often used to control the lap of the catch plates, but it will wear by hammering on the hook and should be renewed from time to time.

Notwithstanding its shortcomings, the Reynolds radial gear has been copied by many engine builders and modified by some; in fact nearly all Corliss releasing gears are based on the radial principle. The designer of the arrangement shown in Fig. 188 got himself into trouble by locating the hook on the steam arm with the dashpot rod. The hook block is fastened in the steam bell crank, the hook is reversed and the action is the opposite to that of the Reynolds gear. On higher speeds and variable loads the design was found to be defective. The hook would shake loose on its pin and fail in picking up, owing to the inertia of the trip lever. So the hook was changed to the position shown in Fig. 189, which corrected the trouble. The pickup takes place on the inside of the hook block, and the inertia of the trip lever assists in hooking up.

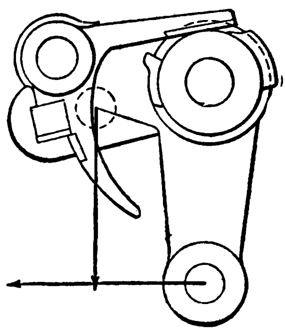


FIG. 189.—Correct position of the hook on the steam lever.

An improvement on the original radial gear is shown in Fig. 190. One of the catch plates is supported directly on the under side of the steam arm, without the use of a hook block. This throws the strain in direct line with the arm, without any lever action as on the gears just described. Overhung pins and hook blocks are likely to become troublesome, since the strains act on a leverage instead of in a straight line. The consequence is that the parts wear out, especially the catch plates, requiring frequent change of cutting edges and renewal. The hook in Fig. 190 is bolted to a rockshaft which rotates in a long bearing in one arm of the steam bell crank and carries at the other end a trip lever with a roller. The knockoff

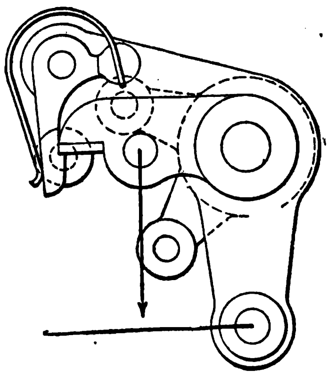


FIG. 190.—Improved radial valve gear.

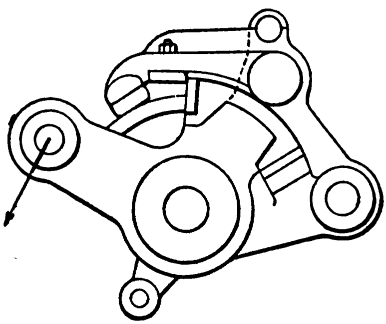


FIG. 191.—Gravity valve gear.

lever is provided with rollers instead of cams, as on the Reynolds gear. The only objection to this design is the curved spring.

Fig. 191 shows a simplified hook with an extension on one side near the end, which acts as a trip lever. In tripping a side strain is thrown on the hook, which will wear the hole out of round. The hook acts partly by gravity; a light spring is also used to prevent the catch plate slipping at high speed. The extension on the hook carries a piece of fiber which rides on the knockoff lever and trips the valve gear. This is a doubtful improvement; it makes the valve gear almost noiseless, but shows

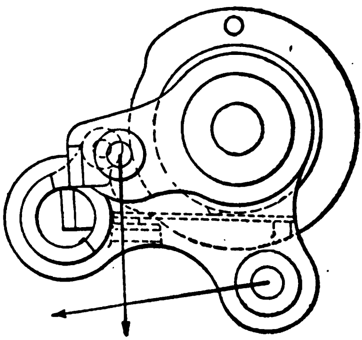


FIG. 192.—Valve gear using a straight spring.

considerable wear after a short time. With a little care this valve gear will give good service for a long time.

The aim of valve-gear designers is to bring all strains as nearly as possible in the same plane and to do away with curved springs. Fig. 192 shows the first successful gear of this kind, which is still much used. There is no hook, and the end of the rockshaft is simply milled out to receive the catch plate. The latter engages its mate, which is screwed to the outer end of the steam arm. Both plates are placed in a vertical position instead of one horizontally and the other vertically. This has its advantage, as will be explained later. With the catch plate on the extreme end of the steam arm, the dashpot pin can be moved out, thereby increasing the lift of the pot and the leverage of the arm, making it more effective, especially for double-ported valves and high steam pressures. There is no appreciable wear of the rockshaft; even after years of service it will remain in excellent condition if the oiling is attended to, because it has full support under the catch plate where the greatest strain is applied. The trip lever at the other end of the rockshaft is provided with a roller that strikes a cam on the knockoff lever. The weak point of this valve gear is the insignificant straight spring that is forced in between two fixed points on the steam bell crank and a lug on the knockoff lever. A heavy initial strain is put on the spring, and as its working length is short it will soon break. A supply of these springs must be kept on hand; if one breaks, a new one can be slipped in place without stopping the engine, and even two, one on top of the other, may be used if one is not strong enough. The roller at the end of the knockoff lever makes this valve gear somewhat noisy when it is run at high speeds.

Fig. 193 shows a similar design, but instead of extending the rockshaft bearing to the outer edge of the lever carrying the catch plate, it is increased in diameter and provided with an adjusting screw by which the lap of the catch plates can be regulated to any desired degree. If the edges wear a little, the lap can be increased, which is often a great convenience. There are no leathers used for regulating the pick-up. A substantial straight spring, not subject to frequent breaking, makes the valve gear hook up, and a straight trip lever effects the release by striking a fiber block. On this valve gear the steam arm is supported and bears in the bonnet, which is unusual. It has the good effect that the valve stem is relieved of all transverse strain and wear at the outer end. The leverage of the dashpot rod pin is short, therefore, it requires powerful dashpots to close the steam valves tightly. On light loads and with high steam pressure the pots will stick, causing the valve gear to rattle disagreeably. If the catch plate on the steam arm should be placed vertically,

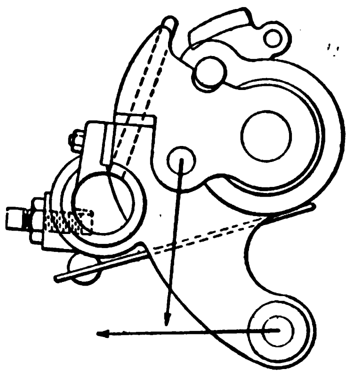


FIG. 193.—Adjusting screw for catch plate is used.

as in Fig. 192, the pin for the dashpot rod could be moved out considerably and the trouble overcome.

Curved or straight springs used on trip motions give more or less trouble and should be avoided if possible. Fig. 194 shows a design which engages the catch plates by gravity and accomplishes the knockoff by a cam and roller without the use of a spring. One of the catch plates is fastened to a long bolt with a sliding motion in the steam arm. For high speeds this gear does not seem to be suitable, the heavy bolt and plate make its action sluggish;

besides, the hardened block wears a shoulder on the steam arm and this in the course of time will impair its action.

A modification of this gear is shown in Fig. 195. The catch plates are transferred from below to above the arm, the sliding bolt is made shorter and lighter and the cut-off is effected by a trip lever with a roller. The catch plates engage by gravity. There is an auxiliary light spring provided to accelerate the motion of the sliding bolt, but

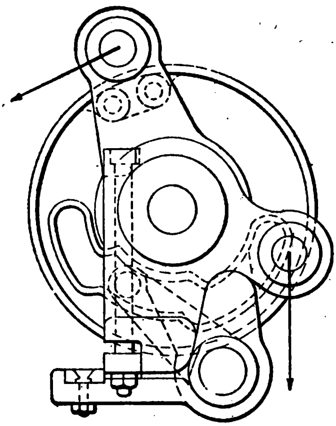


FIG. 194.—Gravity valve gear with cam and roller.

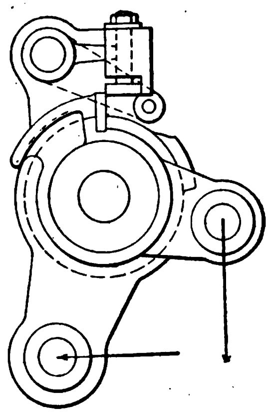


FIG. 195.—Gravity valve gear with trip lever.

this is found to be necessary only on high speeds. The appearance of the whole design is pleasing, but the short sliding bolts give trouble after a short while and the action becomes uncertain, when the catch plates fail to engage occasionally, with the result that the steam valve does not open.

The design shown in Fig. 196 is rather complicated, but no spring is used and the latch and lever carrying the catch plates are well supported; the trip lever is connected to a flat steel arm having two cam surfaces milled on the sides, the motion of which is controlled by two rollers on the knockoff lever. The action seems to be quite positive and satisfactory. The rollers will become noisy after considerable service, especially on higher speeds.

In Fig. 197 the trip arm, trip lever and rockshaft are located on the top of the steam bell crank and arm. There

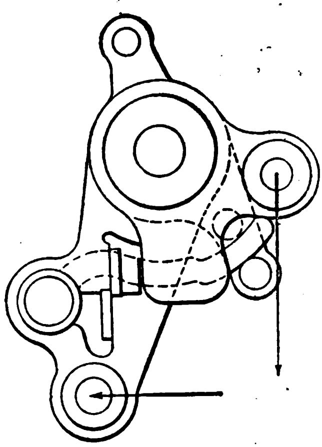


Fig. 196.—Gravity valve gear with curved trip lever and rollers.

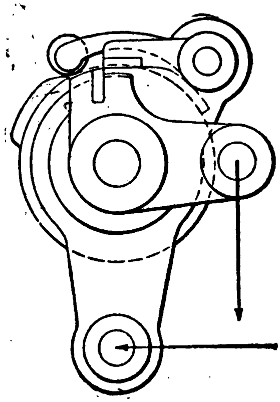


Fig. 197.—A serviceable gear.

are no springs, delicate cam levers or slotted plates used, which will wear out and are expensive; everything works by gravity. The rockshaft has a long bearing in the bell crank and the knockoff cams and safety cams are easily replaced if worn. This valve gear will give good service if properly attended to.

The valve gear shown in Fig. 198 differs from others in its method of tripping the catch plates by a separate motion imparted to a slotted knockoff lever. The motion is derived from an eccentric on the main shaft and is carried through a rocker arm on the governor to the valve gear. There are numerous levers, pins, links and rollers employed, which, of course, will wear out, make the gear noisy, require a lot of lubrication and attention and become generally troublesome. The inertia due to the oscillation of the knockoff levers and connections is a disturb-

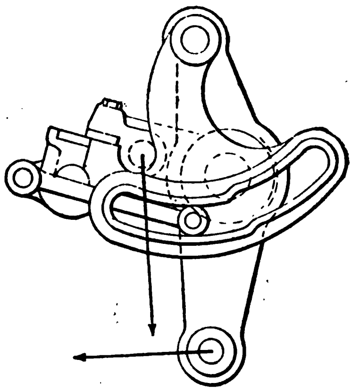


FIG. 198.—Valve gear having a separate motion for knockoff lever.

ing factor. As all the pins and rollers of this trip motion must be hardened and ground and the pivots for the governor connections are hardened and have to run in hardened tool-steel bushings, trouble may be expected. For American practice, where simplicity, reliability, durability and ease of repairs are of paramount importance, this valve gear can hardly be recommended. The claim that it is superior to other long-range gears on account of using only one wristplate while others use two, can hardly be substantiated at the present time, since some Corliss engines have no wristplate at all.

Naturally, the parts that are subject to the greatest shocks and wear on a valve gear are the two catch plates, which engage on a narrow edge every time the valve gear hooks up. On new engines where the steam valves have not yet come to a proper seat, these plates are often overstrained and break off at the edges. New valves are likely to work hard, unless well lubricated. The plates must be made of high carbon steel, well tempered. In Fig. 199

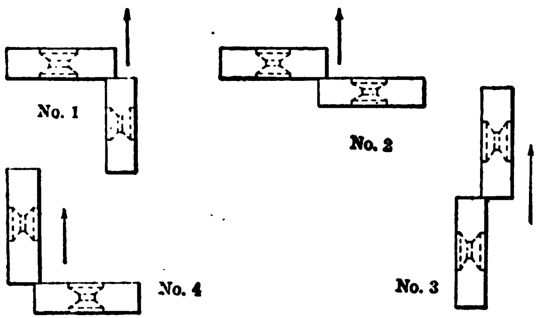


FIG. 199.—Position of catch plates.

the prevailing positions of the plates in regard to one another are shown. In No. 1 the edge of a high side engages an edge of a flat side; in No. 2 the edges of two flat sides come in contact; in No. 3 those of two high sides, and in No. 4 a flat side against a high side. Although it is claimed for high carbon steel that it has no grain and offers the same resistance per unit of area in any direction, it is evident from the way most of it is hammered out into long bars that if there is any fiber or grain it must run lengthwise in the bar, and that the material will offer greater resistance to shock in this direction than in any other. Of the four different positions in Fig. 199, therefore, No. 3 should be the most advantageous for the plates. Experience seems to bear this out. There is, among others, a case on record where two catch plates in this position have been working for over ten years at 120 r.p.m. six to eight hours a day, six days a week, without changing the wearing surfaces. The plates were in the same position when examined; as when the engine was started, showing a smooth sharp edge without any wear. The steam pressure

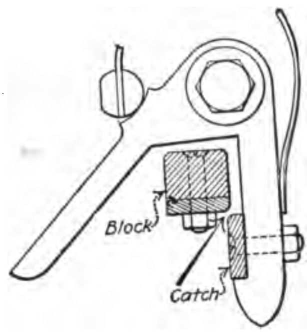


FIG. 200.—Catch plates showing wear of the edges.

carried on the boilers is 120 lbs. This performance has probably never been equalled.

The catch plates should be made square to make eight wearing surfaces available by changing their position. The edges are often broken by allowing too much space between them in hooking up; the closer they come together the lighter will be the shock and the longer they will last. For a nice running valve gear the distance should not be more than $\frac{1}{16}$ in., and the depth to which the two plates engage should also not exceed $\frac{1}{16}$ in., or $\frac{3}{32}$ in.

at the most; if they have to engage deeper to catch, something is wrong and should be corrected. If the plates wear round, as shown in Fig. 200, they will miss in hooking up and no steam will be admitted to the cylinder on that end, the engines will probably slow down taking steam only on one end and the regulation will be bad. The plates should be changed to new cutting edges.

CHAPTER XIII

WRISTPLATES AND VALVE MOTIONS

Single wristplates. Range of cut-off is limited with single eccentric. Valve-gear diagrams. Imperfect release shown by indicator diagrams. Steam lead table. Double eccentric wristplate motions for long range cut-off. Parallel motion for high speeds. Adjustment of valve rods.

THE single wristplate has been a favorite for years. It seems hard to break away from it, with its toggle motion, ease of adjustment and handling. For warming up, starting, stopping and occasionally reversing an engine, nothing simpler has ever been devised. There is a variety of wristplates in existence—old pear-shaped affairs with long throw, significantly called swingplates by their makers; wheel wristplates; round, balanced and spider wristplates—all of which are subject to the same defect, inertia. Some of them are heavy, throwing considerable strain on hook and eccentric rods. Fig. 201 shows the different kinds of wristplates mentioned.

Although the single type, limiting the cut-off in the cylinder to about $\frac{7}{16}$ of the stroke was partly responsible for the remarkable economy of the old Corliss engines, compared with late-cut-off slide-valve engines, on account of the more rational expansion which users of these engines were compelled to adopt, the tendency is to eliminate it and substitute a valve motion which will permit of a long-range cut-off for occasional extremely heavy loads and which is also suitable for higher rotative speeds. This is done in several different ways—by using two wristplates, or one wristplate for the exhaust and parallel motion for the steam, or parallel motion for both steam and exhaust without any wristplate. The objections to wristplates, of course, are greater inertia, more wear, lubrication and repairs. The parallel-motion gear seems to be the only feasible one. It is a trifle more complicated.

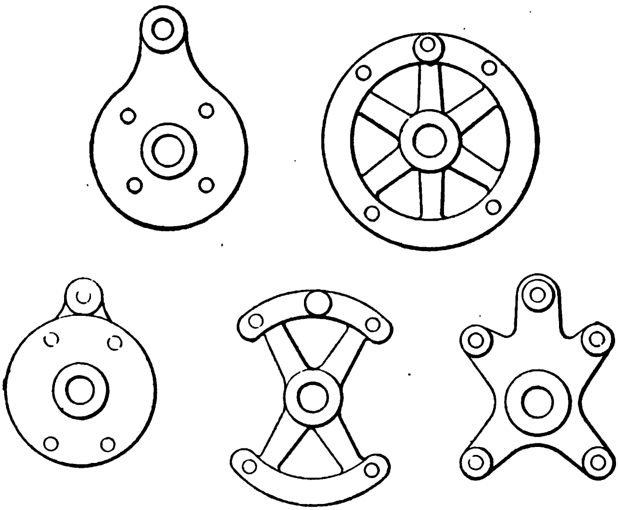


FIG. 201.—Common forms of wristplates found in practice.

On the single-eccentric wristplate motion shown in Fig. 202 any valve can be adjusted independently of the others, for lap with the wristplate central and for lead with the crank on the dead centers. A study of the valve-gear diagram shows its limitation and also the trouble that may be expected if an engine with this motion is overloaded. Single ports are shown for simplicity. The principle is the same for double or any number of ports. With the wristplate in the central position all the valves

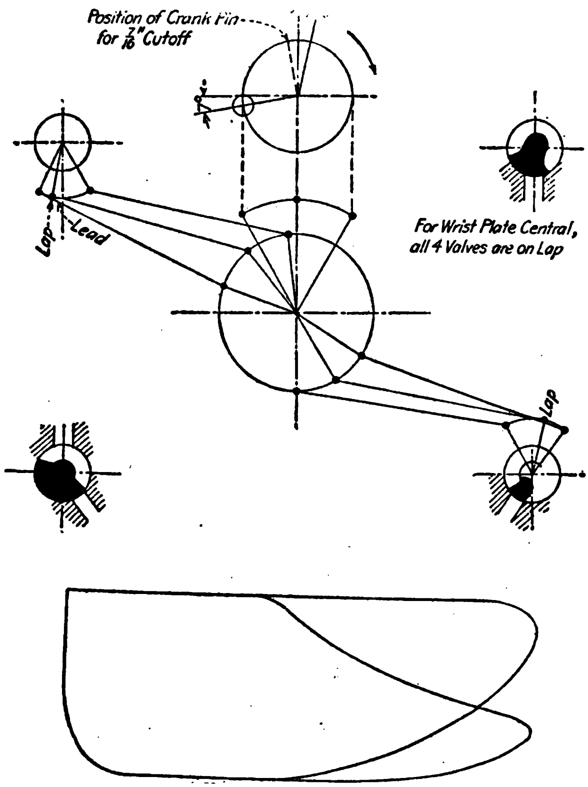


FIG. 202.—Single eccentric wristplate motion and indicator diagram.

must be on the lap. Disregarding all angularities of the connecting-rod and eccentric rod, the relative positions of the crankpin and eccentric for the mid-position of the wristplate are shown in the detail. The eccentric stands for this position at 90° to the horizontal and the crankpin about 7° below. Moving the crankpin to the dead center advances the eccentric 7° from the vertical center line; the head-end steam valve is now on the lead, and if we move the crank around until the eccentric reaches its extreme travel, we find that it has reached a position 7° from the vertical center line. This represents the latest cut-off which the valve gear affords. If the valve gear has not been tripped by the time the eccentric and wristplate have reached their limit, the engine will take steam full stroke; there is no regulation possible between about $\frac{7}{16}$ cut-off and full stroke, and if an overload is put on requiring more steam than the $\frac{7}{16}$ cut-off can admit, the speed of the engine will be irregular. Not only that, but all the working parts will be strained. It takes a certain amount of time to close a steam valve when wide open, therefore, an indicator card may show a cut-off of 50 per cent for a single-eccentric gear, which, of course, is theoretically impossible if the exhaust valves have the requisite amount of lap and lead for compression and release. The steam valve at the head end works simultaneously with the exhaust valve at the crank end and *vice versa*. Therefore, the same distance through which the wristplate travels to move the steam valve from the lap to the lead position is utilized to make the exhaust valve at the other end travel from lap to lead and no more; the motion of one depends on that of the other. This explains in part why a single-eccentric valve gear is limited as to compression and release for the exhaust. The hands of the designer are tied, so to speak, on this valve gear; if he advances the eccentric to give an earlier release and more compression, he reduces the range of cut-off, and if the eccentric is moved back, release and compression will be reduced and the cut-off extended.

A late cut-off indicator diagram is shown in Fig. 202. The effect of imperfect release is clearly noticeable. The diagram for steam full stroke in the same figure shows pronounced wire drawing in the exhaust port. No fixed rule can be established for the amount of lap and lead that should be given the valves of an engine. That depends on the throw of the wristplate and the travel of the valves, which varies with the different makes of engines. Most all Corliss engines have marks on the valves showing the cutting edges and the necessary lap; a few of them indicate the lead. As stated before, with the wristplate central the valves must be on the lap.

If there are no lead marks the following table may be used for the steam valves, with the understanding that it is subject to change to suit conditions:

Size of engine, in.......	2	14	16	18	20	22	24	26	28
Lead of steam valve, in.	$\frac{3}{32}$	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{1}{4}$

An engine better have too little steam lead than too much, for the latter may cause a pound by forcing the piston ahead with a blow instead of gradually. This is especially true for a condensing engine where the compression is low.

If an engineer could have a valve gear diagram of his engine mounted on a board on the wall, showing the ports of the cylinder, cutting edges of the valves, their lap, lead and extreme travel, also the throw of the wristplate and the motion of the pins with the position of the governor rods and cams, distance between valve centers, etc., it would help him to keep the valve gear in good order.

The diagram shown in Fig. 203 represents a double eccentric wristplate motion. In this the steam and exhaust valves can be set independently of each other. Release and compression can be adjusted most advan-

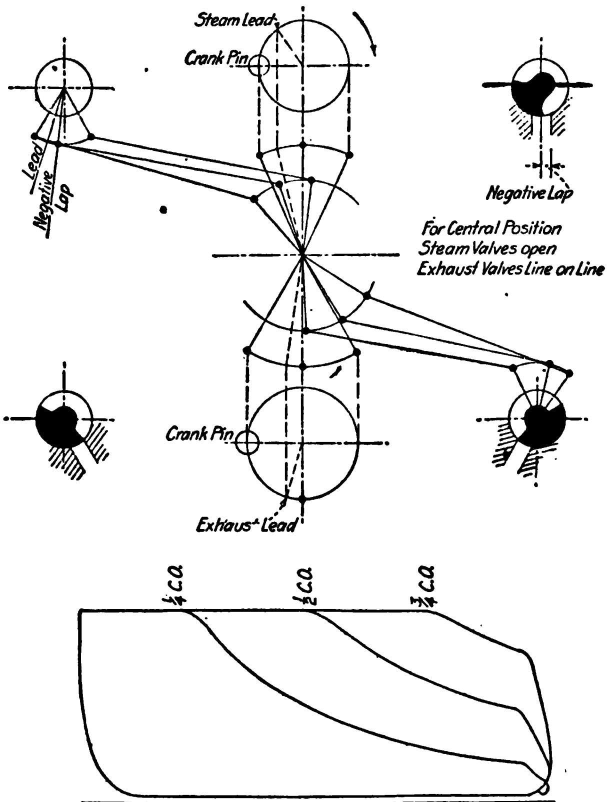


FIG. 203.—Double eccentric wristplate motion and indicator diagram.

tageously, as shown in the indicator diagram. For long range cut-off both steam valves will be open, or they will have what is called negative lap, with the wristplate in the central position. The governor must always trip the valve gear at the end of the stroke, otherwise steam will blow through into the exhaust port when the engine is started or takes steam full stroke. The cut-off can be extended to $\frac{8}{10}$ of the stroke. It is not advisable to go beyond that, for live steam might go to waste through the exhaust. The steam wristplate in this valve gear is, of course, superfluous; it is a convenience in starting and handling the engine, but, otherwise, it is an unnecessary

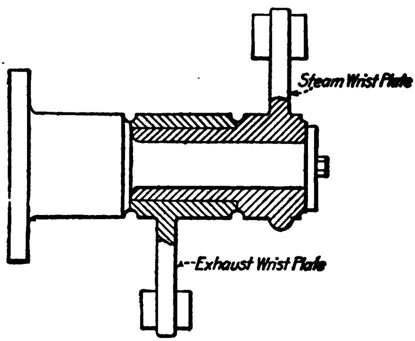


FIG. 204.—Bearing for double wristplates.

adjunct, which requires lubrication and looking after, especially, if the exhaust wristplate has its bearing on the hub of the steam wristplate, as shown in Fig. 204, the way it is done on many engines.

The valve-gear diagram, Fig. 205, shows parallel motion used for the steam valves and a wristplate for the exhaust. Advocates of this system claim that the wristplate motion brings the exhaust valve almost to a stop during the period of admitting steam to the cylinder; others asse t that the live steam pressure on the almost stationary valves will make them stick to their seats, and that the rapid motion in opening imposes an excessive strain on the valve gear parts. Sometimes, instead of using the heavy wristplate, which is objectionable on

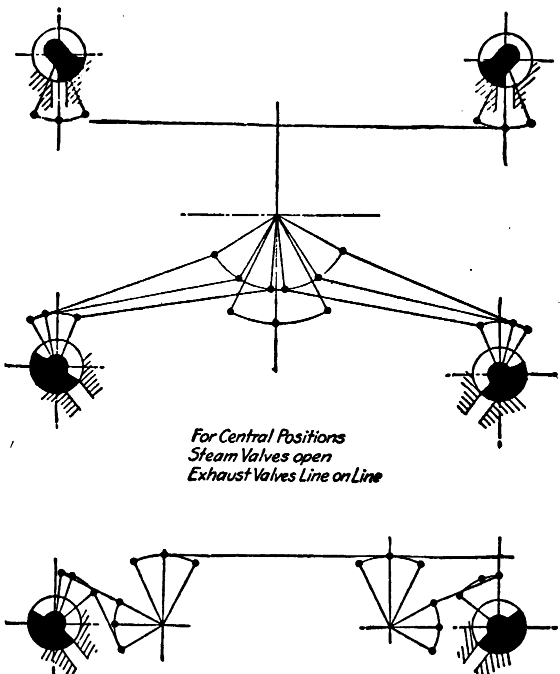


FIG. 205.—Double eccentric valve gears with parallel motion for steam and wristplate or bellcranks for exhaust.

high speeds, each exhaust valve is driven by a bell crank and link, shown below the wristplate motion in Fig. 205. The weight and inertia are probably reduced in this design, but the number of parts has been increased, and it does not seem to be much of an improvement.

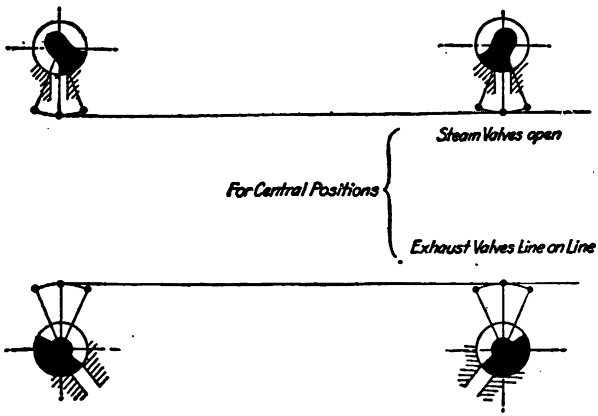


FIG. 206.—Parallel motion valve gear.

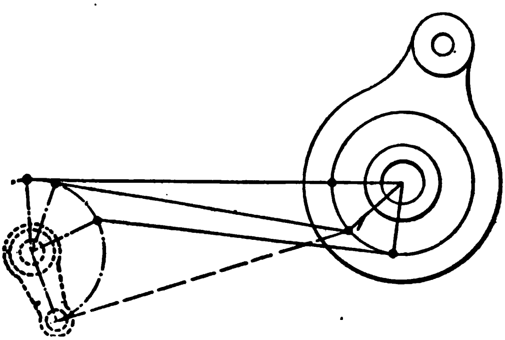


FIG. 207.—Exhaust lever dropped down out of position.

For high speeds the parallel motion shown in Fig. 206 appeals to the exacting engineer. There are no unnecessary wristplates, bell cranks or links used. The motion of the exhaust levers is, like that of the rocker arm, easy and gradual, they are always moving, there are no sudden jerks or stops, and the exhaust valves are not likely to stick to their seats.

In making adjustments of the parallel-motion gear it must be borne in mind that the valves at the head end of the cylinder are affected by changes made in the setting of the valves at the crank end, at least as far as the shortening or lengthening of the parallel rods are concerned, and that they must be adjusted to their original position unless

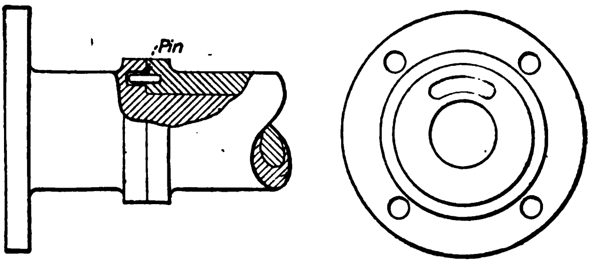


FIG. 208.—Pin in wristplate to prevent excess of travel.

a change of them is also desired. When a change is made in the length of the steam-valve rods, the dashpot rods must also be adjusted; an oversight in this direction may result in a broken bonnet or other trouble.

It will sometimes happen on the wristplate motion that the wristplate is moved over too far by hand and the exhaust lever drops down, as shown in Fig. 207. If this occurs without being noticed, the engine will perform some strange antics, when the steam is turned on and very likely something will be broken or twisted. As a precaution against this, a pin is screwed into the back of the wristplate hub which works in a slot of sufficient length, either milled or cored in the face of the wristplate stand, as shown in Fig. 208. The slot and pin will prevent the wristplate from oscillating too far on either side, but, of course, the operator must be careful in adjusting the length of the hook rod or eccentric rod; otherwise the pin may be sheared off. There must be some play allowed to guard against accidents.

CHAPTER XIV

ROD ENDS AND BONNETS

Threads must be protected. Troublesome designs. Good designs. Hardened pins. Gage for valve rods. Bonnet and valve stem without stuffingbox. Renewable bushing for valve stem. Metallic packings for valve stems not a success. Plastic soft metal packing. Keys for steam arm. Inside arm.

THE ends of the valve rods must be turned and the right- and left-hand threads cut between centers on a lathe. To do this work on a screw machine or turret lathe would be detrimental to the valve gear, since the

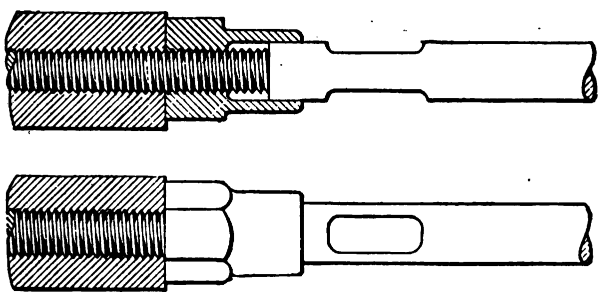


FIG. 209.—Sleeve nut for valve rod.

threads would not be in line with one another. The rods must fit well in the stub ends and the face of the check nuts must be square with the thread. Special nuts turned and threaded in a lathe should be used, and they should have a sleeve, as shown in Fig. 209, to protect the thread on the rods from getting bruised. They should be case-hardened. Commercial nuts are not fit to be used in this class of work. Careful attention should be paid to these small details on which much of the life of an engine depends.

Some of the older designs of stub or rod ends, shown in Fig. 210, are troublesome. The one marked *A* is made of bronze, in two parts bolted together. The only way to adjust it for wear is to remove the cap and reduce

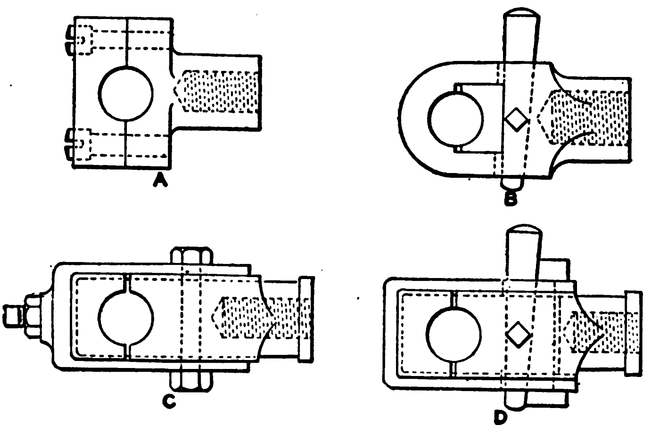


FIG. 210.—Old style stub ends.

it by filing or scraping; this is the worst thing ever devised for the purpose. The stub end *B* is made of bronze, and is provided with a loose half-box, adjusted by a steel wedge. Since most stub ends act by pulling, the greatest wear will be on the body itself, a renewal of which is rather expensive. That shown at *D* consists of the main part and cap, made of brass and held together by a steel strap with gib and key—a complicated and troublesome construction. *C* is of the same order except that it has a bolt instead of the gib and is adjusted by a set-screw on the top, which is a good feature.

The stub ends shown in Fig. 211 are of modern design, are suitable for higher engine speeds, and the adjustments are simple. That at *E* is made of bronze, with a loose box and wedge which may be graduated for accurate adjustment. There is but one objection to this design—the wear of the hole in the stub end itself. The one at *F* is of bronze, with a loose box and screw adjustment in the

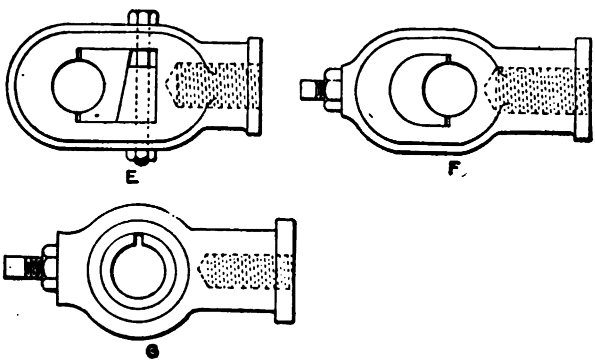


FIG. 211.—Modern stub ends.

top. The body of this stub end is also subject to wear. *G* is made of malleable iron, bronze or forged steel, preferably the last, with a bronze bushing and screw adjustment on the top. A small dowel pin keeps the bushing from turning. The latter is split on one side only, and as it wears the set-screw can be tightened. This is open to criticism from a theoretical standpoint, but it works well in practice.

The valve gear pins are subject to considerable wear. They have a tendency to wear out of round, which makes it difficult to adjust the stub ends properly. They should be case-hardened and ground, which will reduce their wear to a minimum.

As valve settings, through wear and adjustment of the stub ends, undergo certain changes which affect the rod centers, it is advisable to provide a simple check which can be verified at any time. This can be done by marking the valve rods, parallel and eccentric rods, etc., with a center punch any convenient distance, say 3 or 4 ins., from the center of the pins, as in Fig. 212. A gauge can be made for the fixed distance, or a pair of dividers may be used. Simple as this is, it will save a lot of time, trouble and uncertainty.

The bonnets, or as they are often called, the valve

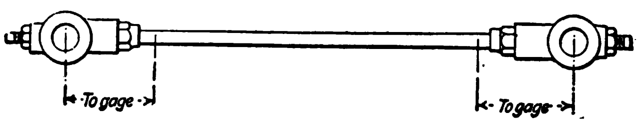


FIG. 212.—Center punch marks for checking wear on rod ends.

brackets, are important members of an engine, since most of the valve-gear parts are supported by them. Some designs give trouble which, with a little care and forethought, could be avoided. The stuffingbox and bearings seem to be the vital points. Attempts have been made to omit the stuffingbox and to substitute a collar and friction disk on the valve stem which bear against the inner surface of the bonnet, as in Fig. 213, where they are beyond the control of the engineer. To lubricate the collar properly is impossible; it will keep on grinding away until worn out. For low pressures and slow speeds the arrangement may have worked fairly well, but it does not answer for present-day requirements.

The combination of bonnet and stuffingbox shown in Fig. 214 is often used. It works all right for a while, but will eventually give trouble in the short bearing back of the stuffingbox. The valve stem must have a close working fit at that place; with the short bearing a groove will be cut around the stem, due to the resistance of the valve in rocking to and fro. A renewable bronze bushing

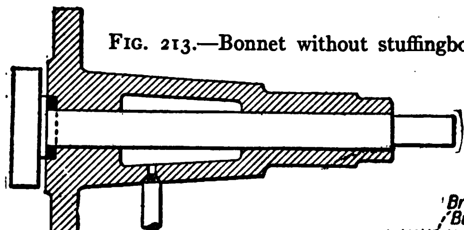


FIG. 213.—Bonnet without stuffingbox.

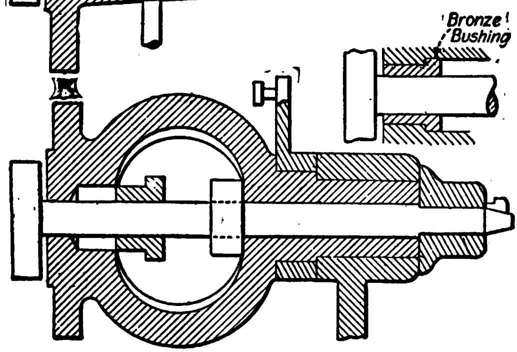


FIG. 214.—Bonnet with stuffingbox and bushing.

equal in length to the diameter of the valve stem should be inserted in the bottom of the stuffingbox, as shown in the detail sketch above Fig. 214. The bushing should be made a close working fit on the stem, avoids the cutting of the stem by the narrow bearing, and is renewable.

The question of packing for the valve stems is problematic. There are metallic valve-stem packings galore on the market, but none of them has met with unqualified success. If a valve stem could be kept central and rotating without any side motion, metallic packing would be all right, but its quick jerky motion combined with a certain amount of vibration gives the steam a chance to work its way through. Plastic soft-metal packing made into suitable rings and forced lightly into the stuff-

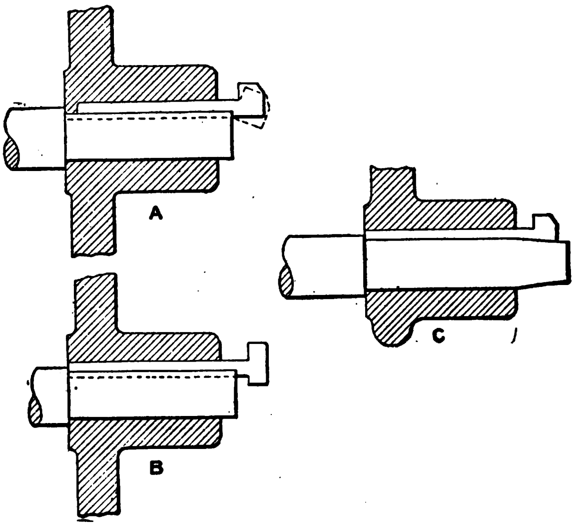


FIG. 215.—Keying of valve stems.

ingbox by the gland is about the best thing for packing Corliss valve stems. Fibrous packing cannot be recommended; it will get hard, wear the stem and need frequent renewal.

Of the three ways of keying the steam arm to the valve-stem shown in Fig. 215, *A* is apt to cause trouble if the key has to be taken out, as the head, being unsupported underneath, will bend over as shown in dotted lines if the key is in tight. The tee-head key shown in *B* is not much better unless two wedges are used to force it back. In *C* the valve stem is extended, with a long taper, giving the head of the key a good support. This is the most approved method.

Fig. 216 shows a design of bonnet in which the steam

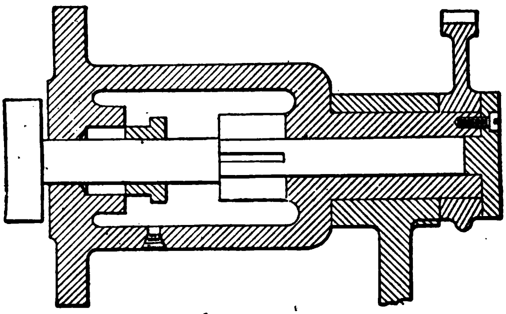


FIG. 216.—Bonnet and valve stem with inside arm.

arm, ordinarily placed at the extreme outer end of the valve stem, is located inside of the bonnet so as to eliminate strains due to the overhanging arm by placing it between two bearings instead of outside of one. Except for large engines the change cannot be recommended, since the arm is less accessible. This design is well suited to heavy engines in rough service.

CHAPTER XV

OILERS

Centrifugal crankpin oilers. Oil holes in crank and crosshead pins. Oil grooves in boxes. Wiper cup. Telescopic crosshead pin oilers. Wiper and tube for crosshead. Oil boat for eccentrics. Oil piping.

It is generally conceded that the crankpin of an engine should be lubricated by means of a so-called centrifugal or center oiler. An outline of the arrangement is shown in Fig. 217. In this way the amount of oil can be regulated at all times. The use of a grease cup on the crankpin will give trouble; the engineer has no control over it as long as the engine is running, and there is always the danger of its breaking off at the shank and hurting someone.

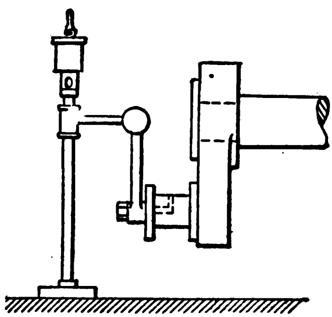


FIG. 217.—Crankpin oiler.

Grease and oil together should never be used on a crankpin for if any grease gets into the oil-grooves and split of the boxes, the oil cannot spread out properly over the bearing, and very likely the pin will get hot. As soon as it gets too hot the grease will melt and run off, and at the same time its lubricating quality will be destroyed. It is better to keep grease off where oil is used.

An enlarged view of a centrifugal crankpin oiler is shown in Fig. 218; *A* shows a short pin with one hole in the middle, pointing inward to the center of the shaft.

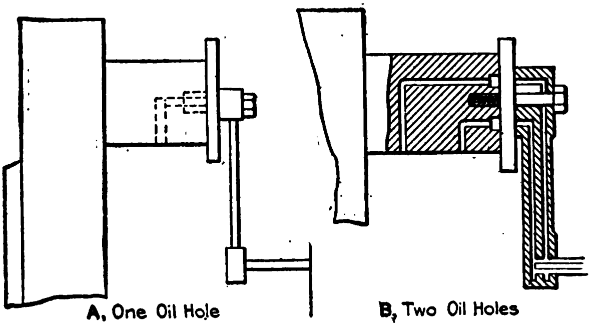


FIG. 218.—Centrifugal crankpin oiler.

If the pin is long and two holes are desired, each must have a separate oil supply, as shown at *B*. The two holes must not be connected, otherwise the first one will take nearly all the oil and none will reach the other. The reason for turning the hole toward the center of the shaft is obvious, if we consider the four positions of the crankpin shown in Fig. 219. The pressure transmitted from the piston to the crankpin affects, during one revolution, only about three-quarters of the circumference, as indicated in heavy lines in the sketch. The part of the pin shown in light lines hardly ever comes in contact with the boxes, and since it offers the least resistance to the flow of the oil, the oil hole should be located within its limits. The most convenient direction is toward the center of the shaft, which makes it right for the engine running over or under. On crankpins with the oil hole drilled toward the outside oil can flow through only when the hole passes the split in the boxes near the upper and lower centers, in which case the centrifugal force will throw most of it against the rod

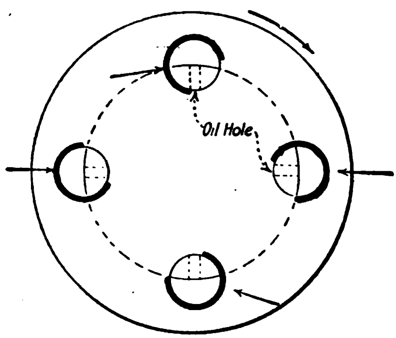


FIG. 219.—Crankpin exposed to wear.

and out at the sides without taking effect in the pin or the boxes. Pins oiled in that way require a large quantity of lubricant. Incidentally, the sketch shows that a crankpin will wear almost entirely on one side provided the engine runs always in the same direction.

The crosshead pin should have the oil hole on the top, as shown in Fig. 220. The upper and lower sides of the pin should be flattened for two reasons: First, to make the oil spread over the length of the pin and, second, to prevent the brasses from wearing shoulders on the pin, owing to the oscillations of the connecting-rod. On large engines with heavy rods, where the wear is considerable, the amount of flat must be carefully determined. If the oil supply is liberal, as it will be if a continuous oiling

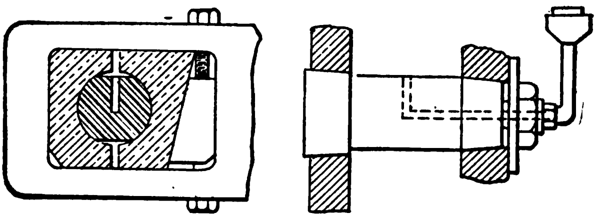


FIG. 220.—Oil hole in crosshead pin.

system is installed, oil grooves in connecting-rod boxes are superfluous; but where the oiling is done by drop feeding from oil cups, grooves will improve the distribution of the oil. In crankpin boxes lubricated by means of a center oiler, the grooves should run crosswise, as shown at

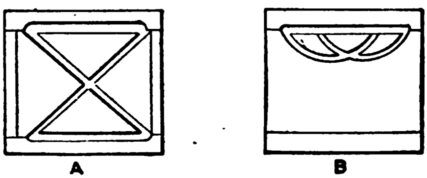


FIG. 221.—Oil grooves for crankpin and crosshead pin boxes.

A, Fig. 221, while the crosshead pin boxes should be grooved only on the upper side, as shown at *B*. The object of oil grooves is to distribute the lubricant over the width of the boxes and to retain part of it for a while. It must be borne in mind that they are also likely to collect dirt, which must be removed from time to time, otherwise they will be worse than useless.

Efficient oiling of the crosshead pin is sometimes quite a problem; although it needs less than one-half the oil that a crankpin requires, it is much more difficult to get the lubricant to the place where it does the most good. On slow-speed engines it has been customary until a few years ago to use a wiper cup which would take a drop of oil at the end of each in- or out-stroke from a pipe con-

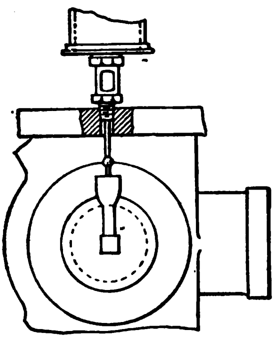


FIG. 222.—Wiper cup.

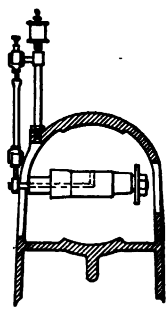


FIG. 223.—Telescope crosshead pin oiler.

nected to a sight feed oiler, as shown in Fig. 222. If the oil is not watched carefully and its flow well regulated it will be thrown over engine and floor.

Telescopic crosshead pin oilers are used to a large extent, they seem to give great satisfaction in some places and a lot of trouble in others. The opinion of engineers as to their reliability and usefulness is divided. When new and applied in the correct way they work so well that an engineer will almost forget there is such a thing as a hot crosshead pin, but in the course of time some parts will wear out and the apparatus becomes unreliable. It is usually less expensive to replace the worn parts with new ones than to try to repair them. There are telescopic oilers which will give good service if well cared for and the proper kind of oil is used others are so poorly designed and badly made that they will give continuous trouble.

A telescopic crosshead pin oiler used on a great many engines is shown in Fig. 223; it seems to be one of the best and most reliable oilers of its kind, can easily be taken apart and cleaned. The principal trouble with telescopic oilers is leakage; the reciprocating motion interferes with the steady flow of the oil, which is thrown fore and back and will ooze out at the joints in spite of the best attention. Next, the sliding of one tube inside of another acts like a pump, the up-and-down movement of the outer tube draws in air and forces it out again, which also interferes with the flow of the oil and will even at times throw it out altogether. To prevent this ball check valves, held on their seats by springs, are used in some instances which makes the device more complicated and its action uncertain. The tubes wear rapidly in places where grit and dirt gets on them and so do the joints.

The most reliable way to oil a crosshead pin is by means of a wiper and tube as shown in Fig. 224. The device consists of a brass tube, closed at both ends and suspended from the upper part of the guide by means of two short pieces of pipe. One of the supports is threaded for an oil cup. A wiper cup of sufficient height is provided for the crosshead pin. There are a number of small holes drilled in each end of the brass tube on the upper side, so that any oil that oozes out will run over the tube and hang in drops on the under side, where the lip of the wiper cup will wipe them off and lead them to the pin. The hole in the crosshead pin must point upward, otherwise no oil will get there. The height of the wiper cup should be at least 3 in. more than half the diameter of the pin. This seems

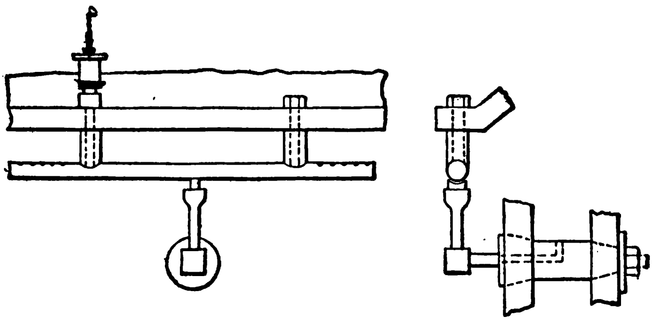


FIG. 224.—Oiler for crosshead pin.

to be the most reliable oiler for the crosshead pin; everything is in plain view of the engineer and under his control.

Telescopic oilers have also been used quite extensively on eccentrics, but after some sad experiences with them the old time oil boat has come into its own again. To

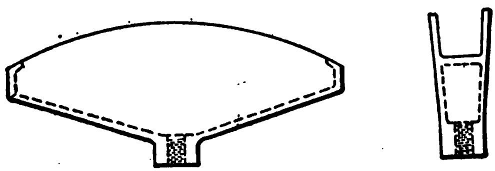


FIG. 225.—Oil boat for eccentric.

make an oil boat effective it should be made very deep, so that no oil can escape or be thrown out. The sides should be very high in the middle, as shown in Fig. 225, to protect the oil from the suction of the flywheel. The oil boat shown in Fig. 226 is too shallow, the oil will run from one end to the other according to which way the eccentric tilts. Oil boats of the shallow kind are used quite frequently and are likely to give more or less trouble unless a continuous stream of oil is used, in which case an efficient guard must be provided to catch the oil that is thrown off.

Sometimes grease is used to lubricate eccentrics, with

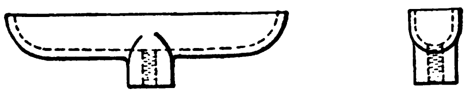


FIG. 226.—A too shallow oil boat.

good results. Two cups should be used on every eccentric to avoid trouble if one should get out of order.

If a gravity oiling system is used on an engine care must be taken to use pipe that is free from scale on the inside. Seamless drawn brass tubing is best for this work. All bends should be made to a long, easy radius, and there should be sufficient tees, crosses and unions, so that the pipes can be readily cleaned, if they should get clogged. Black-iron pipe or galvanized pipe will cause trouble on account of scale.

CHAPTER XVI

RECEIVERS

Size of Receivers. Safety valves must be used. Arrangement for compound engines. Reheaters. Vertical receiver with copper coil. Air must not accumulate. Vibration of pipes causes leaks. Designs for horizontal reheaters.

On compound engines trouble may be experienced with the receiver, which is often too small to give a good steam distribution and cut-off in the low-pressure cylinder, is treated as inferior and superfluous, and is made without regard to strength and efficiency. Accidents to receivers are multiplying, and it is about time that they were watched more closely. They should be of ample size—that is, from $1\frac{1}{2}$ to 2 times the piston displacement of the low-pressure cylinder. This ratio should be carried out on cross-compound and tandem compound engines alike. It is a mistaken idea that tandem compound engines do not need a receiver.

The piping to and from the receiver, the receiver itself and also the low-pressure cylinder should be made strong enough to carry the full boiler pressure—not as a working pressure, but for emergency. The old way of making low-pressure cylinders of light weight and also light receivers and pipe connections is extremely dangerous, for, if anything happens to the high-pressure steam valves or their gear to prevent closing of the valves, steam under full pressure will blow into the receiver through the high-pressure exhaust valves when they open. Every receiver should be provided with at least one safety valve, not a small relief valve to indicate when the receiver pressure has reached the danger point, but one that has an opening about three-fourths the area of the high pressure exhaust; or better, it might have two safety valves of the locomotive type, with large pipe connection to the outside of the building.

A vertical receiver with the high-pressure exhaust

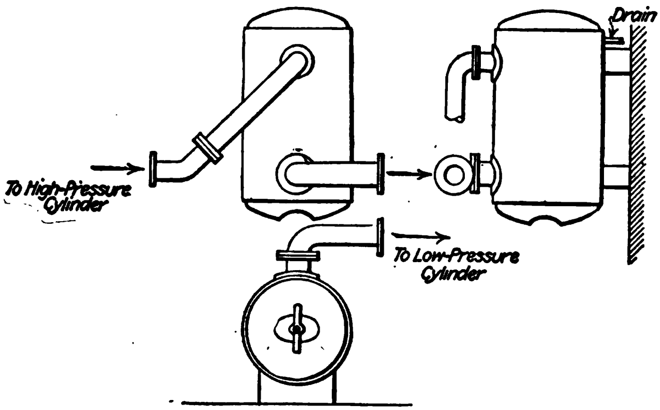


FIG. 227.—Horizontal receiver connections.

entering near the bottom and the low-pressure steam pipe leaving at the top is a good arrangement, but there may be occasions when a horizontal receiver would be more convenient. The advantages of the vertical receiver lie in the ease of draining and of reheating, if this be necessary; its disadvantage in the fact that it cannot be located very conveniently, unless there is a basement that affords sufficient headroom under the engine room. There are no objections to the horizontal receiver, if well drained.

The arrangement shown in Fig. 227 can be recommended for cross-compound engines. The receiver is located in the basement sufficiently low for the exhaust pipe from the high-pressure cylinder to be connected to a flange on the top at one end and for the low-pressure steam pipe to be connected to another nozzle at the other end. Any condensation carried over from the high-pressure cylinder and the water that accumulates from condensation in the receiver itself will collect in the bottom, so there will be no danger if it should slope toward the end at which the high-pressure exhaust enters, where a large drain pipe should discharge the water into a trap. The receiver should be made of steel plates to conform with the specifications adopted by the American Boiler Manufacturers' Association, and the heads should be dished to a radius equal to the diameter of the shell. One of the heads must be provided with a standard manhole. The nozzles should be of heavy design and riveted to the shell, with calking strips under the flanges. The seam of the shell should be made on the upper half, never at the bottom. The receiver may be supported on concrete piers, but not bolted down, with allowance for expansion and contraction.

Reheating receivers are sometimes called for and installed, but except on pumping engines, the reheating part is usually put out of service. If a reheater is desirable, the vertical receiver gives the best results.

The reheater should consist of a copper coil, the steam to enter at the top; the lower end connected to a trap. The area of the reheating surface in square feet should be equal to about twice the volume of the low-pressure piston displacement in cubic feet. The exhaust from the high-pressure cylinder should enter at the bottom, the nozzle for connection to the low-pressure cylinder to be near the top. Fig. 228 shows a vertical receiver with copper coil, the top consisting of a flange riveted to the shell with a cover bolted on, so that the reheating coil can be inspected and removed if necessary. Both the flange and cover should be of cast steel, thoroughly annealed, of about 60,000 lbs. tensile strength. Cast iron must not be used for receiver heads, as it will give trouble. A drain in the bottom will take care of any condensation

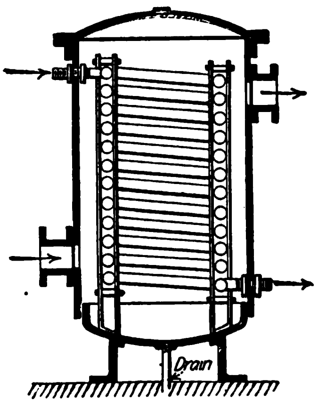


FIG. 228.—Vertical receiver with copper coil.

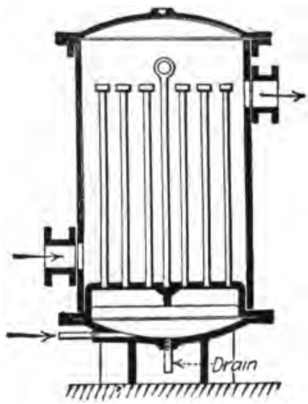


FIG. 229.—Vertical receiver with straight pipes.

Reheating pipes or coils in receivers act about the same as radiators in rooms; in fact, they *are* radiators and subject to similar conditions. Unless properly vented, air contained in the steam will accumulate in certain places and materially reduce the heating capacity. For this reason the arrangement shown in Fig. 229 should be avoided. The vertical tubes, closed at the top with a cap and screwed into the plate at the bottom, will soon fill with air; in fact, there is no way of exhausting the air from them except by a vacuum pump. Having no vent, their heating effect will be destroyed.

If the tubes are arranged as shown in Fig. 230, the air can be forced out so that they will have the full heating effect, but the trouble with this design is that the

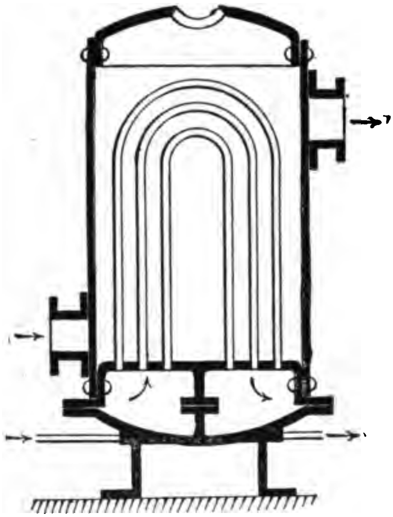


FIG. 230.—Air is easily forced out of the tubes.

tubes are liable to get loose in the plate on account of the constant vibration caused by the inrushing and outrushing steam; if once loose, they will soon begin to leak. There is no saving in steam in a reheater with leaky tubes.

In a horizontal receiver a copper coil seems to be out of place; water will constantly collect in the lower parts and there may be trouble on account of this; the drainage of the coil will be badly impaired. Therefore one of

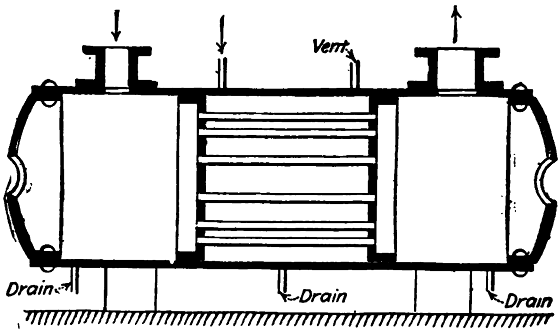


FIG. 231.—Horizontal reheating receiver.

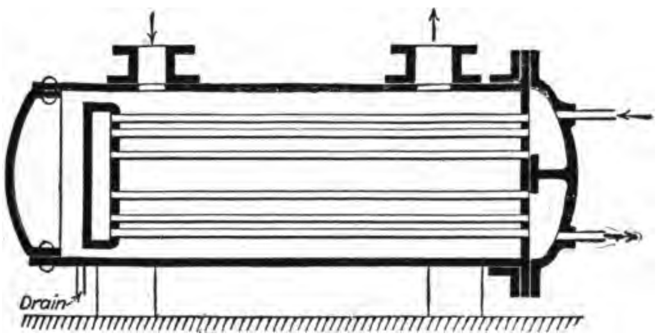


FIG. 232.—Independent pipe system for horizontal receiver.

the two designs shown in Figs. 231 and 232 may be adopted.

Fig. 231 shows two flanged steel heads riveted inside to the shell of the receiver, with steel tubes expanded in place. Brass tubes must not be used, as they will give trouble on account of unequal expansion. Both ends should have a manhole, so that the inside of the receiver and the ends of the tubes can be inspected. Fig. 232 shows another design with horizontal tubes which may freely expand and contract.

CHAPTER XVII

FOUNDATIONS

Weight and area are the principal requirements. Piles supporting foundation. Shallow foundation. Standard construction for direct-connected engine. Danger of ruining an engine by rough usage at the start. Setting the templet. Squaring the center-lines. Foundation bolt boxes. Bolts suspended from templet. Pockets for nuts and washers. Forms for concrete. Mixing of concrete. Filling the forms.

AFTER the most advantageous location for the engine, size of engine room, etc., have been determined, the next thing to be considered is the foundation—its depth, size,

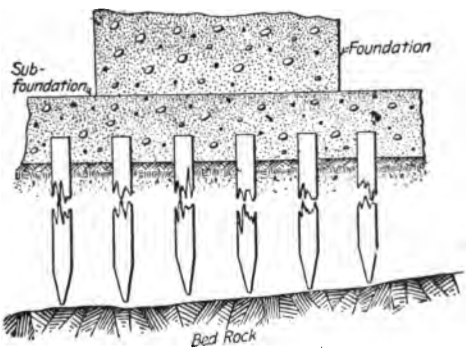


FIG. 233.—Piles supporting foundation.

weight and the material of which it is to be built. The depth depends largely on the ground; on general principles it is better to spread the foundation over a wide surface than to make it narrow and deep; weight and sufficient area for a good hold on the ground are the principal requirements. If bedrock can be reached within a reasonable distance that, of course, is to be preferred, but in most cases it is more likely that the ground is clay or sand, in which case a subfoundation of concrete 12 to 18 ins. thick should be spread over an increased area on which to build the foundation proper. If the ground consists of mud, piles must be driven and covered with a substantial layer of concrete. The piles may be driven about 3 ft. apart to hardpan or bedrock, as shown in Fig. 233, their tops trimmed to an even height and a concrete footing not less than 24 ins. thick spread over the surface. The construction can be much strengthened by placing steel rods crosswise in the concrete near the top and bottom. The upper surface must be left rough to make a good bond with the engine foundation. The piles themselves may be made of reinforced concrete if properly designed.

DIMENSIONS AND WEIGHTS OF FOUNDATIONS

Size of Engine	Main Bearing				Outer Bearing				Depth		Weight of Foundation	Weight of Engine and Wheel
	Length		Width		Length		Width					
In.	Ft.	In.	Ft.	In.	Ft.	In.	Ft.	In.	Ft.	In.	Lbs.	Lbs.
12×30	18	0	5	0	8	0	4	0	5	0	92,000	24,000
14×30	19	0	6	0	9	0	4	0	5	0	114,000	30,000
16×30	20	0	6	0	9	0	4	0	5	6	128,000	34,000
18×30	20	0	6	0	10	0	5	0	6	0	160,000	38,000
18×36	22	0	6	0	11	0	5	0	6	0	180,000	44,000
20×36	22	0	7	0	12	0	5	0	7	0	190,000	55,000
22×42	24	0	7	0	13	0	5	0	7	6	280,000	70,000
24×42	25	0	7	6	14	0	5	0	8	0	320,000	80,000
26×48	30	0	8	0	15	0	5	0	9	0	426,000	100,000
28×48	31	0	9	0	16	0	5	0	9	0	484,000	115,000
30×48	32	0	9	0	18	0	6	0	10	0	600,000	130,000

The weights of both engine and foundation must be taken into account in computing the area over which the subfoundation should extend. Average soil will support about 3 tons per square foot, but to be on the safe side it is best to allow only 2000 lbs. The greatest weight of an engine is located at, and supported by, the bearings, where also the greatest vibration takes place and where the main support in the foundation must be given. Vertical engines, few of which are built now, may be left out of the question, and only the horizontal construction need be considered. The table states the approximate over-all dimensions and weights of foundations for simple, heavy-duty, belted engines and also the weights of the engines themselves. From this table it is seen that the weight of a foundation should be four to five times the weight of the engine, which is considered ample to counteract vibration.

On mushy ground special precautions must be taken to counterbalance the inertia of the reciprocating parts, otherwise the rocking motion may be communicated not only to the engine house, but to all the buildings in the neighborhood on the same swampy soil. For direct-connected engines about 33 per cent should be added to the weights of engines and foundations given in the table, and the piers for the outer bearing and the generator must be connected at the base to the main foundation, giving them a broad bearing on the ground. If the depth of a foundation for a certain size of engine is made less than stated in the table, the length and width must be increased to maintain the weight.

In rocky ground or in places where water is close to the surface or sewers interfere with the work, it is often cheaper and more convenient to make a foundation shallow and extend it over a wide surface, than to make it deep. To give a shallow foundation sufficient strength it should be reinforced, for which purpose old rails, I-beams,

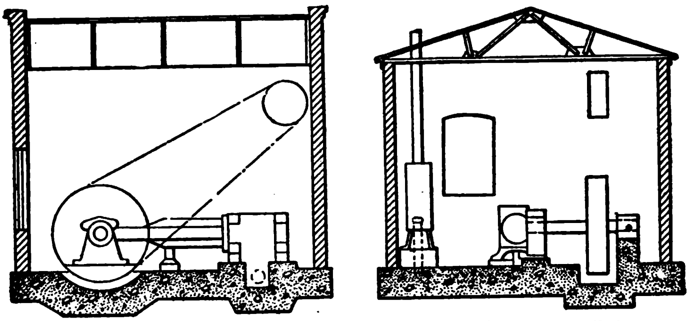


FIG. 234.—Shallow foundation supporting engine and building.

angles or bars may be used to advantage. A little rust on them will do no harm, but they must be free from mud and dirt. The foundation bolts must reach all the way through, with the washer on the under side.

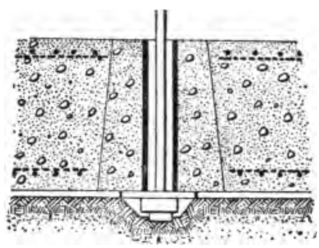


FIG. 235.—Hole in foundation for washer and bolt.

Fig. 234 shows an example of a very shallow foundation which was built for a 12×30-in. belted engine. It was extended under the whole engine room, supporting even the walls of the building. It was made 24 ins. deep, of concrete reinforced by two rows each, at the top and

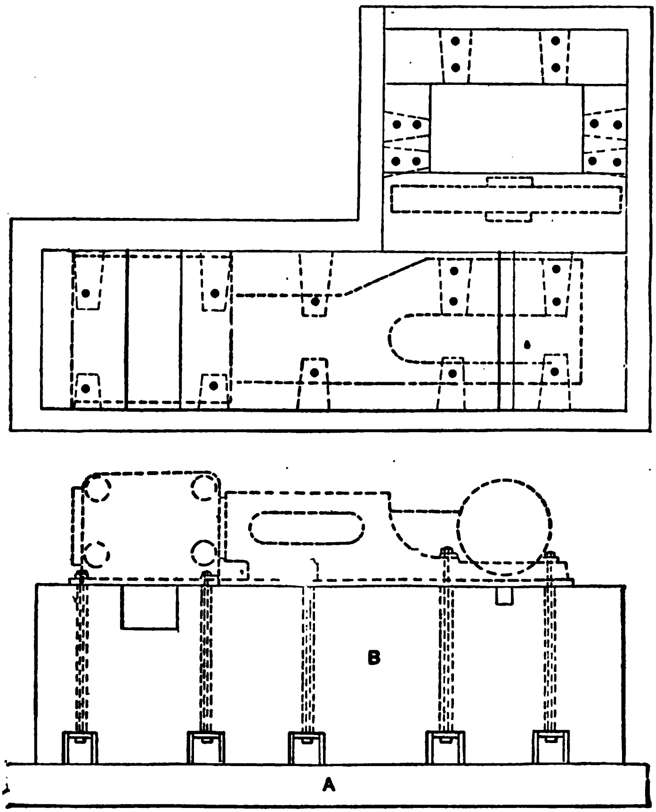


FIG. 236.—Standard foundation for direct-connected engine.

bottom, of ½-in. steel rods laid crosswise and spaced 3 ins. apart. Depressions of sufficient depth were made to form the flywheel pit and the trench for the exhaust pipe. Large holes were left for the foundation bolts and washers, as shown in Fig. 235, which were set later by means of a templet; the holes, which taper toward the top, were filled with concrete. The top of the foundation was finished smooth and forms the floor of the engine room. The reason for the construction was that a large sewer about 4 ft. below the surface of the engine room could not be removed without considerable expense.

Fig. 236 shows a standard foundation for a direct-connected engine. *A* is a slab of concrete which forms the subfoundation, and *B* represents the foundation proper. Some years ago foundations were given various shapes, which sometimes followed the outline of the engine. A favorite construction was to make the sides battered, but now, when foundations are mostly made of concrete, straight lines are followed in all directions. This is done to simplify the forms that have to be built for the concrete.

Erect the Building First

A mistake is often made in having the engine erected before the building for it is completed. As a rule, time is lost on the building and the aim is to make up for it by hurrying the work about the engine. Although an engine will stand a good deal of abuse and will sometimes run under very unfavorable conditions, it is a mechanism designed and built with care to do effective service for a number of years. It is, therefore, not unreasonable to expect that it should be handled with care and not be exposed to dust, dirt and the elements. A sand test is not required, as on a machine gun or cannon, because an engine is not supposed to work under such conditions, but in a nice, clean room, free from dust, rain or sleet. Many an engine is ruined by rough usage at the start.

Assuming, then, that the advice implied in these remarks has been followed and that the walls and roof of the engine room are finished, windows put in, the hole for the foundation dug and the subfoundation constructed, the next thing to do is to locate the center line of the engine, place the targets on the walls or other convenient places and establish a line of strong fishcord from which plumb lines can be dropped to locate the templet, as shown in Fig. 237.

The templet is usually made of 1-in. boards having one long center board (for large engines made of two lengths), with frames for the cylinder and pillow block and cross-pieces for the guide screwed on. The center lines of the engine, cylinder and shaft are marked on the board, blocks of wood are fastened to the frames giving the thickness of the metal at the bolt holes, and the holes for the foundation bolts are drilled through. The under side of the boards constitutes the base of the engine, outer bearing, etc. The templet must be well supported on framework and posts strong enough to carry the heavy bolts and washers. After it is lined up and leveled, the bolts are hung in place with the cast-iron washers at the lower end. The upper ends of the bolts should project about $\frac{1}{2}$ in. above the nuts in order to make sure of having enough thread for full nuts when the engine is put on the foundation. Boxes of $\frac{5}{8}$-in or $\frac{3}{4}$-in boards should be built around the bolts with $\frac{3}{4}$-in. or 1-in. space on each side, giving them sufficient play to allow for irregularities in the castings. They may be made tapering toward the lower end with the object of removing them after the concrete has set, although there is no objection to letting them remain in the foundation. The templet for the outer bearing and, if the engine is direct-connected, that for the generator are located in a similar way, special care being taken to set them level and square with the main templet.

A simple way to square the center line of the shaft with the center line of the engine is to stretch another line across the main line on the same level and adjust it as shown in Fig. 238. Any multiple of the figures 3, 4 and 5

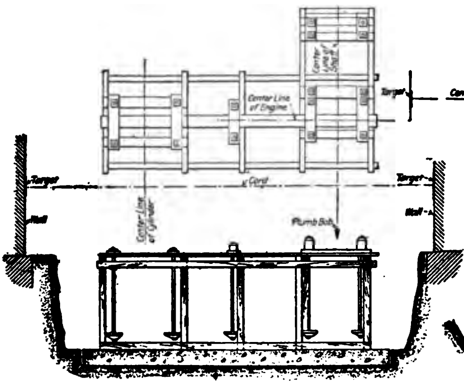


FIG. 237.—Setting the templet.

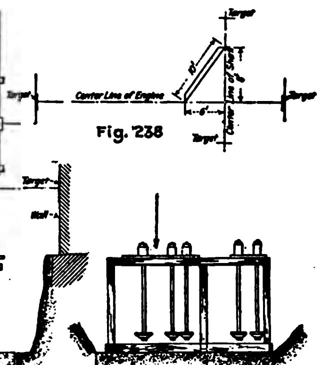


FIG. 238.—Squaring the center line of the shaft.

laid down to any scale will form the two legs and hypotenuse of a right triangle; as a rule 6, 8 and 10 ft. are the accepted standards for lining up an engine; therefore, if two pieces of cord are located 6 and 8 ft. respectively from the intersection of the two lines and distance between the two fixed points is made 10 ft., the two lines will be square to each other. To prevent any error a plank 4 ins. wide and 10 ft. long, pointed at both ends, may be used, with distances of 6 and 8 ft. marked from either end.

This is better than a tapeline or rule. The outer bearing and generator templets are then located by plumbing down to the center lines marked on them. Targets should also be established for the center line of the shaft.

Templets for cross-compound engines are lined up the same way. Parallel center lines are stretched, one for each side of the engine, and the center line of the main shaft is located at right angles to both of them in the manner described. For adjusting the two sides to the same level, a long straight-edge or a transit may be used.

Except on smaller-sized engines and where the founda-

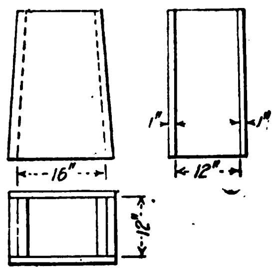


FIG. 239.—Foundation bolt boxes.

tion is crowded against walls or the ground is rocky and hard to excavate, the washers should not be built in the foundation, but located in pockets accessible from the outside. For this purpose boxes, as shown in Fig. 239, will be required; they must be large enough for a person to get into to hold the washers and nuts when the bolts are put in place. The boxes must be secured in position with the holes for the bolts directly under the corresponding holes in the templet. The foundation-bolt boxes are fastened to the templet and lower boxes, the vertical position to be verified by a plumb line.

In erecting an engine much trouble is sometimes experienced if the foundation bolts are cemented in solid before the engine is put in place. There is no need of having the bolts cemented in at all—in fact they should have play in the foundation, so as to accommodate themselves to the holes in the frame and cylinder base. It is well to

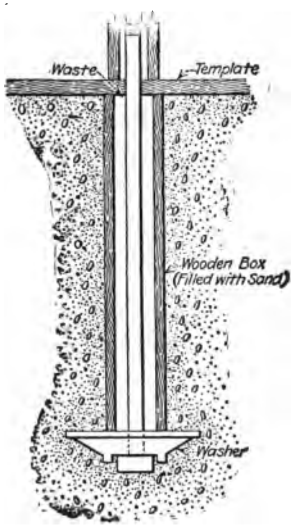


FIG. 240.—Foundation bolt suspended from templet.

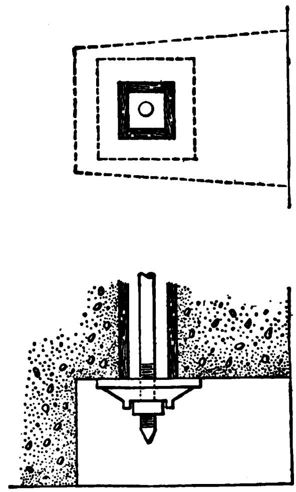


FIG. 241.—Pocket in foundation for washer.

leave about $\frac{3}{4}$ in. play all around, and for bolts 2 ins. in diameter and over a play of 1 in. on the side will be better. The bolts should be long, reaching almost to the bottom of the foundation. In erecting, they are to be suspended from the templet with the cast-iron foundation washers resting on the bolt heads, as shown in Fig. 240. Next wooden boxes of $\frac{1}{2}$-in. or $\frac{5}{8}$-in. boards are built around the bolts and supported on the washers, the top of the boxes to be level with the top of the foundation. The foundation is then built of concrete, or brick laid in cement—preferably concrete. The space in the wooden boxes around the bolts should be filled with sand and some waste put on top to keep out the cement. The sand is filled in loosely so the bolt will yield to one side or the other if necessary. The boxes may be left in the foundation, where they will not do any harm.

Fig. 241 shows the lower end of the foundation bolt and the washer located in a pocket in the foundation. The pocket should be so arranged that it is at all times accessible. In this case the bolt has a long thread with a nut; the end is pointed so it will enter the nut freely. The idea of having these pockets is that the foundation bolts can be inserted after the heavy parts of the engine have been put in place. This is a very convenient method on large frames, where the bolts sometimes project from 2 to 4 ft. and more above the foundation; also, if one should break it can be replaced.

The space around the bolts must not be filled with cement or grouting, but, as previously stated, with sand, sawdust or some other soft material. Anything that will bind the bolt must be kept out of the hole. The only object of the foundation bolts is to hold the engine down tight, so that the friction between the base and the foundation is sufficient to keep it from moving; they should not fit in the holes and be subjected to shear.

The boxes for the flywheel, generator and crank-pits should now be put in place and also those for the trenches for the exhaust pipe and the drain pipes. The form for the foundation should be made of $1\frac{1}{2}$-in. or 2-in. boards running horizontally, well-braced on the outside and, if necessary, held together by means of bolts or wire to withstand the pressure of the concrete and of the tamping, as shown in Fig. 242.

A good mixture of concrete for engine foundations may be made of 1 part of portland cement, 2 parts of sharp sand and 4 parts of gravel or broken stone. The work of mixing the concrete and filling it into the form should be done under the supervision of an experienced mason. Even if the work is rough, it will lead to trouble if only inexperienced help is employed. To prevent sticking,

FIG. 242.—Bracing for concrete forms.

the boards should be made wet before putting in the concrete if the foundation is to be plastered; otherwise grease or soap may be used. Concrete is placed in the form in layers 6 to 8 in. thick and tamped down with a light rammer. If the foundation is not finished in one day, the top layer must be kept wet overnight by covering it with wet rags. The next day the surface must be roughened and a strong mixture of cement and sand used to make a good bond with the next layer. The forms must stay around the foundation for two or three days and on large foundations about a week before they are removed.

On large installations for rolling mills or central power plants the engine foundations are built before the walls of the buildings are constructed, and the center lines and levels are located and the templets set with a transit. In order to avoid carrying the vibrations from the engines into the walls, care must be taken not to connect the foundations of engines and buildings; sand cushions must be provided if they approach one another closely.

The top of the concrete should stop about 2 ins. below the floor line, which includes 1-in. space for grouting. This will locate the base of the engine bed and sole-plates 1 in. below the floor line without any unsightly projections and offsets. The templet can be removed as soon as the foundation is completed, but all the boxes must stay in place until the concrete has set. The foundation-bolt holes must be protected from cement or dirt dropping into them. Pieces of wood can be nailed over the top, or waste, burlap and similar material may be used to close the holes temporarily.

CHAPTER XVIII

ERECTING

Locating targets. Centers in cylinder and guides. Grouting. Erection of flywheel. Oval-ring and tee-head links. Segmental wheels. Flywheel arms. Stresses in the rim of flywheels. Lines for stroke and clearance on guides. Screwing in the piston rod. Excessive clearance is wasteful. Locating the dead centers. Barring holes in rim of flywheel. Eccentrics. Buyer's interest in shipping an engine. Oil guards. Gravity oiling system. Force-feed oil pump for cylinder. Painting an engine.

In erecting the engine the frame is put in place first and leveled. Next, the cylinder and front head, with a gasket between them, are moved to the frame, leveled and bolted on. A plain soft-copper ring of even thickness seems to give the most satisfaction as a gasket; it may

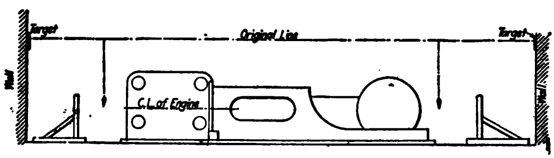


FIG. 243.—Locating center line with aid of targets.

be put in dry or, better, covered with a mixture of oil and graphite. The center line of the engine must be located with the help of the targets. For this purpose two stands made of boards are fastened to the floor—one at the cylinder and the other at the crank end of the frame, as shown in Fig. 243. A fine silk line is stretched between the two as tightly as possible to avoid any sag.

This line must be placed accurately by plumbing down from the original line, and the stands are notched so that the line may be taken off at any time and replaced. Next, the cylinder and guides are adjusted to this line as closely as it can be done by calipering. The line is then removed and centers are established in the rear end of the cylinder and the crank end of the guide by placing strips of wood in them, as shown in Fig. 244, each with a 1-in. hole in the middle covered by a piece of tin. The centers are marked with a pair of centering calipers, as shown; two centering lugs should be placed and machined for this

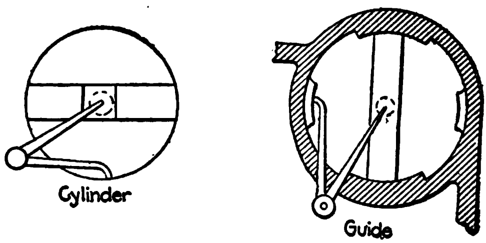


FIG. 244.—Calipering the cylinder and guides.

purpose in every frame. A small hole is drilled with a sharp awl through the centers marked on the tin, and the silk line is stretched through them for a test. If the holes are not in line with the marks on the stands, bed and cylinder must be shifted and wedged up to suit. On long engines a centering strip with a hole should be placed in the piston-rod stuffingbox, and the guide must be checked at the cylinder end for central position. Before placing the frame, cylinder or any other heavy castings on the foundation, it is advisable to check the holes for the foundation bolts and make sure that they correspond with those in the foundation. It sometimes happens that the holes in the frame are not cored all the way through or that they are crooked and must be bored out. Any discrepancy of this kind is much easier made good before the frame is placed in position than after. The safest way is to test every hole with a piece of pipe or a bar of the same diameter as the bolt.

As soon as the engine is aligned and leveled by the use of steel wedges and plates, the foundation bolts are drawn up moderately tight without springing the castings. On large work the anchor bolts should not be put in until the engine is in place. The outer bearing is next placed on its foundation, lined up with the cross-line stretched at right angles to the center line of the engine, as previously described, and adjusted to the same level as the main bearing by the use of a straight-edge. On direct-connected units the sole plates for the generator are placed in position to suit, leveled and squared by the center line of the shaft. The crankshaft is put into the bearings, leveled and squared. In turning the crankpin to either of the dead centers the distance from the face of the crank to the center line of the engine can be checked. Placing the crankpin in the upper and lower vertical positions and measuring the distance from the face of the crank to a plumb-line touching the main center line, will show whether the shaft is level and square with the engine. All four distances must be alike; if there is any difference, the shaft must be adjusted to suit.

The engine is now ready for grouting. A dam of bricks or boards, banked with sand on the outside, is built 3 or 4 ins. from the castings all around the parts to be grouted, and a mixture of one part of cement and one part of sand stirred in water to the consistency of thick cream is poured in. The dam should be about 2 ins. higher than the base of the castings to make the grouting rise well into the inner part of the frame and sole plates. All air pockets must be vented and the grouting stirred while it is flowing to keep the heavy parts from settling. On large frames holes for grouting are sometimes provided in the crank pit, main pillow block and crosshead guides, which are closed afterward by special covers or regular pipe plugs. Ribs are often cast on the under side of sole plates and frames to give them a more substantial grip on the foundation, especially on rolling-mill engines where an end-thrust on the shaft is likely to occur. To plane the under side of bedplates to be grouted is a waste of time; the rougher the surface the better the grouting will hold.

From one to two days is required for the grouting to set; the dam can then be removed and the wedges and plates be drawn out, after which the holes should be filled with cement and the grouting trimmed off all around the outside. In a few days the foundation bolts may be drawn up tight.

Rust and sulphur joints should not be allowed to be put under an engine; they are antiquated, expensive and unreliable—especially sulphur, which gets brittle and is easily affected by oil.

While the masonry is being done, all machined parts of cylinder, frame, etc., should be covered with a coat of cylinder oil; crankpin and shaft must be protected with burlap, and oil holes and handholes should be filled with waste.

The flywheel may now be erected and also the generator if the engine is direct-connected. If the wheel is made in halves, it is advisable to lower one-half into the flywheel pit before the shaft is put into the bearings; it should be blocked up tight against the shaft, the other half placed over it and the two connected by means of the hub bolts, which are generally heated before they are inserted. In cooling they will contract and, with the nuts drawn up tight, will give the wheel a good grip on the shaft. In order to make the hub fit well, it should be bored two or three thousandths of an inch smaller than the diameter of the shaft, and a space of not less than $\frac{1}{2}$ in. allowed between the halves for clamping. A feather secured in

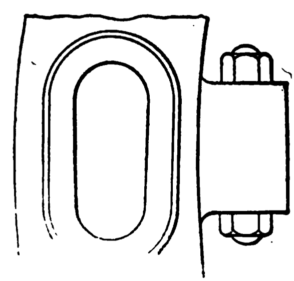


FIG. 245.—Oval ring for flywheel joint.

the shaft and made a good fit on the sides of the keyway in the hub is much to be preferred to a driven key, which should be used only on a solid hub. The rim bolts are put in and tightened and the nuts and bolts are prick-punched to keep the former from backing off. If the rim is held together by links, they must be measured and checked with the recesses to make sure that they will fit. The links are heated and when sufficiently expanded ($\frac{1}{32}$ to $\frac{1}{16}$ in. longer than the recess, according to size) they are put in place.

The old-time links made in the shape of an oval ring, as shown in Fig. 245, are seldom used on new wheels. If welded they are not safe to use; there is never any certainty that a weld is solid even if it appears so on the outside. The rings are sometimes cut out of solid steel, but since they cannot be machined accurately, they can never be made a good fit. Tee-head links, Fig. 246, are the only ones that can be recommended. They should be made of open-hearth forged steel of 0.25 to 0.30 carbon. To use cast steel for this purpose is dangerous for neither

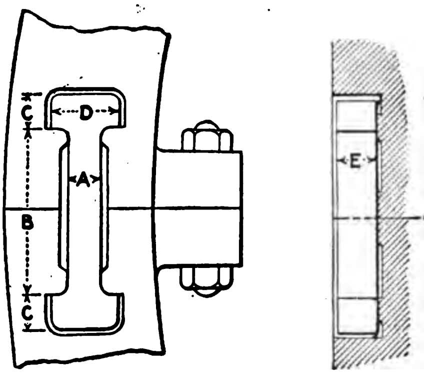


FIG. 246.—Teehead link for square rim wheel.

the quality nor the strength can be relied upon. In order to make the link construction equal in strength to the rim section, the following proportions should be adopted:

$$B = 5A;\ C = A;\ D = 2A;\ E \text{ not less than } A$$

The steel in tension and shear should be 20 per cent of the rim area, and the cast iron in tension and shear should be equal to the rim area. On belt and rope wheels, where bolts are used almost exclusively, the area at the root of the threads in each joint must be not less than 25 per cent of the rim area. The joints in the rim should be planed, and on belt and rope wheels at least one bolt in each joint should be turned and fitted in the hole to prevent side motion. Split joints never make a first-class job.

On a segmental wheel with two solid hub centers on the shaft and arms and rim sections cast separate, the arms are put in first and the rim sections bolted on afterward. It is likely that the wheel was never before in a

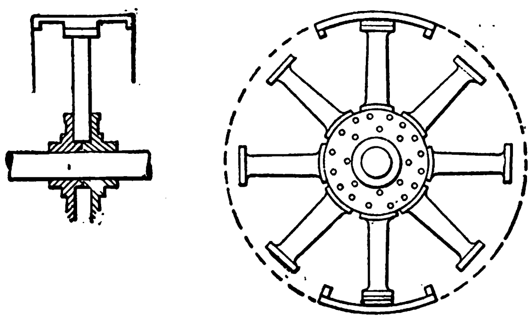


FIG. 247.—Erecting segmental wheel.

vertical position, but was erected on the floor and turned in a vertical boring mill in a horizontal position. It is therefore advisable not to drive home the bolts right away, but to leave about $\frac{1}{2}$ in. between the head and the face of the hub. The bolts and holes should taper $\frac{1}{8}$ in. per foot; straight bolts should not be used on this class of work. With the arms all in place, each segment is put on its respective arm in the vertical position, as shown in Fig. 247, in which it can easily be set to the lines marked in the shop. Two of the bolts connecting the arm and segments must be fitted. It is well to erect the segments opposite each other to keep the wheel as nearly balanced as possible. No bolts connecting the segments should be put in till the last segment is in its place. As the work progresses, a check can be kept on the spaces between the segments by measuring the respective chords, which should be given on the drawing of the wheel. With all the seg-

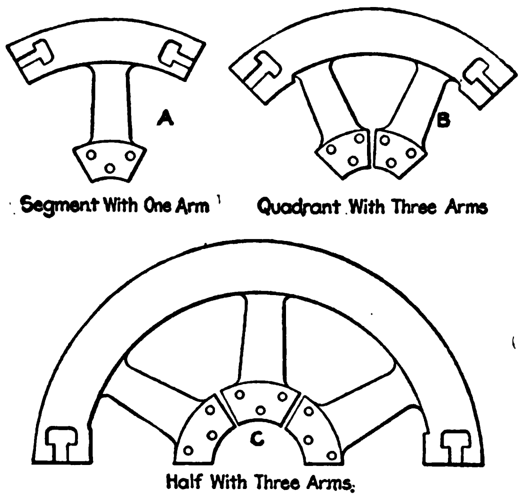


FIG. 248.—Flywheel parts.

ments located, the rim bolts can be put in and tightened and the hub bolts driven home. In a well-designed wheel each arm must be able to support its segment independently of the others and the segments should come together without forcing. Bolts used on flywheels should have a fillet under the head equal to $\frac{1}{16}$ of their diameter.

Shrinkage strains and blowholes in castings and the restrictions of the transportation companies as to size and weight are responsible for built-up wheels. The number of pieces and joints in wheels should be kept as low as possible. On large band wheels it is not feasible to cast the segments and arms together, but on square-rim wheels at least one segment and arm should be cast in one piece. Of the three designs in Fig. 248, *A* shows a segment with one arm, *B* a quadrant with two arms and *C* a half with three arms cast on. There is no connection between the arms at the hub, leaving them free to contract in the mould without offering any resistance

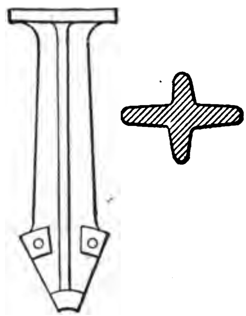


FIG. 249.—Rib section flywheel arm.

to free shrinkage of the rim. It is not good practice to make flywheels with a large number of frail arms; the arms must be in proportion to the rim. Light rims and light arms are gradually disappearing; wherever they are still in use, there is constant danger of accidents. Rib-section arms, as shown in Fig. 249, should be avoided, they may develop cracks near the outer flange and, if run at considerable speed, throw off the air like a fan.

The oval arm has proved to be superior to any other shape and should always be used. Each arm should have three holes for bolting it to the hub centers—two holes, as shown in Fig. 249, are not sufficient. The practice of casting rim and arms hollow cannot be recommended, as the cores might shift in the mould, which would distribute the metal unevenly and make the wheel unsafe. Wheels, like steam cylinders, should be cast of a special grade of close-grained iron, which has no tendency to form blow-holes. Iron showing an open texture is not suitable, even if a test bar of it should show high tensile strength. In fact, test bars hardly ever represent the actual strength of a casting unless they are cut out of solid metal of the same thickness as the segment or arm and cast either in one piece with them or under exactly the same conditions and from the same ladle. Small bars, whether cast on or loose, are of no practical value.

The stress in the rim of cast-iron wheels, resulting from centrifugal force, must not exceed 1000 lbs. per square inch. The simplest kind of calculation that can be made will show whether this figure is exceeded in a wheel or not: Multiply the outside diameter of the wheel in feet by 3.14, which will give the circumference; multiply this by the number of revolutions per second, square the product and divide by 10. As an example, take a 15-ft. wheel running 120 revolutions per minute. On heavy square-rim

$$15 \times 3.14 = 47.1$$

$$47.1 \times \frac{120}{60} = 47.1 \times 2 = 94.2$$

$$\frac{94.2^2}{10} = 887 \textit{ lbs. per square inch.}$$

wheels the bending stresses are negligible, but in belt wheels they must be carefully considered.

The cylinder must be thoroughly cleaned in the ports. Steam and exhaust chests and bore are blown out with steam or air. All core sand left in the casting or grit blown into it in transportation or erecting must be removed. The throttle valve should be put on, making the engine ready for the pipe connections to the boilers. The piston, rod and crosshead should be put in and lined up.

The guide should have marks at each end to indicate the length of the stroke and the clearances in the cylinder by bringing a mark on the crosshead line on line with

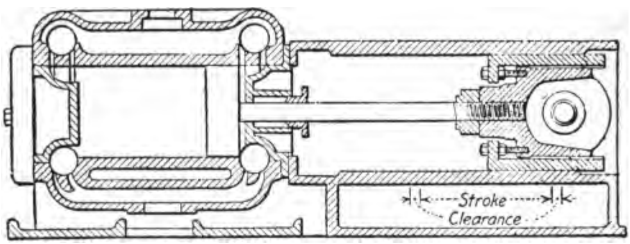


FIG. 250.—Clearance marks on guides.

them, as in Fig. 250. This must be verified by moving crosshead and piston from one end to the other until the piston strikes the heads. If the piston rod has a threaded end, it should never be screwed into the crosshead with pipe tongs; that would leave deep and ineradicable marks on the rod and ruin it. A stout rope should be doubled and wound a sufficient number of times around the rod to give it a good hold, and a bar inserted in the loop; then, using the rod as fulcrum, it is comparatively easy to turn the piston rod. A piece of soft sheet copper inserted between bar and rod will protect the latter from getting bruised. The arrangement is shown in Fig. 251.

The common practice is to give Corliss engines from ¼- to ½-in. clearance at each end of the cylinder. This is

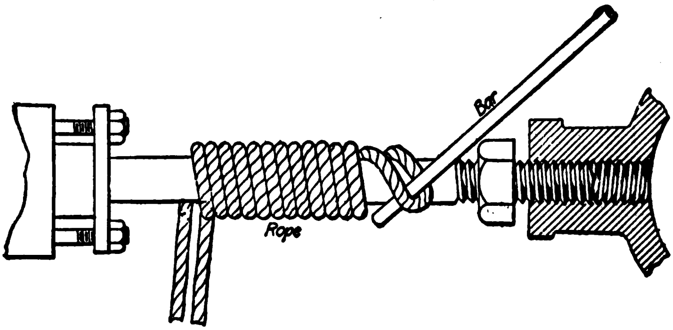


FIG. 251.—Screwing in the piston rod.

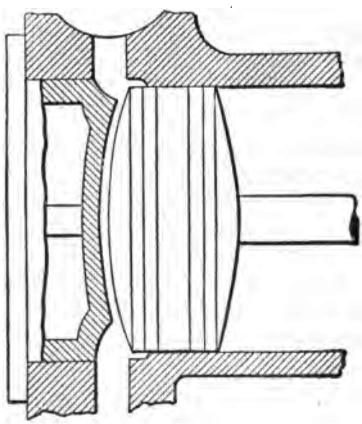


FIG. 252.—Piston with curved surfaces left rough.

too much and results in a loss of steam. Besides, the faces of the pistons and cylinder heads are often not properly machined; in some engines they are left rough, with curved surfaces as shown in Fig. 252. The reason for all this, of course, is to cheapen the product. Large clearances and rough surfaces are detrimental to good steam economy. The loss through clearance and initial condensation often exceeds 25 per cent of the total steam used. On many engines this could be reduced to 10 per cent, making a saving of 15 per cent.

In putting the connecting-rod in place it should be ascertained whether the crosshead pin and crankpin are in line and parallel. For this purpose the crank end of the

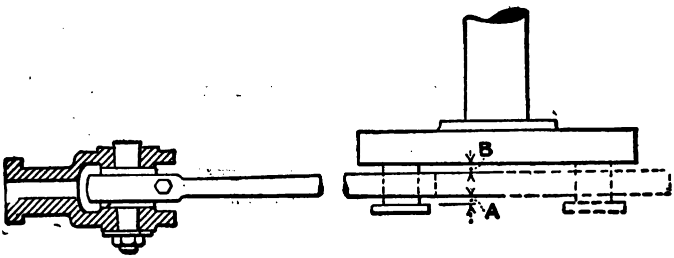


FIG. 253.—Determining if the crosshead and crankpins are in line and parallel.

rod without the boxes is slipped over the crankpin and the crosshead end attached to the crosshead and adjusted tight on the pin by pulling up the wedge. If the two pins are in line the distances *A* and *B* in Fig. 253 will be equal for any position of the crankpin and the sides of the slot in the rod will be parallel with the pin. The crankpin boxes should next be put in and adjusted and the dead centers of the crank located. To do this lay off a certain distance, say 2 or 3 ins., from the line on the guide which indicates the end of the stroke, as shown in Fig. 254, for the crank end. Make a gauge 24 ins. long, shown in detail, and secure a steel pin in the floor near the rim of the flywheel. Now revolve the wheel until the line on the crosshead coincides with the line 2 ins. from the end of the stroke and mark the rim, using the gauge set into a center-punch mark on the pin; then turn the wheel till the line on the crosshead on the return stroke again registers with the line on the guide and make another mark on the rim with the gauge. In dividing the distance between two marks on the rim, a point is located

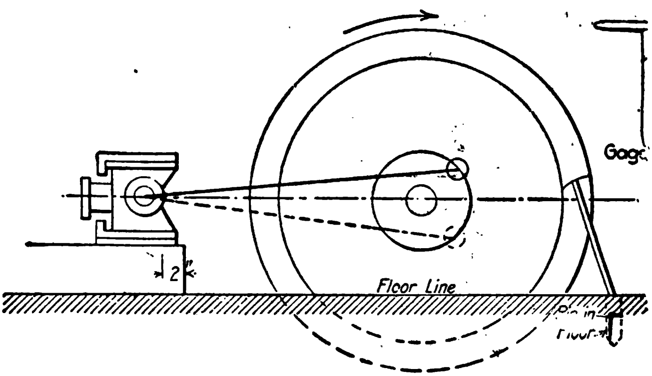


FIG. 254.—Locating the dead centers.

which, when brought to the position in regard to the fixed point in the floor determined by the gauge, will show that the crankpin is on the center at the crank end. The dead-center at the head end is established in a similar way.

To facilitate turning over the engine by hand, the flywheel should have holes cored or drilled in the rim for barring. On square-rim wheels holes $1\frac{1}{2}$ to 2 ins. diameter or square, according to the size of the wheel, spaced 6 ins. from center to center, should be located in the face of the rim, as shown at *A*, Fig. 255, while on band and rope wheels they are drilled in one side or one of the return flanges is cast with teeth of about 4-ins. pitch. Both these methods are shown at *B* and *C*. A simple cast-iron stand fastened on the floor near the rim will act as a fulcrum for the bar, or a pawl and lever supported by a stand on one side of the wheel may be used with the toothed rim.

Eccentrics are made solid or split according to the service for which the engine is intended. Split eccentrics are almost exclusively used on direct-connected engines.

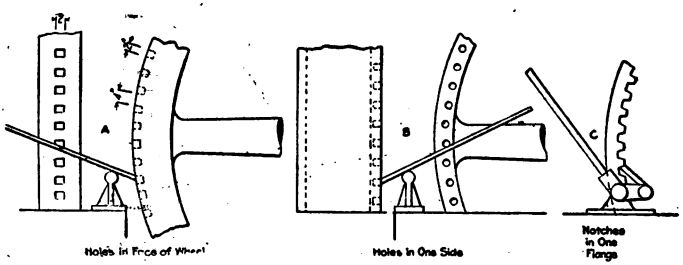


FIG. 255.—Stands for barring the engine over the center.

Corliss-engine eccentrics must not be keyed on the shaft, but should be provided with sets-crews large enough to hold them securely in place without any danger of slipping. The bore should be recessed in the middle about $\frac{1}{16}$ in. deep, as shown in Fig. 256, to allow the eccentric to clear burrs on the shaft, made by forcing the hardened, cup-shaped end of the set-screws into its surface. The straps must have an easy fit on the eccentrics. There is no special advantage in having them babbitted except that when they run hot the babbitt will melt without scoring the eccentric. Cast iron on cast iron makes a good wearing surface if the lubrication is ample and the supply of oil is constant.

Some engine builders do not place the shaft in the bearings in the shop, neither do they adjust the eccentric rods. This is bad practice, dictated by a mistaken idea of saving time. Severe competition is responsible for such methods, and they should not be tolerated. An engine should be finished and erected complete in the factory, with the exception of the flywheel on the larger sizes, and everything should be marked carefully before

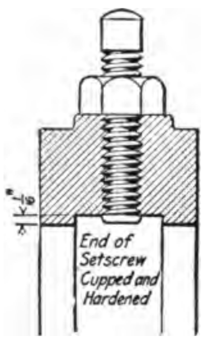


FIG. 265.—Set screw for eccentric.

dismantling. It is unsatisfactory to leave a lot of work to be completed in the field by the erecting engineer. Manufacturers who have not the facilities, especially in the older shops, will sometimes erect one side of a cross-compound engine in one part of the shop and the other side in another. The purchaser should be careful about accepting such work; the best thing for him to do is to employ, as inspector, an experienced engineer, who will make sure that everything is complete before the engine is shipped.

Buyer's Interest in Shipping Engine

The skidding, packing and shipping should also be watched whether the engine is shipped at the risk of the purchaser or not. In case the contract calls for delivery and erection on the purchaser's foundation, it may appear that he is not concerned in that part of the transaction, while, as a matter of fact, it is much to his interest to have the machinery well protected and carefully handled. For instance, if the cylinder and frame are shipped bolted together, the bolts may be unduly strained in transportation and give trouble at some future time. Finished surfaces should not be exposed to the elements or careless handling; they must be coated with heavy oil and protected against damage by boards.

An engine should be provided with ample oil guards over the crank and eccentrics. Large openings in the front of the crank oil guard should be covered with brass-wire gauze of No. 22 mesh to prevent oil from being thrown on the floor. The gauze should be fastened to a frame or door that can be easily removed or opened. Handholes must be provided for feeling the connecting-rod and eccentric straps.

The best material for oil guards is planished steel; it can easily be kept clean and will not show oil or grease like painted plates. Galvanized iron should not be used; the zinc coating is liable to come off in places and spoil the appearance.

A gravity oiling system with filter and storage tank is desirable, for it will save its cost in a short time and give much better results than drop feeding by means of oil cups. It is more advantageous to pour oil in streams over the bearings and pins (thus reducing their tendency to run hot) and to use the oil over and over again than to feed it in small quantities, most of which goes to waste. Before oil fittings or oil cups are put on, the oil holes must be cleaned and the engineer should make sure that oil poured into them will reach the part for which it is intended and not run along the sides of caps and bearings without touching the vital spots.

The most approved method of cylinder lubrication is by means of a force-feed oil pump, which may be driven from the wristplate or other convenient part of the valve gear. On small engines one feed into the steam pipe, 2 or 3 ft. above the throttle valve, is sufficient provided the oil is well diffused by the steam, but on engines with cylinders 18 ins. diameter and over there should be at least three feeds—one into the throttle valve and one into each end of the steam chest near the valves, where the inrushing steam has a chance to spray oil over the length of the port. The hardest places for oil to reach are the ends of the steam valves, especially on low-pressure cylinders. Hand-oil pumps should be provided here for periodical application of oil. Parts of the valve gear that only occasionally require a few drops of oil may be oiled by hand, while others, like the valve rod and rocker-arm pins, may be lubricated by means of grease cups.

Best Paint for the Engine

To get a lasting coat of paint on an engine, the ordinary shop "paint" should be removed, all cavities and holes cleaned and filled and the castings rubbed smooth with sandpaper or pumice stone. An ordinary house painter or decorator, as a rule, has not the ability to do this work; if the paint is to stay on the engine it must be put on by a coach painter. It must not be affected by extreme changes of temperature or by oil and grease. These are severe conditions, which can only be met by excellent work and materials. The varnish especially, of which the engine should receive two coats, should be of the best quality. The color is, perhaps, a matter of taste; dark colors are preferable on account of showing the oil less than lighter shades. Bottle green seems to undergo less of a change when exposed to heat than red or maroon. If any striping is desired, it should be done with gold leaf. Here again, expert work will show; to daub an engine with green, red or yellow stripes is bad taste. On some engines we see even landscapes or other pictures, making them look like circus wagons.

CHAPTER XIX

VALVE SETTING FOR CORLISS ENGINES

Lap lines on valves and port holes. Adjusting dashpots. Adjustment of governor levers. Locating the eccentric. Double eccentrics. Compression. Endplay of valves. Testing governor by hand.

WITH the eccentrics and straps lined up, eccentric rods connected to the straps and rocker arms and the valve-gear parts in position, the setting of the valves can take place. On a wristplate motion, whether driven by single or by double eccentrics, the wristplate or wristplates, as

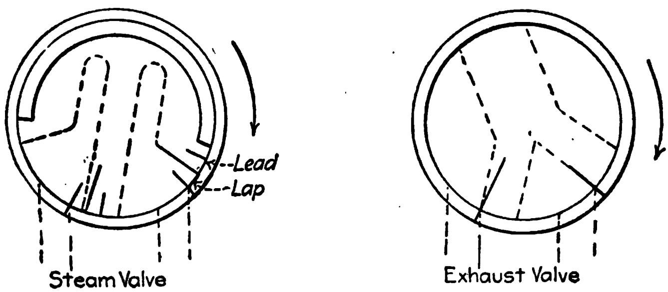


FIG. 257.—Markings of valves and ports.

the case may be, must first be placed in the central position, which is indicated by lines on the hub and stand; the valve rods are then adjusted with all the valves on the lap. There are lines for this purpose on the back of the valves and in the counterbore of the port holes, as shown in Fig. 257, marking the cutting edges and the lap of the valves and the edges of the ports. There should also be a line on the steam valves for the lead; but this is often omitted and causes much trouble if left to the discretion of the erecting engineer, whose aim may be to get a nice-looking indicator diagram, showing a vertical admission line, with the result that a pound will develop in the engine from an excess of lead. Next, the wristplate is moved to the extreme positions marked on the stand, and the dashpot rods are adjusted for length so that with the dashpots in their lowest position the catchplates engage with about $\frac{1}{16}$-in. space between. The nuts on the valve and dashpot rods must be set up tight to prevent their shaking loose when the engine is running.

Adjusting Knockoff Levers

The knockoff levers must be connected to the governor and adjusted. For this purpose the governor is blocked in the lowest working position, as shown in Fig. 258, in which the trip levers should just touch the cams when the wristplate is moved to the extreme positions; no cut-off can take place for this position of the governor; if steam is turned on, the engine will take steam full stroke with the dashpots being moved up and down by the steam levers. As soon as the governor lifts $\frac{1}{8}$ in., cut-off must begin. The cams must be adjusted so that cut-off starts practically at the same time at both ends. This part must be tried several times by hand to make sure that the dashpots come down properly. Advancing the governor to the highest position and working the wristplate forward and backward should result in tripping the valve gear before the steam valves open. This will prevent steam from entering the cylinder. A collar on the spindle keeps the governor from rising above this point. It is next lowered to the safety-stop position and the safety cams adjusted to prevent the valve gear from hooking up, the steam arm and the dashpots remaining stationary. To make sure that the wristplate travels correctly, the hook rod is connected and the eccentric moved around the shaft.

To locate the eccentric in the correct position on the shaft in relation to the crankpin, place the latter on one of the dead-centers and move the eccentric in the direction of rotation around the shaft until the steam valve

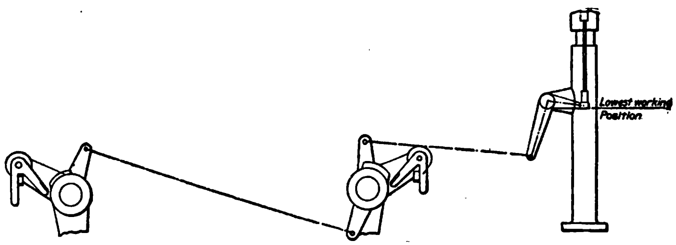


FIG. 258.—Trip levers touching the cams with governor in lowest working position.

that corresponds with the position of the crankpin is on the lead. Now fasten the eccentric and try the other end. If the steam valve at that end also shows the correct lead, the eccentric should be permanently secured, and its position marked on the shaft. On single-eccentric engines, release and compression of the exhaust valves are limited by the lap and lead of the steam valves; therefore, where greater flexibility of the valve motion is desired, double eccentrics must be used. It is safe to state that engines running 100 r.p.m. and over should be provided with double eccentrics to permit setting the valves for an early release and sufficient compression to counteract the inertia of the reciprocating parts.

Function of Compression

Compression weakens the engine, but it serves two purposes: First, to increase the temperature of the steam confined in the cylinder and clearance spaces after the exhaust valve is closed, thus reducing initial condensation and second, to form a cushion for the piston, lessening the shock on the bearing and pins. In the double indi-

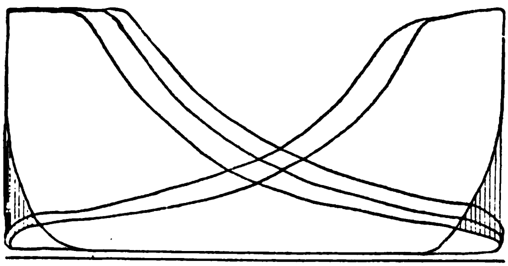


FIG. 259.—Diagram showing cushioning effect of compression and release.

cator diagrams, Fig. 259, the shaded part illustrates the cushioning effect of early release and compression, without which an engine cannot long run successfully. It is quite evident from the diagram that with a later cut-off the cushion is reduced, which accounts for the pound often noticed when a heavy load is suddenly thrown on. With sufficient cushion, an engine does not have to be keyed tight, thus allowing a thicker film of oil to form between the bearing surfaces, which reduces wear.

Before the back bonnets are screwed on, the valves must be checked for end play, which depends partly on the thickness of the gaskets. As a rule, the length of the valves should be equal to the distance between the inner faces of the bonnets and the back bonnets without gaskets. Making the latter of material $\frac{1}{32}$ in. thick will make the clearance $\frac{1}{16}$ in., which is ample. Steam valves expand more than exhaust valves on account of the higher temperature to which they are exposed—a point which must be watched on a new engine. Exhaust valves are more likely to pound if too short, especially on condensing engines. The valves should be well covered with cylinder oil when put in place; graphite dusted over them and the seats is beneficial.

The governor should be examined carefully; all its parts must be free to move without undue friction. A fly-ball or weighted governor running up to 200 r.p.m. can be tested by fastening a handle to the gear shaft and turning it by hand; its action will show whether it will regulate at the predetermined speed. While this is not an accurate test, it will show defects and give a fair idea of the promptness with which the governor will respond to a change of speed. All governor rods must be free on the pins, with at least $\frac{1}{32}$-in. side play. The governor belt must be endless; that is, the joint must be glued. This is usually done in the evening to give it a chance to dry overnight. Belt lacing or metal belt fasteners should not be used.

CHAPTER XX

OPERATION

Warming up an engine. Finding leaks. Short runs in starting. Spare parts and packing material. Tools. Drawings. Indicator piping. Access to indicator must be free. Simple reducing motion. Reducing motion connected to crosshead pin oiler. Inclined plane. Pantograph. Reducing wheels. Checking indicator spring and pressure gage. Pressure losses. Determining the clearance. Indicated horse-power. Several diagrams on one card. Governor regulation. Graduated scale and tachometer to show speed variations. Examine new engine. Crank and crosshead pin boxes must be relieved at the parting. Babbitting boxes. Engine adjustment. Cooling hot boxes. Compounding.

After all the lubricating devices have been put on and tested, the engine is ready to be warmed up. The steam pipe must first be thoroughly drained; next, steam is let into the steam chest either through a by-pass or by opening the throttle about half a turn. Throttle valves over 5 ins. should be provided with by-passes. If water gets into the cylinder it should be drained off through the exhaust by working the valve gear forward and backward by hand. When thoroughly warmed the valves and piston may be tested for tightness. By letting steam into one end of the cylinder and opening the indicator cock at the other, a leak in the piston can be detected; and by closing both steam valves and opening the indicator cocks, a leak in these valves can be noticed. If the exhaust valves leak, it will show by steam blowing out of the exhaust pipe when the valves are closed. A little steam will frequently blow through on a new engine; not much attention need be paid to this unless the leak becomes serious. The governor should now be raised to the starting position and the engine turned over slowly; everything must be watched closely and plenty of oil pumped into the cylinder and poured on bearings, pins and other moving parts.

Adjustments after Short Runs

After running a while the engine should be shut down and all the nuts and bolts about the cylinder examined and tightened. If no other adjustments are found necessary, another start may be made, giving the engine a little more steam and increasing the speed until the dashpots begin to drop and the engine is under the control of the governor. This may be continued for about half a day, which will give everything a chance to get into working condition. A mixture of graphite and cylinder oil pumped at intervals into the steam chest will benefit valves and piston, as stated before.

If everything runs satisfactorily, the engine is ready for a trial run with load, which should be increased gradually. With a light load a new engine should be run either with low boiler pressure or with the throttle partly closed until the valve gear and the dashpots are working freely. The engine may be indicated with a view of adjusting the valves. Of course economical performance must not be expected the first few days after starting.

Before the engineer takes charge of an engine, he should prepare himself for any emergency. Wrenches for the various sizes of nuts should be mounted on a board in a convenient place on the wall; on the same board should be eye-bolts for lifting the main-bearing cap, quarter-boxes and shells, for removing the back cylinder head, piston follower and junk ring, and for taking out the valves. If no spare parts have been furnished with the engine he should see to it that he gets extra sets of cross-head and crankpin brasses with some wedge bolts, a packing ring for the piston and a few follower bolts and nuts. For Corliss engines there should be one spare steam bonnet and one exhaust bonnet with a valve stem for each, an exhaust arm, a full set of catch plates and some springs if any are used on the valve gear.

Sheet-rubber or copper gaskets for joints that may blow out should be kept in stock. There should be an assortment of good files, scrapers and chisels, and a work-bench with a vise and a drawer in which to keep the small tools. Extra pipe fittings and some brass pipe should be provided for small repairs and changes of the oil piping. The builders should furnish detail drawings of all the moving parts with a chart or diagrammatic drawing showing the names and positions of the principal parts of the engine. This the engineer must have, so that no time may be lost owing to misunderstanding if the necessity arises to order quick repairs.

Every engine should be provided with indicator piping. The arrangement, Fig. 260, should show a long-radius elbow cock at each end and a three-way cock in the middle, the most convenient for everyday use, when it is more a question of correct valve setting and average load than of absolute accuracy. For a test two indicators should be used—one of them to be placed at each end of the cylinder with the shortest possible connections. The most direct way is to place the indicators in a horizontal position, with the straightway cocks screwed into the couplings on the short pieces of pipe, as shown in Fig. 261.

Access to the hole in the side of the cylinder must be free; the hole should have beveled or rounded edges,

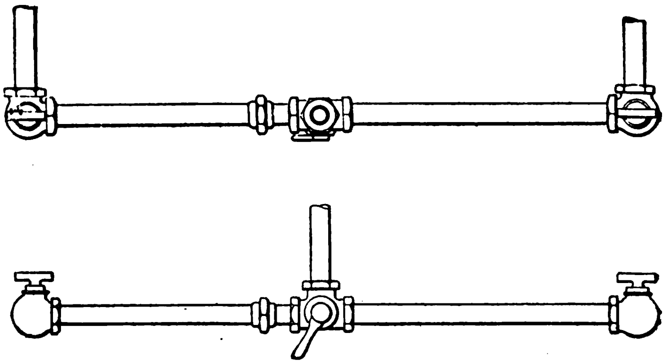


FIG. 260.—Indicator piping.

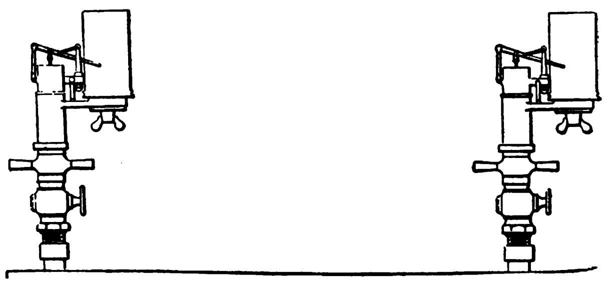


FIG. 261.—Two indicators in horizontal position on the side of the cylinder.

and the nipple should taper on the inside, as shown at *A*, Fig. 262. The piston must not overrun the indicator hole, preventing free access at the beginning of the stroke; neither should there be sharp edges in the cylinder or nipple, as shown at *B*, which will cause a drop in initial pressure in the indicator. These minor details are often lost sight of in indicating an engine. If the horizontal position of the indicators is inconvenient, they may be placed vertically on long-radius elbows turned upward, as shown in Fig. 263 (in which position they are readily manipulated with only a trifling loss in pressure).

Before placing the indicators in position, all pipes,

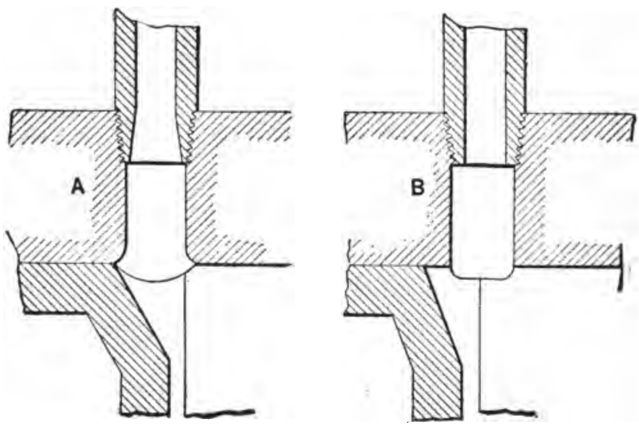


FIG. 262.—Indicator hole and nipple in side of cylinder.

fittings and passages must be blown out to free them from grit, gummy oil or other foreign matter to prevent the instruments being fouled. In long indicator pipes the condensation is often a disturbing factor; the water must always be blown out before a card is taken, otherwise the diagrams will be unreliable. It is evident that, whatever arrangement is adopted there is apt to be some inaccuracy, therefore, the one that is likely to give the least error should be chosen.

An indicator is almost indispensable in an engine room. The medium-priced instruments with inside spring

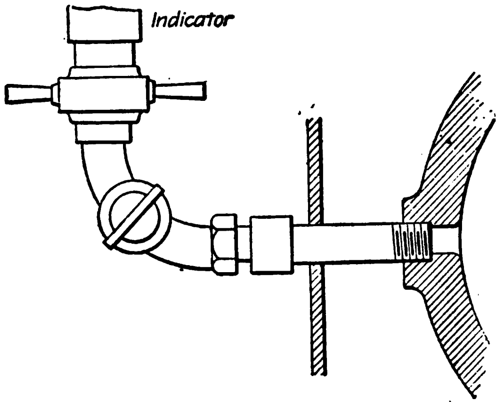


FIG. 263.—Elbow turned upwards to place the indicator in a vertical position.

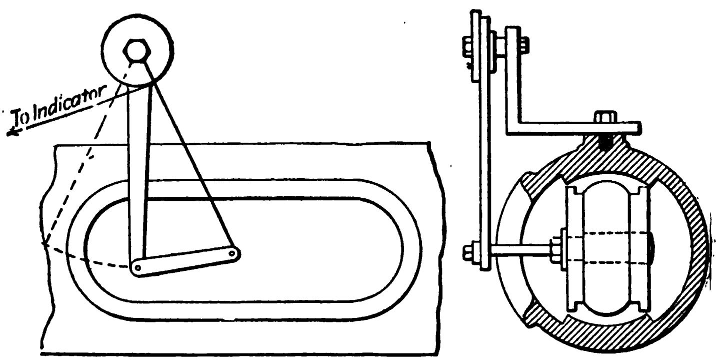


FIG. 264.—Lever reducing motion.

will give fairly good results for everyday use, either with permanent reducing motions furnished with the engine or with reducing motions driven by cords and attached to the indicator. For accurate tests the best instruments, with calibrated outside springs and electrical attachment for instantaneous operation, must be used; the reducing motions must be of the highest order, with the cords running in a straight line to the crosshead.

A simple reducing motion for temporary use may be made as shown in Fig. 264. A bracket made of a bar of

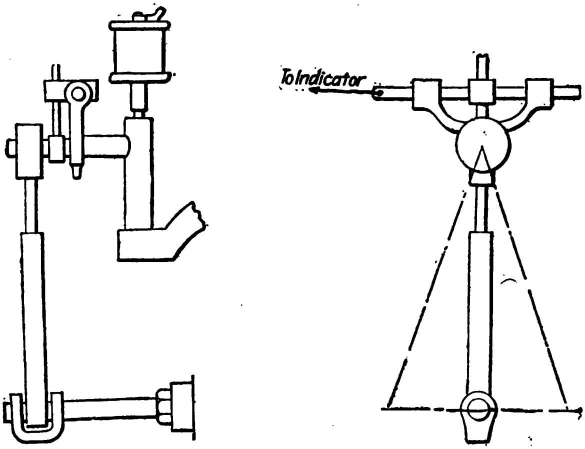


FIG. 265.—Telescopic reducing motion.

flat steel is fastened to the top of the guide; with a wooden lever suspended from a bolt at the upper end. The connection to the crosshead is by means of a horizontal link, also made of a strip of wood, and a sector of the required size to give a motion of about 4 ins. at the circumference is fastened to the lever at the pivot. On high-pressure cylinders the cord can usually be attached directly to the indicator, but on some low-pressure cylinders, on account of their width, a couple of carrying pulleys must be used. If there is no vibration to the bracket and if the bolts are well fitted with brass bushings in the lever and link, the arrangement will give good service. Of course it is not intended for permanent use and does not improve the appearance of an engine.

Something of the same order for a permanent rig is shown in Fig. 265. This works on the telescopic principle and serves the double purpose of crosshead-pin oiler

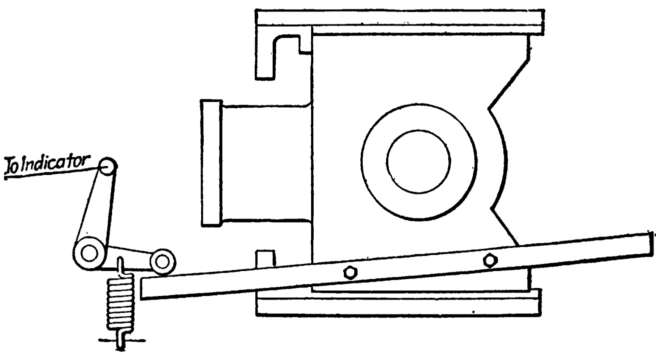


FIG. 266.—Inclined plane reducing motion.

and reducing motion. Running all the time, the device soon begins to wear and loses its accuracy; besides, the telescopic oiler is liable to get clogged and leaky, which may be the cause of a hot pin.

A permanent reducing motion that can be engaged or disengaged at will is shown in Fig. 266. It consists of an inclined plane in the shape of a steel bar fastened to the crosshead; a two-arm lever with a roller on the horizontal arm running on the bar transmits the motion to the indicator. The device can be thrown in and out of gear by securing the lever in such a position that the roller does not touch the bar.

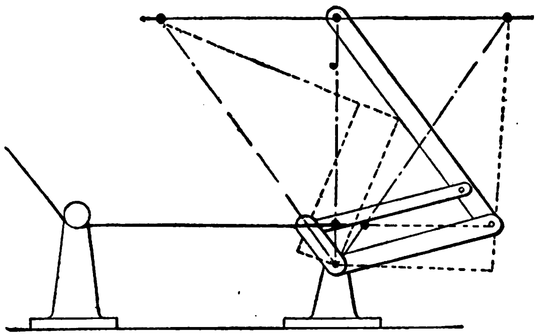


FIG. 267.—Pantograph reducing motion.

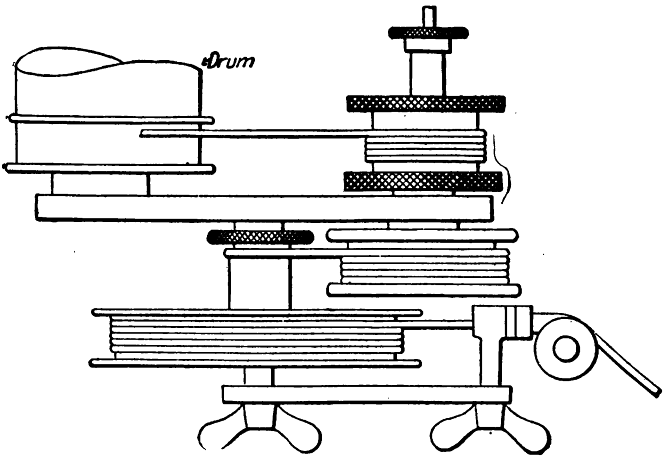


FIG. 268.—Reducing wheel.

Fig. 267 represents a pantograph in its simplest form, with the fulcrum pivoted on a stand fastened to the floor.

The cord is led over rollers to the indicator. The apparatus can be connected to the crosshead or disengaged only when the engine is stopped.

Reducing wheels attached to the indicator and driven by cords, as shown in Fig. 268, are largely used. There are three cords used—one from the crosshead to the large wheel, a second one from a small pulley to a spring case

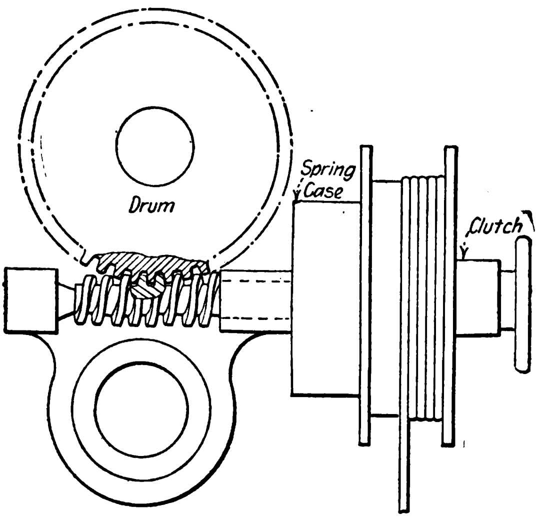


FIG. 269.—Worm wheel reducing motion.

and the third from another pulley to the indicator drum. On account of the number of cords the device may become troublesome; if the cords run unevenly or ride, the indicator cards will be inaccurate.

A more reliable and easily operated apparatus is shown in Fig. 269. Only one cord is used between the crosshead and the spring case. The latter is provided with a clutch that enables the operator to stop the drum of the indicator at any time for removing and renewing the card without disturbing the cord to the crosshead. The paper drum can also be stopped by turning a thumb-piece on top of the drum. The reduction in stroke in this reducing motion is made by means of a worm and wormwheel, the latter forming the base of the drum. There are only a few parts in the mechanism, which materially reduces the chance of anything wearing out or getting out of order.

A new engine should be frequently indicated and samples of the diagrams calculated, analyzed and placed in a record book for reference. The diagrams should be about 4 ins. long and not over 2 ins. high, as this is the limit of the pencil mechanism. In measuring the height of the diagram from the atmospheric line with the same scale as marked on the spring, the initial pressure in the cylinder can be found; comparing this with the boiler pressure will show the drop between boiler and engine. The distance between the exhaust line and the atmospheric line shows the back pressure in the cylinder on the return stroke of the piston. Of course both the steam gauge and the indicator spring may be incorrect. They may be compared as follows:

Place the indicator on a suitable pipe connection at the boiler, then raise steam and as it rises draw lines on a card by moving the drum by hand; when the gauge registers 10 lbs., 20 lbs., etc., up to the working pressure, mark the pressures on the card, as shown in Fig. 270, and compare them with the scale. If there is any marked difference between the two, a test gauge will have to be procured, attached to the boiler and the regular gauge checked with it for the different pressures, as before; comparing the readings on the gauges with the straight lines on the card and the scale will show what allowance should be made in determining the pressures on the diagrams taken from the engine.

The difference between boiler pressure and initial pressure in the cylinder should not exceed 5 lbs., neither should the back pressure be more than 1 lb. above atmospheric, unless the exhaust steam is used for heating. In condensing engines the difference between the vacuum in the condenser and in the exhaust at the engine should not

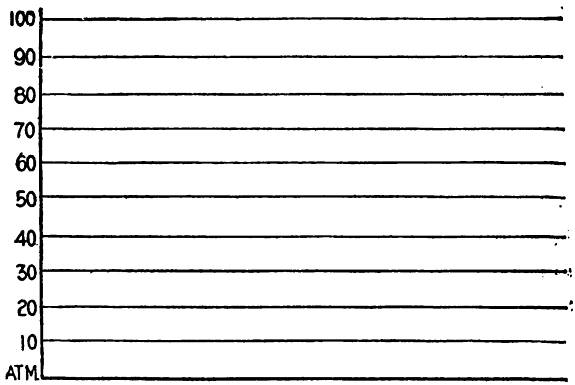


FIG. 270.—Test card for boiler pressures

exceed 2 ins. of mercury. Short and numerous pipe elbows or bends, long pipes or pipes of too small a diameter are often responsible for a reduction in pressure.

A loss in pressure may also occur in the steam chest or in the valves and ports. This can be ascertained by attaching an indicator to the steam chest, taking cards from it and from the cylinder and combining the two, as shown in Fig. 271, in which the drop in pressure is indicated by the difference in height at both ends of the diagram. On well-designed engines the drop should not exceed 2 lbs.

To make a comparison between the expansion curve of a diagram and the theoretical, or hyperbolic, curve, the clearance in the cylinder, ports and valves will have to be determined, either by figuring it from actual measurements or by measuring it with water. The latter method is the more correct one, provided there are no leaks in the piston or exhaust valves. The water may be poured in through the steam port with the steam valve removed

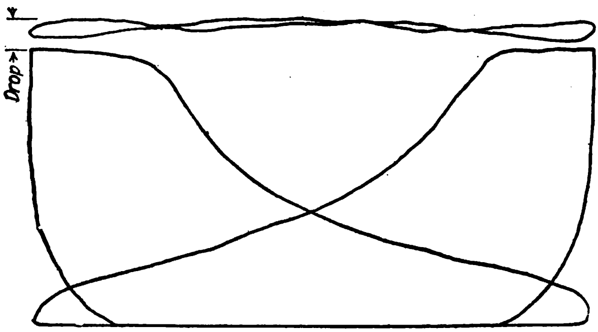


FIG. 271.—Combined cards of cylinder and steam chest.

until it stands level with the port. The piston should, of course, be at the end of the stroke and the corresponding exhaust valve closed. By using a graduated measure for filling the clearance space, the quantity of water in gallons or pints will be known; converting these into cubic inches and dividing by the piston displacement will give the percentage of clearance. With the clearance known, a line representing it and another line for the vacuum must be drawn on the diagram, as shown in Fig. 272, and taking the terminal pressure as the starting point, the isothermal expansion curve is plotted. Any considerable difference between the theoretical and the actual expansion curves of the diagram will indicate leaks in the piston or valves, which can be located by the use of the indicator cocks, as previously explained.

To figure the indicated horse-power of an engine, the mean effective pressure on the piston must be calculated either by means of a planimeter or by drawing a number of ordinates and determining the mean height of the

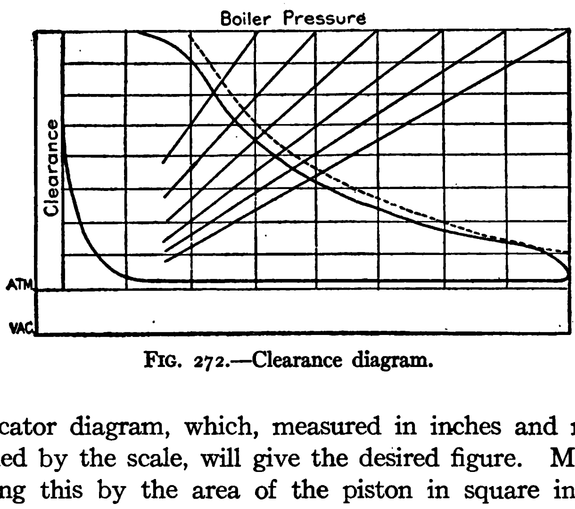


FIG. 272.—Clearance diagram.

indicator diagram, which, measured in inches and multiplied by the scale, will give the desired figure. Multiplying this by the area of the piston in square inches and the piston speed in feet per minute, divided by 33,000 gives the indicated horse-power developed when the diagram was taken.

To get a fair idea of the average load and the fluctuations of the governor, more than one revolution should be recorded on the diagrams by holding the pencil on the card during half a dozen strokes of the piston for each end of the cylinder. This will produce diagrams similar to the one shown in Fig. 273. By repeating the process at intervals of half an hour for a whole working day, a good idea of the load conditions can be obtained.

In factories where the power is transmitted to separate departments, floors or rooms, records may be made of the power required for each of them if the engine is indicated as follows: The first set of cards is taken with the full load on, the next set with one room or department thrown off, then with two off, three off, etc., the last set showing only the friction load. After computing

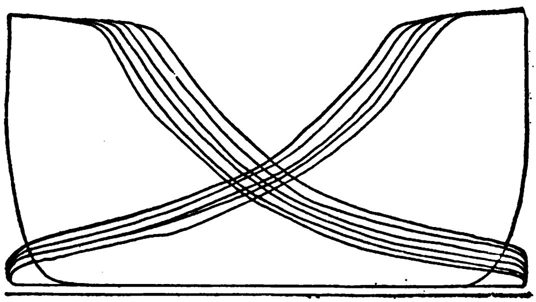


FIG. 273.—Several diagrams on one card.

the average load of each set of diagrams and subtracting them from each other in the same succession as taken, the load for each department can be found.

The speed fluctuation of an engine should be confined between certain limits. This is easily attained for steady loads, but for sudden and great variations of load, special features must be incorporated in the design. The first consideration in this respect is the flywheel, which must be heavy enough to give a uniform motion. It should compensate for the impulse of the steam on the piston during admission and the gradually decreasing pressure during expansion. It must also neutralize the varying effect of the crankpin effort, never the same at any two consecutive moments, and be able to overcome shocks due to instant changes in the load.

Governor Regulation

The governor can begin to act when a change in the velocity of the wheel has taken place, only after one-half of a revolution is completed, when it will admit more or less

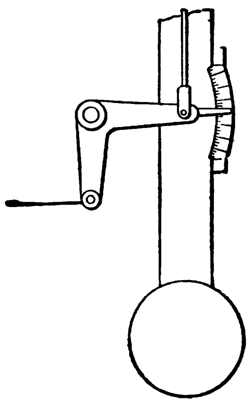


FIG. 274.—Speed indicator on governor column.

steam for the next turn, or stroke. Its object, therefore, is to regulate the amount of steam necessary to keep the shaft and wheel revolving at a certain speed. To what degree of uniformity a governor will regulate the speed of an engine depends on its sensitiveness. Most Corliss-engine governors are provided with a sleeve that moves up and down on a vertical spindle; different positions of the sleeve represent different speeds of the engine. For comparatively steady loads this motion may be utilized to indicate variations in speed, by placing a graduated scale on the governor column with a hand on the lever pointing to it, as shown in Fig. 274, and marking the scale for certain speeds by timing the engine with a watch; but for fluctuating loads and accurate observations a tachometer driven by a belt from the shaft should be used.

Revolution counters cannot be used to show fluctuations of speed; they will only record the number of revolutions that an engine makes during a certain length of time without regard to varying speeds. A recording and indicating tachometer will show the speed of an engine at any moment and also produce a written record on a chart, showing the time of starting and stopping and indicating the fluctuations during twenty-four hours.

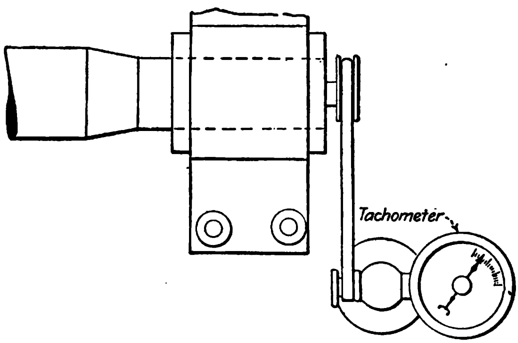


FIG. 275.—Tachometer driven from end of shaft.

On single-cylinder and tandem-compound engines the instrument can be conveniently driven from a pulley fastened to the end of the shaft, as shown in Fig. 275.

This enables the engineer to make a speed test by throwing on the tachometer belt, but on cross-compound engines it is troublesome to connect the driving belt when the engine is running. However, if the speed variation of an engine has once been noted and recorded for a certain length of time and different loads, that will serve as a guide unless some important changes in the valve gear and governor connections have been made since the speed was last tested.

On direct-connected engines fluctuations in speed will also produce a change in voltage. This, however, is not so reliable as a test with a tachometer, since the voltage may also be affected by varying loads of the generator. Recording ammeters and wattmeters afford a convenient and accurate way of keeping records of the load conditions in power plants using direct-connected units exclusively, but indicators and tachometers are of equal importance to keep a check on the performance of the engines. It apparently requires a lot of attention to regulate the performance of an engine, but if a system is once established with a regular routine, it will be found that the work is not only simple, but also interesting.

Examine New Engine

After a week's run a new engine should be thoroughly examined. The valves should be taken out and the high spots " touched up " with a fine file or scraper. The back head must be removed for an inspection of the cylinder bore; if there are signs of cutting, the affected part must be scraped smooth and the junk-ring and the packing ring taken out and carefully smoothed with a file. The crosshead and crankpin and their boxes should be inspected;

if there is any indication of wear or of binding, the boxes must be rescraped and relieved. The effective working surface of the boxes is only about 85 per cent of their projected area; the balance near the parting is of little use—in fact, it may be the cause of pinching and heating the pins. The boxes should, therefore, be relieved at these places, as shown in Fig. 276. This will also allow the lubricant to spread freely over the pins. Before unscrewing the wedge bolts, their position should be marked on the

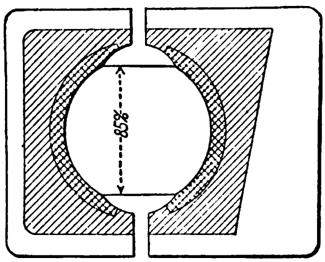


FIG. 276.—Crankpin boxes relieved to allow for wear.

connecting-rod to bring the wedges back to the same place. If a new adjustment is necessary, the wedges may be screwed tight and then backed off to give the boxes sufficient clearance. In most connecting-rods the taper of the wedges is $\frac{1}{8}$ in. per inch; if 1-in. bolts with eight threads per inch are used, one turn will adjust the boxes $\frac{1}{64}$ in. With proper marks on the heads of the bolts and on the rod, a close check can be kept on the adjustment.

Babbitting Boxes

The crankpin boxes must be lined with babbitt of the best quality. For this purpose the recess in the castings must be thoroughly cleaned, scraped and tinned. Before the babbitt is poured the boxes must be heated, and after cooling it should be expanded by hammering. Babbitt poured into boxes not previously tinned and heated is liable to shake loose and break into small pieces. Genuine babbitt is composed of 3 parts copper, 89 parts tin, and 8 parts antimony—an expensive mixture, but the extra expense will be saved over and over again through its excellent wearing qualities. Inferior grades of anti-friction metal are principally composed of lead; they wear rapidly, unless the pressure per unit of surface is low. On babbitted boxes no other part except the babbitt must come in contact with the bearing surface of the pin.

The shells and quarter-boxes of the main and outer bearings should be examined after the engine has been run several days, and if they show unequal wear they must be lightly rescraped. It is not only a needless waste of material to babbitt the caps of main and outer bearings the full length on horizontal engines, but also a detriment to efficient lubrication. There is no tendency for the shaft to lift, therefore, a narrow strip at each end will be sufficient to keep it in place. The wide space between the two strips and between cap and shaft makes it possible for the oil to spread over the whole surface and lessens the danger of dry places being formed. If the edges of the quarter-boxes and lower shells are well beveled, there is no need of oil grooves. Some engine builders state that the lower shell of the main bearing on their engines can be removed " by lifting the shaft a fractional part of an inch," which is a vague expression. If it means that the shaft has to be lifted $\frac{15}{16}$ in., for instance, that would be unsatisfactory, since it would, in many cases, involve quite an expenditure of time and money; $\frac{1}{8}$ in., or, on large engines, $\frac{1}{4}$ in. should be the highest lift required for this purpose. An engineer will do well to investigate this matter before an engine is accepted.

Engine Adjustment

Any part that has been adjusted must be watched closely for a while, and the engineer must be prepared for an emergency. One of the most disagreeable things that can happen is a hot pin or bearing. On a new engine this may come quickly and not much time can be lost in applying remedies. If the crankpin gets hot, it should be flooded with oil through the center-oiler; should the heating continue, the engine must be slowed down and the wedge backed off. The same thing should be done if the crosshead pin heats unduly. On main and outer bearings the capbolts should be loosened, the quarter-boxes backed off, and plenty of oil poured on the shaft.

Light oil has no lubricating effect on hot surfaces. It is, therefore, of little use on a hot bearing or pin except to carry off some of the heat if used in considerable quantities. To be beneficial it must be mixed with cylinder oil, but cylinder oil alone is to be preferred. Filtered water mixed with oil may be used for cooling, but never hydrant or well water that contains grit, which would embed itself in the surfaces and quickly ruin them.

Cooling Hot Boxes

It is dangerous to throw water over or " turn the hose on " the connecting-rod and boxes if the crosshead pin or crankpin gets hot; they will contract if suddenly cooled and close in on the pins pinching them that much tighter, which may result in a serious breakdown. The cooling process should be done from the inside to make the pins cool first. If the babbitt begins to run, the engine must be turned over slowly but not stopped suddenly, otherwise the babbitt will "freeze," making it difficult to remove the boxes or shells. The engine must be kept moving and the hot parts cooled until the babbitt is safe, when the engine may be shut down and examined. It depends, of course, on circumstances whether it may be started again without removing the parts affected, or whether they must be taken down and overhauled or even rebabbitted and refitted.

Compound engines are cared for much the same as simple engines. In a cross-compound, two engines are connected to one shaft, while in a tandem-compound both cylinders have a piston rod, crosshead, connecting-rod frame and crank in common. The object of compounding is to attain better economy by the use of higher steam pressures, a greater ratio of expansion and reduced cylinder condensation. The steam is expanded in two cylinders, which lessens the range of temperatures in each of them. Steam has the tendency to give up heat the moment it strikes a colder surface, which is the cause of much loss of steam in single-cylinder engines running with high steam pressure.

Assuming, for instance, an initial pressure of 150 lbs. gauge with a temperature of 366° F., if the engine exhausts against 1 lb. back pressure, the incoming steam will come in contact with surfaces which have been exposed to a temperature of 216° F., a difference of 150°, provided there is no compression. Now if the process is carried out in two cylinders, one high- and the other low-pressure, the temperature in the high-pressure cylinder will range between 366 and 285° with a difference of 81°, and in the low pressure between 276 and 216°, with a difference of 60°.

If a condenser is used, the single-cylinder engine will be at a greater disadvantage. Compression will reduce initial condensation, but it weakens the engine so that for the same load more steam has to be admitted. The question whether it is more advantageous to increase the compression and extend the cut-off, or to reduce the compression and lose steam through initial condensation, can, in many cases, be decided only by the coal pile. In a Corliss engine there will usually be quite enough compression from this point of view if sufficient is used to give the necessary cushioning effect.

Most Economical Results

An interesting problem for an engineer is to find out under what conditions an engine will give the most economical results. Proper records will enable him to notice when peak loads or light loads occur and during which periods the load is normal. If there is any regularity in the changes of the load it will be possible to make careful tests as to steam consumption and to regulate the boiler pressure to suit. A high boiler pressure is not always the most economical; in fact, on a light load it may be wasteful. Neither is it good practice, as some people think, to overload an engine. If an economy test is made on an engine it should concern only the engine itself and not the evaporation and leaks of the boilers or the condensation in the steam pipe, which can be taken into account only in a test of the whole power plant or of each individual part.

An engineer should be concerned not only in the smooth and perfect running of his engine, but also in the greatest possible saving of steam. If he has an indicator, he can determine the steam consumption by taking diagrams at certain intervals and figure the amount of steam accounted for by each card in the following manner: Divide the number 859,375 by the product of the volume of the steam at the terminal pressure and the mean effective pressure; the quotient will give the steam consumed per indicated horse-power per hour.

The actual performance of steam engines in everyday use has been found to be about as follows:

Automatic high-speed engine..	50–60	lbs. per I.H.P. hr.
Slide valve engine............	40–50	"
Corliss engine................	24–34	"

This is for simple non-condensing engines and boiler pressures from 80 to 100 lbs. per square inch.

For compound engines and boiler pressures from 120 to 150 pounds the steam consumption may be assumed to be about 50 per cent less.

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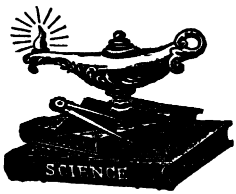

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