CHAPTER XXXVII.—THE STEAM ENGINE.

THE high pressure steam engine, in whatever form it exists, consists of a frame or bed plate carrying two distinct mechanisms, first, the driving or power-transmitting mechanism, and second, the valve gear or valve motion, and to these are added such other mechanisms as the nature of the duty the engine is to perform may require.

The most prominent of these additional mechanisms is a governor for regulating the speed at which the engine is to run; nearly all steam engines require a governor in some form or other, while for electric lighting and some other purposes it constitutes the main feature in the design of the engine.

In a locomotive the air brake and the sand box are elements not found in other engines.

In a jet condensing engine, the condenser and injection water, or condensing water mechanism, is a part of the engine.

In a surface condensing engine, the air pumps and circulating pumps are a part of the engine.

In marine engines there are mechanisms for turning the engine around when no steam is up; for moving the reversing gear quickly, and for varying the point of cut off, and therefore the amount of expansion, and various other and minor mechanisms.

Referring now to the simplest form of high pressure stationary steam engine, such as represented in Figs. 3293, 3294, and 3295, its valve gear or valve motion consists of the eccentric and its strap, the eccentric rod, the valve rod guide A, the valve rod or valve spindle, and the valve *v*, these parts controlling the admission of steam to one side of the piston, and the exhaust from the other.

The piston, piston rod, cross head, connecting rod, crank, crank shaft, main shaft or driving shaft, and the fly wheel constitute the driving or power-transmitting mechanism.

The steam side of the piston is that against which the steam is pressing, as side S in Fig. 3295. The exhaust side, E, of the piston is that on which the steam is passing out or exhausting.

The governor for a common D valve engine regulates the engine speed by varying the opening in the bore of the pipe through which the steam passes from the boiler to the steam chest, leaving a wider opening in proportion as the engine runs slower, and reducing the opening when the engine runs faster. Assuming the engine to be running at its slowest, or its load to be so great that a full supply of steam is required in order to keep the engine up to its proper speed, and the governor will be open at its widest, so that all the further action the governor can have is to reduce the steam pipe opening, and thus cause the pressure in the steam chest to be less than that in the steam pipe.

This action is called wire-drawing the steam, and the governor is called a throttling governor.

An engine bed or bed plate is a frame that is seated or bedded to its foundation along its whole length.

An engine frame is seated to its foundations at two or more places, but not continuously throughout its length.

THE CYLINDER.

Cylinders are secured to the engine frames in three principal ways, as follows: by bolting them down to the bed plate; by bolting them to one end of the bed plate, so that they may expand and contract without springing the bed plate; and in vertical engines, by bolting them to the top of the frames.

The bores of cylinders require to be parallel, so that the piston rings may fit to the bore without requiring to expand and contract in diameter at different parts of the stroke.

Cylinders are designated for size by the diameter of the cylinder bore and the length of the stroke; thus, a 10 × 12 cylinder has a piston of ten inches diameter and 12 inches stroke.

The wear of a cylinder bore is (if the engine is kept in proper line and the piston rings, or packing rings as they are sometimes termed, fit to the bore with an equal pressure throughout the stroke) greatest near the middle of the length and least at the ends of the stroke. But when the piston rings are set out by the steam pressure, and the point of cut off occurs early in the stroke, the wear may be greatest at the ends of the cylinder bore, because of the pressure of the steam diminishing during the expansion.

The counterbore of a cylinder is a short length at each end of the cylinder, that is made of larger diameter than the rest of the bore, so that the piston head may travel completely over the working bore, and thus prevent the formation of a shoulder at each end of the cylinder. Such a shoulder forms when there is a part of the bore over which the piston does not pass. The length of the counterbore should exceed the amount of the taper on the connecting rod key, so that as the connecting rod length alters from the wear, the piston shall not strike the cylinder head.

The clearance of a cylinder is the amount of space that exists between the face of the piston when it is at the end of its stroke and that of the valve when it covers the port, the piston being at the end of the stroke, and as this space exists at each end of the cylinder, the total clearance for a revolution is twice the above amount.

The clearance at the crank end of the cylinder is reduced by the piston rod passing through it.

The amount of clearance may be measured by the following method, which has been given by Professor John E. Sweet:

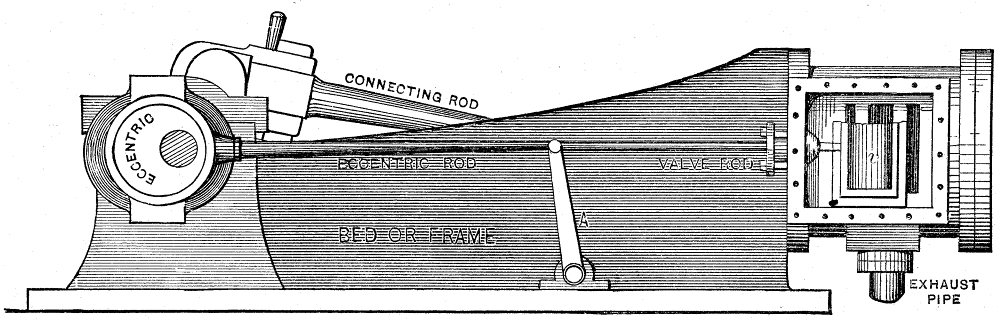


Fig. 3293.

See that the piston and valves are made tight, and the valves disconnected; arrange to fill the clearance spaces with water through the indicator holes, or holes drilled for the purpose. Turn the engine on the dead centre; make marks on the cross-head and guide that correspond; weigh a pail of water, and from it fill all the clearance space. Weigh the remaining water, so as to determine how much is used. Then weigh out exactly the same amount of water, turn the engine off the centre, pour in the second charge of water, and turn back until the water comes to the same point that it did in the first case. Make another mark on the cross head, and the distance between these marks is exactly what you really wish to know; that is, it is just what piston travel equals the clearance. This gives the proportion that the clearance space bears to the space in the cylinder occupied by the steam at the end of the piston stroke. Thus, if it takes one pound of water to fill this space, and to admit the one pound of water the piston must be moved one inch, then the clearance bears the same relation to the capacity of the engine as one inch bears to the stroke of the piston. Thus, under these circumstances, in an engine of ten-inch stroke, it would be said the engine had ten per cent. clearance.

When a cylinder is to be rebored, the boring bar should be set true or central to the circumference of the counterbore, so that the bore of the cylinder may be brought to its original position with reference to the bore of the stuffing box.

Cylinders require lubricating, both to avoid friction and wear of the cylinder bore, as well as of the valve and valve seat. The amount of lubrication required depends upon the degree of tightness of the piston rings, upon the speed of the piston, upon the amount of pressure of the valve to its seat, and upon the method of operating the side valve.

Cylinders with releasing valve gears require freely lubricating, because the closure of the valve depends upon the dash pot, and undue friction retards the closing motion.

The less the movement of the valve at the moment of its release, the easier it is to move it, because the friction is less, and less lubrication is required.

Cylinders are lubricated by automatic oilers placed on the steam pipe of the engine, the oil being distributed over the surfaces by the steam.

Cylinder oilers sometimes have a pump to force the oil in, and in others the steam in the oiler condenses, and the water thus formed floats the oil over the top of a tube, or up to an orifice through which the oil gradually feeds as the condensation proceeds.

In other oil feeders, the feed is regulated by increasing or diminishing the opening through which the steam passes from the cup to the steam pipe.

Sight oil feeders are those in which there is a glass tube or body, in which the passage of the oil can be seen as it drops.

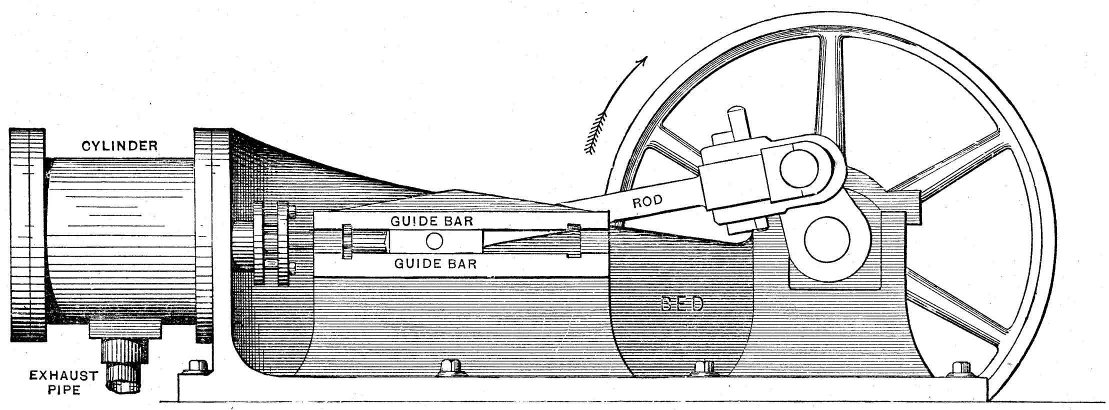


Fig. 3294.

Cylinder cocks are employed at each end of the cylinder to let out the water that condenses from the steam when admitted to a cold or partly cooled cylinder. The two cocks are usually connected together by a rod, so that both may operate together.

Cylinder relief valves are valves at each end of the cylinder to relieve the cylinder from the charges of water that sometimes enter from the boiler with the live steam.

Steam ports give a quicker admission in proportion as their length is increased, and this reduces the amount of valve travel, and are sometimes given a length equal to the diameter of the cylinder bore.

The bottoms of the steam ports are sometimes so placed as to be below the level of the cylinder bore, so as to drain off the water of condensation of the steam.

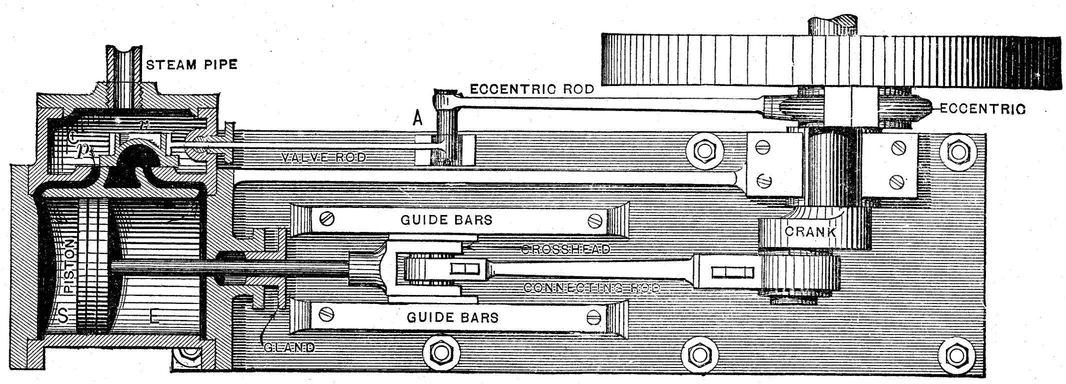


Fig. 3295.

Rule to find the required area of steam port.

Multiply the area in square inches of the piston, by the number opposite to the given piston speed in the following table:

Speed of piston in feet per minute.	Number by which to multiply the piston area.
100	0.02
200	0.04
300	0.06
400	0.07
500	0.09
600	0.1
700	0.12
800	0.14
900	0.15
1,000	0.17

The cylinder exhaust port must be open when the valve is at the end of its travel, to an amount equal to the width of the steam port, but what this width will be in any given case depends upon the width of the bridges, the amount of the steam lap and the travel of the valve, as will be explained with reference to the slide valve.

Jacketed cylinders are those in which there is a space around the cylinder that is filled with live steam.

The object of jacketing is to prevent the loss of heat from the steam within the cylinder by radiation. The steam in the jacket should be received direct from the boiler, and should not be drawn from the jacket into the steam chest because the jacket reduces its temperature and condenses it.

The water of condensation of a steam jacket should not be allowed to accumulate in any part of the jacket, but should drain off and pass back to the boiler. To render the jacket as effective as possible, it should extend from end to end of the cylinder, the exhaust steam pipe leading directly away, so as to have as little communication with both the cylinder and the jacket as possible.

The jacket should have open communication with the boiler at all times, so as to have the pressure in the jacket at the same pressure as that in the steam chest, while the cylinder being kept hot, it will be unnecessary to blow steam through in order to warm the cylinder when starting the engine. The steam should enter the jacket at the highest point, so as to prevent the accumulation of air in the jacket. Or, if the steam is admitted at some other point, it should be so arranged as to permit its thorough circulation in the jacket. When a jacket is used, the metal of the cylinder body should be as thin as possible, because the transmission of heat through the metal is, both in time and quantity, inversely as the distance or thickness passed through.

The steam in the jacket should be as dry as possible, so that all wet steam admitted during the live steam period may be evaporated by the heat received from the steam in the jacket. The outside of the jacket should be thoroughly protected from cooling by being lagged or clothed with felt or some other material that is a non-conductor of heat.

From experiments made by Mr. Charles A. Smith, of St. Louis, it was found that the amount of variation of temperature that occurred during the stroke in a locomotive cylinder was inversely proportional to the speed of engine revolution, which shows the advantages of jacketing cylinders and of lagging them, as well as the advantage of a high rotative speed.

A lagged cylinder is one clothed, which is sometimes done with wood or metal strips, leaving an air space around the cylinder, while in others this space is filled with felt or some non-conducting material.

Experiments made by Charles E. Emery gave the following general results: The thickness of the pipes and of the non-conducting materials was kept constant.

Hair felt was the best non-conducting material of all those tested, and the value of a thickness of two inches of hair felt was taken as unity and the maximum.

The value of two inches of mineral wool as a non-conductor was 0.832 of hair felt; two inches of mineral wool and tar was 0.715. Two inches of sawdust, 0.68; two inches of a cheaper grade of mineral wool, 0.676; charcoal, 0.632; two inches of pine wood, across the grain, 0.553; two inches of loam, 0.55. This was from the Jersey flats, and almost all vegetable fibre not yet become compact. Slaked lime from the gas works, expressed decimally, with hair felt as unity, 0.48; coal ashes, 0.345; coke, only 0.277, the same as used for fuel; two inches of air space, only 0.136, which dashes a great many people's hopes, and is as interesting as any part of the data; two inches of asbestos, 0.363; two inches of Western coke, about the same as the other coke; two inches of gas house charcoal, 0.47.

These are very interesting, particularly so this matter of an air space. It has been supposed that an air space around a pipe is as good as anything we can have. The fact is, convection or circulation takes place; the air is cooled on one side of the space, descends, and rises on the other, and it is necessary to break up the air space, and that undoubtedly accounts for the efficiency of these different materials. It is the air probably that is the non-conductor; but it should be kept quiescent instead of being allowed to circulate. The air space itself is of very little value until the circulation is prevented.

THE PISTON.

In calculating the power of an engine it is the piston speed that is taken into account, and not the length of the stroke, the latter being used merely in order to obtain the piston speed.

Long strokes are usually employed upon engines running at moderate piston speeds, as from 300 to 500 feet per minute, and short strokes for piston speeds from 400 to 800 feet per minute.

The Porter Allen engine has been run noiselessly at 1,100 feet per minute.

In determining the stroke of an engine the nature of the valve-operating mechanism is taken into account.

In releasing mechanisms, or those in which connection between the eccentric rod and valve spindle is broken in order to permit the valve to close quickly, too high a speed of revolution may cause the tripping mechanism to fail to act, hence a high piston speed is obtained by means of employing a comparatively long stroke.

In positive valve gears, or those in which the valve is controlled throughout the whole of its movement by the eccentric, the valve mechanism may operate quicker without danger of missing, hence the piston speed may be greater.

When the stroke equals the diameter of the cylinder bore, the cylinder presents the least amount of exposed surface in proportion to its cubical contents.

To obtain the same amount of expansion in a short as in a long stroke engine, the steam must be expanded through an equal proportion of the stroke; thus, if the steam is cut off at half stroke in both cases, the amount of this expansion will be equal.

Pistons are made an easy fit to the cylinder bore, a steam-tight fit between the two being obtained by means of the piston rings.

Solid pistons are provided with snap piston rings.

A snap piston ring is one that is larger in diameter than the cylinder bore, and is closed in to get it into the cylinder, while it depends on its own spring outwards for its fit to the cylinder bore, having no supplementary rings or springs to force it out.

Piston rings that are expanded by supplementary springs should be tapering in thickness, the thickest part being opposite to the split, and the thinnest at the split. This causes the ring to conform itself to the cylinder bore, and makes it sit more evenly around its whole circumference. These rings are made larger in diameter than the cylinder bore, in proportion of about $\frac{1}{8}$ inch per foot of diameter, the split being closed when the ring is sprung into place in the cylinder. But if made of brass, the split must be left open enough to allow for the expansion, or otherwise the ring expanding more than the cylinder will seize and cut single.

The split of a piston ring should be placed on the bottom of the piston (in a horizontal engine), so that the piston head, in resting on the cylinder bore, will cover up the opening of the ring.

When two or more rings are employed, the splits may be placed on the lower half of the cylinder, so as to cover up their splits as much as possible.

The follower of a piston is a plate or cover that is employed to hold the piston rings in place, and the piston rings should be so fitted that the follower should be bolted firmly up, or otherwise the bolts may come loose and work out, and getting between the piston and the cylinder cover, may cause the piston to knock the cylinder cover out.

Piston followers are necessary when the rings are set out by springs or other parts adjustable within the piston head. Snap piston rings, however, permit the use of a solid piston, dispensing with the need for a follower.

The effectiveness of a piston ring may be tested, when the construction of the engine will permit it, by disconnecting the valve for the head end, setting it so that it covers the port, and then taking off the cylinder cover at the head end and admitting steam through the crank end steam port, when any leak in the piston rings will be seen by the escape of the steam.

THE PISTON ROD.

Piston rods should be of slightly diminishing diameter at the ends, so that the wear shall not leave a shoulder at each end of the rod.

In determining the diameter of the piston rod, allowance is made for turning it occasionally in the lathe to restore its parallelism, the wear reducing its diameter more in the middle than at the ends. The diameter of a piston rod is found in practice to range between one-sixth and one-tenth the diameter of the cylinder bore.

Steel piston rods wear better than those of wrought iron, being free from scaly seams which are apt to cut the packing and cause the rod to wear in grooves.

The best method of securing a piston rod to a piston head and to the cross head is by a taper seat and a key, so that no nut is needed, and the cylinder cover need not have a recess to receive the nut when the piston is at the end of the stroke, and the amount of clearance is correspondingly reduced.

Piston head keyways are sometimes given so little clearance that the key completely fills the keyway when driven fully home. This prevents the edges of the keys from bulging into the clearance space in the keyway, which action is apt to cause the key to loosen in time. The key should have a safety pin at its small end.

When piston rods are threaded into the cross head, or into the piston, the threads are made an easy fit, and taper seats or split hubs secured by clamping screws are relied upon to keep the rod true to the cross head or piston, it being found that the screw alone cannot be relied upon for this purpose.

PISTON ROD PACKING.

Piston rod packing, of fibrous or similar material, should be cut in rings that will not quite fully envelop the piston rod, and the first ring should be placed with its split upwards. After two or three rings have been inserted, each having its split at a different part of the bore, so as to "break joints," the gland should be screwed up enough so as to carry the packing home to the back of the stuffing box. This process should be continued until the stuffing box is filled for about two-thirds of its depth, when the gland may be screwed home.

The gland should be screwed up quite evenly, so that the packing in the stuffing box shall be compressed equally all around the rod, and will not cause the gland to bind on the rod or in the stuffing box bore.

The wrench should be applied first to one nut, giving it a turn or two, and then to the other, and after the gland is firmly home the nuts should be eased back about two turns.

When a gland requires packing, it is proper to take out all the old packing that has become hard and set.

A leak in piston rod packing may sometimes be remedied by taking out three or four rings of the packing and reversing it.

If the packing is tightened up while the engine is running, it should be done very gently and evenly, as a very little screwing up may stop the leak, while excessive screwing produces undue friction.

Piston rods are in some of the most advanced practice packed with metallic packing, or packing composed of soft metal. In some forms of metallic packing the construction is such that the gland and packing do not attempt to restrain the line of motion of the piston rod, this duty being left to the guide blocks and guide bars, where it properly belongs.

THE CROSS HEAD.

In engines having Corliss frames, the cross head is provided with shoes and adjusting screws, to take up the wear.

When guide bars are shaped thus ⌐ the cross head is provided with gibs (usually of brass composition) to take up the wear.

In either case care must be taken to make the adjustment correct, and thus keep the piston rod in line. The shoes or gibs should not bear hard upon the guides, but be an easy sliding fit without lost motion.

Cross head pins should be kept eased away on the two parts of their circumference which are within the connecting rod brasses or boxes and near the joint faces of the same. This is necessary because the wear is greatest on the crowns of the boxes, and the pins are apt to wear oval. In some engines, the surface of the pin is cut away, but if it is not, and the pin can be revolved in the cross head, it is a good plan to give it half a turn occasionally, which will keep it round.

THE GUIDE BARS.

The guide bars of an engine require to be set exactly in line with the axis of the cylinder bore, so that they may guide the piston to travel in a straight line. They should be an easy sliding fit to the cross-head guide.

The top bar is more difficult to lubricate than the bottom one, especially when it receives the most pressure, as is the case when the top of the fly-wheel runs towards the cylinder.

Cast iron guide bars wear better than either brass, iron, or steel ones, so long as they are properly lubricated. The face of each guide bar should be cut away, so that the ends of the cross head guides will travel past it. This will prevent a shoulder forming at the ends of the bar as the face wears away. Such shoulders are apt to cause a knock as the connecting rods are lined up, because in the lining the connecting rod is restored to its original length, and the path of the cross-head guides along the bars may be altered.

THE CONNECTING ROD.

There are two principal kinds of connecting rods, the "strap ended" and the "solid ended." The solid ended wear the best, but are more difficult to get on and off the engine.

Connecting rod straps are secured to the stub ends (as the ends of the rod are called), either by bolts or by one or two gibs, and the brasses are set up by a taper key or wedge.

The taper for connecting rod keys is about an inch per foot.

The angularity of a connecting rod is a term that applies to its path of motion, which is (during all parts of the stroke except on the dead centre) at an angle to the line of engine centres. The effect of this angularity is to cause the piston motion to be accelerated at one part of the stroke and retarded at another, thus causing the point of cut-off to occur at different points of the two strokes.

The direction of the variation is to cause the point of cut-off to occur later on the stroke when the piston is moving from the head end of the cylinder towards the crank.

The amount of variation caused in the two points of cut off by the connecting rod depends upon the proportion that exists between the length of the crank and that of the connecting rod, and is less in proportion as the length of the connecting rod is greater than that of the crank.

An ordinary length of connecting rod is six times the length of the crank, or *six cranks*, as it is commonly termed.

Fig. 3296 represents a cylinder, piston and rod, cross head, connecting rod, and crank.

The piston *b* is shown in the middle of the cylinder, the cross head at E, and the crank pin at B, instead of being at G′, as it would but for the connecting rod, or if the connecting rod was infinitely long.

Now take a pair of compasses and set it from *b* to E, and then try it from *a* to D, and from *c* to F, and it will be seen that the three cross head positions D, E, and F correspond correctly to the three piston positions *a*, *b*, *c*. Then take a pair of compasses and set them to the length of the connecting rod (from E to B) and try them from D to A, from B to E, and from C to F, and it will be seen that crank pin positions A, B, and C correspond to cross head positions D, E, and F, and therefore that the crank is not at half stroke when the piston is in the middle of the cylinder. Take these same compasses, and resting one point at (G′) mark the arc H, and that is where the cross head would be when the crank was at (G′). Now then we see that the connecting rod causes the piston to move slower while running from *a* to *b* than it does while running from *b* to *c*.

THE D SLIDE VALVE.

The various events which are governed by the D slide valve of a steam engine are as follows:

The live steam period is that during which the steam is admitted from the steam chest into the cylinder and the steam admitted during this period is termed *live* steam.

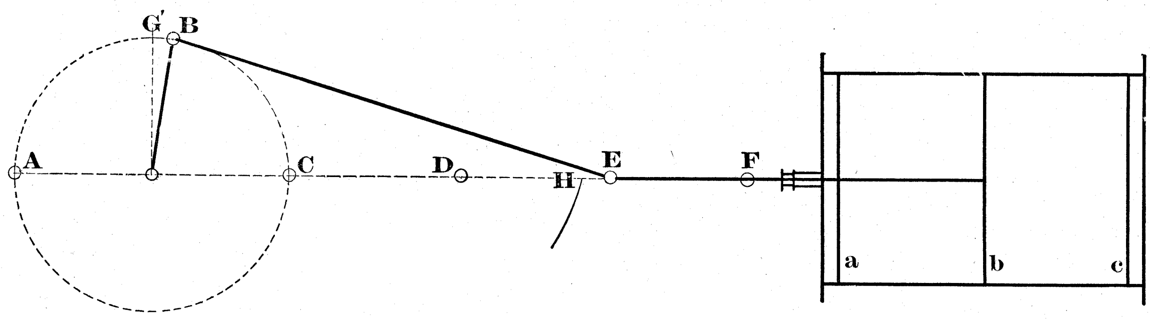


Fig. 3296.

The point of cut off is that at which the valve closes the steam port, and the admission of steam into the cylinder is stopped, hence the point of cut off is at the end of the live steam period.

The period of expansion is that during which the steam is allowed to expand in the cylinder, and therefore begins at the point of cut off, and ends at the point of release.

The point of release is that at which the valve opens the port and permits the steam to escape.

The point of compression is that at which the exhaust port is closed, which occurs before the piston has reached the end of its stroke; the steam that has not passed out of the cylinder is therefore compressed, the compression continuing until the valve opens for the lead.

The lead of the valve is the amount the port is open to the live steam when the crank is on the dead centre.

The point of admission is that at which the port opens for the live steam to enter, and it follows that the lead and compression both act as a cushion, arresting the motion of the piston when it reaches the end of the stroke.

Cushioning begins, however, at the time the exhaust port is closed enough to arrest the escape of the steam, while compression begins when the valve has closed the exhaust port.

The construction of a common slide valve is shown in Fig. 3297,

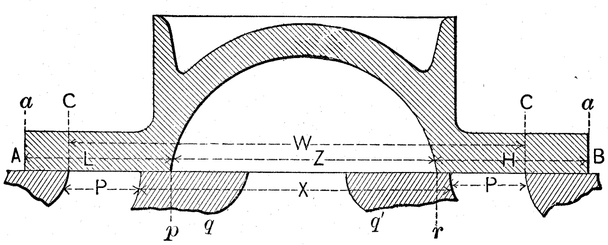


Fig. 3297.

in which the valve is shown in its mid-position. P P are the cylinder steam ports (as the openings through which the steam passes from the steam chest to the cylinder are termed), and at X is the cylinder exhaust port, through which the steam escapes from the cylinder. Z is the valve exhaust port or exhaust cavity.

The lip of a valve is the width of its flange face, or the distance L, which is measured from the steam edge A to the exhaust cavity Z. At the other end of the valve, H is the lip extending from the steam edge B to the exhaust cavity.

Steam lap is the distance the steam ends (or the steam edges as they are called) A, B overlap the steam ports, this distance being shown on the ends of the valve at *a* C. If the valve had no steam lap, its steam edges would just cover the ports, as denoted by the dimension W.

Exhaust lap is the amount the exhaust cavity Z overlaps the bridges *q q′*, as at *p*, *r*.

Unequal steam lap is given to cause the point of cut off to occur at equal points in the piston stroke; thus in the figure there is more steam lap at the head end than at the crank end of the valve. But unequal lap could also be given in order to greatly vary the points of cut off for the two piston strokes, if such was desired.

Unequal exhaust lap may be given to equalize the point of release, or to equalize the points of compression.

The head end of the valve (or of the cylinder) is that which is furthest from the crank shaft, the other end, or that nearest to the crank shaft, being termed the crank end.

THE ACTION OF A COMMON SLIDE VALVE.

The action of a common slide valve may be traced as follows:

Port *a*, open to the amount of the lead.

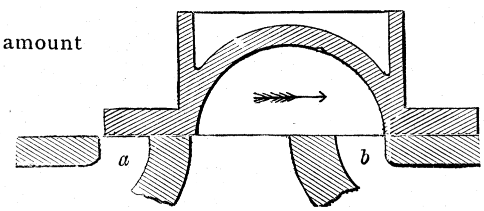


Fig. 3298.

Port *a*, full open for the admission.

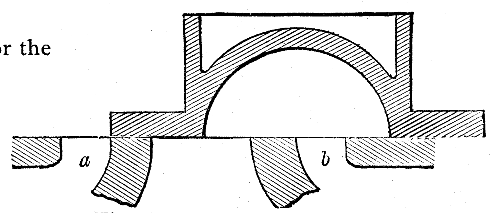


Fig. 3299.

Port *a*, closed for cut off.

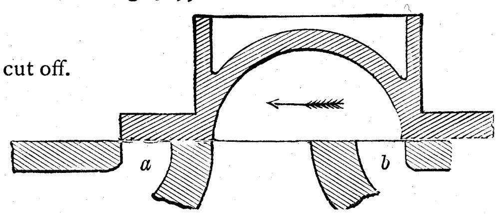


Fig. 3300.

Valve opening port *a*, for the exhaust.

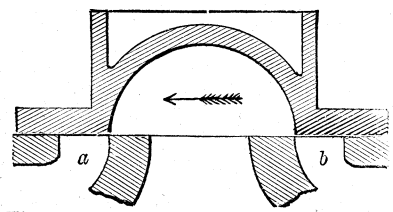


Fig. 3301.

Port a, full open for the exhaust.

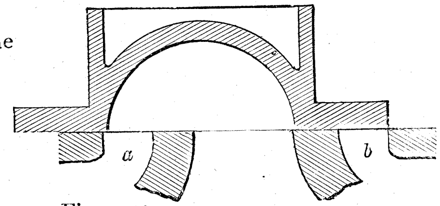


Fig. 3302.

Suppose the port a to be at the head end of the cylinder and open to the amount of the lead with the crank on the corresponding dead centre, and if the valve travel be made equal to twice the lap and the lead, the various positions of the valve will be as marked in Figs. from 3298 to 3302; the event corresponding to each valve position being stated in the figures.

DOUBLE PORTED VALVES.

The term *port* applies strictly to the area of opening of the steam passage where it emerges upon the valve seat. The term *steam passage* includes the full length of the opening from the cylinder bore to the face upon which the valve is seated.

A double ported steam port is one in which there are *two* openings or steam ports, leading into *one steam passage*.

A double ported valve is one in which there are two ports *at each end* of the valve. These two ports in some cases admit steam to a single cylinder port, and in others to two steam ports, terminating in one steam passage.

A griddle valve is one that has two or more ports at each end upon a seat that has two or more ports for each steam passage.

Double ported valves are employed in some cases to increase the admission of live steam to the cylinder, and in others to increase the exhaust openings also. The effectiveness of a double ported valve is mainly valuable at the beginning of the stroke, and is especially valuable in cases when the travel of the valve is diminished to hasten the point of cut off, because in such cases the outer edges of the valve do not open the steam port to its full width, and a single port is apt to wire draw the steam. By the employment of more than one port, or several ports, a sufficient admission may be obtained with less valve travel.

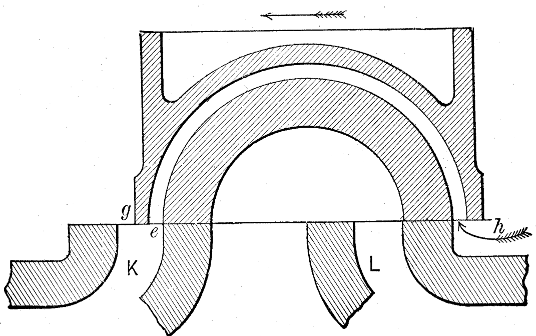


Fig. 3303.

The Allen double ported valve is one in which the second port increases the port opening for the admission only, as shown in Fig. 3303, in which the valve is moving in the direction of the arrow; the port K will receive steam through the opening at g, and from a port passing through the valve, the steam entering it as shown by the arrow. The second port forms part of the lap of the valve, and enables the travel to be short enough to be cut off at early points in the stroke, without employing so much steam lap as to widely distort the points of cut off, this latter being a defect of the D valve.

Webb's patent slide valve is circular, and is so arranged as to be free to revolve in the hoop of the valve rod, the effect being that the valve moves around, or to and fro in the hoop, without any special mechanism to produce such movement, and the result is, that the valve and port facings wear smooth and even without any tendency to become grooved.

BALANCED VALVES.

A balanced valve is one in which means are employed to relieve the back of the valve of the steam pressure, and thus prevent its being forced to its seat with unnecessary pressure.

In some of the most successful balanced valves this is accomplished by providing a cover plate, which may be set up to exclude the steam from the back of the valve which works (a sliding fit)

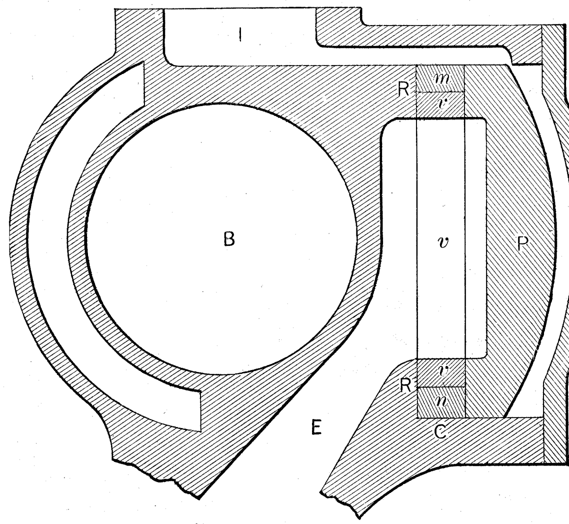


Fig. 3304.

between the valve face and the face of the cover plate. Such a method of balancing is sufficiently effective for all practical purposes, if the following conditions are observed: The valve rod must be accurately guided so as to avoid side strains; the valve must fit accurately to its seat and to the cover plate, and the adjustment so made that the valve slides freely at first, being steam tight, and yet allowing room for lubrication to enter. When the travel of a valve, balanced by a cover plate, is varied to alter the point of cut off, the construction must be such that the ends of the valve at the shortest stroke pass over the ends of the seat and cover plate faces, or otherwise the middle of the seat and cover plate faces will wear hollow.

The Buckeye, Porter-Allen, and Straight-Line Engines are examples of practically balanced valves. The first of these has a balancing device that follows up the wear; the second has an adjustment whereby the cover plate may be set up to take up the wear; and in the third the wear is reduced to a minimum, by accurately fitting and guiding the parts.

The construction of the valve in the Straight-Line Engine is shown in Fig. 3304, in which B represents the cylinder bore; the valve v rests on a parallel strip n, and on its top rests the parallel strip m, the pressure relieving plate P is set up firmly against the pieces m n, whose thicknesses are such as to leave the valve a working fit between the faces of R R and of P.

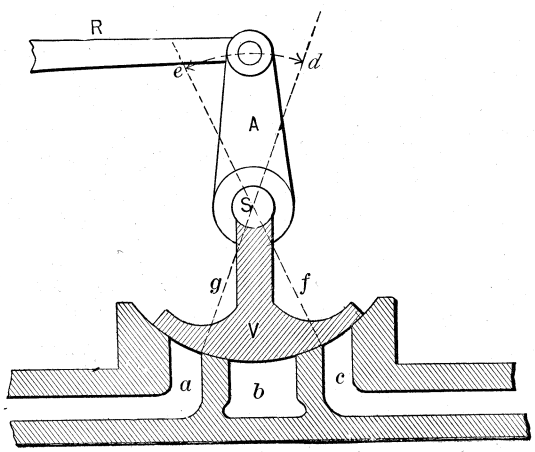


Fig. 3304 *a*.

Instead of the valve sliding on a flat face, it may work upon a shaft or spindle as a centre, its face moving in an arc of a circle, and its action will be the same as a flat valve having the same proportions. Fig. 3304*a* represents a valve V of this construction, whose shaft is at S, A being an arm fast on S, and driven by the eccentric rod R. To find the necessary amount of travel for such a valve, we draw lines, as *f*, *g*, from the inner edges of the steam ports, through the centre of the shaft S, and also draw an arc through the centre of the eye of arm A, and where lines *f g* cut the arc, as at *d* and *e*, are the extremes of motion of A.

PISTON VALVES.

A piston valve acts the same as a flat or plain (D) valve, having the same amount of lap lead and travel. In Fig. 3305 we have a cylinder with a flat valve on one side and a piston valve on the other, the head end ports being about to take steam, and it is seen that the eccentrics occupy the same positions for the two valves.

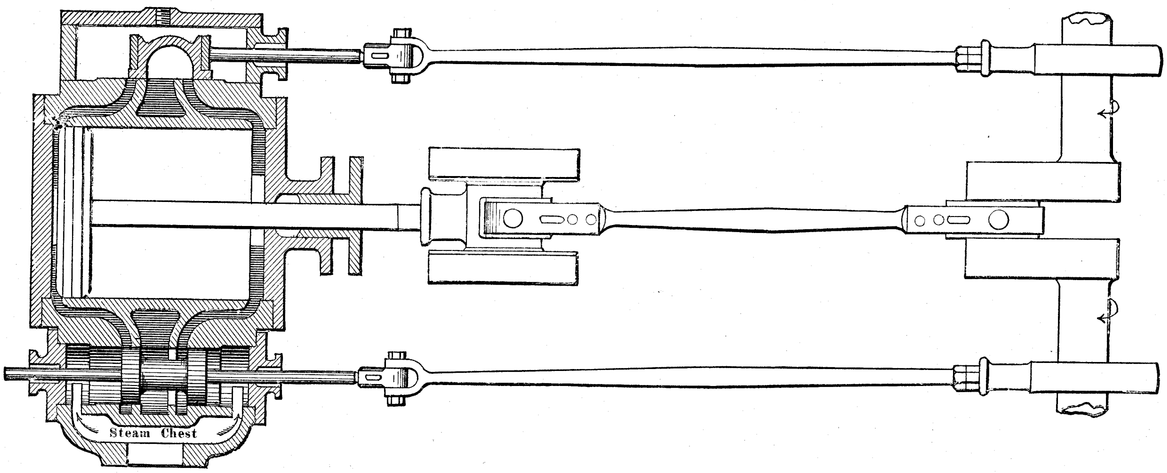


Fig. 3305.

The steam ports are, for the piston valve, annular grooves provided in the bore in which the valve fits. The piston valve is balanced because it receives its steam pressure on the ends, but it will not follow up its wear as the flat valve does, hence it is liable to leak.

SEPARATE CUT OFF VALVES.

Meyer's cut off valve is constructed as shown in Fig. 3306, M being the main valve, and *v v* the two cut off valves, whose sole duty is to cut off the steam at an earlier point than the main valve would do.

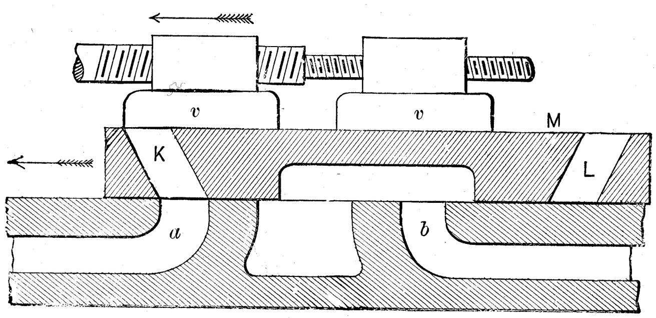


Fig. 3306.

If the engine is to have a fixed point of cut off, or, in other words, if the cut off is always to occur at some one particular point in the stroke, the valves may be set to do so, and equalize the points of cut off.

Variable points of cut off with the Meyer's valve may be obtained by shifting the position of the eccentric that operates the cut off valve, but it is usually done by means of moving the valve by a right and left hand screw, such as shown in Fig. 3306. The cut off eccentric is set ahead of the main eccentric, so that the cut off valve will close the ports before the main valve would do so; thus, in the figure the cut off valve is shown to have effected the cut off for port *a* by the time the main valve has fully opened port *a*, and is reversing its motion. If the engine requires to reverse its motion, the cut off eccentric is set exactly opposite to the crank, but otherwise, it may be set 8 or 10 degrees either ahead of or behind the crank, but if set too little ahead of the crank, the port may reopen after the cut off has been effected.

Gonzenback's cut off valve is constructed as in Fig. 3307, the steam chest having two compartments. A, A are the cylinder steam ports, C the main valve, and E the cut off valve, whose ports (as G) are made wider than the ports F.

Reducing the travel delays the point of cut off in the Gonzenback valve, whereas in the common slide valve it gives an earlier cut off.

THE ECCENTRIC.

When a single eccentric is used, it is simply termed *the* eccentric. If a cut off valve (or two cut off valves) are used upon the engine, then the eccentric that works the main valve is called the

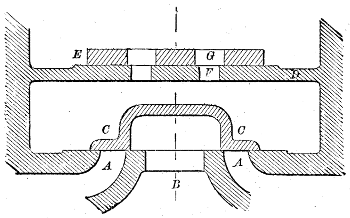


Fig. 3307.

main eccentric, while that which works the cut off valve or valves is called the cut off eccentric. The main valve is that which works on the cylinder face; the cut off valve is that which effects the cut off.

A shifting eccentric is one that is *moved across* the shaft so as to alter its amount of throw, and, therefore, the amount of valve travel, the effect being to vary the point of cut off.

In engines where a constant amount of lead is given, or in other words, when the eccentric position is intended to be fixed, the eccentric should be secured to the crank shaft by a feather or key sunk into the crank shaft so as to prevent the eccentric from moving, while enabling it to be taken off and replaced without requiring any operations to adjust its position with relation to the crank.

The feather should fit tight on the sides, as well as on the top and bottom, and may have a slight taper on the sides, which will make it easier to fit the featherway or keyway to the feather, and easier to put the eccentric on or take it off.

By this means the eccentric cannot shift, and may be replaced after being taken off without having to set the whole valve motion over again.

When the amount of valve lead or of compression is varied to suit the speed at which the engine is to run, or to aid the counterbalancing of the engine, a feather cannot be used because it will not permit the eccentric to be moved to effect the adjustment.

Set screws possess disadvantages, inasmuch as that the point of the set screw may leave an indentation, which, if the eccentric is moved a trifle, may cause the set screw point to fall back into the old indentation, thus rendering it difficult to make a small adjustment of eccentric position.

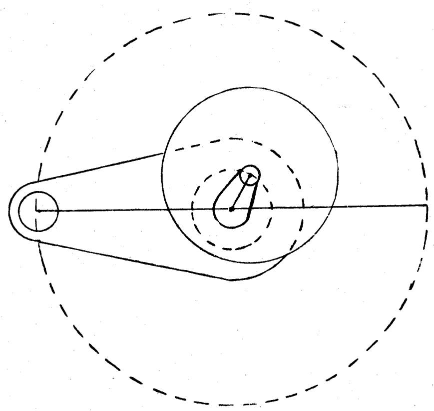


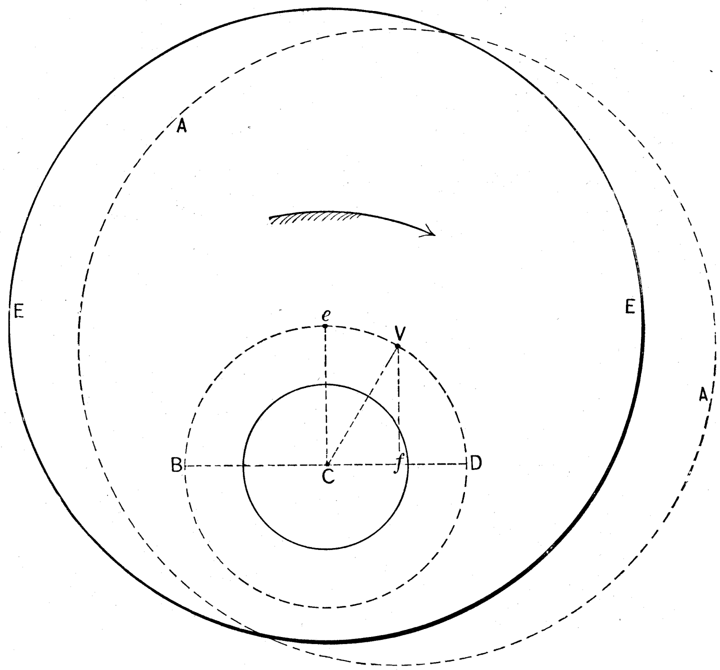
Fig. 3308.

An eccentric is the exact equivalent of a crank having the same amount of throw, as may be seen from Fig. 3308, in which the outer dotted circle represents the path of the crank and the inner one the path of the centre of the eccentric. A small crank is marked in, having the same throw as the eccentric has, and the motion given by this small crank is precisely the same as that given by the eccentric whose outer circumference is denoted by the full circle.

Considering the motion of both the crank and the eccentric, therefore, we may treat them precisely the same as two levers, placed a certain distance apart, revolving upon the same centre (the centre of the crank shaft), and represented by their throw-lines.

In Fig. 3309, let the full circle E E represent an eccentric upon a shaft whose centre is at C, and let the centre of the eccentric be at *e*. The path of revolution of the eccentric centre will be that of the dotted circle whose diameter is B, D. As the eccentric is in mid-position (*e* being equidistant from B and D), the valve will be in mid-position as denoted by the full lines at the bottom of the figure. Now suppose the eccentric to be revolved on the centre C, until its centre moves from *e* to V, its circumference being denoted by the dotted circle A A, and if we draw from V a vertical line cutting the line B, D at *f*, then from C to *f* will be the distance the eccentric would move the valve, which would then be in the position denoted by the dotted lines at the bottom of the figure. It becomes clear then that if we suppose the eccentric to have moved from mid-position to any other position, we may find how much it will have moved the valve by first drawing a circle representing the path of the centre of the eccentric, next drawing a line (as B D) through its centre, and then drawing a vertical line as (C *e*) through its mid-position and also a vertical line from the eccentric centre in its new position, the distance between these two vertical lines (as distance C *f* in the figure) being the amount the eccentric will have moved the valve.

It may have been noticed that the diameter of the eccentric does not affect the case, the distance B D, or the diameter of the circle described by the centre of the eccentric, being that which determines the amount of valve motion in all cases. This being the



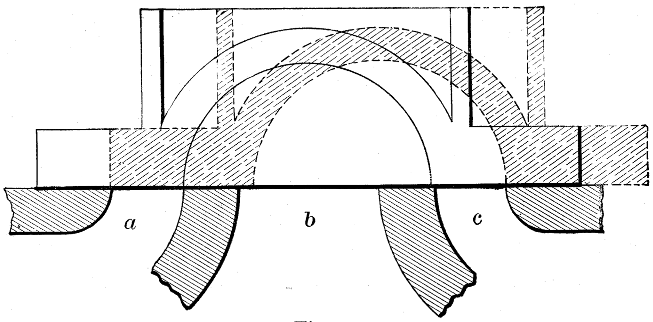


Fig. 3309.

case, we may use the circle representing the path of the eccentric centre for tracing out the valve movement without drawing the full eccentric, and the diameter of that circle will of course equal the full travel of the valve.

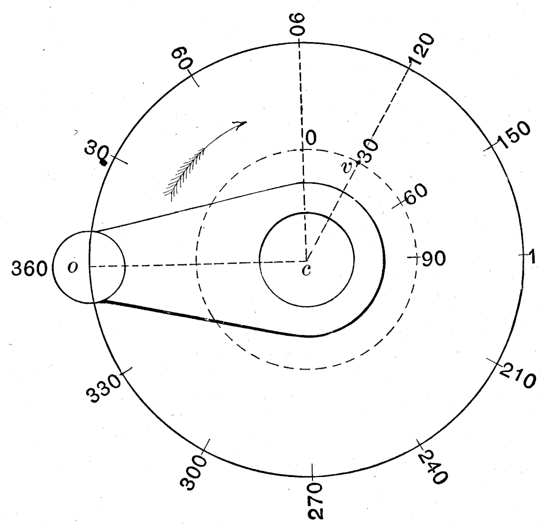


Fig. 3310.

The position of an eccentric upon a shaft is often given in degrees of angle, which is very convenient in some cases. If a valve has no lap or lead, the eccentric will stand at a right angle or angle of 90 degrees when the crank is on the dead centre.

The division of a circle into degrees may be explained as follows:

Suppose we take a circle of any diameter whatever and divide its circumference into 360 equal divisions, then each of these divisions will be one degree. The number 360 has been taken as the standard, and this being the case, there are 360 degrees in a circle, in a quarter of a circle there will therefore be 90 degrees, because 90 is one quarter of 360. By means of dividing a circle in degrees therefore we have a means of measuring or defining any required portion of it.

In Fig. 3310 the degrees of a circle are applied for defining the relative positions of a crank and an eccentric. As the zero position of the crank is on a dead centre, it is so placed in the figure, while as the zero position of the eccentric (which is for a valve having no steam lap) is at 90 degrees from the crank, therefore the dotted circle representing the path of the eccentric centre has its O or zero point at 90 degrees from the crank. Now suppose the eccentric centre stood at v and the eccentric throw line at $c\ v$, and it will stand at 30 degrees from O, hence the angular advance of the eccentric is in this case 30 degrees, or in other words, it is 30 degrees in advance of its zero position, or the position it would occupy when the crank is on the dead centre and the valve has no lap and no lead.

If we measure the distance apart of the crank and the eccentric in degrees, we find it is 120 degrees, hence place the crank where we may, we can find the corresponding eccentric position because it is 120 degrees ahead of the crank. The sign for degrees is a small ° placed at the right hand of the figures and slightly above them; thus, thirty degrees would be written 30°.

FINDING THE WORKING RESULTS GIVEN BY A D SLIDE VALVE.

Although not strictly within the line of duty of an engineer or engine driver, he is nevertheless sometimes called upon to find out how a valve of given proportions will dispose of the steam, or what proportions to give to a valve to accomplish certain results.

This is easy enough when either the travel of the valve or the amount of the lap and the width of the port are given, but if the width of the port alone is given, and all the other elements are to be found, it becomes a more difficult problem.

An engineer, however, is rarely called upon to solve the question from this stand-point, which properly belongs to the draughtsman or engine designer.

If the amount of valve travel is given, however, all the other elements may readily be found by the following construction:

Suppose that in Fig. 3311 a D valve is to be designed to cut off the steam when the piston has travelled from position B′ to R,′ or at three-quarters of its stroke. Then to find the position the crank pin will be in when the cut off occurs, we draw a circle, B D, representing the path of the crank on the same scale that the length of the piston stroke is represented. The straight line from B to D will, therefore, represent the piston stroke without drawing the piston or cylinder at all (this being done in the figure to make the explanation clear). When the crank is on its dead centre, B, the piston, will be at B′, and the valve in the position shown (supposing it to have no lead). As soon as the crank and valves begin to move, the steam will enter steam port a, and to find where the crank will be when the piston is at three-quarters stroke, and is, therefore, in position R′, we mark a point at R three-quarters of the distance from B to D. Then, taking no account of the length of the connecting rod, we draw a vertical line Y from R to the circle, and this line gives at H the position the crank will be in when the piston is at R. We have so far, therefore, that while the piston travels from B′ to R′, the crank will travel from B to H. Now, it will be found that if we set a pair of compasses from B to F, which is half-way from B to H, and then rest the compasses at D, and mark an arc V, then a line from V to the centre of the crank will give us the proper position of the eccentric. As the centre of the crank pin and also the centre of the eccentric both travel in a circle, we may, therefore, take a circle having a diameter equal to twice the throw of the eccentric, (or, what is the same thing, equal to the full travel of the valve), and let it represent the paths of both the eccentric centre and the crank pin centre, the latter being drawn to a scale that is found by dividing the length of the piston stroke by the travel of the valve; thus, if the travel is 3 inches and the stroke 30 inches, the diameter of a 3 inch circle will represent the valve travel full size, and the piston stroke one-tenth full size, because $30 \div 3 = 10$. It has been shown on page 376 that the length of the connecting rod affects the motion of the piston by distorting it, and it is necessary to take this into account in constructing the actual diagram, which may be done as follows:

The valve travel and point of cut off being given, to find the required amount of lap, there being no lead, draw a circle equal in diameter to the travel of the valve, and draw the line of centres B D, Fig. 3312; mark on the line of centres a point R, representing the position the piston is to be in at the time the cut off is to take place.

Set a pair of compasses to represent the length of the connecting rod on the same scale as the circle B D represents the path of the crank; thus, if the connecting rod is three times the length

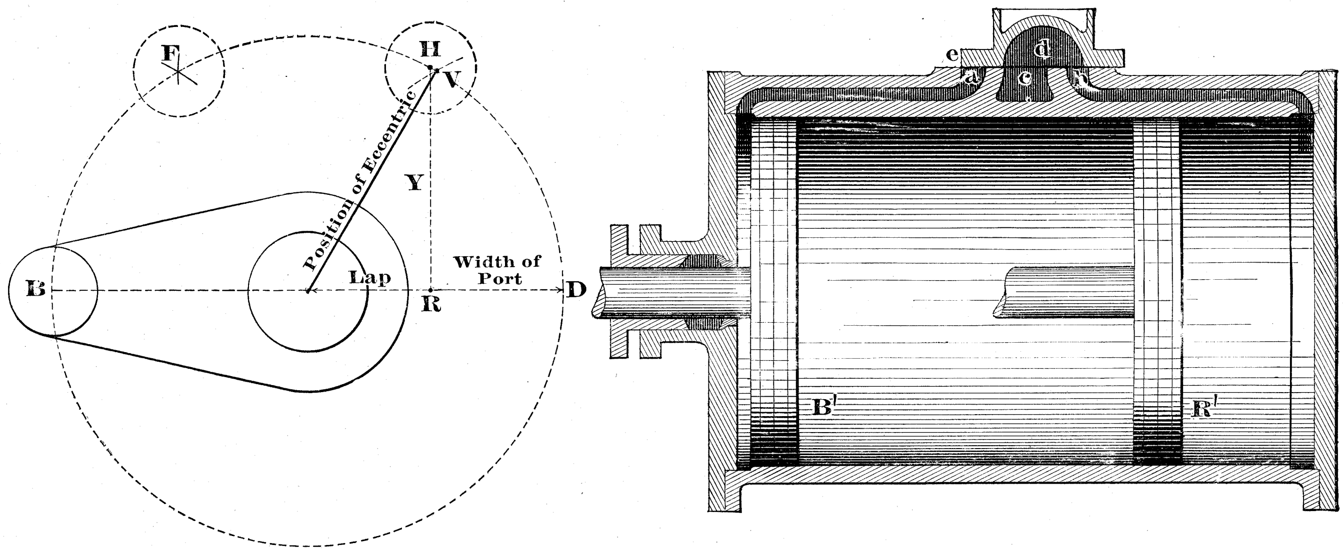


Fig. 3311.

of the stroke, the compasses would be set to three times the diameter of the circle B D.

A straight line from B to D and passing through the centre C of the crank will represent the line of centres of the engine, which

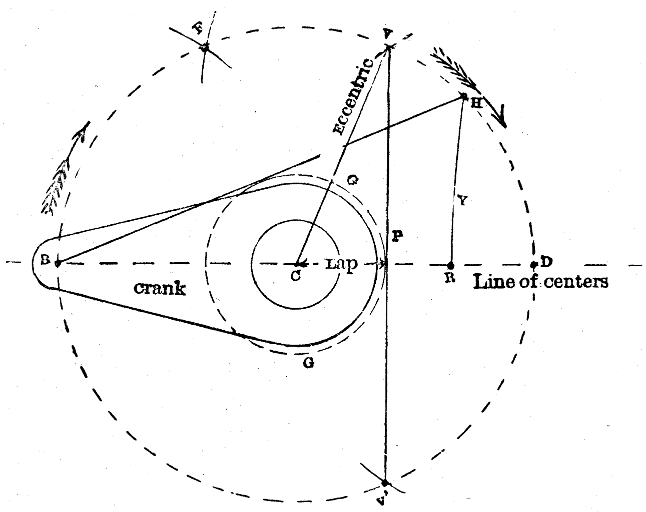


Fig. 3312.

must be prolonged to the right sufficiently to rest the compasses on it and draw the arc Y, which will give at H the position of the crank when the piston is at R, and the cut off is to occur.

We have thus found that the amount of circular path the crank will move through from the dead centre to the point of cut off is from B to H, and as the eccentric is fast upon the same shaft, it will, in the same time, of course, move through the same part of a circle.

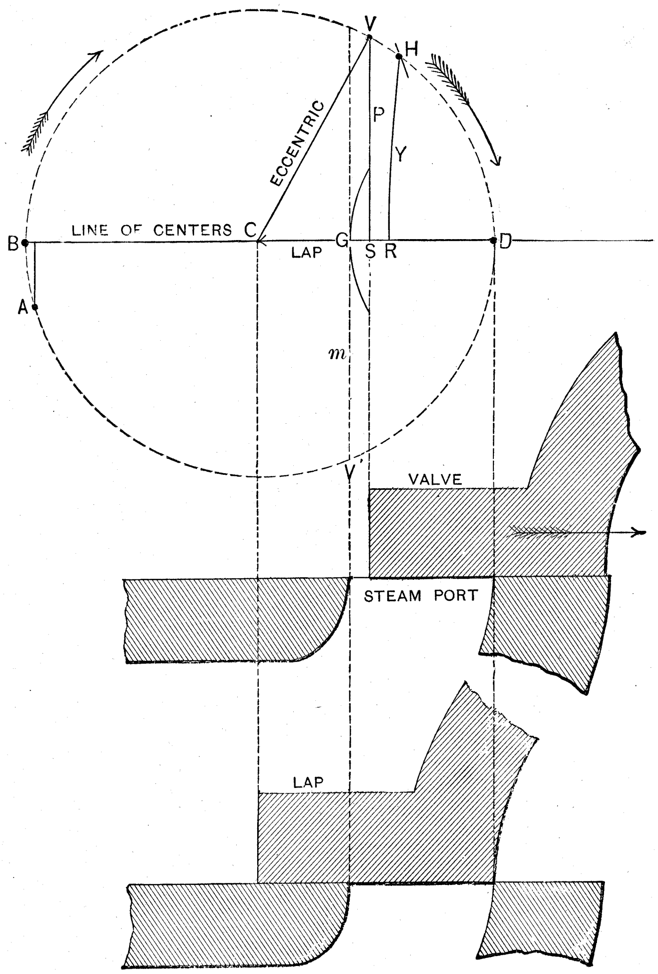


Fig. 3313.

One half of its motion will be to open and one half to close the port, so that we may by means of the arcs at F get the point F, which is midway between B and H, and with the compasses set from B to F, mark from D the two arcs V and V' whose distance apart will obviously be the same as from B to H.

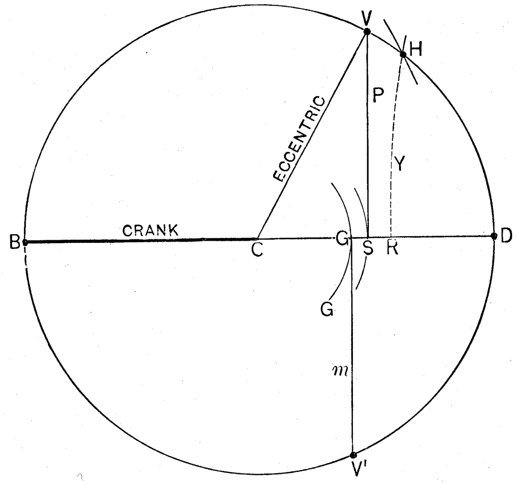
Then from V to V' draw the line P, and from this line to the centre C of the crank shaft is the amount of steam lap necessary for the valve, while from this line (P) to D is the width of the steam port.

The proof of the diagram is as follows:

When the crank is on the dead centre, the centre of the eccentric is at V, its throw line being represented by the line from V to C, and the valve is about to open the port as shown in the figure.

While the eccentric is moving from V to D, the valve will move in the direction of the arrow and will fully open the port, while the crank pin will move from B to F.

Then, while the crank moves from F to H, the eccentric will have moved the valve to the position it occupies in the figure, having closed the port and effected the cut off.



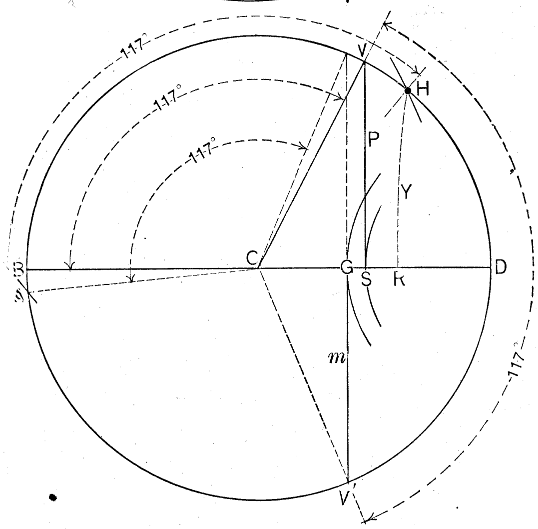


Fig. 3314.

We have here found the amount of lap and the position of the eccentric necessary for a given point of cut off when the latter is given in terms of the piston stroke. If, however, the point of cut off had been given in terms of the crank pin position, we might find the required amount of lap at once, by simply drawing a line from the centre B, the point to H where the crank pin is to be when the cut off occurs.

From this line we could then draw the dotted circle G, and just meeting the line P, which would give the eccentric position.

To find the piston position, the arc Y would require to be drawn by the same means as before.

If the valve is to have lead, the diagram may be constructed as in

Fig. 3313, in which the circle has a diameter equal to the travel of the valve and the cut off is to occur when the piston is at R and the crank at H.

When the valve is at the end of its travel and has fully opened the port, the eccentric will be at D, hence from D we mark an arc G distant from D to an amount equal to the width of the steam port, drop the vertical *m* from G, and at its lower end V′ is the position of the eccentric centre at the point of cut off. Then draw a line P, distant from *m* equal to the lead, which will give at V the position of the eccentric when the crank is on the dead centre, and the valve is open to the amount of the lead. The lap is obviously the distance from the centre C of the crank shaft to the arc G.

We have here found all the points necessary except the point at at V′ the eccentric position at the point of cut off. Then with the compasses set to the points V V′, we may mark from B an arc, locating at H the position of the crank at the point of cut off, and from this, with compasses set to represent the length of the connecting rod on the same scale as the circle represents the path of the crank, we may, from a point on the line of centres, mark an arc Y giving at R the piston position at the point of cut off.

When, therefore, the lap is given, we mark it from the center C of the crank shaft, and find the other elements from it, whereas, when the lap is to be found, we mark the width of the port from the end D of the valve travel, and find the other elements from that.

A proof of all the constructions is given in Fig. 3314, in which

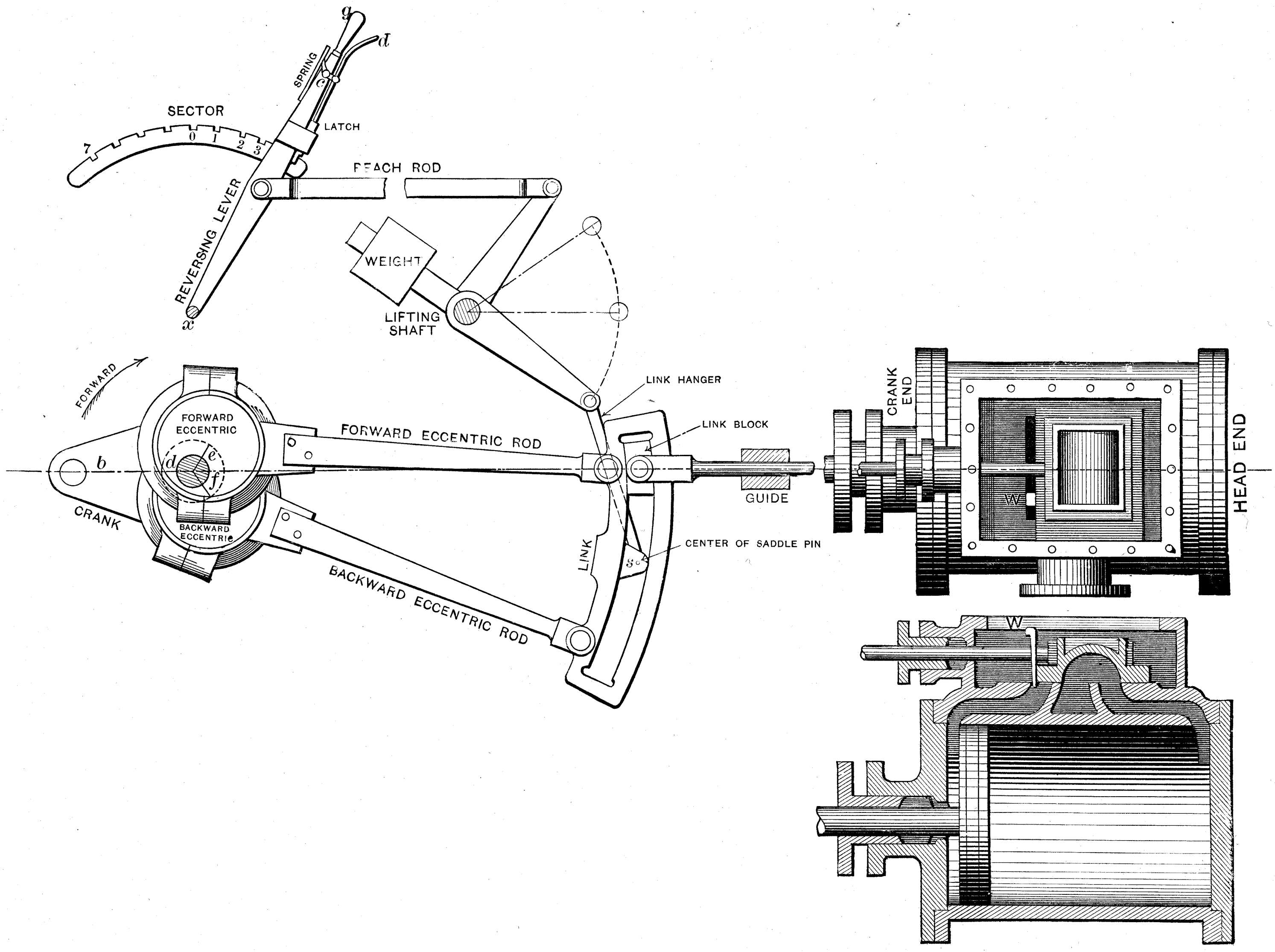


Fig. 3315.

which the valve will open the port for the lead, and this we may find by setting a pair of compasses to the radius B H (or to radius V V′, as both these radii are equal), and from V as a centre, mark at A an arc, which will give the crank pin position at the time the port first opens for the lead, or in other words it will give the position. The proof of the construction is, that if we set the compasses to the distance between the crank pin position on the dead centre and the point of cut off (or from B to H), we may apply the compasses to the points V, V′, which represent the eccentric position when the port is opened to the amount of the lead, and when the cut off occurs.

If the point of cut off only is to be found, we mark from C, Fig. 3314, an arc G representing the amount of valve lap and arc S representing the lead. A vertical P gives the eccentric position V when the crank is on the dead centre at B, and a vertical *m* from G gives the letters of reference correspond to those in the previous figures, and the positions of the parts are marked in degrees of angle.

To find the piston position at the point of cut off, measured in inches, of the piston stroke it must be borne in mind that as the circle B D represents the full travel of the valve, the diagram gives all the positions of the eccentric and valve full size, but that as it represents the crank path on a reduced scale, therefore we must multiply the measurement on the diagram by that scale.

Suppose, for example, that the piston stroke is 10 inches, and the valve travel $2\frac{1}{2}$ inches, and the circle being $2\frac{1}{2}$ inches in diameter, is, when considered with relation to the eccentric motion, full size, but when considered with relation to the piston or crank motion, it is only $\frac{1}{4}$ the size, hence to find the piston position at the time of cut off, we must multiply the distance from B to R by 4.

LINK MOTION FOR STATIONARY ENGINES.

The ordinary mechanism employed to enable a stationary engine to be reversed or run in either direction is the Stephenson link motion. Other forms of link motion have been devised, but the Stephenson form has become almost universal.

Fig. 3315 represents this link motion or reversing gear with the parts in position for the full gear of the forward motion, and Fig. 3316 represents it in full gear for the backward motion.

The meaning of the term full gear is that the parts are in the position in which the steam follows the piston throughout the longest or greatest part of the stroke. When in full gear the link motion operates the valve almost precisely the same as if the eccentric rod was attached direct to the valve spindle and no link motion was used.

Besides enabling the engine to run in both directions, however, the link motion provides a means of reducing the amount of valve travel and thus causes the live steam to be cut off earlier in the piston stroke, thus using the steam more expansively. This is done by moving the reversing lever more upright, the earliest point of cut off being obtained when it is upright and the latch is in the notch marked O on the sector in Fig. 3315. If with the engine standing still we move the link motion from full gear forward to full gear backward and watch the valve, we shall find that the valve lead increases as the reversing lever approaches the upright position, or mid gear as it is termed, and that after passing that point it gradually diminishes again, the valve being so set that the lead is the same for full gear forward as it is for full gear backward.

The reversing lever is used to move the link into the required position and to hold it there (the end of the latch fitting into the notches in the sector being the detaining or locking device); as the link is suspended by its saddle pin S and the link hanger, therefore its motion is to swing or partly rotate on the pin S, and at the same time ending in the arc of a circle whose centre of motion is in the pin at the upper end of the link hanger which is pivoted to the lower arm of the lifting shaft (which is sometimes termed the tumbling shaft). It will clearly be seen that with the position the parts occupy in Fig. 3315, and the crank motion being in the direction of the arrow, the forward eccentric will move the top of the link to the right and therefore the valve will move to the right, while the backward eccentric will move the bottom end of the link to the left.

In full gear, however, the bottom eccentric rod has but a very slight effect indeed on the motion of the valve because both the link hanger and the link block will permit the link to swing on centre of the link block pin as a pivot. If now we turn to Fig. 3316 for the full gear backward, we shall see that these conditions are reversed and the backward eccentric becomes the effective one, being in line with the valve spindle. By shifting the link from one gear to the other, therefore, we have merely changed the direction in which the link will move the valve, and, therefore, the direction in which the engine would run.

In Fig. 3315 for the full gear the parts are shown in position, with the piston at the crank end of the cylinder, and the crank pin on the dead centre, and the eccentrics must be set as shown in the cut, the eccentric rods being open and not crossed. When, however, the crank is on the other dead centre and the piston at the head end of the cylinder, the rods will cross each other, and it is necessary to remember that the rods should be open when the piston is at the crank end of the cylinder. If, however, the running gear contains a rock shaft, or rocker (as is the case in American locomotives), then these conditions are reversed, and the eccentric rods will cross when the piston is at the crank end of the cylinder.

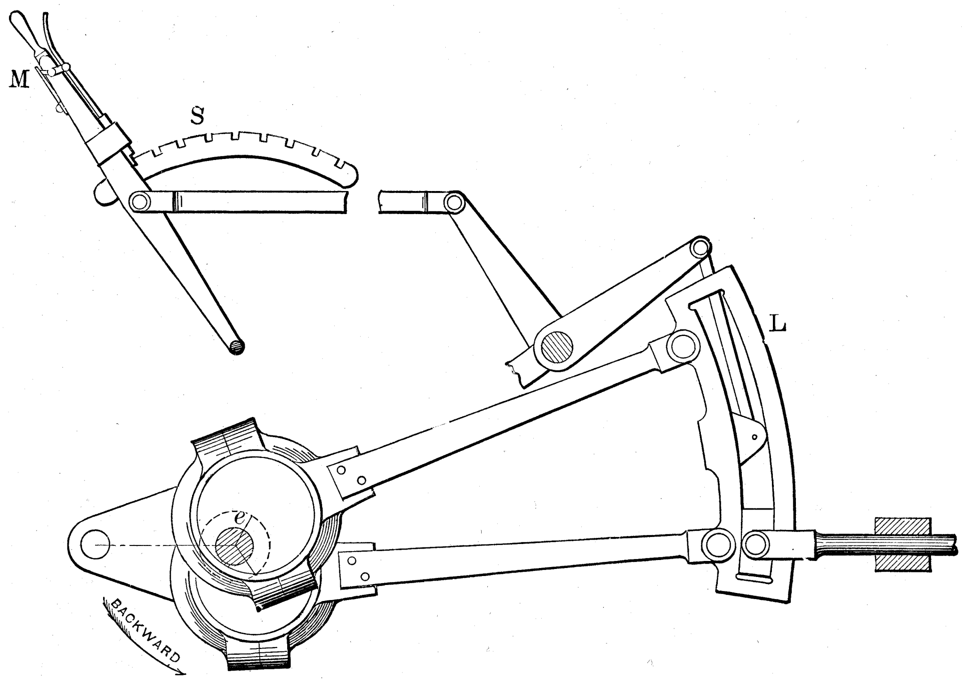


Fig. 3316.

In setting the slide valve of an engine having a link motion, there are two distinct operations. First, to put the crank on the respective dead centres, which will be fully described on page 394, and need not be repeated; and second, to set the eccentrics in their proper positions on the shaft, and correct, if necessary, the lengths of the eccentric rods. The crank being on the dead centre, with the piston at crank end of the cylinder, the eccentric should be moved around on the shaft by hand until there is the desired amount of lead at the crank end port, and temporarily fastened there, a set screw usually being provided (in the eccentric) for this purpose. The lead is best measured with a wedge, W, Fig. 3315. The crank is then put on its other dead centre, and the lead for the head end port is measured. If the lead is to be made equal for the two ports (as is usually the case in horizontal engines) and it is found to come so, the valve setting for the forward gear is complete. If the lead is not equal, the forward eccentric rod or else the valve spindle must be altered so as to make the lead equal. In some engines adjusting screws are provided for the purpose of regulating the length of either the eccentric rod or else of the slide spindle; it does not matter which is altered. The link motion is then put in full gear for the backward motion, and, with the crank on the dead centre (it does not matter which dead centre), the eccentric is moved by hand upon the crank shaft until there is the required amount of valve lead. The eccentric is then fastened on the shaft and the crank put on the other dead centre, and the lead tried for the other port, and made equal by lengthening or shortening the backward eccentric rod. It is to be noted that altering the length of the eccentric rod or of the valve spindle makes it necessary to reset the eccentric, as it affects the amount of lead at both ports; hence, if any alteration of rod length is made, the whole process here described must be repeated after each alteration of rod length.

FLY BALL OR THROTTLING GOVERNORS.

An isochronal governor is one in which the two opposing forces are equal throughout the whole range of governor action, or, in other words, equal, let the vertical height of the plane in which the balls revolve or swing be what it may.

A dancing governor is one that acts spasmodically. Such an action may occur from undue friction in the parts of the governor or of its throttle valve.

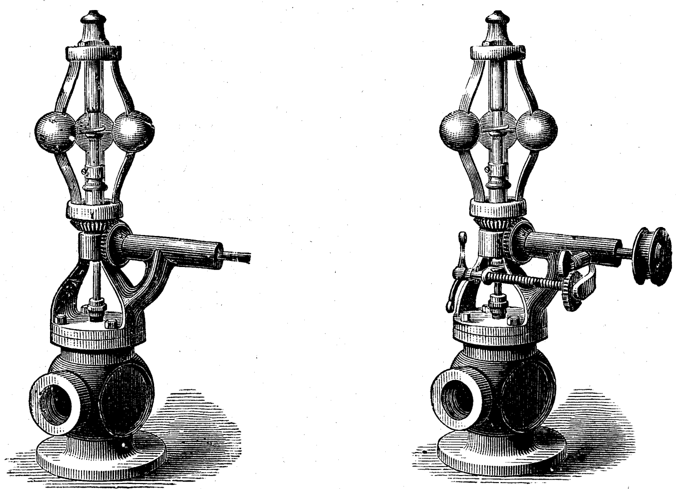
The friction offers a greater resistance to starting the parts in motion than it does to keep them in motion after being started; hence, the parts are apt to remain at rest too long, and to move too far after being put in motion.

Rule to find the number of revolutions a governor should make. Divide the constant number 375.36 by twice the square root of the height of the cone in inches. The quotient is the proper number of revolutions per minute.

Example.—A governor with arms $30\frac{1}{2}$ inches long, measuring from the centre of suspension to the centre of the ball, revolves, in the mean position of the arms, at an angle of about thirty degrees with a vertical spindle forming a cone of about $26\frac{1}{2}$ inches high. At what number of revolutions per minute should this governor be driven? Here the height of the cone being 26.5 inches, the square root of which is 5.14 and twice the square root 10.28, we divide 375.36 by 10.28, which give us 36.5 as the proper number of revolutions per minute at which the governor should be driven.

The construction of the Pickering governor is as follows:

In Fig. 3317 it is shown in its simplest form, and in Fig. 3318



Figs. 3317, 3318.

with the driving pulley and speeder (or engine speed regulating device) attached. This speeder consists of a spiral spring whose tension may be adjusted to more or less resist the rise of the governor balls, and thus enable the engine to run at a greater speed for a given amount of rise of the governor balls, hence by increasing the tension the engine speed is increased.

THE SPRING ADJUSTMENT.

The adjustment of the spring tension is made by a worm actuating a worm wheel on a rod passing through the spring, and to which one end of the spring is attached, the other acting on an arm that projects into a slot in the governor spindle. It is obvious that the speeder can be adjusted while the engine is running.

In Fig. 3319 the governor is shown with the speeder and Sawyer's valve, the latter enabling the governor valve to be opened or closed without affecting the rise and fall of the governor balls, which is done by operating the arm shown on the right, whose

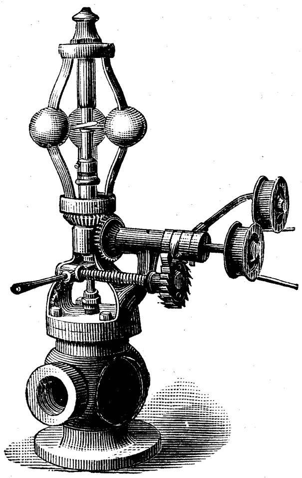


Fig. 3319.

ends are provided with loops, so that a cord may be attached, enabling the engineer to operate the governor from a distance.

The safety stop or stop motion is shown on the right, Fig. 3320.

It acts to close the governor valve and stop the engine in case the belt that drives the governor should get off the pulley or break. This stop motion consists of a pulley suspended by a rod, and riding on the belt which supports its weight. Should the governor belt break, this pulley falls and severs the connection between the valve and the governor, closing the valve, and holding it closed. Fig. 3321 shows the governor in section to expose the construction of the valve. The valve V is what is termed a poppet or poppet valve, which is balanced, because the steam entering at I, and taking the course denoted by the arrows, acts equally on both ends of the valve and does not press it in either direction, while as the steam surrounds the valve it is not pressed sideways.

At B is a gland or stuffing box to keep the spindle or rod steam-tight. At A is the slot for receiving the arm from the speeder and from the stop motion.

P is obviously the driving pulley, imparting motion to the bevel wheels G, which drive the outer spindle S, the inner spindle *s'* being connected to A. The balls are upon ribbon springs D, which are secured at their lower ends to a link fast to the spindle S.

The centrifugal force generated by the balls causes them to move outwards, their upper ends pulling down the cap to which they are secured, and this cap operates the valve.

Governors of this class are sometimes termed *fly-ball* governors.

STARTING A PLAIN SLIDE-VALVE ENGINE.

The method to be pursued before starting a plain slide-valve engine depends upon what the engineer knows about the condition of the engine.

If he knows the engine is in proper running order, all that is necessary is to first attend to the oil cups and start them feeding.

Then, if it is necessary, move the crank into the required position to start it easily; open the waste water cocks to relieve the cylinder of the water that will be condensed from the steam when it enters a cool cylinder, and turn on the steam; giving the throttle valve enough opening to start the engine slowly.

The best position for the crank pin to be in to enable its starting easily is midway between the horizontal and vertical position (or, in other words, at an angle of 45° to the line of centres) and inclining toward the cylinder, so that when the engine moves the piston will travel toward the crank shaft.

There are two reasons why this is the best position for starting.

The first applies to all engines because there is a greater piston area for the steam to act on when the piston is moving toward the crank than there is when it is moving away from it. This occurs because the piston rod excludes the steam from a part of the face of the piston. The second applies to all plain slide-valve engines whose slide valves have equal laps and both steam ports of equal widths, because the live steam follows further on the stroke when the piston is moving toward the crank than it does when it is moving away from it, and it follows that more piston power is developed, and the engine is less likely to stop when passing the dead centre.

When first taking charge of an engine, it is proper, before starting it, to ascertain that it is in fair working order.

A complete examination of an engine should include a test of the fit of the piston to the cylinder bore, of the cross head to the guide bars, of the connecting rod brasses to the crank pin and cross head journals, and of the crank shaft to its bearings. It would also include a testing of the alignment of the crank shaft and of the guide bars, as well as the set of the valves and the adjustment of the governor.

The least examination permissible with a due regard to safety would be to move the engine throughout at least one full revolution by hand, and to see that the connecting rod brasses and the main bearings do not fit too tight to their respective journals, and to then start the engine slowly by giving it only enough steam to move it, keeping the hand on the throttle valve so as to be able to shut off steam instantly should it become necessary.

A thorough examination should be made in the following order:

First, slightly loosen the nuts on the crank shaft bearings and also the connecting rod keys.

Then move the fly wheel around until the crank points straight to the cylinder, which will bring the piston up to the outer end of the cylinder bore.

Take off the cylinder cover and also the follower from the piston head, and see that the piston rings are set out to fit the cylinder bore but not to bind it tight. Then bolt the follower up firmly in place again.

Take off the connecting rod and move the piston until it touches the cylinder cover at the other or crank end of the cylinder, and then draw a line across the side face of the cross head guide and on the guide itself.

Put on the cylinder cover and push the piston back until it abuts against it, and then make another line on the cross head guide and the guide bar, and these two lines will show the extreme positions to which the piston can be moved when the connecting rod is disconnected.

Next put on the connecting rod, carefully adjust the keys or wedges, so that the bores of the brasses fit easily to the crank pin and cross head pin, seeing that the oil holes are clear, and that oil will feed properly to the journals.

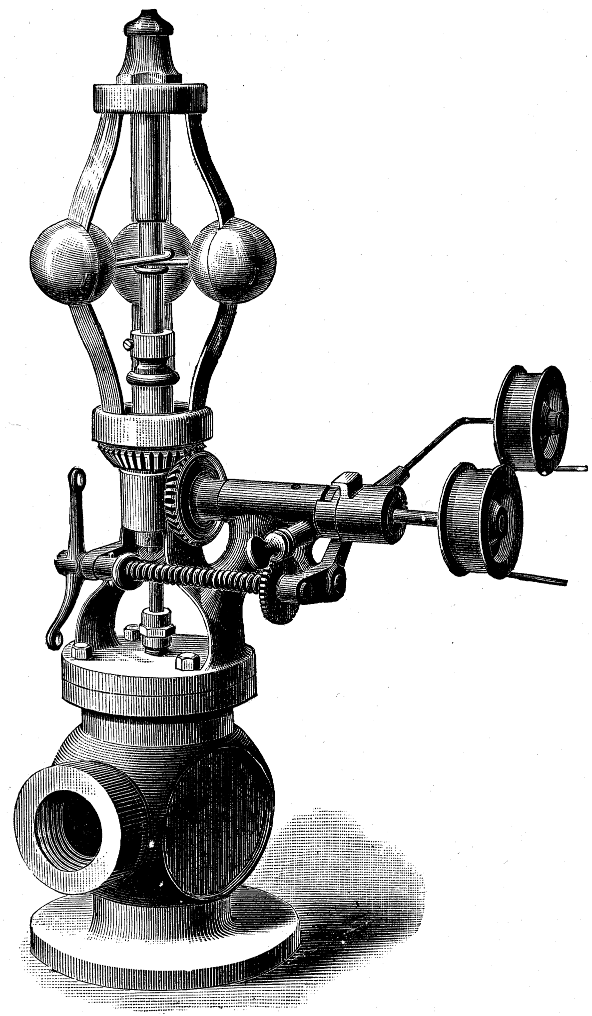


Fig. 3320.

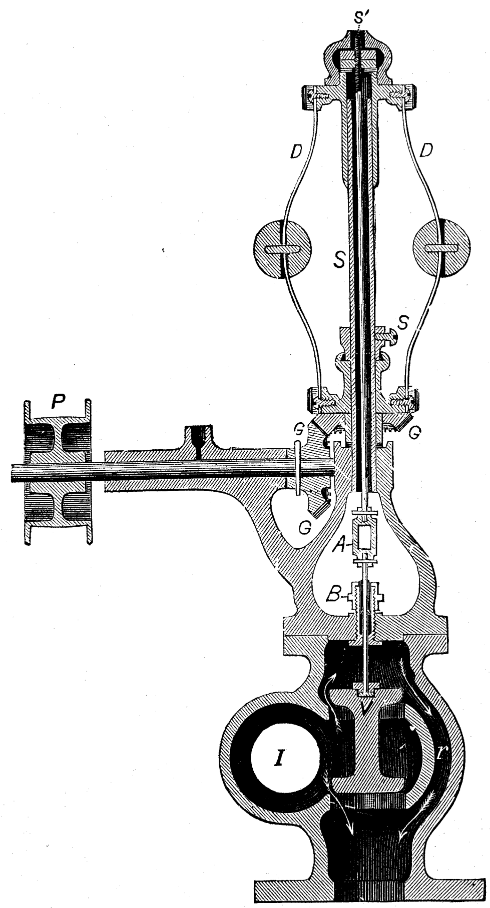


Fig. 3321.

In making this adjustment it is a good plan, if there is any end play of the brasses on the crank pin, to set up the key or wedge until the rod can just be moved by hand on the pin, by first pulling the rod to one end of the pin, and then pushing it to the other.

In putting on the rod, it will be necessary to move the piston a trifle towards the crank.

In making the adjustment of the crank pin fit to the rod brasses, it is a good plan to drive the key home until the brasses are known to bind the crank pin, and then mark a line across the side face of the key and fair with the top face of the connecting rod strap, to then slacken back the key enough to ease back the brasses to a proper fit, and then mark another line on the key.

The first line will form a guide as to how much to slacken back the brasses to adjust the fit, and the second one will form a guide as to how much the key is moved when making a second adjustment, if one should be found necessary after the engine has been running.

Similarly in adjusting the main bearing boxes to the crank shaft, either the nuts, or what are called leads, may be taken to adjust the fit. Leads are necessary when the joint faces of the brasses do not meet, but are left open so that the wear can be taken up while the engine is running.

It is better, however, to let the brasses abut together, so that it may be known that the fit is correct when the nut is screwed firmly home.

The method of taking a lead is as follows: The top brass is loosened, and between the joint faces of the brasses or boxes on each side of the shaft a piece of lead wire is inserted. For a shaft of, say, four inches in diameter, the lead wire will be about $\frac{7}{16}$ inch in diameter, or for 10 inch shaft the wire should be $\frac{1}{8}$ inch in diameter, and should be as long as the brass. The nuts are then screwed firmly home, and the wire will be squeezed between the brasses and thus flattened on two opposite sides, the thickness showing how far the joint faces of the brasses are apart when the bore grips the journal.

A liner, fit strip, distance piece, or shim (all these names meaning the same thing) is a strip of metal placed between the joint faces of the brasses to hold them the proper distance apart to make a working fit of the journal and brasses, when the latter are firmly bolted up.

The fit of the top brass therefore depends upon the fit strip being of the proper thickness from end to end.

Now the lead wire is the gauge for the thickness of the fit strip, the latter being made a trifle thicker than the flattened sides of the lead.

If the lead is thicker one end than the other, or if one lead is thicker than the other, the fit strips must be made so, and the leads must be marked so that it may be known which way they were placed between the brasses so that the proper fit strip may be on the proper side of the brass, and the proper end towards the crank.

Another method that is adopted in the case of large brasses is to screw down the nuts until the brasses bind the journal, and then make a mark on the nut and on the bolt thread. The nut is then slackened back as much as the judgment dictates, and a note made of how much this is, the marks forming a guide.

As the wear takes place, and the nuts screw farther down, a new mark is made on the nut, so that it may always be known how much to screw up or unscrew the nut, to make a light adjustment.

To avoid heating, it is a good plan to press some tallow into the bottom or in one corner of the oil cup, and then pour in the oil used for ordinary lubrication. So long as the bearing remains cool, the oil will feed and the tallow remain.

If the bearing heats, the tallow will melt, and, having a heavier body, will give a more suitable lubrication.

To find if the connecting rod is of the right length to give, as it should do, an equal amount of clearance (or space between the piston and the cylinder cover) at each end of the stroke, move the fly wheel a trifle in either direction, and then move it back until the crank is on the dead centre, and draw a line across the cross head guide and guide bar, and the distance between this line and that drawn when the connecting rod was disconnected, shows the amount of clearance at that end of the cylinder. Then move the crank pin over to its other dead centre, and mark a line across the cross head guide and the guide bar, and the distance between this line and that drawn before the connecting rod was put on will show the clearance at this end of the cylinder.

If the clearance is not equal for the two ends, it should be made so by putting liners behind the connecting rod brasses so as to lengthen or shorten the connecting rod (according as the case may require), and equalize the clearance, while at the same time bringing the connecting rod keys up to their proper heights.

To test the set of the valve, the steam-chest cover must be taken off, the crank placed alternately on each dead centre, and the lead measured for each port.

An unequal or an equal degree of valve lead may be given by suitably altering the length of the eccentric rod, but when the lead is equal for the two ports, its amount must be regulated by moving the position of the eccentric upon the crank shaft.

SQUARING A VALVE.—A method not uncommonly pursued in setting a valve is to what is called *square it* before trying it.

This squaring process consists in so adjusting the length of the eccentric rod that the valve travels an equal distance over or past the steam edge of each steam port; but since the valve does not, when set to give equal lead, travel equally past each port, therefore the work done in squaring a valve is all thrown away, and may result in altering the eccentric rod from its proper length to an improper one, necessitating that it be altered back again in order to set the lead right.

The proper method is to adjust both the length of the rod and the position of the eccentric, by testing the lead at once, lengthening the eccentric rod to increase the lead at the crank end, or vice versa.

Each alteration of eccentric position may render necessary an alteration of rod length, or vice versa, each alteration of rod length may render it necessary to alter the eccentric position, hence the lead should be tried at both ends of the cylinder after each alteration of either rod length or eccentric position.

In vertical engines the weight of the crank shaft causes it to wear the bottom brass or part of the bearing box the most, thus lowering its position, while the eccentric straps and pins wear most in the same direction; hence the wear increases the lead at the head end of the cylinder when the latter is above the crank, and at the crank end when the crank is above the cylinder.

When the cylinder is above the crank, the weight of the piston, cross head and connecting rod is counterbalanced at the end of the downward piston stroke by giving the crank end port more lead; but when the cylinder is below the crank, it is the head end port that must be given increased lead to prevent a pound or knock, or to allow for the wear downwards of the parts.

After an engine is started, the pet cocks should (if they are not automatic) be closed as soon as dry steam issues, and if this cannot be seen, it may be assumed to occur after the engine has made about 20 revolutions.

The parts that will then require particular attention are the crank pin, main bearings, cross head guides and the pump, if there is one. The former must be kept properly lubricated, so that they may not get hot and the cylinder lubricator (which is usually placed on the steam pipe) must be set to self feed properly.

If the crank shaft bearings should begin to heat, loosen the cap bolts and lubricate more freely, or, if it is at hand, some melted tallow may be applied with the oil, as a heavier lubricant may stop the heating.

The crank pin requires the most attention and is the most difficult to keep cool and to examine, because of its circular path rendering it difficult to feel it. This may be done, however, in two ways, first by standing at the end of the engine bed and gradually extending the hand, until the end of the rod meets it as it passes, and, second, by placing the hand on the connecting rod as near to the end of the guide bar as possible where its motion is diminished and moving the hand towards the crank pin, by which means the end of the crank pin may be approached gradually.

If the end of the rod is hot, the engine speed should be reduced or the engine should be stopped so that the connecting rod key or wedge may be eased back and the oil feed made more copious. Then, after the engine has been stopped for the night, the brasses should be taken out and any rough surface, either on the brasses or on the pin, smoothed down with a file.

Hot crank pins may occur from several causes, but by far the most common ones are from improper oiling, or from the engine being out of line.

A heavier oil will often stop, or at least modify, the heating, but its cause should always be discovered and remedied.

Engines that are used out of doors or are exposed to temperatures below the freezing point must be left so that steam leaks may not condense in any of the parts or pipes and burst them.

Leaky throttle valves may, for example, cause water to accumulate in the steam chest and freeze, perhaps bursting the steam-chest cover.

To prevent this let the engine stand with the crank just past the dead centre, so that the steam port will be open, and open the waste water cocks on the cylinder, and also on the steam chest if there is any.

If the cylinder is jacketed all the drain cocks for the jacket should also be opened.

A leaky check valve may cause the steam to condense in the pump and freeze it up solid or burst it or the pipes. To avoid this, open the pump pet cock.

Open all the drain cocks on the heater and water pipes.

If the water is left in the boiler all night it is liable to freeze.

To prevent this leave a well banked fire.

In extreme weather remember that on exposed engines the oil, if of such quality as sperm or lard oil, may freeze and prevent feeding until the bearings get hot and melt the oil.

To prevent this use a lighter oil, as, for example, a mineral oil. Or, in case of freezing, melt the oil in the cups with a piece of wire made red hot while getting up steam in the morning.

A good plan to prevent oil from freezing and yet have a good quality of oil is to mix two parts of lard oil with one part of kerosene.

Portable engines should stand as nearly level as possible, so that the water will stand level above the tubes and crown sheet of the fire box.

When feed water is drawn from a natural supply, as from a stream, the strainer at the end of the suction pipe should be clear of the bottom of the stream, where it is liable to be choked.

When the exhaust steam is used to feed the boiler, do not open the valve that lets the exhaust steam into the feed-water tank until a little while after the engine has started, because the oil fed to the cylinder will otherwise pass into the feed tank and may cause priming.

In engines having plunger pumps for feeding the boiler it is essential to keep the plunger properly packed, as a leak there impairs or stops the pump from acting.

A gauge glass may be cleaned when the engine is cold by shutting off the cocks leading from the boiler and filling the glass with benzine, allowing it to stand two hours; the benzine must be let out at the bottom of the glass tube, and not allowed to enter the boiler.

In starting a new engine be careful to let the bearings be slightly loose.

At first give only enough steam to just keep the engine going, and keep the hand on the throttle valve ready to shut off steam instantly if occasion should require.

PUMPS.

Pumps are divided into the following classes:

Lift pumps, in which the water flows freely away from the pump, which performs lifting duty only.

Force pumps, which deliver the water under pressure.

Plunger pumps, in which a "plunger," or "ram," as it is sometimes termed, is used.

Piston pumps have a piston instead of a plunger.

A double acting pump is one in which water enters into and is delivered from the pump at each stroke of its piston or plunger, or, in other words, one in which, while water is being drawn in at one end of the pump, it is also being forced out at the other.

A single acting pump is one in which the water enters the pump barrel during one piston or plunger stroke, and is expelled from the pump during the next stroke, hence the action of the suction and of the delivery is intermittent, although the pump is in continuous action.

For very heavy pressures plunger pumps are generally used, the plunger being termed a *ram*.

The advantage of the plunger or ram is that it gives a positive displacement, whereas in a piston pump a leaky piston permits the water from the suction side to pass through the leak in the piston, to the delivery side.

Piston pumps possess the advantage that there is less difference between the contents of the pump and the displacement than is the case in plunger pumps.

The displacement of a piston pump is found by multiplying the area of the pump bore by the length of the piston stroke.

The displacement of a plunger pump is less than the above, by reason of there being a certain amount of clearance or space between the circumference of the plunger and that of the cylinder bore.

It is desirable to keep the clearance space in all pumps as small

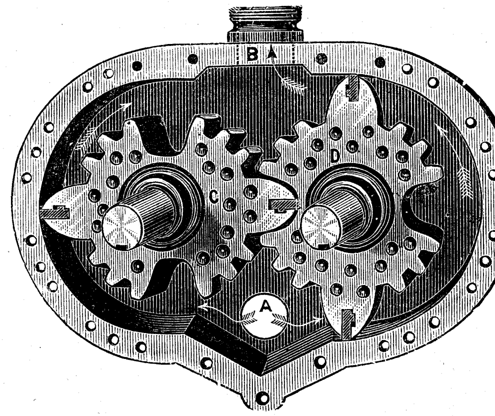


Fig. 3322.

as the conditions will allow, especially if the pump is liable to lose its water.

Losing the water means the falling of the suction water back into the source of supply, which may occur when the engine has to stop temporarily, and there is a leak in the suction valves.

Rotary pumps are those in which the piston revolves, an example of the most successful form of rotary pump being shown in Fig. 3322, which is that used by the Silsby fire engine.

The advantage possessed by a rotary pump is that it keeps the water passing through the suction in a continuous and uniform stream, as it has no valves.

It may therefore be run at a high velocity or attached direct to the engine shaft.

If a rotary pump leaks, the efficiency is not impaired so much as

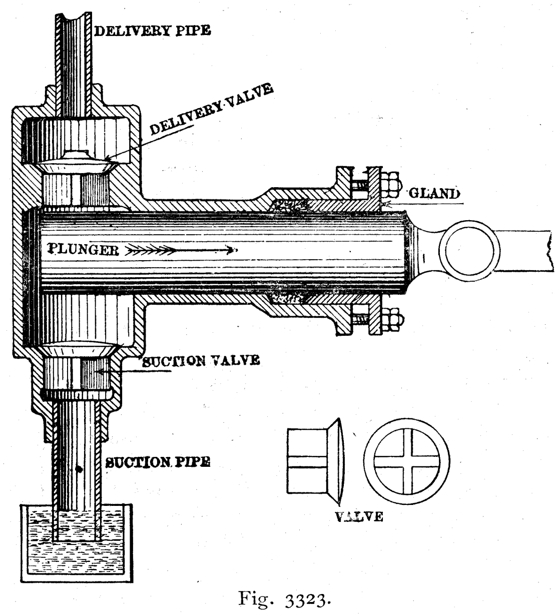


Fig. 3323.

in a piston or plunger pump, all that is necessary being to run the pump at a high speed.

The principles of action of a pump may be understood from Fig. 3323, which represents a single acting plunger pump shown in section, and with the suction pipe in a tank of water, the pump being empty.

The surface of the water in the tank has the pressure of the atmosphere resting upon it, and as the pump is filled with air, the surface of the water within the pipe is also under atmospheric pressure.

Now suppose the plunger to move to the right, and as no more air can get into the pump, that already within it will expand, and will therefore become lighter, hence there will be less pressure on the surface of the water within the suction pipe than there is on the outside of it, and as a result the water will rise up the pipe, not because the plunger draws it, but because the air outside the pipe presses it up within the pipe.

The water inside the pipe will rise above that outside in proportion to the amount to which it is relieved of the pressure of the air, so that if the first outward stroke of the plunger reduces the pressure within the pump from 15 lbs. to 14 lbs. per square inch (15 lbs. per square inch being assumed to be its normal pressure), the water will be forced up the suction pipe to a distance of about $2\frac{1}{4}$ feet, because a column of water an inch square and $2\frac{1}{4}$ feet high is equal to 1 lb. in weight. In Fig. 3324 the pump plunger is shown to

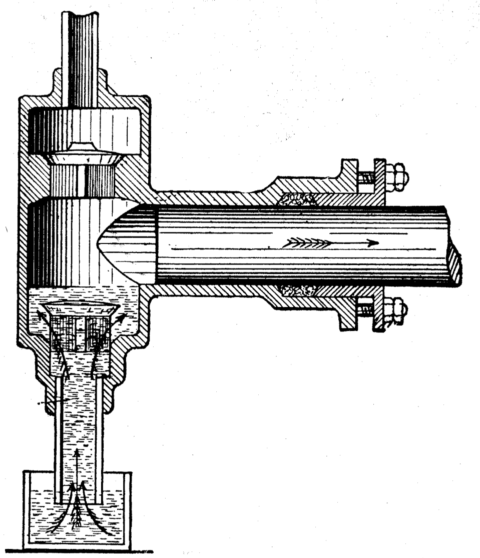


Fig. 3324.

have moved enough to have permitted the water to rise above the suction valve, and it will continue to rise and enter the pump barrel as long as the plunger moves to the right.

When the plunger stops, the suction valve will fall back to its seat and enclose the water in the pump; but as soon as the plunger moves back to the left hand and enters the barrel pump further, the delivery valve will rise, and the plunger will expel from the pump a body of air or water equal in volume to the cubical contents of the plunger, or rather of that part of it that is within the barrel, and displaces water.

If the plunger was at the end of its first stroke to the right and the pump half filled with air, then this air will be expelled from the pump before any water is; whereas if the pump was filled with water, the latter only will be delivered.

Now suppose the first plunger stroke reduces the air pressure from 15 to 14 lbs., and that the second drawing stroke of the plunger reduces the air pressure in the pipe to 13 pounds per inch, the water will rise up it another $2\frac{1}{4}$ feet, and so on until such time as the rise of a column of water within the pipe is sufficient to be equal in weight to the pressure of the air upon the surface of the water without; hence it is only necessary to determine the height of a column of water that will weigh 15 lbs. per square inch of area at the base of the column to ascertain how far a suction pump will cause water to rise, and this is found by calculation or measurement to be a column nearly 34 feet high.

It is clear then, that however high the pump may be above the level of the water, the water cannot rise more than 34 feet up the suction pipe, even though all the air be excluded from it and a perfect vacuum formed, because the propelling force, that is, the atmospheric pressure, can only raise a column of water equal in weight to itself, and it is found in practice to be an unusually good pump that will lift water thirty feet.

Fig. 3325 shows the plunger making a delivery stroke, the suction valve being closed, and the delivery valve open where it will remain until the plunger stops.

To regulate the quantity of water the pump will deliver in cases where it is necessary to restrict its capacity, as in the case of maintaining a constant boiler feed without pumping too much water in the boiler, the height to which the suction valves can lift must be restricted, so as to limit the amount of water that can enter the pump at each drawing stroke.

The delivery valve should lift no more than necessary to give a free discharge without causing the valve to seat with a blow; but if the pump has a positive motion, the delivery valve must open wide enough to let the water out, or pressure enough may be got up in the pump to break it.

A check valve is merely a second delivery valve placed close to the boiler and serving to enable the pump to be taken apart if occasion should arise, without letting the water out of the boiler.

The lift and fall of both valves act to impair the capacity of the pump. Thus, while the suction valve is falling to its seat, the water already in the pump passes back into the suction pipe, and similarly, while the delivery valve is closing, the delivery water passes back.

A foot valve is virtually a second suction valve placed at the bottom or foot of the suction pipe.

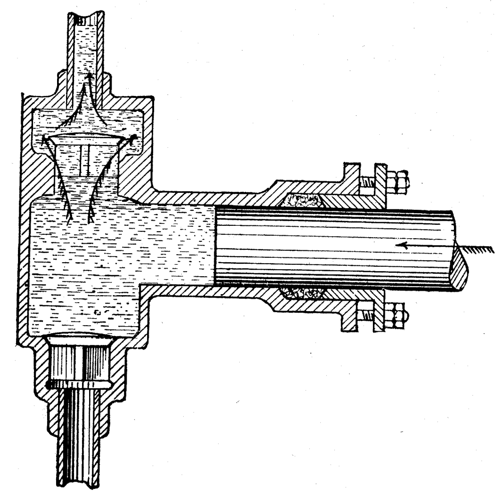


Fig. 3325.

The capacity of a pump is from 70 to 85 per cent. of the displacement of the plunger or piston, and varies with the speed at which the plunger or piston runs.

If a pump runs too fast, the water has not sufficient time to follow the piston or plunger, especially if the suction pipe has bends in it, as these bends increase the friction of the water against the bore of the pipe.

The speed of the piston or plunger should not exceed such as will require the water to pass through the suction pipe at a speed not greater than 500 feet per minute, and better results will be obtained at 350 feet per minute.

An air chamber placed above the suction pipe of any pump causes a better supply of water to the pump by holding a body of water close to it, and by making the supply of water up the suction pipe more uniform and continuous. Air chambers should be made as long in the neck as convenient, so that the water in passing through the pump barrel to the delivery pipe could not be forced up into the chamber, as, if such be the case, the air in the chamber is soon absorbed by the water.

Belt pumps are more economical than independent steam pumps, because the power they utilize is more nearly the equivalent of the power it takes to drive them, whereas in steam pumps there is a certain amount of steam, and therefore of power, expended in tripping the valves and in filling the clearance spaces in the cylinder. Furthermore, the main engine uses the steam expansively, whereas the steam pump does not.

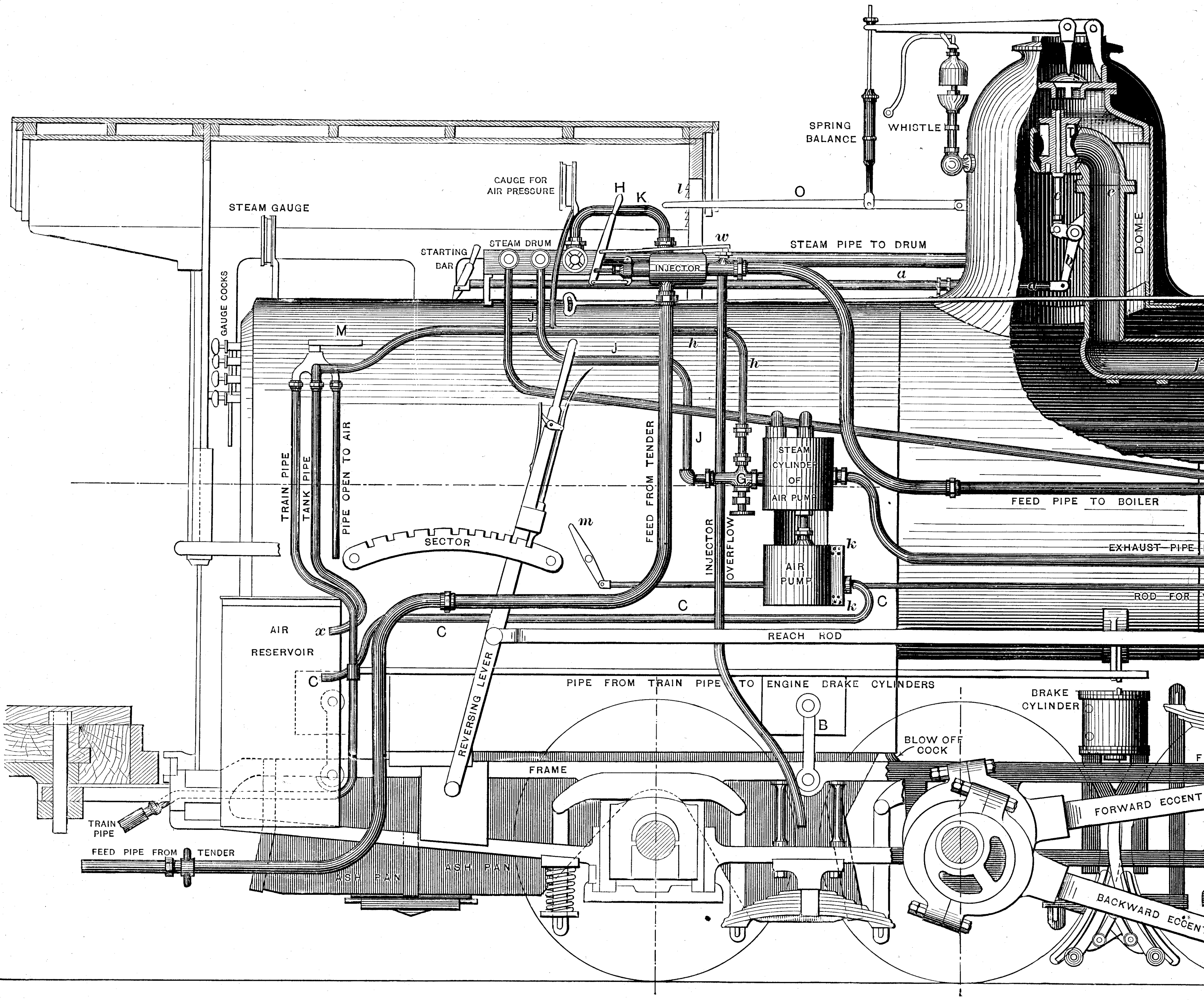
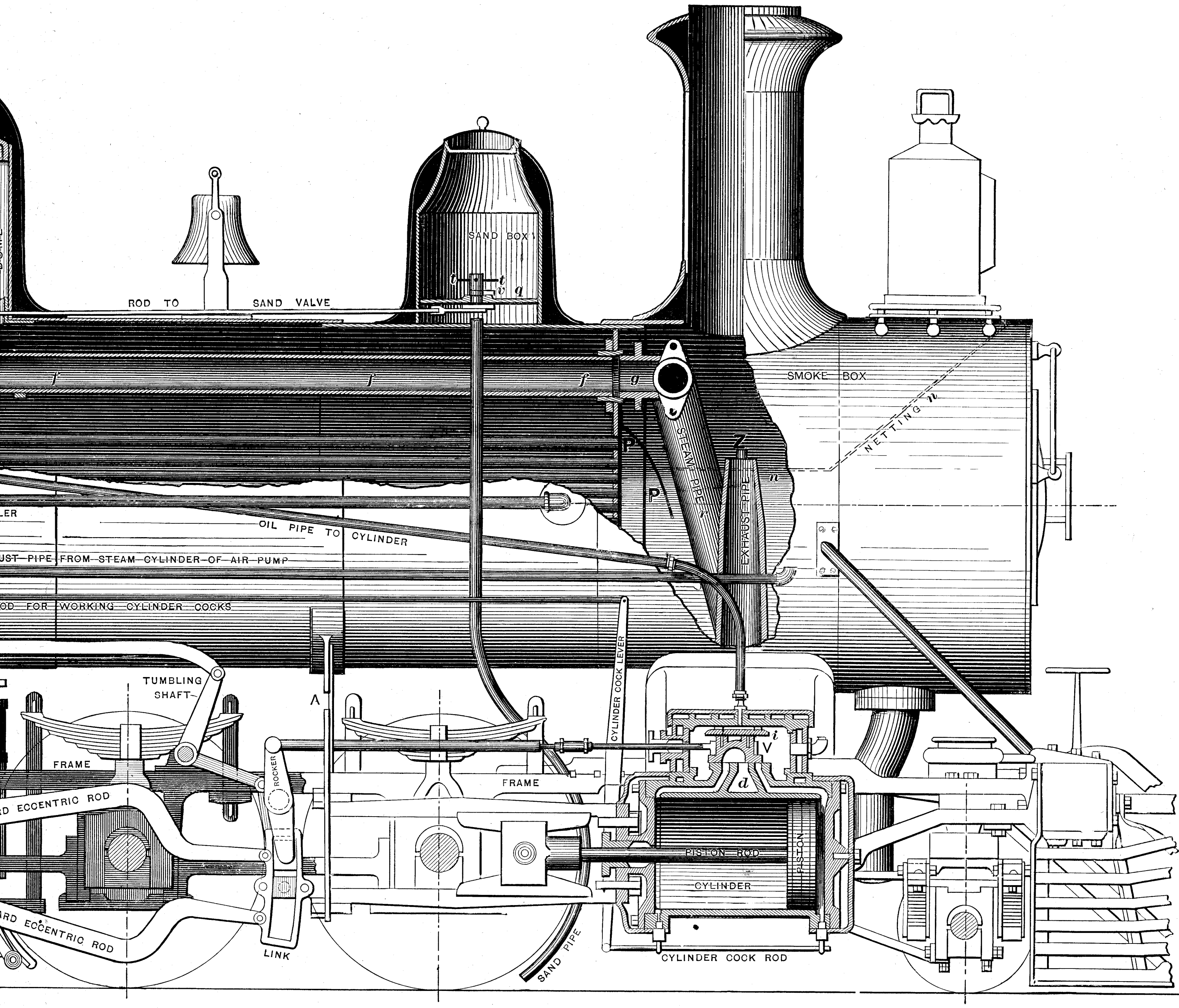

SPRING BALANCE
WHISTLE
CAUGE FOR AIR PRESSURE
H
K
O
STEAM GAUGE
STEAM DRUM
STARTING BAR
INJECTOR
STEAM PIPE TO DRUM
DOME
GAUGE COCKS
M
TRAIN PIPE
TANK PIPE
PIPE OPEN TO AIR
FEED FROM TENDER
STEAM CYLINDER OF AIR PUMP
FEED PIPE TO BOILER
SECTOR
INJECTOR OVERFLOW
AIR PUMP
EXHAUST PIPE
AIR RESERVOIR
REACH ROD
REVERSING LEVER
PIPE FROM TRAIN PIPE TO ENGINE BRAKE CYLINDERS
BRAKE CYLINDER
BLOW OFF COCK
FRAME
FORWARD ECCENT
BACKWARD ECCENT
TRAIN PIPE
FEED PIPE FROM TENDER
ASH PAN

Fig. 33



ROD TO
SAND VALVE
SAND BOX
SMOKE BOX
NETTING n
STEAM PIPE
EXHAUST PIPE
OIL PIPE TO CYLINDER
UST PIPE FROM STEAM CYLINDER OF AIR PUMP
OD FOR WORKING CYLINDER COCKS
TUMBLING SHAFT
FRAME
FRAME
RD ECCENTRIC ROD
RD ECCENTRIC ROD
ROCKER
LINK
CYLINDER COCK LEVER
PISTON ROD
PISTON
CYLINDER
CYLINDER COCK ROD
SAND PIPE

Fig. 3326.