steam distribution are determined by drawing lines AB and CD parallel to the piston's path and distant from it by the amount of the outside and inside lap respectively. Then a, b, c, and d, and the corresponding points p, q, r, and s determine the four events as in former diagrams. Fig. 65 shows at a glance the amount of steam-opening at any part of the period of admission. AE is the lead. The events for the other side of the piston are determined by drawing AB above and CD below the middle line.

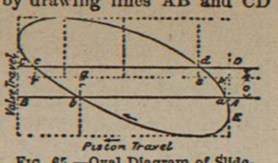

FIG. 65.—Oval Diagram of Slide-Valve Motion.

147. The graphic construction most usually employed in slide-valve investigations is the ingenious diagram published by Dr G. Zeuner in the *Civilingenieur* in 1856.[1] On the line AB (fig. 66), which represents the travel of the valve, let a pair of circles (called valve-circles) be drawn, each with diameter equal to the half-travel. A radius vector CP, drawn in the direction of the eccentric at any instant, is cut by one of the circles at Q, so that CQ represents the corresponding displacement of the valve from its middle position. That this is so will be seen by drawing PM (as in fig. 59) and joining QB, when it is obvious that CQ = CM, which is the displacement of the valve. The line AB with the circles on it may now be turned back through an angle of $90° + \theta$ (θ being the angular advance), so that the valve-circles take the position shown to a larger scale in fig. 67. This makes the direction of CQ (the eccentric) coincide on the paper with the simultaneous direction of the crank, and hence to find the displacement of the valve at any position of the crank we have only to draw CQ in fig. 67 parallel to the crank, when CQ represents the displacement of the valve to the scale on which the diameter of each valve circle represents the half-travel of the valve. CQ_0 is the valve displacement at the beginning of the stroke shown by the arrow. Draw circular arcs ab and cd with C as centre and with radii equal to the outside lap o and the inside lap i respectively. Ca is the position of the crank at which preadmission occurs. The lead is a_0Q_0. The greatest steam opening is a_1B. The cut-off occurs when the crank has the direction Cb. Cc is the position of the crank at release, and Cd marks the end of the exhaust.

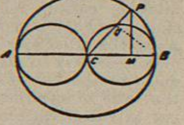
Fig. 66.

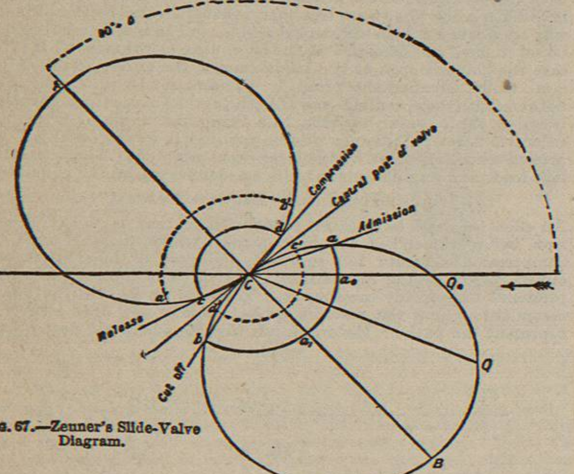

FIG. 67.—Zeuner's Slide-Valve Diagram.

148. In this diagram radii drawn from C mark the angular positions of the crank, and their intercepts by the valve circles determine the corresponding displacement of the valve. It remains to find the corresponding displacement of the piston. For this Zeuner employs a supplementary graphic construction, shown in fig. 68. Here ab or $a'b'$ represents the connecting rod, and bc or $b'c$ the crank. With centre c and radius ac a circle ap is drawn, and with centre b and radius ab another circle aq. Then for any position of the crank, as cb', the intercept pq between the circles is easily seen to be equal to aa', and is therefore the distance by which the piston has moved from its extreme position at the beginning of the stroke. In practice this diagram is combined with that of fig. 67, by drawing both about the same centre and using different scales for valve and piston travel. A radius vector drawn from the centre parallel to the crank in any position then shows the valve's displacement from the valve's middle position by the intercept CQ of fig. 67, and the piston's displacement from the beginning of the piston's motion by the intercept pq of fig. 68.

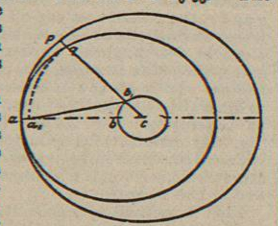
Fig. 68.

[1] Zeuner, *Treatise on Valve Gears*, transl. by M. Müller, 1868.

149. In all the figures which have been sketched the events refer to the front end of the cylinder, that is the end nearest to the crank (see fig. 63). To determine the events of steam distribution at the back end, the lap circles shown by dotted lines in fig. 67 must also be drawn, Ca' being the outside lap for the back end, and Cc' the inside lap. These laps are not necessarily equal to those at the other end of the valve. From fig. 65 it is obvious that, especially with a short connecting-rod, the cut-off and release occur earlier and the compression later at the front than at the back end if the laps are equal, and a more symmetrical steam distribution can be produced by making the inside lap greater and the outside lap less on the side which leads to the front end of the cylinder. On the other hand, an unsymmetrical distribution may be desirable, as in a vertical engine, where the weight of the piston assists the steam during the down-stroke and resists it during the up-stroke, and this may be secured by a suitable inequality in the laps.

150. By varying the ratio of the laps o and i to the travel of the valve, we produce effects on the steam distribution which are readily traced in the oval diagram of fig. 65 or in the other figures. Reduction of travel (which is equivalent to increase of both o and i) gives later preadmission, earlier cut-off, later release, and earlier compression; the ratios of expansion and of compression are both increased. The effect of a change in the angular advance is more easily seen by reference to Zeuner's diagram, which shows that to increase θ accelerates all the events and causes a slight increase in the ratio of expansion.

151. In designing a slide-valve the breadth of the steam ports in the direction of the valve's motion is determined with reference to the volume of the exhaust steam to be discharged in a given time, the area of the ports being generally such that the mean velocity of the steam during discharge is less than 100 feet per second. The travel is made great enough to keep the cylinder port fully open during the greater part of the exhaust; for this purpose it is 2½ or 3 times the breadth of the steam port. To facilitate the exit of steam the inside lap is always small, and is often wanting or even *negative*. During admission the steam port is rarely quite uncovered, especially if the outside lap is large and the travel moderate. Large travel has the advantage of giving freer ingress and egress of steam, with more sharply-defined cut-off, compression, and release, but this advantage is secured at the cost of more work spent in moving the valve and more wear of the faces. To lessen the necessary travel without reducing the area of steam ports, double- and even treble-ported valves are often used. An example of a double-ported valve is shown in fig. 85. Fig. 69 shows the Trick valve, an ingenious device for the same purpose.

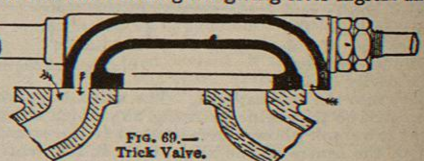
FIG. 69.—Trick Valve.

152. The eccentric must stand in advance of the crank by the angle $90° + \theta$, as in fig. 70, where CK is the crank, and CE the corresponding position of the eccentric when the engine is running in the direction of the arrow a. To set the engine in gear to run in the opposite direction (b) it is only necessary to shift the eccentric into the position CE′, when it will still be $90° + \theta$ in advance of the crank. In the older engines this reversal was effected by temporarily disengaging the eccentric-rod from the valve-rod, working the valve by hand until the crank turned back through an angle equal to ECE′; the eccentric meanwhile remaining at rest, and then re-engaging the gear. The eccentric sheave, instead of being keyed to the shaft, was driven by a stop fixed to the shaft, which abutted on one or other of two shoulders projecting from the sheave. In some modern forms of reversing gear means are provided for turning the eccentric round on the shaft, but the arrangement known as the link-motion is now the most usual gear in locomotive, marine, winding, and other engines which require to be often and easily reversed. Reversal of motion with slide valve.

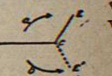
Fig. 70.

153. In the link-motion two eccentrics are used, and the ends of their rods are connected by a link. In Stephenson's link-motion—the earliest and still the most usual form—the link is a slotted bar or pair of bars curved to the same radius as the eccentric rods (fig. 71), and capable of being shifted up or down by a suspension rod. Link-motion. Stephenson's.

The valve-rod ends in a block which slides within the link, and when the link is placed so that this block is nearly in line with the forward eccentric rod (R, fig. 71) the valve moves in nearly the same way as if it were driven directly by a single eccentric. This is the position of "full forward gear." In "full backward gear," on the other hand, the link is pulled up until the block is in nearly a line with the backward eccentric rod R′. The link-motion thus gives a ready means of reversing the engine,—but it does more than this. By setting the link in an intermediate position the valve receives a motion nearly the same as that which would be given by an eccentric of shorter radius and of greater angular advance, and the effect is to give a distribution of steam in which the cut-off is earlier than in full gear, and the expansion and compression are greater. In mid gear the steam distribution is such that scarcely any work is done in the cylinder. The movement of the link is effected by a hand lever, or by a screw, or (in large engines) by an auxiliary steam-engine. A usual arrangement of hand lever, sketched in fig. 71, has given rise to the phrase "notching up," to describe the setting of the link to give a greater degree of expansion.

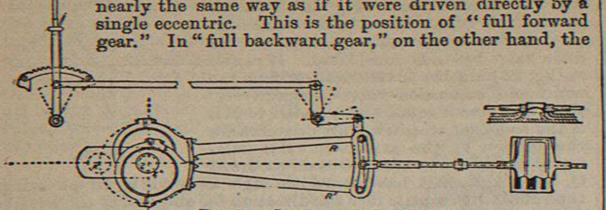
FIG. 71.—Stephenson's Link-Motion.

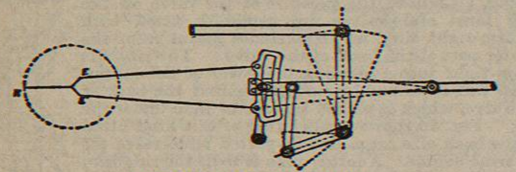
FIG. 72.—Gooch's Link-Motion.

Gooch's link-motion. 154. In Gooch's link-motion (fig. 72) the link is not moved up in shifting from forward to backward gear, but a radius rod between the valve-rod and the link (which is curved to suit this radius rod) is raised or lowered—a plan which has the advantage that the lead is the same in all gears. In Allan's motion (fig. 73) the change of gear is effected partly by shifting the link and partly by shifting a radius rod, and the link is straight.

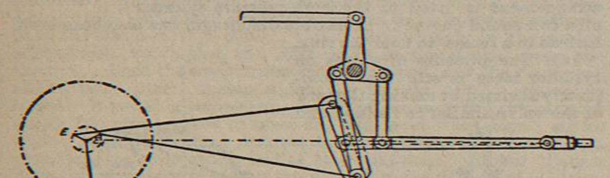
FIG. 73.—Allan's Link-Motion.

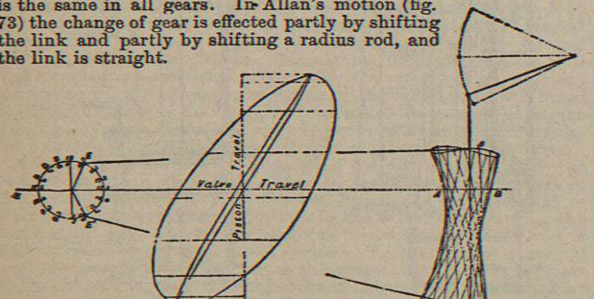

Fig 74.

155. The movement of a valve driven by a link-motion may be very fully and exactly analysed by drawing with the aid of a template the positions of the centre line of the link corresponding to a number of successive positions of the crank. Thus, in fig. 74, two circular arcs passing through e and e' are drawn with E and E′ as centres and the eccentric rods are radii. These are loci of two known points of the link, and a third locus is the circle a in which the point of suspension must lie. By placing on the paper a template of the link, with these three points marked on it, the position of the link is readily found, and by repeating the process for other positions of the eccentrics a diagram of positions (fig. 74) is drawn for the assigned state of the gear. A line AB drawn across this diagram in the path of the valve's travel determines the displacements of the valve, and enables the oval diagram to be drawn (as in fig. 65), which is shown to a larger scale in another part of fig. 74. The example refers to Stephenson's link-motion in nearly full forward gear; with obvious modification the same method may be used in the analysis of Gooch's or Allan's motion. The same diagram determines the amount of slotting or sliding motion of the block in the link. In a well-designed gear this sliding is reduced to a minimum for that position of the gear in which the engine runs most usually. In marine engines the suspension-rod is generally connected to the link at the end of the link next the forward eccentric, to reduce this sliding when the engine is in forward gear. A less laborious, but less accurate, solution of link-motion problems is reached by the use of what is called the equivalent eccentric—an imaginary eccentric, which would give the valve nearly the same motion as it gets from the joint action of the actual eccentrics. The following rule for finding the equivalent eccentric, in any state of gear, is due to Mr M'Farlane Gray:—

Connect the eccentric centres E and E′ (fig. 75) by a circular arc whose radius $= \dfrac{EE' \times \text{length of eccentric rod}}{2 \times ee'}$

Then, if the block is at any point B, take EF such that EF : EE′ :: eB : ee'. CF then represents the equivalent eccentric both in radius and in angular position. If the rods of the link-motion are crossed instead of open,—an arrangement seldom used,—the arc EFE′ is to be drawn convex towards C.

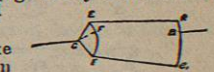
Fig. 75

156. Many forms of gear for reversing and for varying expansion have been devised with the object of escaping the use of two eccentrics, and of obtaining a more perfect distribution of steam than the link-motion can often be made to give. Hackworth's gear, the parent of several others, has a single eccentric E (fig. 76) opposite the crank, with an eccentric-rod EQ, whose mean position is perpendicular to the travel of the valve. The rod ends in a block Q, which slides on a fixed inclined guide-bar or link, and the valve-rod receives its motion through a connecting rod from an intermediate point P of the eccentric-rod, the locus of which is an ellipse. To reverse the gear the guide-bar is tilted over to the position shown by the dotted lines, and intermediate inclinations give various degrees of expansion without altering the lead. The steam distribution is excellent, and the cut-off is sharper than in the usual link-motion, but an objection to the gear is the wear of the sliding-block and guide. In Bremme's or Marshall's form this objection is obviated with some loss of symmetry in the valve's motion by constraining the motion of the point Q, not by a sliding-guide, but a suspension-link, which makes the path of Q a circular arc instead of a straight line; to reverse the gear the centre of suspension R of this link is thrown over to the position R′ (fig. 77). In the example sketched P is beyond Q, but P may be between Q and the crank (as in fig. 76), in which case the eccentric is set at 180° from the crank. This gear has been applied in a number of marine engines. In Joy's gear, which is extensively used in locomotives, no eccentric is required; and the rod corresponding to the eccentric rod in Hackworth's gear receives its motion from a point in the connecting rod by the linkage shown in fig. 78, and is either suspended, as in Marshall's form, by a rod whose suspension centre R is thrown over to reverse the motion, or constrained, as in Hackworth's, by a slot guide whose inclination is reversed. Fig. 79 shows Joy's gear as

Other reversing gears. Hackworth's. Bremme or Marshall's.

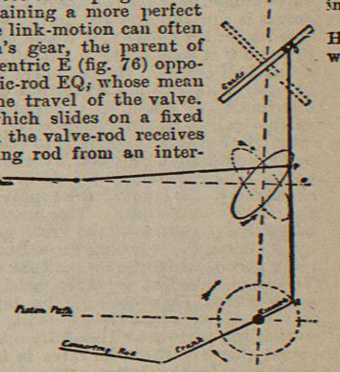

FIG. 76.—Hackworth's Valve-Gear.

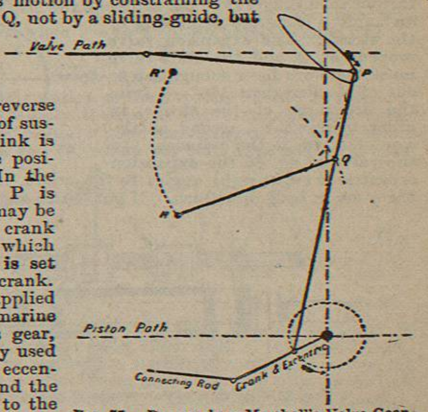

FIG. 77.—Bremme's or Marshall's Valve-Gear.

applied to a locomotive. A slot-guide E is used, and it is curved to allow for the obliquity of the valve connecting-rod AE. C is the crank-pin, B the piston path, and D a fixed centre. The reversing gears of Walschaert, Brown, and Kitson also dispense with eccentrics, and are closely related to the invention of Hackworth.[1] A

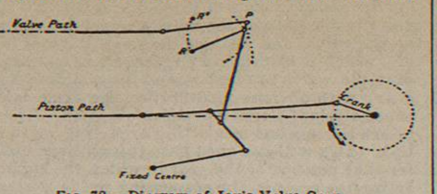

FIG. 78.—Diagram of Joy's Valve-Gear.

method of reversing with a common slide-valve, which is used in steam steering engines[2] and some others, is to supply steam to

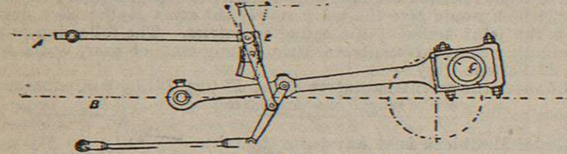
FIG. 79.—Joy's Gear as applied to a Locomotive.

what was (before reversal) the exhaust side of the valve and connect the exhaust to what was the steam side. This is done by means of a separate reversing valve through which the steam and exhaust pipes pass.

Separate expansion valves. 157. When the distribution of steam is effected by the slide-valve alone the arc of the crank's motion during which compression occurs is equal to the arc during which expansion occurs, and for this reason the slide-valve would give an excessive amount of compression if it were made to cut off the supply of steam earlier than about half-stroke. Hence, where an early cut-off is wanted it is necessary either to use an entirely different means of regulating the distribution of steam, or to supplement the slide-valve by another valve,—called an expansion-valve, usually driven by a separate eccentric,—whose function is to effect the cut-off, the other events being determined as usual by the slide-valve. Such expansion-valves belong generally to one or other of two types. In one the expansion-valve cuts off the supply of steam to the chest in which the main valve works. This may be done by a disk or double-beat valve (§ 163), as in the Proëll gear mentioned in § 175 below, or by a slide-valve working on a fixed seat (furnished with one or more ports), which forms the back or side of the main valve-chest. Valves of this last type are usually made in the "gridiron" or many-ported form to combine large steam-opening with small travel. Expansion-valves working in a fixed seat may be arranged so that the ports are either fully open (fig. 80) or closed (fig. 81) when the valve is in its middle position. In the latter case the expansion-valve eccentric is set in line with or opposite to the crank, if the engine is to run in either direction with the same grade of expansion. Cut-off then occurs at P, fig. 82, when the shaft has turned through an angle ϕ from the beginning of the stroke. The expansion valve reopens at Q, and the slide-valve must therefore have enough lap to cut off earlier than $180° - \phi$ from the beginning of the stroke, in order to prevent a second admission of steam to the cylinder. In the valve of fig. 80 the expansion eccentric is set at right angles to the crank, if the action is to be the same in both directions. If not, these angles may be deviated

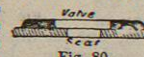

Fig. 80.

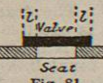

Fig. 81.

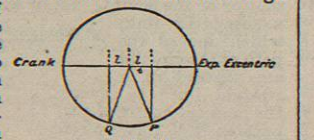

Fig. 82.

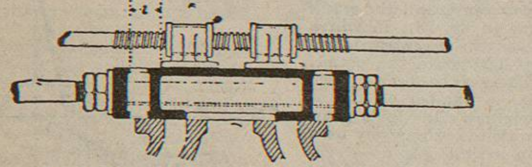
FIG. 83.—Expansion-Valve on back of Main Slide-Valve.

from, and in this way a more rapid travel at the instant of cut-off may be secured for one direction of running.

[1] Reversing gears of this type are generally termed radial gears. A discussion of Mr Joy's and other arrangements will be found in *Proc. Inst. Mech. Eng.* 1880. Mr Kirk, Mr Bryce-Douglas, and others have designed forms which more or less resemble those mentioned in the text.

[2] *Proc. Inst. Mech. Eng.*, 1867.

158. The other and much commoner type of expansion-valve is one sliding on the back of the main slide-valve, which is provided with through ports which the expansion-valve opens and closes. Fig. 83 shows one form of this type. Here the resultant relative motion of the expansion-valve and main-valve has to be considered. If r_a and r_b (fig. 84) are the eccentrics working the main and expansion valves respectively, then CR drawn equal and parallel to ME is the *resultant* eccentric which determines the motion of the expansion-valve relatively to the main-valve. Cut-off occurs at Q, when the shaft has turned through an angle ϕ, which brings the resultant eccentric into the direction CQ and makes the relative displacement of the two valves equal to the distance l. Another form of this valve (corresponding to fig. 81) cuts off steam at the inside edges of the expansion-slides.

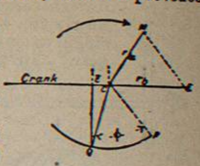

Fig. 84.

159. Expansion-valves furnish a convenient means of *varying* the expansion, which may be done by altering their lap, travel, or angular advance. Alteration of lap, or rather of the distance l in the figures, is often effected by having the expansion-valve in two parts (as in fig. 83) and holding them on one rod by right- and left-handed screws respectively; by turning the valve-rod the parts are made to approach or recede from each other. In large valves the adjustment is more conveniently made by varying the travel of the valve, which is done by connecting it to its eccentric through a link which serves as a lever of variable length. Variation of cut-off.

160. To relieve the pressure of the valve on the seat, large slide-valves are generally fitted with a steam-tight ring, which excludes steam from the greater part of the back of the valve. The ring fits steam-tight into a recess in the cover of the steam-chest, and is pressed by springs against the back of the valve, which is planed smooth to slide under the ring. Fig. 85 shows a relief ring of this kind fitted on the back of a large double-ported slide-valve for a marine engine. Another plan is to fit the ring into a recess on the back of the valve, and let it slide on the inside of the steam-chest cover. Steam is thus excluded from the space within the ring, any steam that leaks in being allowed to escape to the condenser (or to the intermediate receiver when the arrangement is fitted to the high-pressure cylinder of a compound engine). A flexible diaphragm has also been used, instead of a recess, to hold the ring. Relief rings.

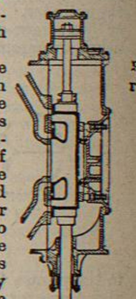
Fig. 85.

161. The pressure of valves on cylinder faces is still more completely obviated by making the back of the valve similar to its face, and Piston slide-valve.

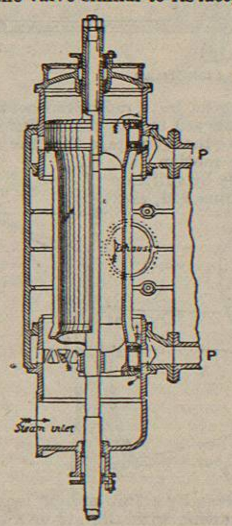

FIG. 86.—Piston Slide-Valve.

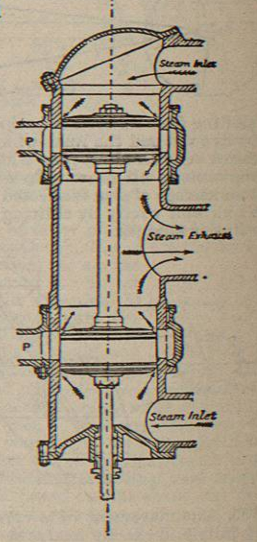

FIG. 87.—Piston Slide-Valve.

causing the back to slide in contact with the valve-chest cover, which has recesses corresponding to the cylinder ports. This arrangement is most perfectly carried out in the piston slide-valves now very largely used in the high-pressure cylinders of marine engines. The piston slide-valve may be described as a slide-valve

in which the valve face is curved to form a complete cylinder, round whose whole circumference the ports extend. The pistons are packed like ordinary cylinder pistons by metallic rings, and the ports are crossed here and there by diagonal bars to keep the rings from springing out as the valve moves over them. Figs. 86 and 87 show two forms of piston valve designed by Mr Kirk for the supply of high-pressure steam to large marine engines. P, P are the cylinder ports in each.

Fig. 85 illustrates an arrangement common in all heavy slide-valves whose travel is vertical—the *balance-piston*, which is pressed up by steam on its lower side and so equilibrates the weight of the valve, valve-rod, and connected parts of the mechanism.

162. The slide-valve sometimes takes the form of a disk revolving or oscillating on a fixed seat, and sometimes of a rocking cylinder (fig. 88). This last kind of sliding motion is very usual in stationary engines fitted with the Corliss gear, which will be described in the next chapter, in which case four distinct rocking slides are commonly employed to effect the steam distribution, one giving admission and one giving exhaust at each end of the cylinder (see fig. 127).


FIG. 88.—Rocking Slide-Valve.

Double-beat valve. 163. In many stationary engines *lift* or *disk* valves are used, worked by tappets, cams, or eccentrics. Lift valves are generally of the Cornish or double-beat type (fig. 89), in which equilibrium is secured by the use of two conical faces which open or close together. In Cornish pumping engines, which retain the single action of Watt's early engine, three double-beat valves are used, as steam-valve, equilibrium-valve, and exhaust-valve respectively. These are closed by tappets on a rod moving with the beam, but are opened by means of a device called a cataract, which acts as follows.

Cornish cataract. The cataract is a small pump with a weighted plunger, discharging fluid through a stop-cock which can be adjusted by hand when it is desired to alter the speed of the engine. The weighted plunger is raised by a rod from the beam, but is free in its descent, so that it comes down at a rate depending on the extent to which the stop-cock is opened. When it comes down a certain way it opens the steam and exhaust valves, by liberating catches which hold them closed; the "out-door" stroke then begins and admission continues until the steam-valve is closed: this is done directly by the motion of the beam, which also, at a later point in the stroke, closes the exhaust. Then the equilibrium-valve is opened, and the "in-door" stroke takes place, during which the plunger of the cataract is raised. When it is completed, the piston pauses until the cataract causes the steam-valve to open and the next "out-door" stroke begins. By applying a cataract to the equilibrium-valve also, a pause is introduced at the end of the "out-door" stroke. Pauses have the advantage of giving the pump time to fill and of allowing the pump-valves to settle in their seats without shock.

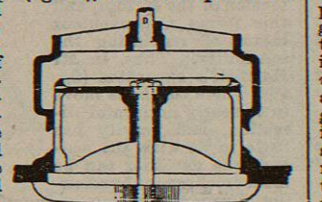
FIG. 89.—Double-Beat Lift-Valve.

IX. GOVERNING.

Methods of regulating. 164. To make an engine run steadily an almost continuous process of adjustment must go on, by which the amount of work done by the steam in the cylinder is adapted to the amount of external work demanded of the engine. Even in cases where the demand for work is sensibly uniform, fluctuations in boiler-pressure still make regulation necessary. Generally the process of government aims at regularity of speed; occasionally, however, it is some other condition of running that is maintained constant, as when an engine driving a dynamo-electric machine is governed by an electric regulator to give a constant difference of potential between the brushes.

The ordinary methods of regulating are either (*a*) to alter the pressure at which steam is admitted by opening or closing more or less a throttle-valve between the boiler and the engine, or (*b*) to alter the volume of steam admitted to the cylinder by varying the point of cut-off. The former plan was introduced by Watt and is still common, especially in small engines. From the point of view of heat economy it is wasteful, since the process of throttling is essentially irreversible, but this objection is to some extent lessened by the fact that the wire-drawing of steam dries or superheats it, and consequently reduces the condensation which it suffers on coming into contact with the chilled cylinder walls. On the other hand, to hasten the cut-off involves a gain rather than a loss of efficiency unless the ratio of expansion is already very great. The second plan of regulating is much to be preferred, especially when the engine is subject to large variations of load, and is very generally followed in stationary engines of the larger types.

165. Within certain limits regulation by either plan can be effected by hand, but for the finer adjustment of speed some form of automatic governor is necessary. Speed governors are commonly of the *centrifugal* type: a pair of masses revolving about a spindle which is driven by the engine are kept from flying out by a certain controlling force. When an increase of speed occurs this controlling force is no longer able to keep the masses revolving in their former path; they move out until the controlling force is sufficiently increased, and in moving out they act on the regulator of the engine, which may be a throttle-valve or some form of automatic expansion gear. In the conical pendulum governor of Watt (fig. 90) the revolving masses are balls attached to a vertical spindle by links, and the controlling force is furnished by the weight of the balls, which, in receding from the spindle, are obliged to rise. When the speed exceeds or falls short of its normal value they move out or in, and so raise or lower a collar C which is in connexion by a lever with the throttle-valve. The suspension-links may be hung from a cross-bar (figs. 94, 95) instead of being pivoted in the axis of the spindle.

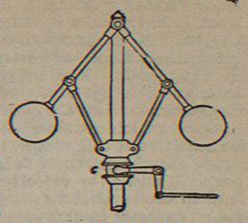

FIG. 90.—Watt's Governor.

Loaded governor. 166. In a modified form of Watt's governor, known as Porter's, or the loaded governor, a supplementary controlling force is given by placing a weight on the sliding collar (fig. 91). This is equivalent to increasing the *weight* of the balls without altering their *mass*. In other governors the controlling force is wholly or partly produced by springs. Fig. 92 shows a governor by Messrs Tangye in which the balls are controlled partly by their own weight and partly by a spring, the tension of which is regulated by turning the cap A.

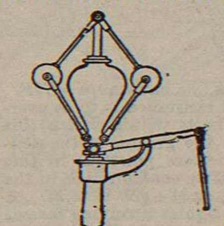
FIG. 91.—Loaded Governor.

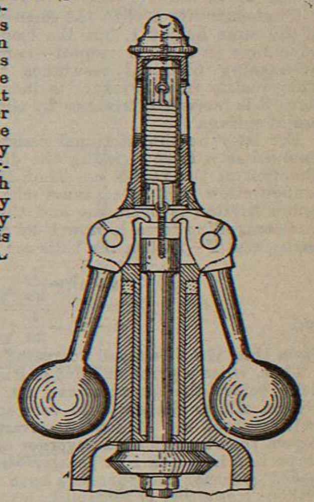
FIG. 92.—Spring Governor (Tangye).

Equilibrium governor. 167. In whatever way the revolving masses are controlled, the controlling force may be treated as a force F acting on each ball in the direction of the radius towards the axis of revolution. Then, if M be the mass of the ball, n the number of revolutions per second, and r the radius of the ball's path, the governor will revolve in equilibrium when $F = 4\pi^2 n^2 r M$ (in absolute units), or

$$n = \frac{1}{2\pi}\sqrt{\frac{F}{Mr}}$$

In order that the configuration of the governor should be stable, F must increase more rapidly than r, as the balls move outwards. In the simple conical pendulum governor, any of the three forms shown in figs. 93, 94, and 95, where the balls have no load to raise

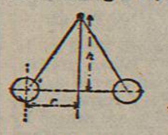
Fig. 93.

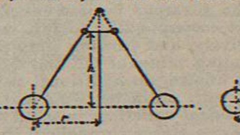
Fig. 94.


Fig. 95.


Fig. 96.

but their own weight, the controlling force F is the resultant of T, the tension in the link, and Mg, the weight of the ball (fig. 96). Let the height of the pendulum, that is, the distance above the plane of the balls of the point where the suspending-link, or the link produced, cuts the axis, be called h. Then $F : Mg :: r : h$. Hence

$$F = \frac{Mgr}{h}, \text{ and } n = \frac{1}{2\pi}\sqrt{\frac{g}{h}}$$

Any change of n tends to produce a change of h, and, if the governor itself and the regulating mechanism attached to it were free from friction, only one position of the governor would be possible for any one value of n. It is obvious that neither this governor nor any other *stable* governor maintains a strictly constant speed in the engine which it controls. If the boiler pressure or the demand for work is changed, a certain amount of permanent displacement of the balls is necessary to alter the steam supply, and the balls can retain their displaced position only by virtue of a permanent change in the speed. The maximum range of speed depends on that amount of change of n which suffices to alter the configuration of the governor from the position which gives no steam-supply to the position which gives full steam-supply; and the governor is said to be sensitive if this range is a small fraction of n.

168. If the governor is loaded, let M′ be the amount of the load per ball, and q the velocity ratio of the vertical movement of the load to the vertical movement of the ball. Then $qM'g$ is the equivalent increase in the weight of each ball. The effect of the load is to increase the controlling force F from Mgr/h to $(M+qM')gr/h$, and the speed at which the governor must now turn, to maintain any assigned height h, is

$$n=\frac{1}{2\pi}\sqrt{\frac{(M+qM')g}{Mh}}$$

The speed of the loaded governor must therefore be greater than that of an unloaded governor of the same height in the ratio $\sqrt{(M+qM')}$ to $\sqrt{M}$.

The sensibility is then the same as that of an unloaded governor of the same height h, but the loaded governor has an important advantage in another respect—namely, its *power* or capability of overcoming frictional resistance to a change of configuration. This quality in a governor is increased whenever the controlling force F is increased, whether by the addition of a load or by the use of springs.

For let f be the frictional resistance to be overcome per ball, resolved as a force resisting the displacement of each ball in the direction of the radius r. Then if n be the speed normal to any configuration this speed must change by a certain amount Δn before friction is overcome and the balls begin to be displaced. The controlling force is now $F+f$ when the balls are moving outwards, and $F-f$ when the balls are moving inwards. Hence

$$n+\Delta n=\frac{1}{2\pi}\sqrt{\frac{F+f}{Mr}},$$

and

$$n-\Delta n=\frac{1}{2\pi}\sqrt{\frac{F-f}{Mr}}.$$

From this, if Δn be small compared with n, we have $\Delta n/n=f/2F$.

Thus, when a given amount of frictional resistance is to be overcome before the governor can act, the limits within which this friction allows the speed to vary are less the greater is the controlling force F. A loaded governor is more powerful in this respect than an unloaded governor of the same configuration in the proportion in which F is greater—namely, as $M+qM'$ is to M. A loaded governor may therefore have much lighter revolving masses without loss either of sensibility or of power.

General solution.-graphic method. 169. The same results are applicable to governors in which the controlling force is supplied by springs as well as by gravity, or by springs alone. To find the configuration which the governor will assume at any particular speed, or the speed corresponding to a particular configuration, it is only necessary to determine the whole controlling force F per ball acting along the radius towards the axis for various values of r. Let a curve ab (fig. 97) be drawn showing the relation of F to r. At any assigned value of r set up an ordinate $QC=4\pi^2n^2rM$. Join OC. The point c, in which OC cuts the curve, determines the value of r at which the balls will revolve at the assigned speed n. Or, if that is given, and the value of n is to be found, the line Oc produced will determine C, and then $n^2=QC/4\pi^2rM$. The sensibility of the governor is determined by taking points a and b corresponding to full steam and no steam respectively, and drawing lines through them to determine the corresponding values of QA and QB.[1] When the frictional resistance f is known, an additional pair of curves drawn above and below ab, with ordinates $F+f$ and $F-f$ respectively, serve to show the additional variations in speed which are caused by friction. The governor is stable throughout its whole range when the curve ab has a steeper gradient than any line drawn from O to meet it.

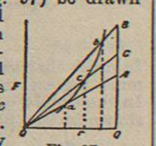

Fig. 97.

170. By § 167 it is evident that, if, when the balls are displaced, the controlling force F changes proportionally to the radius r, the speed is constant. In other words, the equilibrium of the governor is then neutral; it can revolve in equilibrium at one, and only at one, speed. At this speed it assumes, indifferently, any one of its possible configurations. The slightest variation of speed drives it to the extremity of its range; hence its sensibility is indefinitely great. Such a governor is called *isochronous*. A gravity governor is isochronous when h is constant for all positions of the balls (since $n\propto\sqrt{g/h}$). This will be the case if the balls are constrained to move in a parabolic path (fig. 98), it being a property of the parabola that the subnormal QM, which is h, is constant. A useful approximation to the same condition, through a limited range, is secured in Farcot's governor by the device of hanging the balls by crossed links from the distant ends of a T piece (fig. 95). If each centre of suspension were at the centre of curvature of a parabolic arc which coincided with the actual circular locus of the balls at the position of normal speed, the governor would be sensibly isochronous at that speed; by taking the centres of suspension rather nearer the axis, a suitable margin of stability is secured, but the governor is still nearly enough isochronous to be exceedingly sensitive.[2] Where springs furnish the controlling force, an approach to isochronism can be secured by adjusting the initial tension of the springs, and this forms a convenient means of regulating the sensibility. Thus, in Mr Hartnell's apparatus (fig. 99), where the balls move in a nearly horizontal direction, and gravity has little to do with the control, the governor can be made isochronous by screwing down the spring, so that the initial force exerted by the spring is to its increase by displacement of the balls as the initial radius of the balls' path is to the increase of radius by displacement. When the initial force is increased beyond this the governor becomes unstable. In fig. 97 the condition of isochronism is secured when the line ab coincides with a straight line through O.

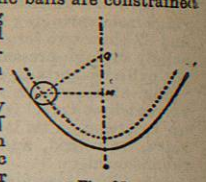

Fig. 98.

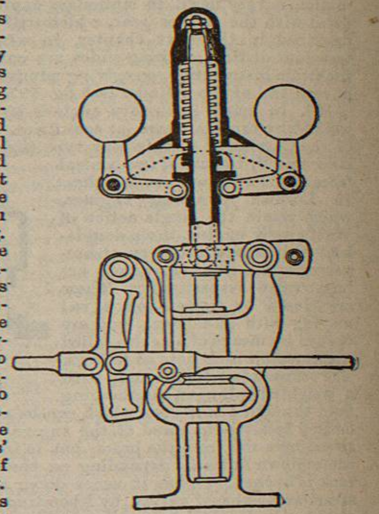
Fig. 99.—Hartnell's Governor.

171. In practice no governor can be absolutely isochronous. It is indispensable to leave a small margin of stability for the sake of preventing violent change in the supply of steam, especially when there is much frictional resistance to be overcome by the governor, or where the influence of the governor takes much time to be felt by the engine. An over-sensitive governor is liable to fall into a state of oscillation called *hunting*. When an alteration of speed begins to be felt, however readily the governor alters its form, the engine's response is more or less delayed. If the governor acts by closing a throttle-valve, the engine has still a capacious valve-chest on which to draw for steam. If it acts by changing the cut-off, its opportunity is passed if the cut-off has already occurred, and the control only begins with the next stroke. This lagging of effect is specially felt in compound engines, where that portion of the steam which is already in the engine continues to do its work for nearly a whole revolution after passing beyond the governor's control. The result of this storage of energy in an engine whose governor is too nearly isochronous is that, whenever the demand for power suddenly falls, the speed rises so much as to force the governor into a position of over-control, such that the supply of steam is no longer adequate to meet even the reduced demand for power. Then the speed slackens, and the same kind of excessive regulation is repeated in the opposite direction. A state of forced oscillation is consequently set up. The effect is aggravated by the momentum which the governor balls acquire in being displaced, and also, to a very great degree, by the friction of the governor and the regulating mechanism. Hunting is to be avoided by giving the governor a fair degree of stability, by reducing as far as possible the static frictional resistances, and by introducing a *viscous* resistance to the displacement of the governor, which prevents the displacement from occurring too suddenly, without affecting the ultimate position of equilibrium. For this purpose many governors are furnished with a *dash-pot*, which is an hydraulic or pneumatic brake, consisting of a piston connected to the governor, working loosely in a cylinder which is filled with oil or with air. Hunting

172. In some high-speed engines the governor balls or blocks revolve in a vertical plane, about a horizontal axis, and the control is given wholly by springs. An example is shown in fig. 100, which is the governor of the Armington and Sims engine referred to in § 197 below. Another example is furnished by the governor of Brotherhood's engine (§ 203, fig. 128).

[1] See a paper by Mr W. Hartnell, "On Governing Engines by Regulating the Expansion," *Proc. Inst. Mech. Eng.*, 1882.

[2] See also a paper by Mr J. Head *Proc. Inst. Mech. Eng.*, 1871.

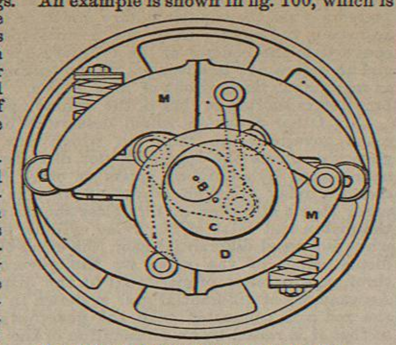

Fig. 100.—Governor of Armington & Sims Engine.

173. The throttle-valve, as introduced by Watt, was originally a disk turning on a transverse axis across the centre of the steam-pipe. It is now usually a double-beat valve (fig. 89) or a piston-valve. When regulation is effected by varying the cut-off, and an expansion-valve of the slide-valve type is used, the governor generally acts by changing the travel of the valve. Fig. 99 illustrates a common mode of doing this, by giving the expansion-valve its motion from an eccentric-rod through a link, the throw of which is varied by the displacement of the governor balls. In fig. 100, the governor acts on the main slide-valve of the engine (there being no separate expansion-valve), and the displacement of the revolving masses M, M changes both the throw and the angular advance of the eccentric, thereby producing a change in the steam supply similar to that produced by "notching up" a link-motion. The eccentricity B is altered by the relative displacement of two parts C, D into which the eccentric sheave is divided. In other forms of automatic expansion-gear the lap of the valve is altered; in others the governor acts by shifting the expansion-valve eccentric round on its shaft, and so changing its angular advance.

174. In large stationary engines the most usual plan of automatically regulating the expansion is to employ some form of trip-gear, the earliest type of which was introduced in 1849 by G. H. Corliss of Providence, U.S. In the Corliss system the valves which admit steam are distinct from the exhaust-valves. The latter are opened and closed by a reciprocating piece which takes its motion from an eccentric. The former are opened by a reciprocating piece, but are closed by springing back when released by a trip- or trigger-action. The trip occurs earlier or later in the piston's stroke according to the position of the governor. The admission-valve is opened by the reciprocating piece with equal rapidity whether the cut-off is going to be early or late. It remains wide open during the admission, and then, when the trip-action comes into play, it closes suddenly. The indicator diagram of a Corliss engine consequently has a nearly horizontal admission-line and a sharply defined cut-off. Generally the valves of Corliss engines are cylindrical plates turning in hollow cylindrical seats which extend across the width of the cylinder. Often, however, the admission-valves are of the disk or double-beat type, and spring into their seats when the trip-gear acts. Many forms of Corliss gear have been invented by Corliss himself and by others. One of these, the Spencer Inglis[1] trip-gear, by Messrs Hick, Hargreaves, & Co., is shown in figs. 101 and 102. A wrist-plate A, which turns on a pin on the outside of the cylinder, receives a motion of oscillation from an eccentric. It opens the cylindrical rocking-valve B by pulling the link C, which consists of two parts, connected to each other by a pair of spring clips a, a. Between the clips there is a rocking-cam b, and as the link is pulled down this cam places itself more and more athwart the link, until at a certain point it forces the clips open. Then the upper part of the link springs back and allows the valve B to close by the action of a spring in the dash-pot D. When the wrist-plate makes its return stroke the clips re-engage the upper portion of the link C, and things are ready for the next stroke. The rocking-cam b has its position controlled by the governor through the link E in such a way that when the speed of the engine increases it stands more athwart the link C, and therefore causes the clips to be released at an earlier point in the stroke. A precisely similar arrangement governs the admission of steam to the other end of the cylinder. The exhaust-valves are situated on the

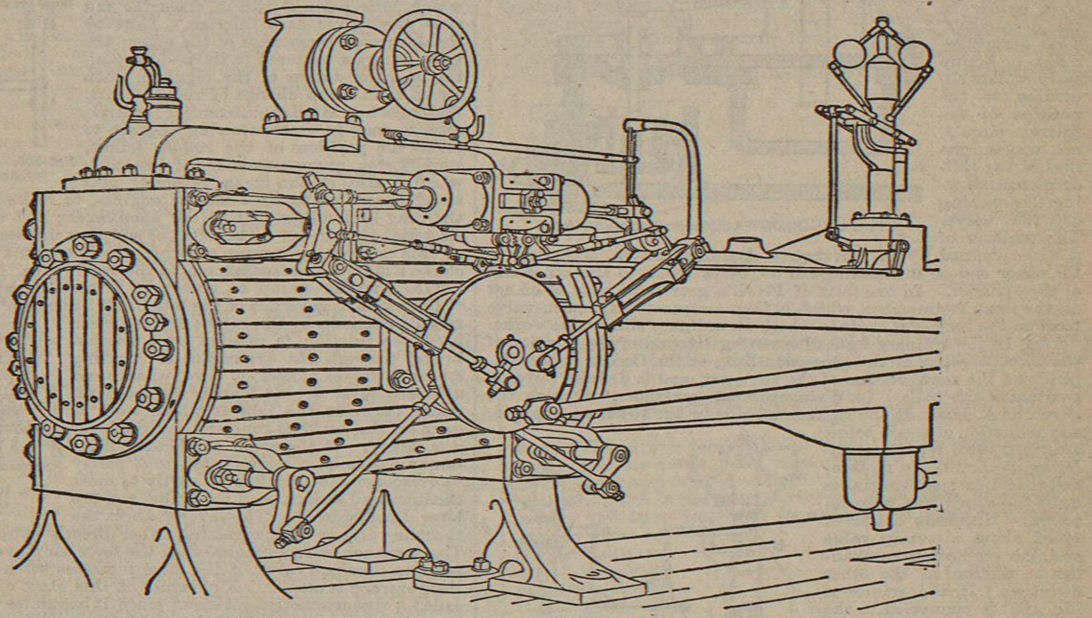
Fig. 101.—Corliss Engine, with Spencer Inglis Trip-Gear.

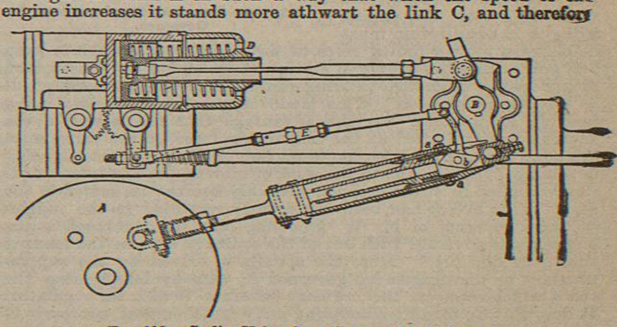

Fig. 102.—Corliss Valve-Gear, Spencer Inglis form.

[1] *Proc. Inst. Mech. Eng.*, 1868.