in which the block in its extreme positions is exactly opposite the ends of the eccentric rods. Sometimes the link is suspended from the middle, and sometimes from one or other of the ends. Sometimes the eccentric rods are worked in the open position as shown in fig. 94, and sometimes the rods are crossed. All of these details effect variations in the distribution of the steam. The other link motions best known in this country are Gooch's, in which the link is not shifted up or down by the reversing gear, but the block U slides within a link suspended to a fixed point; and Allen's, in which both link and block are made to shift in opposite directions.

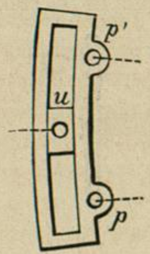

Fig. 97.

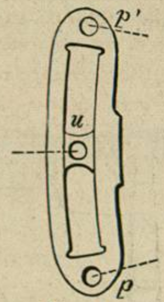

Fig. 97A.

It will be readily perceived from the description that the link motion is a very efficient method of reversing an engine, and a convenient means of regulating the rate of expansion. Its performance of the latter function is, however, attended with certain drawbacks. In addition to altering the point at which the valve cuts off the steam, each new position of the link also alters the lead, and the periods of release and compression of the exhaust steam. As, moreover, the travel of the valve is altered while the lap remains constant, the extent to which the steam port is opened is very seriously curtailed as the sliding block approaches its central position. These peculiarities, coupled with the fact that all simple slide valves open and close the ports with a comparatively slow motion, thus causing the corners of the indicator diagrams to assume a rounded instead of a sharp appearance, have led to the adoption of modifications of the above, or other methods of effecting the dis-

tribution of the steam when economy of fuel is much sought after.

Meyer's valve gear.—The method in most common use to obviate the disadvantages belonging to the common D slide valve and link motion is the adoption of a second valve to control the cut-off, while the ordinary valve, or a modification of it, regulates all the other points connected with the steam distribution. There are several varieties of these double valve motions in use, but the one selected for description, on account of its simplicity and efficiency, is that known as Meyer's valve gear, illustrated in fig. 98.

In fig. 98, A, A′, E are the two steam and the exhaust ports, which are worked by a valve B B, differing from an ordinary slide valve only in the fact that it is much longer, and that two steam passages *a a′*, for enabling the fresh steam to get to the ports, are made through the substance of the prolonged portions. This valve is driven in the usual manner either by one eccentric or, when the engine is required to reverse, by two eccentrics set opposite to each other in the ordinary way and connected by a link, which latter, however, is not used for altering the rate of expansion. The valve B B is called the distribution valve, and effects the admission, the release, and the closing of the exhaust precisely in the same way as an ordinary slide valve. The cut-off is effected by closing the passages *a a′* at the proper point in the stroke by means of the expansion valve C C′, which slides on the back of B B. The expansion valve is formed in two halves, which, by means of the threaded valve spindle D can be caused to approach or recede from each other, thus varying the point at which they close the passages *a a′* and cut off the steam. The expansion valve is usually driven by a fixed eccentric,

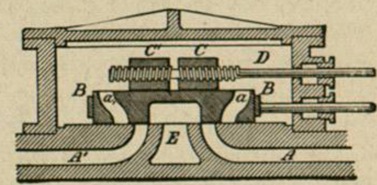

Fig. 98.

and means are provided exterior to the valve box for turning the spindle D by hand, so as to put the point of cut-off under the control of the driver, or, in some cases, of the governor.

It is evident that, with this gear, the lead and the points of admission, release, and compression are quite independent of the rate of expansion, and, when once fixed, are invariable. Moreover, as the expansion valve, when closing the passages *a a'*, is always moving in the opposite direction to the distribution valve, the cut-off can be effected much more rapidly than with the ordinary D slide. The proper method of proportioning the two valves, and of setting the eccentrics so as to provide for any desired range of expansion, will be explained in the theoretical portion of the chapter (see page 302).

Corliss valve motion.—Meyer's valve gear, though it possesses many advantages, does not remedy the evil of admitting the fresh steam to the cylinder through the passages which have been cooled down by the low temperature of the exhaust steam. Moreover, the extra power required to drive it, in consequence of the friction of two sliding surfaces is so considerable, that many eminent authorities doubt the advantage of using a comparatively complicated gear, the chief object of which is to prevent the compression curve increasing too rapidly with the rate of expansion. In order to provide against the evil of admitting the fresh steam through comparatively cold passages, and at the same time to diminish friction and retain the good points of the Meyer class of gearing, the Corliss valve motion was brought out, and is now very often employed, in some of its modifications, in non-compound expansive engines when fuel economy is a point of primary importance.

The leading peculiarities of the Corliss gear are that there are four steam ports and four valves, the two of the latter which control the steam admission and cut-off being driven by a special mechanism which ensures a very sharp closing of the steam ports when the cut-off takes place. The

two upper ports are intended solely for the admission of the steam, the remaining two serving for the exhaust. Thanks to this arrangement, the fresh steam on entering the cylinder does not come in contact with surfaces which have just been cooled down by the passage of the comparatively cold exhaust steam. The valves do not belong to the class of D slides, but are formed of portions of cylinders, A A, B B, fig. 99,

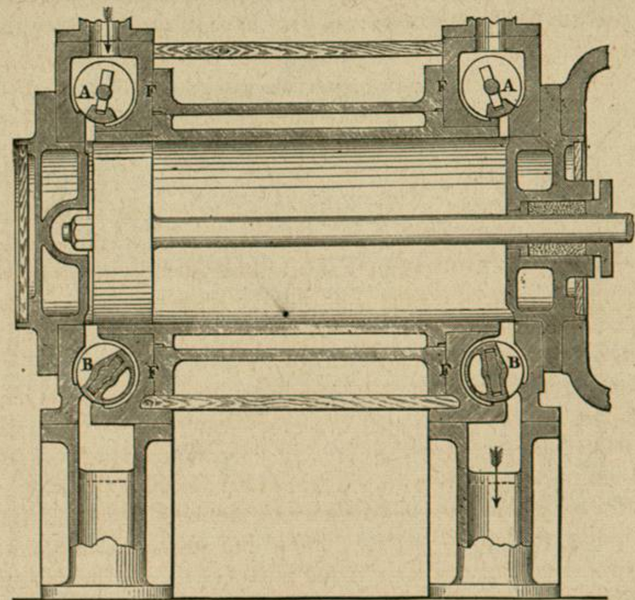

Fig. 99.

which oscillate on a cylindrical face. The valves are so arranged that the pressure of the steam forces them against their seats only when the port is closed. In all other positions there is no friction due to steam pressure, consequently the operation of these valves absorbs very little of the power of the engines. Fig. 99 is a section of a cylinder provided with Corliss gear. It shows very clearly the four steam passages, and the cylindrical valves with the peculiarities of seatings just explained.

Fig. 100 gives a general view of the mechanism by which the valves are worked. The disc L is caused to oscillate in an arc of a circle round its own centre by means of the eccentric and the eccentric rod F. To the disc are jointed four valve rods, G G and H H, which are also attached to the four valves. The two belonging to the lower valves B B are perfectly simple ; all they have to do is to open the

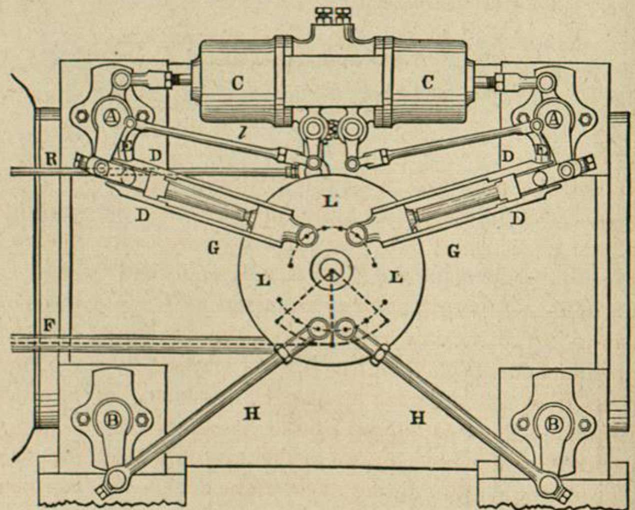

Fig. 100.

exhaust ports at the proper moment, and keep them open during the required fixed period of the stroke. The two upper rods are more complicated. Their function is, not only to open the steam ports at the proper moment, but also to keep them open during whatever period of the stroke is required by the rate of expansion, and then to liberate the valve so as to permit of its being closed sharply by an independent piece of mechanism which will be described hereafter. The upper rods are made in two independent pieces, one of which slides within the other. One of these pieces

is attached to the disc L, and the other to the rocking arm of the steam valves A A. By means of two side-clip springs D D, fig. 101, which are permanently fastened to the disc end of the rod, and which engage with the projecting shoulders *e e*, on the valve end of the rod, the two halves can at will be united into one rigid rod, or be disunited, so that the motion of the disc no longer controls that of the valve. At a certain

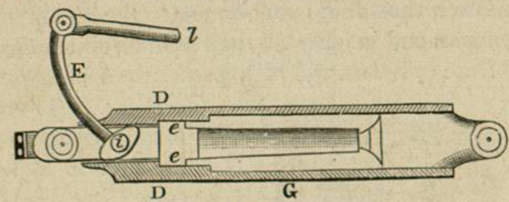

Fig. 101.

period in each stroke, dependent on the action of the governor, the clip springs D D are prised apart ; the valve rod ceases to act as a whole, the valve is consequently liberated from the control of the disc L, and is instantly closed by the action of the mechanism contained in C. The mechanism by which the clip springs D D are prised apart is shown on an enlarged scale in fig. 101 ; it consists of a toe lever centred on *i*, and having an arm E, the inclination of which is capable of being altered by the rod *l*, which in its turn is under the immediate control of the governor rod R, fig. 100. The toe lever rocks during each stroke of the valve rod, and the period that it reaches its extreme position, and prises open the clip springs, is determined by the inclination of the arm E, and consequently by the governor. Hence we see that in this type of valve gear the rate of expansion is controlled automatically. When the clip springs are opened the valve is closed in the following manner. The valve spindle A, fig. 100, is provided with a second arm which is attached to a piston or plunger contained in C. The back of this piston is always in communication with the condenser, while the front receives the atmospheric pressure ; and, consequently, whenever the valve is released from the action of the valve rod, the piston is sharply forced inwards by the atmospheric pressure and actuates the arm which closes the valve. There is also an air buffer, or dash pot, contained in C, which prevents the concussion, due to the very sharp action of the piston, which would otherwise ensue when the valve is closed.

Varieties of valves.—There are many varieties of valves besides the short D slides which have been described above. These latter, though in common use in the smaller types of land engines and in locomotives, would be quite inapplicable to the large cylinders and high pressures of modern marine engines. It will be readily understood that the friction to be overcome in driving the ordinary type of D slide valve is very considerable when high pressures are used, as is the case in locomotives. In fact, it frequently happens that each of the valves of a large locomotive is pressed against the surface on which it slides with a total net pressure of nine or ten tons, so that the mere driving of the valves probably absorbs some five or six per cent. of the total power exerted by the engine. This defect would be very much exaggerated if the D type of slide valve were applied without modification to marine engines.

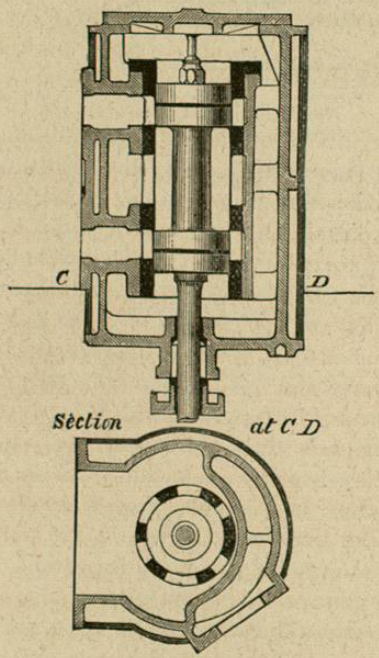

Fig. 102.

The types of valve generally made use of nowadays in marine engines are either the double ported slide valve or else what is known as the piston valve, an example of which is shown in fig. 102. Here it will be seen the valve consist of two pistons, one to each port, connected together by a common spindle. The pistons are provided with the ordinary spring ring packings to keep them steam-tight; they move up and down in the two cylindrical spaces shown, in which are cast the openings to the steam ports. The latter are not made continuously open as is the case with ordinary slide valves, but are cast with bars of metal as shown in the section, to prevent the packing rings of the pistons from springing out into the ports, and also in order to afford a continuous guide to the pistons. It will be seen that this type of valve is perfectly balanced, as far as the steam pressure is concerned. The only friction to be overcome in its motion is that due to the pressure of the spring packing rings. In the piston valve shown in fig. 102 the steam is admitted between the two pistons, the inner edges of which cut off the steam.

The double ported slide valve is an ingenious modification of the ordinary D slide already described. Examples are shown in section in the valves of the low-pressure cylinder, pages 456 and 458, and also in fig. 103. It will be noticed that the steam passages of these cylinders have each two ports or openings on the valve face. The steam is admitted to the outside ports in the usual way, over the outer edges of the valves, but the two inner ports get their steam from the passages cast in the body of the valve, and which are shown in section in the figures above referred to. The advantage of this arrangement is, that for a given movement of the valve twice the area of steam port is uncovered as with the ordinary single ported valve, and consequently for a given area of opening the travel of the valve may be greatly reduced.

In order to relieve the faces of the slide valves of marine engines of a portion of the pressure brought to bear on them by the action of the steam on their backs, it is usual to fit relief rings on to these latter, in order to cut off a portion of their area from the pressure of the steam. It will be readily understood that if the back of the slide valve were planed perfectly flat, and if it could be so arranged as to work in perfect contact with the inner face of the valve box cover, so that no steam could get between the two faces, then the valve would only be held against its seating by the unbalanced portion of the pressure acting on the lips of the valve

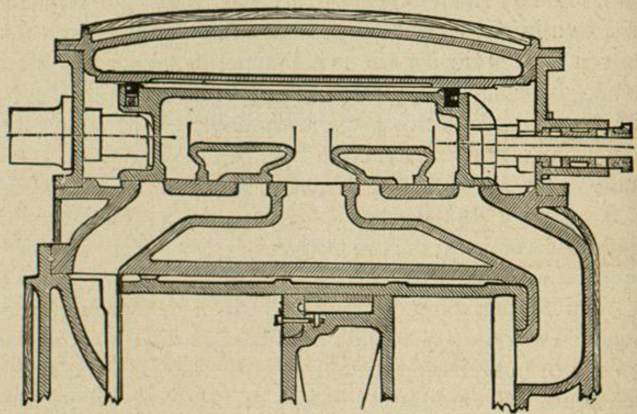

Fig. 103.

c c, fig. 89. Such an arrangement could not however be made to work satisfactorily in practice, because, in consequence of the different expansions under heat of the metals of the valve and the box, the former would either jamb or work loose. The relief ring is intended to get over this difficulty. It consists of a flat ring on the back of the slide valve, which is pressed outwards against the face of the valve box cover by means of a spring. The ring fits steam-tight into the back of the valve and works steam-tight against the face of the valve box, and thus excludes a large portion of the back of the valve from the direct pressure of the

steam. In case any steam should leak into the hollow space within the ring, the latter is generally placed in communication with the condenser. Fig. 103 is a section of a double ported valve fitted with a relief ring, which is shown in section in black.

Joy's valve gear.—There are other methods in common use for driving valves besides the eccentric system which has been described. It has constantly been an object with inventors to get rid of the complications of the two eccentrics, link, &c., required for an expansion gear. Several successful gears have been brought out, both in this country and the Continent, in which the valve is driven from some moving part of the engine. One of the best known of these

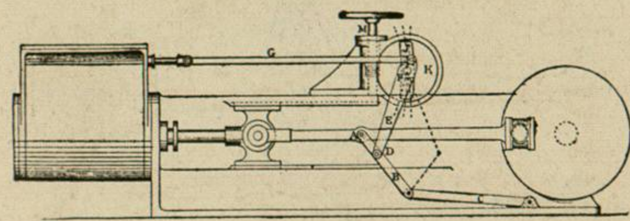

Fig. 104.

is the Joy valve gear, which has been largely used both for locomotives and marine engines. Fig. 104 illustrates a simple form of this gear applied to a horizontal stationary engine. A vibrating rod or link, B, is attached at one end to a point A, near the middle of the connecting rod; while the lower end is jointed to the radius rod C, which compels B to move in a vertical plane. To a point D in the link B is jointed the end of the long arm of a lever E F, of which the end of the small arm works the valve rod G, and the fulcrum F is attached to a block which slides in the curved slot J. This slot is formed in a disc, the centre of which is the position of the fulcrum F when the piston is at either end of its stroke. The radius of the slot is equal to the length of the valve rod G. The disc can be made to rotate through an arc by means of the worm and wheel shown. Thus the slot can be inclined to either side of the vertical. The slot allows the fulcrum of the lever to move up and down with the motion of the point A of the connecting rod. The forward or backward motion of the engine, and the rate of expansion, are controlled by inclining the slot to one or other side of the vertical, the central position corresponding with mid-gear. If the end D of the lever were attached direct to the connecting rod, the motion of the fulcrum F about the centre of the slot would not be symmetrical, and the result would be that the cut-off would

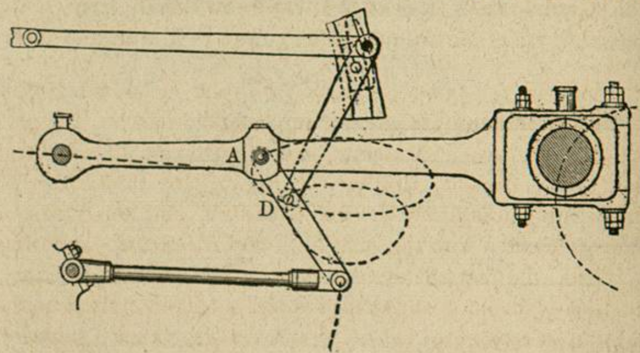

Fig. 105.

be unequal in the two strokes. This error is corrected by attaching the end of the lever to the point D of the vibrating link. For while the point A on the connecting rod describes a nearly true ellipse, as shown in fig. 105, the point D describes a bulged figure, and the amount of the bulge is so regulated as to correct the unequal motion of the fulcrum above and below its central position. It is obvious that by shifting the point D the amount of the bulge may be altered, and thus the error may be corrected too little, or too much, and by taking advantage of this circumstance a later cut-off may be given to either end of the cylinder if found desirable.

Several advantages are claimed for this type of gear over the ordinary link motion driven by eccentrics. Foremost among these is the fact that it gives an almost mathematically correct motion to the valve, which the older gear does not. It is also considerably cheaper; and from the peculiarity that the valve boxes are on the top of horizontal cylinders, and in front of vertical marine engine cylinders, instead of being at the sides, as is the case when the valves are driven by ordinary eccentrics, a considerable saving of space is effected.

Geometrical Representation of Action of Slide Valve.

There are few points connected with the successful working of steam engines of greater importance than the design of the valve gearing. Mathematical calculations intended to show the connection between the dimensions of the valve, the position and throw of the eccentric, and the various points connected with the distribution of the steam—such as the lead, the period of admission, the cut off, release, duration of exhaust and compression—are too complicated and cumbersome to be of general use. Many geometrical diagrams have been designed to meet the drawbacks of mathematical calculation, and of these the most simple and comprehensive are those designed by Dr. G. Zeuner, which we will now proceed to describe and illustrate.

Zeuner's valve diagrams.—It will be assumed for the sake of simplicity that the obliquity of the eccentric rod has no appreciable effect upon the position of the valve. The diagram will be made to show the particular angles of the crank at which the various critical points of the steam distribution take place, from which the corresponding positions of the piston can be deduced for the forward and backward strokes when we know the ratio of the length of the connecting rod to the arm of the crank.

Referring to fig. 85, which represents a valve without lap or lead and an eccentric set at right angles to the crank, let AO, fig. 106, represent the position of the crank at one of its dead centres, and OE at right angles to AO, the corresponding position of the eccentric, the valve being then in its central position; also let the amount of eccentricity or half throw of the eccentric equal OE. If the crank now occupies successively the positions OB, OC, &c., the eccentric will take up the corresponding positions OF, OG, &c.; and if from the points F, G, &c. we let fall perpendiculars F*f*, G*f'*, &c., upon the diameter AOH, the lengths O*f*, O*f'*, &c., intercepted between the centre of the circle and the feet of the perpendiculars, will represent the distances by which the valve has been moved from its central position when the eccentric takes up successively the positions OF, OG, &c.

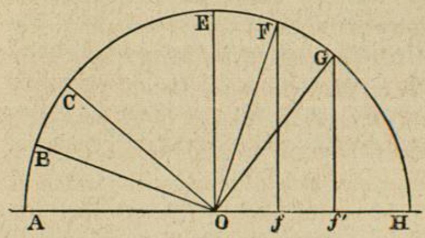

Fig. 106.

Now, however, suppose the eccentric to be fixed in position independently of the crank shaft and suppose the latter to revolve with the engine cylinder and all the moving parts attached round the centre O. It is evident that the fixed eccentric will in this case impart exactly the same motion to the valve as was done in the former case, only

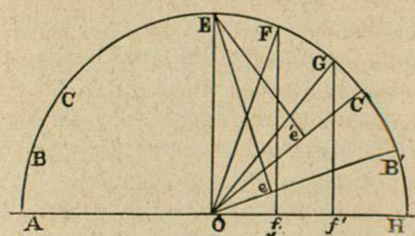

Fig. 107.

that in this instance the revolution will have to start from the dead centre H, fig. 107, and proceed in the reverse direction, so that when the crank occupies successively the positions OB, OC, &c., it will be represented in the diagram as occupying the positions OB′, OC′, &c. From the centre of the eccentric E let fall perpendiculars E*e*, E*e′*, &c. upon the lines OB′, OC′, &c.; then the lines O*e*, O*e′* represent the distances travelled by the valve from its central position when the crank occupies the respective positions OB′, OC′, &c. For in the diagram, fig. 106, we saw that the distances moved by the valve were represented by the lines O*f*, O*f′*; and it can easily be proved that O*f*, O*f′* are respectively equal to O*e*, O*e′*. In the triangles OE*e* and OF*f* we have OF = OE, both being radii of the same circle; also the right angle O*e*E = the right angle O*f*F. Also by construction the angle EOF = the angle B′O*f*. Therefore, adding to each of these the common angle FO*e*, we have the whole angle EO*e* = the whole angle FO*f*, and consequently the two triangles are equal, and the side O*e* = the side O*f*, and similarly O*e′* = O*f′*, and so on for every other position of the crank; therefore O*e*, O*e′*, &c. represent the distances travelled by the valve from its central position when the crank occupies positions opposite OB′, OC′, &c. Hence we see that the distances moved by the valve for any positions of the crank OB, OC may be found graphically by dropping perpendiculars from the centre of the eccentric E upon the opposite positions OB′, OC′, and measuring the lines intercepted between the feet of these perpendiculars and the centre O.

Now it will be noticed that in this way a series of right-angled triangles, O*e*E, O*e′*E, &c. are constructed upon a common base OE; and it is a well-known fact (depending upon Prop. 31, Euclid, Bk. iii.) that when such a series of right-angled triangles is constructed, their apices, *e*, *e′*, *e″*, &c., all lie upon the circumference of a circle of which the base-line is the diameter.