condensation and partial re-evaporation of some of the entering steam.

4. The steam contained in the clearance spaces which affects the curve of expansion.

5. The gradual opening of the exhaust port, which makes it necessary to release the steam too early in the stroke.

6. The friction of the exhaust passages, which in the case of condensing engines prevents the attainment in the cylinder of the same degree of vacuum as in the condenser, and in the case of non-condensing engines adds to the back pressure.

7. The momentum of the moving parts, which, combined with cause 4, and also with the unavoidable nature of the simple slide valve driven by an eccentric, renders a curve of compression necessary.

Examples of diagrams from actual engines.—We shall afterwards see that the indicator possesses other important uses in addition to those named above, but it is first proposed to give a few examples of good and bad diagrams, taken from various engines, and to point out some of the peculiarities which they reveal.

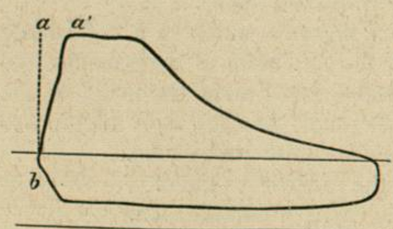

Fig. 136.

The diagram represented by fig. 136 shows a very late admission of steam, the maximum pressure not being attained till the piston has traversed a portion of the stroke represented by *aa'*. The proper position of the line *ba'* is the dotted line *ba*. The fault is evidently due to the valve having been badly set, either through becoming displaced, or else through the eccentric not having been given sufficient advance (see page 255).

Fig. 137 shows that the steam pressure during admission is injuriously affected by throttling, owing to insufficient opening or area of the port.

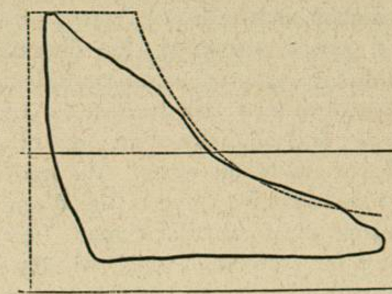
Fig. 137.

Fig. 138 represents a diagram which in addition to numerous other defects shows a very high back pressure. This was due to the fact that the steam, instead of being allowed to escape directly into the atmosphere, was passed first into a feed water heater. With some classes of feed heaters it happens that much more power is lost by the increase in the back pressure than is gained by raising the temperature of the feed.

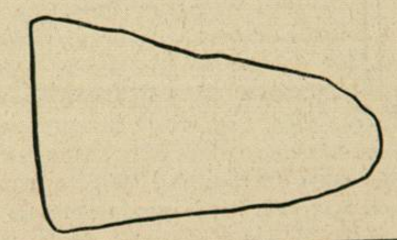
Fig. 138.

How to draw the hyperbolic curve of expansion.—The diagrams (figs. 139 to 142) are given to show the effects of condensation and re-evaporation on the curves of expansion. In order to be able to mark this effect more accurately it is advisable in all cases to lay down on the diagram the hyperbolic curve of expansion, which is the graphic representation of Boyle's law; for though this curve represents neither the curve of expansion of steam nor the curve of its relative volumes, it is found, nevertheless, that it is the line to which the expansion of steam in the best types of engines most closely approximates, and is for this reason the best curve to use as a standard of comparison.

In order to draw the curve of expansion for a given diagram, such as fig. 139, erect a perpendicular *ab*, to the line of perfect vacuum *ac*, the distance *aa'* representing the clearance reduced to an equivalent fraction of the stroke. The lines *ab*, *ac*, will be then the asymptotes of the hyperbola; and *ad*, drawn at an angle of 45° with *ab* and *ac*, will be the axis of the curve. We must now select some point in the expansion curve of the diagram from which to commence the hyperbolic curve. This latter will in general vary for every point which we may choose, for if there be condensation at the commencement and re-evaporation towards the end of the stroke, it is evident that there may be a higher pressure of steam in the cylinder at the end of the stroke than there should be if the true curve of saturated steam expanding and doing work were followed. It is usual to choose a point either at the commencement of the curve, such as *e* (fig. 139), when the steam port has been completely closed, or else a point *f*, just before the exhaust is opened. The co-ordinates, *eg* and *ek*, of the point *e* (or of *f* if we select the latter point) must then be measured and multiplied together. The points e', e'', e''', &c., corresponding to the positions of the piston, g', g'', g''', are such that the products of their co-ordinates equal the product of the co-ordinates of the original point, *e*.

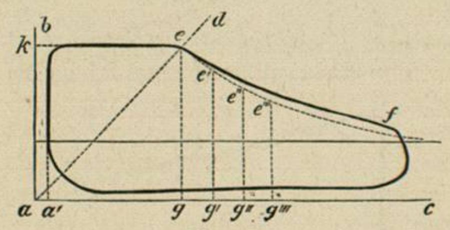

Fig. 139.

Thus $e'g' \times g'a = eg \times ga. \therefore e'g' = \dfrac{eg \times ga}{g'a}$.

Initial condensation and re-evaporation shown by diagrams.—In the diagram, fig. 139, we see that the actual

expansion curve, *ef*, of the steam lies throughout its whole length above the hyperbola line, showing a considerable re-evaporation of water, which has either been formed by condensation at the commencement of the stroke or carried over in the form of spray from the boiler.

Fig. 140 shows the same effect in a still more marked degree. In this case it was ascertained that a large quantity of water was carried over into the cylinder from the boiler, which was partially re-evaporated by the end of the stroke. That the water was not wholly re-evaporated, before the return stroke commenced, is shown by the bad vacuum line. The effect of water in the cylinder in increasing the back pressure is most marked, and is, no doubt, in part due to the re-evaporation which goes on during the exhaust, when the diminished pressure must enable considerable quantities of the highly heated water to burst into steam.

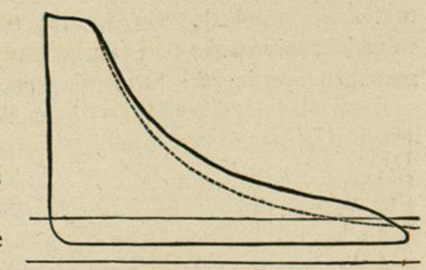
Fig. 140.

The diagram on fig. 141 is given to illustrate the case of condensation at the commencement and re-evaporation at the end of the stroke. Here, it will be noticed, the expansion curve falls below the hyperbola at the commencement, then crosses it, and for the remainder of the stroke lies above it.

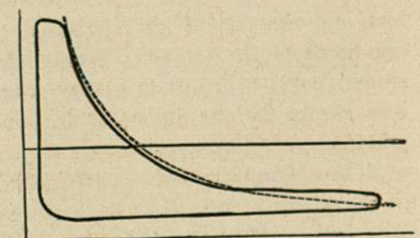
Fig. 141.

Leaky pistons and slide valves.—In engines which have been long at work, the expansion curve may be injuriously affected by leakage of steam through the valve or piston. The best method of ascertaining whether this is going on is to block the fly-wheel at any point in the stroke while the admission port is open, then to admit steam, and to open the lubricating cock at the other end of the cylinder. If steam continues to pour steadily forth from the cock, it shows that there is a leak. By blocking the fly-wheel when the valve is at mid-stroke, and consequently covering both ports, and then opening the lubricating cock, or looking at the mouth of the exhaust pipe, we can ascertain whether the leakage is through the valve.

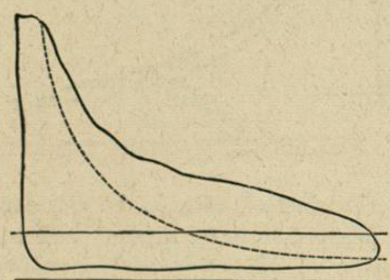

Fig. 142.

Fig. 142 is the diagram of an engine in which a very considerable leak took place past the valve. The initial pressure of steam in the cylinder was about 67 lbs. absolute, and the cut-off was supposed to take place at about one-tenth of the stroke. Hence the pressure at release should have been about seven pounds absolute, whereas it is shown by the indicator to have been nearly twenty pounds.

The diagrams on fig. 143 are intended to show differences of the exhaust line. In No. 1 the exhaust port is opened at *a* before the end of the stroke, and by the end of the stroke the steam pressure has fallen very low. In No. 2, which was taken from the same engine as No. 1, but with the eccentric badly set, so as to cause a late admission of the steam, the exhaust is not opened till just before the very end of the stroke, and the terminal pressure, is much higher

than in the case of No. 1, although the initial pressure is less, and the rate of expansion the same. The vacuum line is inferior to that of No. 1 at the commencement, as the steam has not had time to escape before the return stroke begins. It very often happens that what is gained by postponing the release till the stroke is finished is lost again through the increase in the back pressure. The terminal pressures are, however, very different in the two cases, and consequently, also, the twisting moments on the crank towards the end of the stroke.

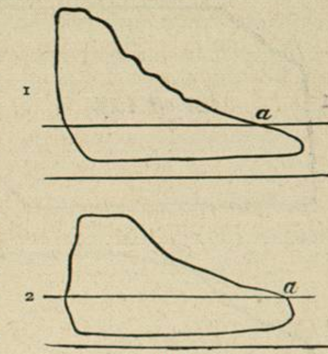

Fig. 143.

Fig. 144 is taken from a cylinder provided with exhaust valves and ports quite independent of the steam valves and ports. As will be seen from the sharp corner at *a* and the sudden fall of the pressure, the exhaust port opens sharply and fully, thus allowing the steam to escape very readily.

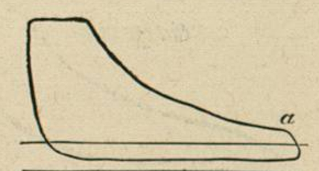

Fig. 144.

The diagram, fig. 145, shows as bad a distribution of the

Fig. 145.

steam as it is possible to conceive. The valve is deficient both in lap and lead, consequently the admission of the steam is late; the engine works with full steam during the whole stroke; the port is opened so gradually that the full pressure is not attained till the end of the stroke. The exhaust opens so late and so gradually that the pressure of the exhaust steam had not fallen to its proper point till nearly the end of the return stroke.

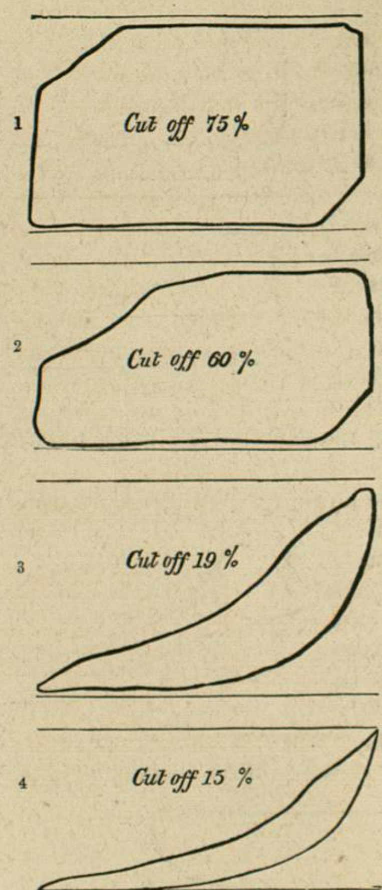

Fig. 146.

The diagrams on fig. 146 have reference to the line of compression. In an engine with a single slide valve to regulate the admission and exhaust of the steam from each end of the cylinder, the point of the stroke where the exhaust closes is dependent on the point where it opens — *i.e.* the point of release; and in proportion as the latter occurs early in the forward stroke, so will the former take place early during the return stroke. This effect is more particularly apparent in the diagrams of locomotives which are driven by a single slide, the rate of expansion being varied by means of the link motion. Nos. 1, 2, 3, and 4 are taken from the same locomotive engine, working at different rates of expansion, and with varying points of release and compression. In Nos. 3 and 4, the exhaust is closed so early that the curve of compression rises to a great height before the end of the return stroke, so much so that the back pressure at the end of the return stroke rises nearly to the initial pressure at the commencement of the forward stroke. The advantage of compressing the exhaust steam has already been explained.

It would be impossible to give in this chapter examples of all the peculiarities which may arise in diagrams from defective valve setting, or leaky valves and pistons, priming boilers, and unjacketed cylinders; but, as the nature of the best attainable diagram has been explained, and also the principal faults and peculiarities which occur in ordinary engines, enough has been said to enable the student to investigate the condition of most engines from an inspection of their indicator diagrams.

Gross and net indicated power.—The indicator diagram gives us, as we have seen, an exact account of the working of the engine and of the power which is being exerted and which is available for transmission, either to the working parts of the engine proper or to some external train of mechanism. It also gives the total power which is being exerted by the engine which includes the power which is being thrown away in overcoming back pressure. Take, for instance, the diagram, fig. 147, which is taken from a high-pressure expansive engine, the line AE being the line of absolute vacuum. It is evident that during the forward stroke the work done by

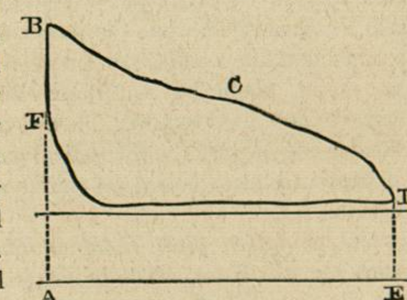

Fig. 147.

the steam on the piston is equal to the area ABCDE. During the back stroke the return of the piston is opposed by the pressure of the atmosphere and of the exhaust steam, on which it does work, measured by the area DEAF, so that the useful work available for transmission to the parts of the engine and external objects is represented by the difference between these two figures—*i.e.* BCDF.

The work represented by ABCDE is called the gross indicated power, and BCDF is called the net indicated power. This latter, again, is divisible into two parts, one being the work necessary to overcome the friction of the moving parts of the engine, and the other being the remainder, which is called the useful power, and which is all that is available for doing external work.

The above explanation will render clear the reason why the economy due to expansion in non-condensing engines is so very limited. For instance, taking fig. 148, it is evident that the power thrown away in overcoming the back pressure is about forty per cent. of the total power exerted. If a still greater degree of expansion were used, the gross power would be diminished, while the power wasted would remain the same, and the comparison between the useful and the gross power would be still more disadvantageous. Of course, condensation removes this evil to a great extent, but in condensing engines the theoretical gain due to high rates of expansion is also limited by causes which are explained in Chapter XI.

How to deduce from indicator diagrams the effective pressure on the piston.—The indicator diagram, though it shows the pressure at every point of the stroke, does not show the pressure which is available for transmission through the piston and connecting rods to the crank of the engine. This pressure is the difference between the total pressure, as shown by the diagram, on the driving side of the piston, and the simultaneous back pressure on the other side of the piston, which latter can only be obtained by taking a separate diagram from the other end of the cylinder. In well-designed horizontal engines the diagrams from the two cylinder ends should in most cases be as nearly as possible alike, but in many engines, especially those in which the valve setting is defective, or in which the connecting rod is short compared to the length of the crank, the diagrams differ from each other very considerably, and consequently a pair of diagrams is often absolutely essential to enable us to compute the net pressure transmitted by the piston.

In order to obtain the pressure available for transmission externally at any point of the stroke we must construct a new diagram, which shall show the differences between the simultaneous forward and back pressures on the two sides of the piston.

Let ABCD, A′B′C′D′, fig. 148, represent diagrams taken from the opposite ends of a cylinder. The diagrams are arranged so as to overlap, the point D′ marking the end of the forward stroke, being in the same vertical straight line as the point A, which marks the commencement of the back stroke. This arrangement is convenient, as it enables the simultaneous forward and back pressures to be seen at a glance. Thus at the commencement of the forward stroke the pressure on the working face of the piston is measured by the distance of the point A from the line of absolute vacuum GG′. At the same moment the other side of the piston is acted on by a back pressure measured by

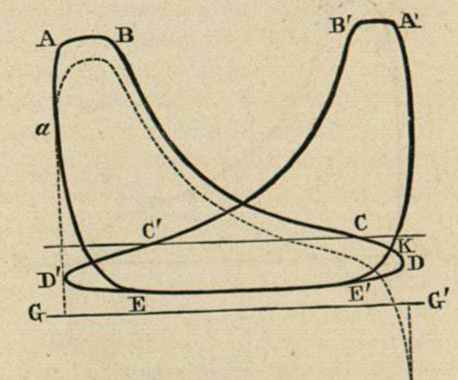

Fig. 148.

the distance of the point D′ from *gg′*, and therefore the effective pressure is the difference between these two, and is measured by the line AD′. When the piston has reached the position K, being the place where the exhaust line of one diagram crosses the line of compression of the other diagram, the two pressures are equal, and consequently the piston is urged neither forwards nor backwards by the steam, and continues its forward motion only by reason of the energy stored up in the moving parts of the engine and the fly-wheel. From the point K to the end of the stroke D, the pressure urging the piston forward is actually less than the back pressure, which latter consequently tends to bring the piston to a state of rest.

The true diagram of resultant pressure on the piston is formed by setting off the differences between the forward and back pressures as ordinates, measuring from the line of absolute vacuum as a base and drawing a curve through the ends of the ordinates. Whenever the forward pressure is in excess, the ordinate is drawn above the line GG′, and whenever the back pressure is in excess it is drawn below. We thus get the dotted curve of fig. 148 which is reproduced for the sake of clearness as a separate diagram in the curve D′AB*k*DA′, fig. 149, which shows the resultant pressure on the piston at any point of the stroke.

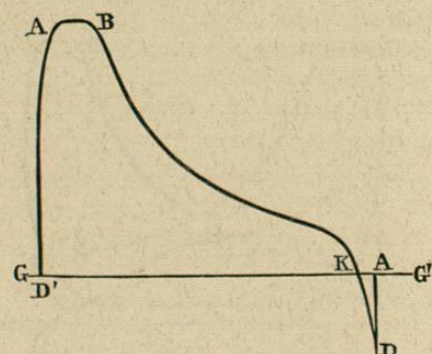

Fig. 149.

We are enabled by this method to see what important modifications of the resultant diagram may be caused by

alterations in the positions of the points of release C and compression E, fig. 148. If the release occurred at the end of the stroke instead of at the point C, the pressure between the points C and D would be greater than that shown in the figure. On the other hand, the vacuum at the commencement of the return stroke would not be so good, and consequently the back pressure between the points D and E′ would be greater than is represented. Moreover, if the steam distribution were controlled by an ordinary slide valve the points of release and of compression would be interdependent, and in the case under consideration there would be no appreciable compression. The general result would be that the positive pressure at the commencement and the negative pressure at the end of the stroke would each be diminished, and consequently the pressure on the piston would fluctuate between less wide limits.

The resultant diagram will be found of great use when we require to calculate the twisting moment on the crank-pin throughout a revolution (see Chapter V.).

How to ascertain the expenditure of steam accounted for by the diagram.—Another and most important use of the indicator diagram is to enable us to account for the expenditure of the steam and the heat supplied from the boiler to the engine. In order to measure, from the diagram, the steam consumed, we must ascertain the pressure at a point *a*, fig. 150, just before the release takes place. By reference to Col. 5 of Table I. we can deduce the weight of a cubic foot of steam of this pressure. We

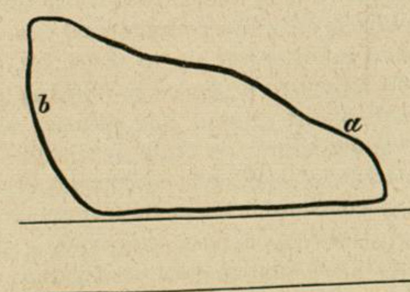

Fig. 150.

must ascertain the cubic contents of the cylinder, including the clearance, in cubic feet, up to the point *a*, and multiplying this number by the weight of the cubic foot of steam, we obtain the weight of steam present in the cylinder immediately before the release. When there is any compression, we must deduct the quantity of steam saved by the early closing of the exhaust. To do this we have only to measure the pressure at any point *b*, after the exhaust has closed, and to ascertain the weight of the contents of the cylinder up to the point *b*. The difference between the steam spent and saved is the quantity accounted for by the indicator. By comparing the quantity thus accounted for in an hour with the weight of water which leaves the boiler in the same time we can ascertain how much of it is lost by the combined effects of priming, condensation in the pipes, and condensation in the cylinder itself.

If we ascertain the pressure and weight of the steam contained in the cylinder at a point immediately after the admission is closed, we can, by the help of Table I., ascertain the number of thermal units contained in the steam at that point of the stroke. This number of thermal units will always be less than the total heat which has been supplied to the engine up to that point, for a certain number will have been expended in restoring the temperature of the ends of the cylinder and piston. Also during the expansion the steam loses the heat which has been converted into mechanical work, and consequently, if we were to measure the heat contained in the steam at any point, say *a*, fig. 150, before the release takes place, we should expect to find that the number of thermal units was less than that contained in the steam at the point of cut-off by the number converted into mechanical work As a matter of fact, however, the number is never less, and is often considerably greater, thus showing that a great deal of the heat which passes from the steam to

the sides of the cylinder during the early part of the stroke is, during the remainder of the stroke, re-transmitted from the cylinder to the steam, and passes out with the exhaust and is partly wasted.

The diagrams of compound engines will be considered in Chapter XI.