ment of the stroke the piston is actually being dragged forward by the crank-pin, instead of pushing the latter round, as it should do. When the curve *defg* rises above the line there is a small amount of pressure transmitted to the crank-pin, and consequently the strains in the piston and connecting rods are reversed from tension to compression, and a knock must occur if there is the least wear in the brasses. Between *e* and *f* the steam pressure is again insufficient, and the piston is again drawn round by the crank-pin, and at each of these points a knock will occur.

It will be noticed that the general effect of the action of the reciprocating parts is to completely reverse the pressures as deduced from the indicator diagram; for whereas these are greatest at the commencement of the stroke and dwindle down to nearly nothing at the end, when the reciprocating parts are taken into account, the pressures are greatest at the end of the stroke and are actually negative at the beginning.

During the return stroke the state of things at the commencement will not be so bad because the pressure required is only $56 \cdot 5 - 9 \cdot 17 = 47 \cdot 33$ lbs., and there is sufficient steam power available for this purpose and to leave a balance over to transmit to the crank pin. On the other hand, at the end of the stroke the state of things will be very unfavourable for easy running because of the enormous accumulation of pressure.

In order to improve this engine, one of three plans might be adopted. The initial steam pressure might be increased: the piston speed might be diminished; or, lastly, the weights of the moving parts might be reduced. A combination of these methods would, of course, also be effectual.

Effect of distribution of steam on the action of the reciprocating parts.[1]—Great care must be taken when designing the valve gear of high-speed engines to see that the distribution of the steam is properly effected, otherwise the ill effects of

[1] Students who have no knowledge of indicator diagrams should read this section after studying Chapter VIII.

a bad distribution may be greatly aggravated. Take, for example, the very common fault of a late exhaust, as represented by fig. 44 Assuming that the diagram of the return stroke is equally bad, the result will be that there will be a back pressure $a'b' = ab$ at the commencement of the stroke, and the effective steam pressure will only be Aa', which, if the piston speed is high, and the weight of the reciprocating parts considerable, may not be sufficient to impart the desired velocity to these latter. A late admission of steam will, of course, produce a similar result to a still greater degree.

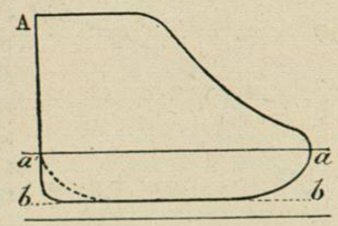

Fig. 44.

On the other hand a good compression curve, which, as we shall hereafter see, is an excellent feature in an indicator diagram, is productive of great good from the point of view of steady running, when its effect on the action of the reciprocating parts is taken into account. We have seen that the tendency with high-speed expansive engines is that the effective pressure on the crank-pin is transferred to the end of the stroke. Now it is very undesirable that the pressure on the pin should be very high at the *extreme* end of the stroke, as it causes heavy strains on many parts of the machinery; but the effect of a marked curve of compression is to cause a considerable back pressure at the end of the stroke, which pressure must of course be deducted from the pressure on the crank-pin, due to the combined effort of the steam and the retardation of the reciprocating parts. This action is of course much assisted if the exhaust opens early. In fact, in such cases it often happens that when a strong compression curve exists the back pressure on the piston is much in excess of the direct pressure.

This effect is clearly shown in fig. 43, and is also illustrated by fig. 45. The indicator diagram is clearly recognisable ; FGH, the curve showing the effect of the reciprocating parts, is indicated by the full line. The irregular dotted line AB is the resultant curve, giving the true nett pressures on the piston, and showing a negative pressure at the end of the stroke, on the assumption that a diagram taken from the other end of the cylinder would give a similar line of back pressure to that shown on fig. 45.

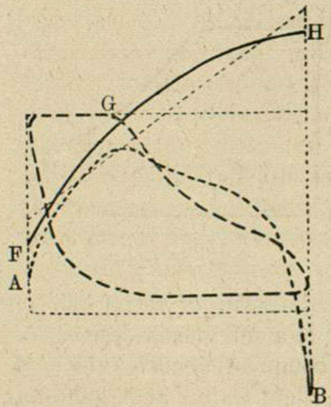

Fig. 45.

The student is now in possession of the means of constructing a curve of twisting moment, or tangential effort on the crank-pin, taking into account all the principal modifying circumstances. It will have been observed that the effect of the connecting rod is twofold. Firstly, by its varying inclination it alters the length of the tangential component of the pressure transmitted to the crank-pin (see p. 173) ; and secondly, it modifies the action of the reciprocating parts in the manner just explained.

The various steps to be taken to produce a complete curve of effort on the crank-pin are as follows :—

1st. To obtain a pair of indicator diagrams, viz. one from each end of the cylinder.

2nd. By taking account of the back pressure at each point in the stroke to deduce a pair of resultant diagrams showing the true pressures on the driving sides of the piston, as explained pages 190 and 340.

3rd. To find the effect of the weights and velocities of the reciprocating parts as modified by the length of the connecting rod, and to correct the last found diagrams accordingly, as explained pages 183 to 192.

4th. To deduce from the last diagrams the circular diagrams of effort on the crank-pin, taking into account the obliquity of the connecting rod, as explained pages 172 to 175.

These various steps have been explained at such length that it will not be necessary to give a final example illustrating them. The student will have no difficulty in applying the principles to any example whatever.

In the case of two or more cylinders being coupled on to one crank shaft, the diagram of effort will have to be made for each cylinder separately, unless they are identical in dimensions and steam distribution, and a resultant diagram formed by adding their separate effects together; see page 175.

How to approximate to uniformity of effort on the crank-pin.—In all of the examples given the engines have had single cylinders, and though it is possible in such engines to attain to great uniformity of driving power throughout the greater portion of the revolution, even when very high rates of expansions are made use of, by running them at high speeds so as to utilise the action of the reciprocating parts, nevertheless, the difficulty exists that at the beginning of each stroke there is no rotative effort whatever on the crank. There are two methods of overcoming this defect.

First. Two or more cylinders may be made use of coupled on to the same crank shaft, but with the cranks set at angles to each other, in such a manner that one engine is producing its maximum rotative effort when the other is at the dead centre. Each of these cylinders may be identical in dimensions and in steam distribution, as is the case with locomotives and many types of land engines ; or the engine may be compound, one cylinder being much larger than the other, and the steam which has been partially expanded in the small cylinder being used over again in the larger. This type of engine, which presents many advantages from the point of view of fuel economy, will be more particularly described in Chapter XI. It has, on account of its economical properties, come into very general use for marine purposes.

Second. A fly wheel may be used which, by absorbing energy when the turning power is in excess of the resistance, gives it out again when this condition is reversed, and thus enables the engine to pass the dead centre.

The double cylinder type of engine is invariably used where there is much stopping and starting and reversing to be done, because it may be easily started from any position of the cranks. The single cylinder engine with a suitable fly wheel is used with advantage whenever there is much steady running to be done in one direction. This is the case with the majority of factory engines. The single cylinder type with a fly wheel, and a high rate of expansion. has even been used successfully for marine purposes.

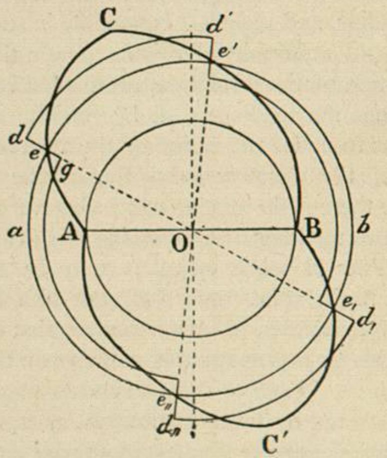

Fig. 46.

Graphic representation of the action of a fly wheel.—In cases where the resistance is uniform the power applied tangentially to the crank-pin in single cylinder engines is in

excess of the resistance during two portions of the revolution and is less than the resistance during the other two portions. This is illustrated in fig. 46, where AB is the crank circle, *ab* the circle of uniform resistance. The curve showing the twisting moment on the crank is denoted by ACB, BC′A. The tangential pressure is in excess of the resistance from e to e' and from $e_{,}$ to $e_{,,}$ and during the other two arcs the resistance is in excess. The average value of the excess of twisting moment is represented by *ed*, and of the resistance by *eg*.

The same thing may be illustrated also by a diagram constructed on a straight base similar to fig. 30A. The line ABA′ is equal in length to the circumference of the crank-pin circle. ACB and BC′A′ are the curves of twisting

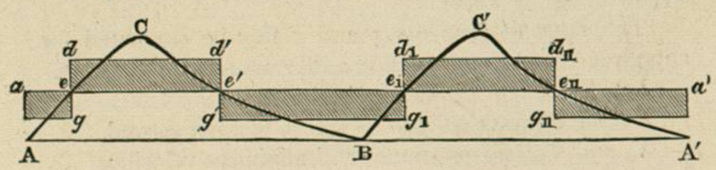

Fig. 47.

moment during the forward and back strokes respectively. The uniform resistance is shown by the line aa', the ordinate aA being the mean of all the ordinates of the curves ACB, BC′A′. The excess of tangential pressure between ee' and e_1e_{11} above the resistance is shown by the ordinates of the curves $e\mathrm{C}e'$ and $e_1\mathrm{C}'e_{11}$, and the excess of resistance above tangential pressure during the remainder of the stroke is shown by the ordinates of Ae, e'B, Be_1, and e_{11}A′. At the four points e, e', e_1, e_{11}, the resistance exactly equals the tangential pressure. The average values of the excess of tangential pressure is given by the lines ed and e_1d_1, the areas $edd'e'$ and $e_1\,d_1\,d_{11}\,e_{11}$ being equal respectively to the areas $e\mathrm{C}e'$ and $e_1\mathrm{C}'e_{11}$. Similarly the average values of the excess of resistance are given by the ordinates eg and $e'g'$. Also the sum of the areas $edd'e'$ and $e_1\, d_1\, d_{11}\, e_{11}$ is exactly equal to the sum of the areas ag, $e'g_1$, and $g_{11}a'$.

The weight of fly wheel necessary in any given case depends on the following conditions :—

1. The mean diameter of the fly wheel rim.
2. The velocity at which this rim moves.
3. The variation from uniform speed which is to be permitted.
4. The fractions of the entire revolution during which the tangential force is above or below the uniform resistance.
5. The average amount of the excess of power of resistance.

When these conditions are known we can compute the weight of the fly wheel.

The value of elements 4 and 5 may be computed for any given case by constructing a diagram similar to fig. 47.

Let W=weight of the fly wheel in lbs.

V=the mean velocity of its rim in feet per second.

V_1 and V_2=the maximum and minimum velocities.

$\frac{1}{k}$=the fraction of the mean velocity allowed for variation, that is to say the difference between V_1 and $V_2 = V \times \frac{1}{k}$.

Then the energy stored up in the fly wheel when at its maximum velocity

$$= \frac{WV_1^2}{2g}$$

and when at its least velocity the energy remaining

$$= \frac{WV_2^2}{2g}$$

Therefore the work given out by the fly wheel while its velocity falls from V_1 to V_2

$$= \frac{W(V_1^2 - V_2^2)}{2g}$$

Now this work is measured by the product $ed \times ee' \times A$ the area of the piston (see fig. 47), ed being the average excess of tangential pressure in lbs. per square inch of piston.

$$\therefore\ ed \times ee' \times A = \frac{W(V_1^2 - V_2^2)}{2g}$$

$$= \frac{W\,(V_1 + V_2)\,(V_1 - V_2)}{2g} = \frac{W \times 2V \times \frac{V}{k}}{2g}$$

$$\therefore\ W = \frac{g \times k \times ed \times ee' \times A}{V^2}$$

Now V, the mean velocity of the fly wheel rim in feet per second, may be expressed as follows in terms of the diameter D, and the number of revolutions per minute R

$$V = \frac{\pi . D . R}{60}$$

Substituting this value of V in the above equation and reducing to tons, we have very nearly

$$W\ (\text{tons}) = 5{\cdot}25\ \frac{k \times ed \times ee' \times A}{D^2R^2}$$

It will generally be found that the value of ed differs in the two strokes; we must therefore take the value which shows the greatest inequality in making the above calculation for the weight of the fly wheel, otherwise it will permit a greater fluctuation than is desired. The value of k varies according to the kind of work the engine has to do. When employed in driving dynamo machines or spinning machinery, or in any kind of work where very uniform motion is required, the variation from the mean velocity should not exceed from 1 to $1\frac{1}{4}$ per cent., that is to say the value of k would vary from 100 to 80. In other cases the variation may be as much as one twentieth, or 5 per cent. or k=20. It will be seen from the foregoing formula that the weight of the fly wheel varies directly as k, that is to say the less the permissible variation from the mean speed the greater the weight of the wheel. It also varies inversely as the square of the diameter and inversely as the square of the number of revolutions per minute.

The foregoing investigation applies to the case where the resistance is practically uniform. If the resistance fluctuates during each revolution, as for instance when the engine is driving a two-bladed screw propeller, a proper curve of resistance would have to be drawn in lieu of the line *a a'*, fig. 47. When the resistance is liable to sudden fluctuations, as for instance in the case of factories where a large number of machines are occasionally thrown on or off, the fly wheel by itself is powerless to produce even an approximation to steady running. To effect this in such cases is the duty of the governor, which controls the actual power developed by the engine, either by throttling the steam on the way from the boiler, so that its pressure is reduced considerably by the time it enters the cylinder; or else, by varying the rate of expansion so that the engine develops more or less power in proportion to the work it has to do. For description of various governors see p. 239.

Read to 273

CHAPTER VI.

THE MECHANISM AND DETAILS OF STEAM ENGINES.

Cylinders with their fittings—Clearance—Steam passages—Valve boxes—Jacketing—Lubricators—Pistons—Piston packings—Piston rods—Cross heads and slide bars—Connecting rods—Cranks and eccentrics—Eccentric rods—The strains in crank shafts—Journals—Shaft bearings and pedestals—Axle boxes—Governors—Fly wheels.

It is intended in this chapter to give an account of the separate parts which constitute the mechanism of the steam engine, excepting only the valves and valve-gear, which require separate treatment. The variety of form and arrangement of the parts of steam engines is so great, that it will only be possible to give a few representative examples. A more extended knowledge of the mechanism can only be gained by close observation of numerous engines, or working drawings.

The Cylinder.—The cylinder of a steam engine is the closed vessel in which the piston works backwards and forwards. It is so called because the interior is cylindrical in shape, though the form of the exterior is complicated by sundry additions. Examples of stationary engine cylinders are given in figs. 5 to 7, and of marine engine cylinders in figs. 194, 195, 197 to 200. Fig. 48 is a longitudinal section of a steam cylinder of a locomotive engine. It is made of cast iron, the interior being carefully bored so as to form a smooth and cylindrical surface for the passage of the piston. It consists of the following principal parts. The cylindrical body AA, which is cast in one piece;—the valve box BB, in the thickness of which are formed the two