the crank-pin. Suppose the piston to be at the commencement or end of the stroke as shown in the upper diagram fig. 62. Corresponding to these positions, the crank lies in the direction of the axis of the piston rod prolonged. If pressure be applied to the piston when at rest in either of these positions, that pressure will be transmitted direct to the bearings of the crank axle, there will be no turning force applied to the crank, and consequently the latter will have no tendency to revolve. Hence these two positions of the crank are called its dead centres. Let, however, the crank occupy the position *a′c*, fig. 62. The connecting rod will now be inclined to the directions of both piston rod and crank. If when in this position pressure be applied to the

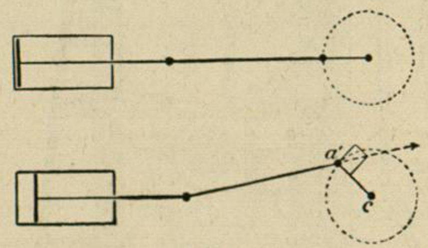

Fig. 62.

piston, the pressure will be transmitted through the connecting rod to the crank-pin, and can there be resolved in two directions, viz. along the crank and at right angles to it, i.e. tangential to the crank circle. The former component merely exerts pressure on the bearings of the crank axle, but the latter tends to cause the crank to revolve. The two components are constantly varying in size and in relation to each other, as is fully explained in Chapter V. When the crank is on either dead centre the tangential component vanishes, while the radial one is a maximum; and *vice versâ*, when the crank occupies such a position that the axis of the connecting rod forms a tangent with the crank circle, the radial component vanishes, while the tangential is a maximum.

This position, which occurs twice in each revolution, depends upon the length of the connecting rod relatively to that of the crank. If the connecting rod were of infinite length it would form a tangent with the crank circle when the crank was at right angles to the direction of the axis of the piston rod. The shorter the connecting rod relatively to the crank, the nearer to the dead centre will be the position of the crank when the tangent is formed. This is

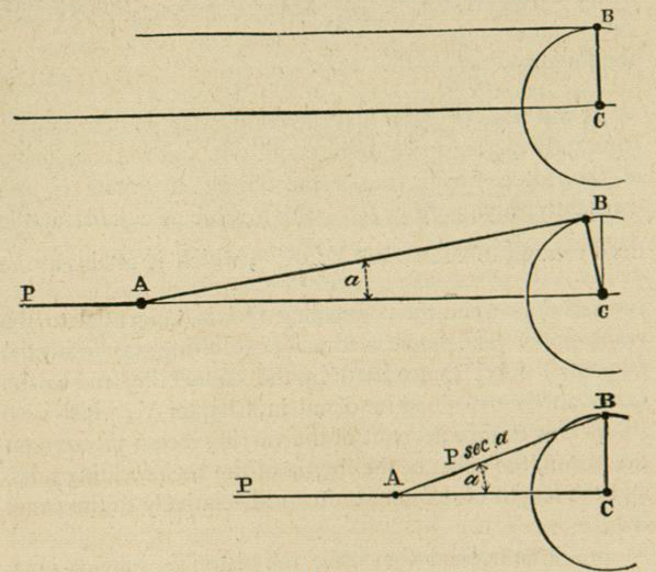

Fig. 63.

illustrated by the three diagrams, fig. 63, which represent the tangential positions for connecting rods having respectively lengths equal to infinity, and to six times and three times the length of the crank.

The force transmitted by the connecting rod is estimated as follows. Let P be the force acting on the piston rod, and a the angle which this latter forms with the connecting rod, then P sec a is the force in the connecting rod. If P were constant the value of this expression would increase with a. The maximum value attained by a is when the crank is at right angles with direction of the line PC. Let the connecting rod be r times the length of the crank arm. In this position ABC is a right-angled triangle and sec $a = AB \div AC = \frac{r}{\sqrt{r^2-1}}$. Suppose that $r=6$, then the force in the connecting rod $= P\frac{6}{\sqrt{35}} = P \times 1{\cdot}0142$. If $r=3$, then the force $= P\frac{3}{\sqrt{8}} = P \times 1{\cdot}06$.

It will thus be seen that the connecting rod transmits a constantly varying force to the crank-pin, and the component of the force which causes the crank to rotate is also constantly varying from zero, which value it attains at the dead centres, to P sec $a = P\frac{\sqrt{r^2+1}}{r}$, which it attains at the two positions when the connecting rod is tangential to the crank circle. The proper method of exhibiting graphically the tangential effort transmitted by the connecting rod to the crank-pin is explained in detail in Chapter V., which also shows how to take account of the varying steam pressure on the piston, the effect of the inertia of the reciprocating parts, and the length of the connecting rod relatively to the crank arm.

It will be noticed that while the piston is moving once backwards and forwards in the cylinder, the crank-pin is describing a circle, having a diameter equal to the length of stroke. Consequently the relative lengths of the paths travelled by piston and crank-pin are as the diameter to the semicircumference of a circle, or as $2r : \pi r = 2 : 3{\cdot}14159$. Now, by the principle of work, the pressure on the piston multiplied by the length of the path it travels equals the tangential pressure on the crank pin multiplied by the length of its path; hence the average pressure on the piston

is to the average tangential pressure on the crank pin as $3{\cdot}14159 : 2$. The fact that the average tangential pressure on the crank-pin was less than that on the piston led old writers on the steam engine into the fallacy of asserting that the oblique action of the connecting rod caused a loss of power in the steam engine. They forgot, however, that though the full pressure on the piston does not in general appear tangentially to the crank, the path travelled by the latter is correspondingly greater than that of the piston.

In addition to the stress due to the pressure of steam on the piston, which is alternately one of compression and tension, the connecting rod is also subject to stresses, due to the inertia of the piston and piston rods and also due to its own inertia. These stresses, which are dealt with in Chapter V., may become very considerable in high-speed engines. The rapid vibration of the rod from side to side, somewhat after the manner of a pendulum, also tends to produce a peculiar motion of the rod called whipping. This arises from a bending stress due to the resistance of the rod to acceleration in a direction at right angles to the motion of the piston. The tendency to whip is greatest in rods of great relative length. It may be guarded against by sparing unnecessary weight in the rod, by making it strongest where the tendency to whip is greatest, and by so distributing the material in the cross-section of the rod that its resistance to bending stresses may be a maximum, For instance the rod is usually tapered from the piston rod end to the crank-pin end; and instead of being made round, it is usually given a rectangular section, the side of the rectangle which lies in the plane of the motion being the greatest. In this manner a given weight of material is distributed so as best to resist a bending stress. Sometimes the rod is given a section resembling that of a flanged girder, the greater part of the material being distributed in the two flanges.[1]

[1] The rules for estimating the stresses on and fixing the proportions of the various moving parts of engines are given with great clearness in Professor Unwin's Elements of Machine Design, published in this series.

Fig. 64 represents a connecting rod, and its various details as used in a stationary engine. The parts surrounding

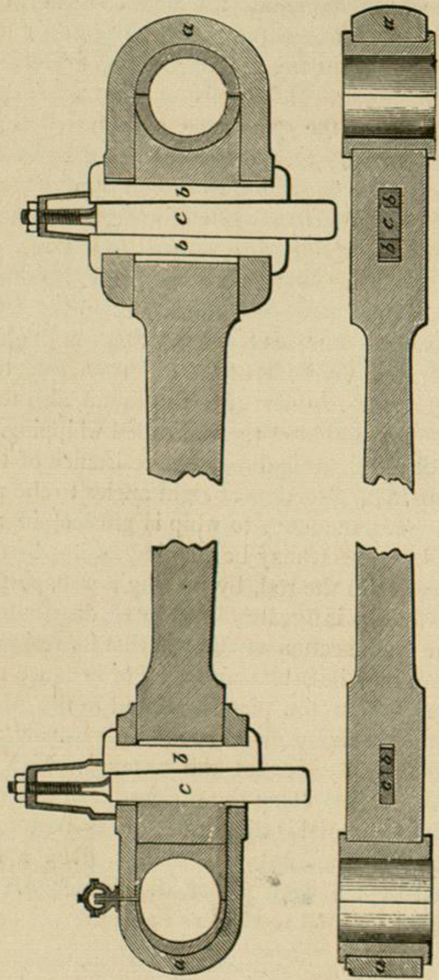

Fig. 64.

the crank and cross-head pins are made of gun metal, brass, or white metal so as to diminish friction. They are made in separate pieces, called steps, and are held in place by the straps *aa*, which are fastened to the rod by means of the gibs *bb*, and the cotters *cc*. When the brasses wear they can be tightened up by driving in or screwing up the cotter, which draws up the strap, and thus tends to shorten the rod.

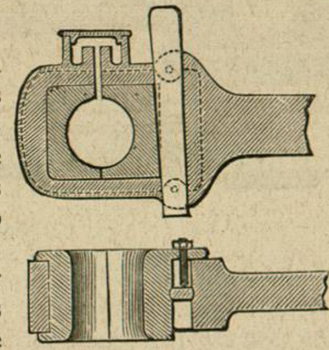

Fig. 65.

Fig. 65 shows a solid ended rod. In this case the brass steps are placed in the rectangular opening formed in the end of the rod, their flanges being partly cut away so as to allow of their being inserted. They are held in position by a cotter, which in its turn is prevented from moving by one or more set screws. When the brasses wear they can be tightened up by driving in the cotter, which has the effect of lengthening the rod.

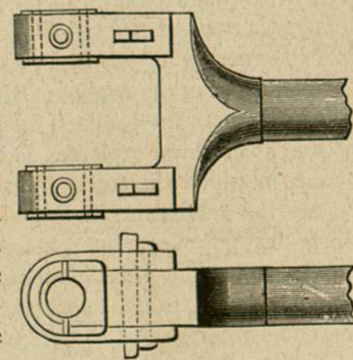

Fig. 66.

Fig. 66 shows a fork-ended connecting rod which is necessary with certain sorts of cross-heads. Fig. 67 is the ordinary type of marine connecting rod. In this rod the brass steps are not held by gibs and cotters, but by bolts, which are clearly shown in the sketch.

The lubrication of the surfaces of the brass steps where they rub on the crank and cross-head pins must be very carefully attended to.

The ends are for this purpose provided with lubricators, which are sometimes of brass, and can be removed, but are more commonly forged solid on to the straps or ends. If

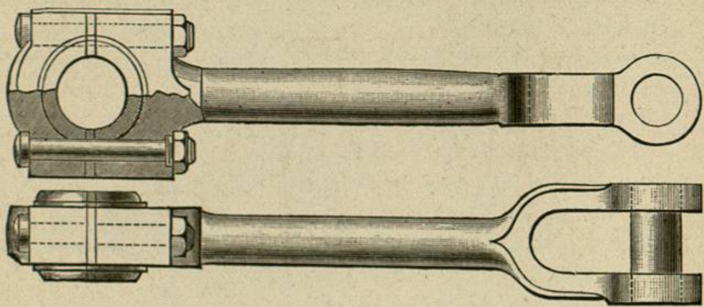

Fig. 67.

the lubrication is well attended to, and if the diameters and lengths of the pins are made sufficiently large so as to keep the pressure per square inch within moderate limits, the wear of the brass steps should give very little trouble.

In old-fashioned engines the connecting rod was often made of cast iron, but nowadays it is invariably of case-hardened wrought iron, or of mild steel.

Cranks and eccentrics.—The crank is simply a lever of the first order, either attached to, or forged in one piece with the main shaft of the engine. By means of it, the reciprocating motion of the piston is finally converted into circular motion, as has been explained in the previous section. The manner in which the force, transmitted by the connecting rod to the crank, is resolved into a radial component which has no effect but to exert pressure on the main bearing, and a tangential component which alone tends to revolve the shaft, is also explained in that section, and more fully in Chapter V.

Fig. 68 shows two views of one of the simplest forms of crank; *a* is the crank shaft, *c* the crank-pin. The distance from the centre of *a* to the centre of *c* is the length of the

crank arm, which is, of course, equal to half the stroke of the piston. *b* is the web of the crank, *d*, *d'* the bosses. Cranks of this form are generally of cast iron, and are attached to the

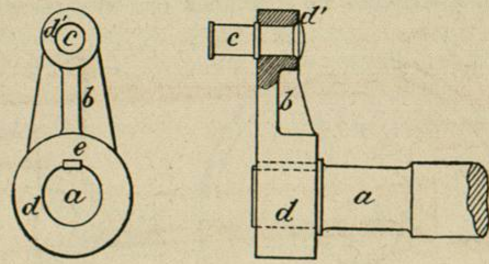

Fig. 68.

main shaft by means of a key *e*. The fastening of a moveable crank on a shaft requires the greatest care, because all the stresses thrown on the crank are liable to reversion during each stroke, especially in the case of a slow-running engine working with a considerable cushion of steam. Very severe reversals of pressure also occur if water is allowed to accumulate in the cylinder. In such cases the piston is brought up dead before the end of the stroke is reached, while the crank endeavours to pull it on, thus throwing a heavy strain on all the connections, and amongst others on the key. Cranks of this type in addition to being keyed are generally shrunk on to the shaft, or else are forced on by hydraulic pressure.

Fig. 69 shows an end elevation and cross-section of another form of cast-iron crank, called a disc crank. It is, as its name implies, formed of a disc of cast iron, attached to the shaft by the methods just described, and provided with a wrought-iron or steel crank-pin. The portion of the disc opposite to the pin is usually much thicker and heavier than the remainder of the disc, this extra weight being used as a balance to the weights of the reciprocating parts.

Very often it is impossible to arrange that the crank shall overhang the end of the shaft, and in such cases it has to be forged in one piece with the latter. A great number of engines also work with two or more cylinders, and a corresponding number of cranks. In the case of two-cylindered engines, the cranks are, as a rule, set at right angles to each other, so that when one of them is on its

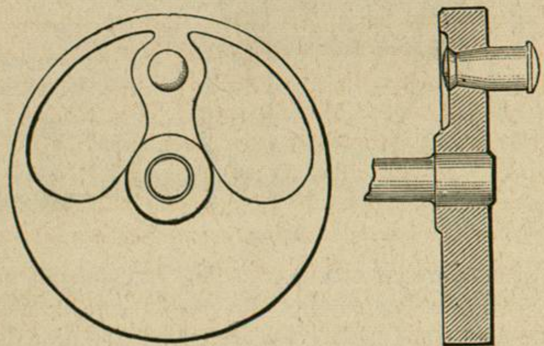

Fig. 69.

dead centre, the other is in the most favourable position, and thus, the resultant force tending to turn the shaft is made more uniform. Fig. 70 shows two views of the crank

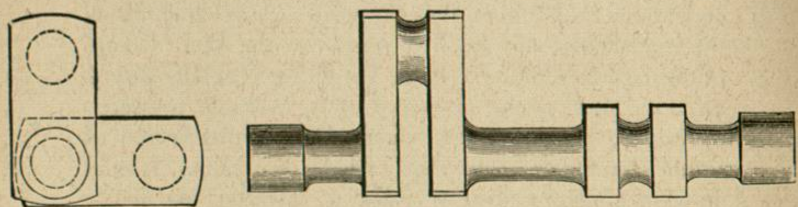

Fig. 70.

shaft of a locomotive engine, with two cranks forged in one piece with the shaft, the whole being made of steel, which metal, on account of its great strength, is especially useful in cases where, from want of space, the thickness of the webs or side pieces of the crank is limited. It will be noted

that in the case illustrated in fig. 70 the want of thickness in the webs is made up for by increasing their width, which is nearly double the diameter of the shaft.

The crank-pin is the portion of the engine which receives the greatest stress, and special care must therefore be given to its design and lubrication. The pressure which comes upon it varies in practice, according to the type and speed of the engine, from 500 lbs. to 2,000 lbs. per square inch. This latter figure is far too high for safety. The diameter and length of the pin should be so chosen that the pressure may not exceed 1,000 lbs. per square inch. It is desirable to make the pin as long as is consistent with strength and other considerations, because bearings retain their lubrication better when long than short. In high speed-engines, there is a great tendency to expel the lubricating oil from the crank pin by centrifugal force, and this tendency must be carefully guarded against, otherwise excessive wear of the connecting rod brasses will be the result.

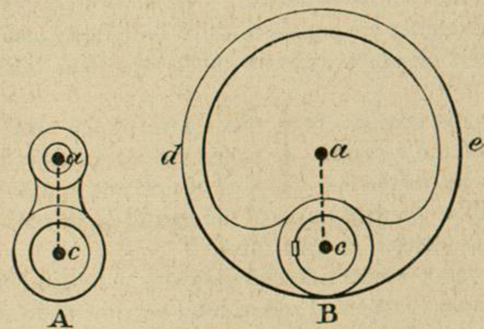

Fig. 71.

In the crank proper, the pin *a*, fig. 71 (A), is of such a diameter that it does not come in contact with the shaft. There is, however, a species of crank called the *eccentric*, in which the pin is so large that it completely envelopes the shaft. Such a crank is shown at fig. 71 (B). The distance