ac is the same as at A, but the pin has assumed the diameter of the outer circle *de*. In such a case, of course, the web is impossible and unnecessary. A crank is used for converting the rectilinear motion of the piston into circular motion; the eccentric, on the other hand, is usually employed for converting the circular motion of the main shaft back into rectilinear motion. The ordinary crank is, of course, equally capable of effecting this conversion, but it would in many cases be impossible to apply it to such a purpose. A crank always weakens a shaft except when overhung; hence, the fewer the number of cranks made use of, the better. Moreover, when circular motion is converted into rectilinear, as a rule, the rectilinear path travelled is short, relatively to the diameter of the crank shaft; hence a crank would be an impossibility, because the crank-pin would fall either wholly or partly within the section of the shaft. With an eccentric, however, the case is different, the distance *ac*, fig. 71 (B), corresponding to the length of the crank arm, may be as small as we please. The most frequent uses to which eccentrics are put are to drive slide valves and pumps, the travels of which are very much less than that of the piston.

The distance *ac*, from the centre of the shaft to the centre of the eccentric, is called the eccentric radius, the eccentricity, or the half-throw of the eccentric, and is equal in length to the half-travel of the part to be driven, such as the pump plunger, or slide-valve.

Fig. 72 represents side elevation and a longitudinal section of an eccentric and rod as used for driving an ordinary slide-valve. The circular portion *a*, which corresponds to the crank pin, is called the sheave of the eccentric. It is usually made of cast iron in two halves, which are bolted together round the shaft, and keyed on in the proper position. The piece *bb* is called the strap, and corresponds with the big end of a connecting rod. The strap is made of cast iron, wrought iron, or steel, according to circumstances,

and is lined with a brass or white metal ring, where it comes in contact with the sheave. This ring is grooved, as shown in the section at *cc*, so as to prevent it from getting off the

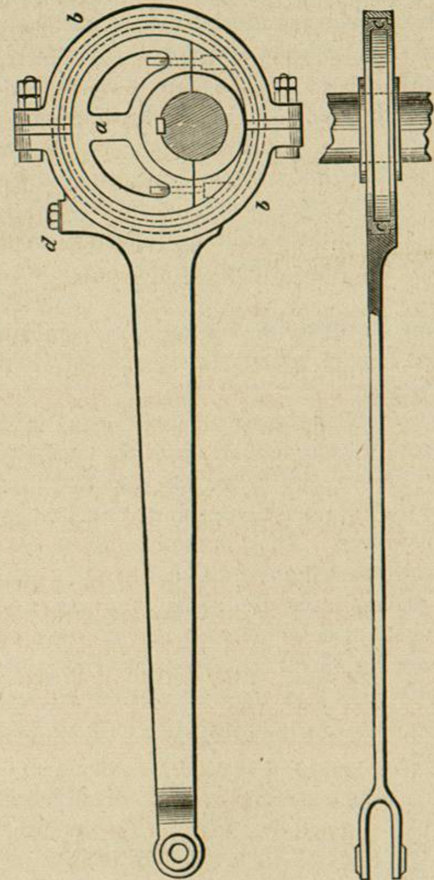

Fig. 72.

sheave. The strap is made in two halves bolted together so that it can be readily put on, or taken off the sheaf. It should be made sufficiently rigid not to spring when the engine is running, as the effect of this would be to cause a great deal of local friction between the sheave and strap. An oil cup *d* is usually forged solid on one half of the strap, and particular care must be given to the lubrication, the friction of eccentrics being much greater than that of cranks. The eccentric rod, which corresponds with the connecting rod of a crank, is sometimes forged solid with one half of the strap; it is, however, often made separate, and attached by bolts to the strap. The length of the eccentric rod, relatively to the half throw of the eccentric, is always much greater than that of the connecting rod, relatively to the crank arm. Hence, the disturbing influence on the point driven, due to the finite length of the rod, is usually very slight.

The manner in which the eccentric, or combinations of eccentrics, are used to actuate the valves, will be explained in the next chapter.

Crank Shafts.—The shaft of the engine is the part which receives circular motion from the crank and the reciprocating pieces. By means of the shaft, the power generated in the cylinder is transmitted to the machinery intended to be driven. Thus, in the case of factory engines, a pulley is usually keyed on to the shaft, and by means of a leather belt passing over this pulley, the various lines of shafting throughout the building are driven. In locomotive engines, the driving wheels are keyed direct on to the shaft, and rotate with it, and in the case of marine engines, the paddles or screw are also attached direct on to the shaft or its prolongation.

Strains in Crank Shafts.—Shafts are subjected to a variety of strains. In the first place, they undergo bending stresses from any weights which may be attached to them, the most considerable of which is that of the fly-wheel, acting vertically downwards. Also the pull of the driving belt causes a bending stress, which acts in the line joining the driving and the driven shaft. The most important stresses,

however, are due to the direct thrust and pull of the connecting rod, or rods, which, when at their maximum, act in the line of the axes of the cylinders. These various bending stresses may act in such directions as either to partly neutralise or to reinforce each other. In any case a resultant stress can always be found which represents their combined action. A shaft under the action of these stresses behaves in exactly the same manner as a loaded girder.

The foregoing strains act in the planes of the axis of the shaft. Shafts are, however, subjected to a totally different class of strains, which act in planes at right angles to the axle, viz. those which result from the twisting or torsional effects of the power and resistance acting at the ends of levers represented by the length of the crank arm, and the radius of the pulley, over which the main belt passes in the case of factory engines; or the radius of the driving wheels, or of the centres of effort of propellers, in the cases of locomotive and marine engines respectively. The tendency of these torsional strains is to shear the metal, composing the shaft, in a plane at right angles to the axis. They are consequently opposed, by the resistance of the metal to

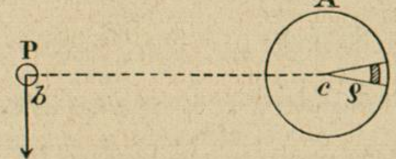

Fig. 73.

shearing. In fig. 73 let A represent a section of a shaft. Let $bc = l$ be the length of the crank arm, and P be the maximum force in lbs. transmitted to the crank pin by the connecting rod, estimated in the direction of the arrow. Then, $P \times l$ is the twisting moment applied to the shaft. If l is expressed in feet, then the moment is in foot-pounds, and if in inches, then the moment is in inch-pounds. The forces at work are prevented from shearing the shaft by the resistance to shearing of all the metal in the section A. If the shaft were just upon the point of shearing, the moment got by multiplying the resistance of the metal by the distance of its centre of resistance from c, would be exactly equal to the twisting moment Pl. To calculate the value of this moment of resistance, we must proceed in the following manner: Conceive the section of the shaft to be divided up into a number of triangles (one of which is shown), and the triangle to be divided up by lines parallel to its base into a series of bands (one of which is also shown shaded); then, the power of any one of these bands to oppose the shearing effect of the twisting moment Pl, relatively to the corresponding power of the exterior band, depends upon the area of the given band, and its distance from the centre, relatively to the exterior radius of the shaft. Call the variable distance of the bands from the centre ρ, and the angle at the apex c of the triangle, $d\theta$, then the breadth of the band, measured along the radius, is $d\rho$, and its length $\rho.d\theta$; consequently, its area is $\rho.d\rho.d\theta$, and if the band were carried right round the circle, its area would be $2\pi.\rho.d\rho$. The distance of this annular band from the centre, relatively to that of the exterior circumference of the shaft $=\frac{\rho}{r}$; hence, the power of the strip to resist shearing, relatively to that of the outside, $=2\pi.\rho.d\rho.\frac{\rho}{r}$ $=\frac{2\pi\rho^2}{r}d\rho$, and its moment about the centre $c=\frac{2\pi\rho^3}{r}d\rho$. Similarly with every other circular band into which the section of the shaft can be divided. Hence, the moment of resistance of the whole shaft to resist shearing about the centre c, is proportional to the above expression integrated between the limits $\rho=0$ and $\rho=r$, or

$$\frac{2\pi}{r}\int_0^r \rho^3 d\rho = \frac{2\pi}{r}\frac{r^4}{4} = \frac{\pi r^3}{2}.$$

If, instead of the radius r, we take the half diameter $\frac{d}{2}$ the above expression becomes $\frac{\pi d^3}{16}=0{\cdot}196d^3$.

In order to make this expression practically useful we require to know first, the ultimate resistance to shearing of the metal of which the shaft is made; and second, the factor of safety to be employed, that is to say, the number of times which the moment of resistance of the shaft should exceed the twisting moment, in order that it may be safe in practice. The following are usually taken as the ultimate shearing strengths of the metals usually employed in making shafts:

Cast iron	28,000	lbs. per square inch.
Wrought iron	54,000	
Steel	80,000	

As a rule the factor of safety employed for shafts is 6, therefore in designing a shaft for a given purpose we must only take one sixth of the above figures. Thus to find the proper diameter for a wrought-iron shaft subject to a given twisting moment Pl in inch lbs., we have

$$\cdot 196d^3 \times \frac{54000}{6} = Pl,$$

$$\therefore d \text{ (in inches)} = \cdot 08275 \times \sqrt[3]{Pl}.$$

Sometimes the horse-power (HP) transmitted by the shaft and the number of revolutions (N) are given. One horse-power = 33,000 foot-pounds per minute. The total horse-power equals the average pressure (P) applied to the end of the crank arm (l) × by the path of the crank pin in feet $\left(\frac{2\pi l}{12}\right)$ × by the number of revolutions (N) per minute,

$$\therefore 33{,}000 \times HP = \frac{P\ 2\pi l\ N}{12},$$

$$\therefore Pl = 63{,}024 \times \frac{HP}{N}$$

and, equating this latter expression to $\cdot 196d^3 \times 9000$,

we obtain d (in inches) $= 3\cdot295\sqrt[3]{\dfrac{HP}{N}}$.

Journals.—The part of the shaft which is supported by the bearing is called the journal. The usual form of the journal of an engine crank is shown in fig. 74. The part

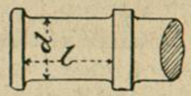

Fig. 74.

which runs in the bearings is turned so as to be truly cylindrical. The end play of the shaft is limited by the two raised collars. The length of the journal, or the distance between the inner faces of the collars, relatively to the diameter depends principally upon the number of revolutions which the shaft has to make per minute. For slow-running engines the length is sometimes equal to the diameter, whereas in cases of high speed it may be as much as from two to three times the diameter of the journal. The following figures give the proportions usually adopted[1] for wrought-iron journals.

No. of revolutions per minute	50,	100,	150,	200,	250,	500.
Ratio of length to diameter	1·2,	1·4,	1·6,	1·8,	2·0,	3·0.

Great care must be taken in designing journals not to pass abruptly from one section of the metal to another. All such differences should be gradually rounded off as shown in fig. 74.

The strains to which the journals of crank shafts are subjected are due to the combined action of the twisting forces and the transverse loads.

Shaft Bearings and Pedestals.—The bearing usually consists of brass steps supported by a cast-iron pedestal or

[1] *Elements of Machine Design.* By Professor W. Cawthorne Unwin.

plummer block. Fig. 75 shows three views in half elevation and half section of a common form of pedestal which is used with a masonry foundation. It consists of a wall plate

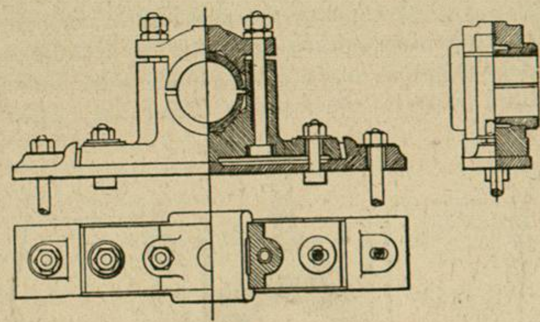

Fig. 75.

which is bolted to the foundation and on which is fixed the pedestal proper. The nature of the arrangement and the means by which the steps are adjusted and secured are sufficiently explained by the drawing.

In most stationary engines one or both of the pedestals are attached to the cast-iron framework as shown in fig. 76, which represents the principal pedestal of a horizontal engine. In this case the steps are not divided horizontally, but in an oblique plane, so that the direction of the resultant force of the pull or thrust in the connecting rod and of the other forces which act on the shaft, may pass through the solid metal of the step and not through the junction between the steps.

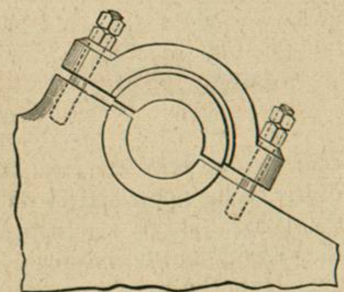

Fig. 76.

In the case of locomotives the bearings are not fixed, but are free to slide up and down in a vertical plane, within the limits allowed by the springs. These bearings are called axle boxes. The whole weight of the engine is transmitted through them to the journals by means of the springs. Fig. 77 explains the structure of an axle box, which consists of an outer case, arranged so as to be capable of sliding up and down, between guides called horn plates which are bolted to the frame of the engine. The casing

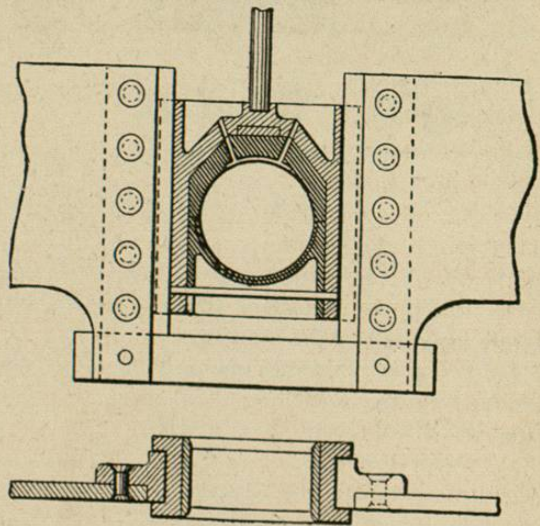

Fig. 77.

contains one brass step, the whole of the pressure being of course on the upper half of the journal. The lower part of the casing contains a receptacle for the oil which escapes after lubricating the bearings. The upper portion contains the oil box, and has also a socket formed in it which receives the foot of a spindle by means of which the pressure from the springs is transmitted.

Governors.—If, during the working of a steam engine, the load were wholly or partially removed while the supply

of steam to the cylinder remained undiminished, the engine would commence to race. If, on the contrary, the load were increased, the speed of the engine would be reduced below the proper rate. To prevent such variations in the speed, a contrivance called a governor is made use of which acts upon the steam supply in one of two ways; viz. either by partially closing or opening the throttle valve which regulates the flow of steam from the boiler; or else, by acting directly on the valve gear in such a way as to vary the point in the stroke where the steam is cut off, and thus alter the rate of expansion.

The most common form of governor was invented by Watt. It consists (see fig. 78) of two heavy metal balls A,D, attached to two inclined arms, which latter are jointed at the point E, to the central vertical spindle. The latter is connected by gearing with the main shaft of the engine so as to revolve at a rate strictly proportional to that of the shaft. The effect of rotation is that the balls tend to fly away from the vertical spindle and, being controlled by the arms, they can only rise and fall in arcs of circles about the centre E.

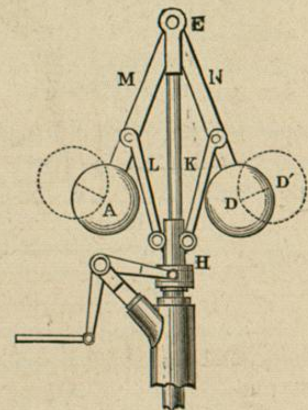

Fig. 78.

Supposing that the velocity of rotation were increased beyond the normal rate, the balls would fly out and occupy some new position D′, at the same time lifting the collar H which slides on the central spindle and which is attached by the links L and K and to the ball arms M and N. Into the collar H gears the forked end of a bell crank lever which is connected by a link with the throttle valve. When H is lifted the link acts upon the throttle valve, partly closing