it, and reducing the supply of steam. Conversely when the balls fall, H falls also and the throttle valve is opened.

The theory of the conical pendulum governor is as follows. Let us suppose that the weight of the arms may be neglected. When the balls occupy any position as in fig. 79, each of them is maintained in position by the three following forces. The weight of the ball W acting downwards; the tension T in the inclined arm; the centrifugal force $\frac{Wv^2}{gr}$ (where v is the velocity of rotation of the ball), acting radially and horizontally outwards. Since the ball is in equilibrium, the three forces may be represented in magnitude and direction by the three sides of the triangle ABC. Let the radius BC be denoted by r, and the height of the cone of revolution AC by h feet,

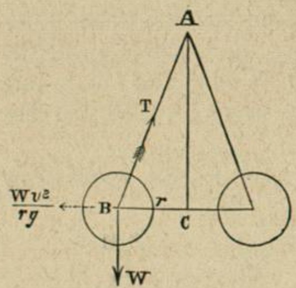

Fig. 79.

then $$\frac{h}{r}=\frac{W}{\frac{Wv^2}{gr}}=\frac{gr}{v^2}, \quad \therefore \frac{r^2}{v^2}=\frac{h}{g}, \quad \therefore \frac{r}{v}=\sqrt{\frac{h}{g}}.$$

Also since the ball is supposed to move in a circle in a horizontal plane with a uniform velocity, let $t=$ the time in seconds occupied in making one revolution,

then $$\frac{2\pi r}{v}=t, \quad \therefore t=2\pi\sqrt{\frac{h}{g}}.$$

Consequently the time of a revolution is proportional to the square root of the height of the cone of revolution.

If we are given the number of revolutions N per minute,

then $$Nt=60, \quad \therefore \frac{60}{N}=2\pi\sqrt{\frac{h}{g}},$$

$$\therefore N=\frac{30}{\pi}\sqrt{\frac{g}{h}}=\frac{54\cdot 29}{\sqrt{h}}.$$

If h be given in inches instead of feet, the above formula becomes

$$N=\frac{188\cdot 2}{\sqrt{h}}.$$

As the speed of rotation of the governor and consequently of the engine is inversely proportional to the square root of the height of the cone of revolution, it is clear that the possible variations in the height of the cone have a very direct influence upon the sensitiveness of the governor.

For instance, if the governor were so contrived that the height of the cone were a constant quantity, the speed of the engine would remain constant. The object aimed at in the practical design of governors is to keep the variations in the height of the cone of revolution within convenient limits. It will be noted that in fig. 78 the ball arms are jointed on the axis of the vertical spindle, while in other cases the joints are at some distance from the axis; in other instances the arms are crossed as in fig. 81, so that the joints are on the sides of the spindle opposite to the corresponding balls.

Each of these arrangements affects the height of the cone of revolution for a given position of the balls.

It is evident, from a mere inspection of the figures, for a given deviation of the balls from the axis of revolution, that the variation in the height is greatest in fig. 78, and least in fig. 81.

It is quite possible to make a governor so inconveniently sensitive, that it is never still for a moment, but is affected by even the small periodic changes of velocity, which occur in each revolution of the engine. As the governor can, by its nature, never act until after the change of velocity actually occurs, which it is designed to control, and as, moreover, the periodic changes of velocity above referred to are only momentary in their duration, it may easily happen that the effect of a hypersensitive governor is only felt by the engine when it has resumed its normal speed of rotation, or even attained a rate of speed which has fluctuated in the opposite sense to that which had affected the governor. In such cases, only harm can be done by the sensitiveness of the governor: for, supposing that it is affected at a period of the revolution when the crank-pin velocity is slightly above the average of the revolution, the supply of steam will be diminished when the crank-pin has re-attained its average speed, or even sunk below it, in which case the effect of the governor will be to aggravate the evil it was designed to cure.

Many advantages are found to attend the use of high-speed governors. They are more sensitive to alterations in speed, the parts may be made lighter and move with less friction. In order, however, to prevent the balls from flying out too far, in consequence of the increased speed of rotation,

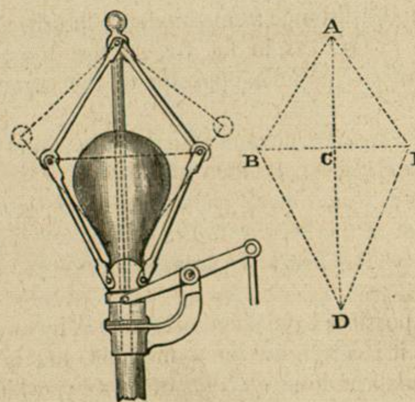

Fig. 80.

a weight, or else a spring, is so arranged as to act on the ball arms in such a manner as to develope a radial force in the contrary direction to the line of action of the centrifugal force. Fig. 80 shows a loaded high-speed governor. Each ball is attached to two sets of links. The weight is arranged

to slide on the central spindle, and presses directly upon the lower pair of ball links. To find the height of the cone h, corresponding to a given speed of rotation, we reason as follows : Each ball is at rest under the action of its weight acting downwards,—the centrifugal force acting radially outwards, and the tensions in the two ball arms due to the weights which they support.

Calling the weight of each ball, W, that of the load, W′, the radius BC$=r$, and the height of the cone AC$=h$, the tension in BA=T, in BD=T′, the angle BAC$=\alpha$, and BDC $=\beta$, we have :—

The centrifugal force in BC is balanced by the components of the tensions in the two arms, estimated in the direction BC

$$\therefore \frac{Wv^2}{gr} = T\sin\alpha + T'\sin\beta.$$

Also, the vertical component of the tension in BA balances the weight of the ball W, and the vertical component of the tension in BD

$$\therefore T\cos\alpha = W + T'\cos\beta.$$

The tension in BD is due to half the weight W′

$$\therefore \tfrac{1}{2}W' = T'\cos\beta, \quad \therefore W' = 2T'\cos\beta,$$

and, finally, $r = BA\sin\alpha = BD\sin\beta$, from which equations it is possible to eliminate the values of the tensions in the ball arms, and also the angles α and β, so that h, for any velocity v, may be expressed in terms of the weights of the balls and load, and of the radius r. The simplest case is when $\alpha=\beta$, whence

$$\frac{Wv^2}{gr} = (T+T')\sin\alpha,$$

$$W = (T-T')\cos\alpha,$$

$$W' = 2T'\cos\alpha.$$

Also, $h = r\cot\alpha$.

Eliminating T, T′, and $\cot\alpha$, we obtain

$$\frac{Wv^2}{gr}=\frac{r(W+W')}{h},$$

$$\therefore\ h=\frac{r^2g}{v^2}\left(1+\frac{W'}{W}\right).$$

It has been already shown how the governor can be arranged to act on the throttle valve. In many modern

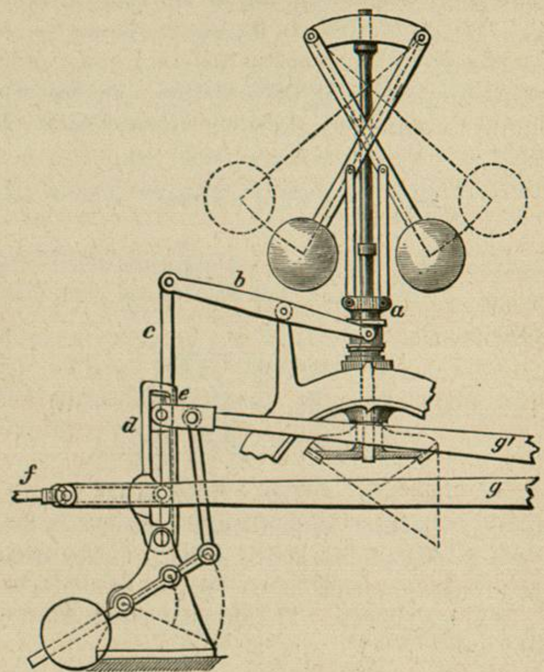

Fig. 81.

engines the throttle valve is, however, not interfered with during the working, and the governor is arranged to act directly on the expansion gear of the slide valves. Fig. 81 shows a simple method of effecting this object. The collar

a, on the vertical spindle of the governor, works a lever *b*, which is connected by a link *c* with the end of the sliding block *d* which works in a rocking link as shown. The block *d* is attached to the end of the eccentric rod *g′*. On the position of *d* depends the amount of swing given to the rocking link. An expansion valve, working on the back of the main valve (see p. 263) is driven from another point of the rocking link, and on the travel of this expansion valve depends the point at which the steam is cut off. The main valve is driven by the eccentric rod *g* in the ordinary manner.[1]

The forms of governors are so numerous, that it has been impossible here to do more than explain the principles upon which they act.

Locomotive engines are never fitted with governors, but in marine engines they are very necessary, as racing may ensue whenever the propeller is partially out of water, or whenever the propeller or crank shaft may give way. On account of the motion on board ship, the forms of governors used on land engines could not be employed for marine purposes. Marine governors are of two principal sorts, viz. those that are actuated by variations in the water pressure at the stern of the ship, and those which depend for their motion on variations in the velocity of the engine. The former class only provide for cases due to the incomplete immersion of the propeller, but the latter will guard against every contingency. In consequence of the great size of the throttle valves and expansion gear of marine engines, an ordinary governor cannot conveniently be employed to act directly on the controlling parts; hence, in this class of engines, what are called steam governors are now generally employed. The governor proper is arranged to move the slide valve of a small steam cylinder, which, in its turn, actuates the throttle valve.

[1] This will be better understood after reading the succeeding chapter on valve gearing.

Fly-wheels.—The functions of fly wheels have been explained in Chapter V., pp. 157 and 196. It is only necessary now to consider the principles involved in their construction.

The greater portion of the mass of a fly wheel is concentrated in its rim, and when revolving, every particle of the rim is under the action of centrifugal force, and tends to fly away radially from the centre; hence the rim, when in a state of revolution, resembles the condition of a ring put in a state of tension by a force from within acting outwards. The tension developed in the rim is opposed by the tensile strength of the metal of which it is formed, and should the former exceed the latter the rim will inevitably burst asunder, just as a boiler would burst if the steam pressure were too great for the strength of the shell plates.

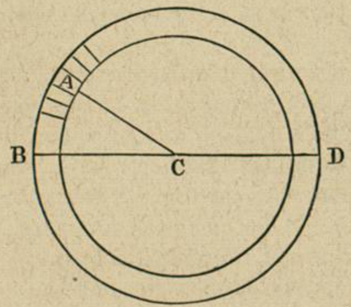

Fig. 82.

Suppose the rim in fig. 82 to be divided up into a number of segments. The centrifugal force on any one of them, such as A, acts radially along the line AC, and may be resolved into two components, one along the diameter BD, and the other at right angles to it, and similarly for all the other segments. The sum of all the components at right angles to BD is the force which tends to tear the ring asunder at the sections B and D.

It is well known that if a force press uniformly outwards all along the semicircumference of a ring, the components at right angles to a given diameter equal the total radial force multiplied by the ratio of the diameter to the semicircumference. In the case of a fly wheel, the radial pressure acting along any given semicircumference is half the

centrifugal force of the entire wheel, and the sum of the components at right angles to the diameter BD

$$= \tfrac{1}{2} \text{ centrifugal force of wheel} \times \frac{2r}{\pi r}.$$

Hence, the tension at either B, or at D, is half the above quantity=half centrifugal force of wheel $\times \frac{1}{\pi}$.

If W=the weight of the wheel in lbs., r its mean radius in feet, and N the number of revolutions per minute, we have (see p. 161)

$$\text{Tension at B, or at D} = \frac{W \times N^2 \times r \times \cdot 00034}{2 \times \pi}.$$

Example.

A fly wheel, the mean radius of which is 8 feet, weighs 15,000 lbs., the whole of which weight is supposed to act at the mean radius; what is the tension on the metal of the rim at either end of any diameter, the wheel making 60 revolutions per minute?

Answer. $\text{Tension} = \frac{15000 \times 60 \times 60 \times 8 \times \cdot 00034}{2 \times 3 \cdot 14159} = 23376$ lbs.

Taking the tensile strength of cast iron at 15,680 lbs. per square inch and allowing a factor of safety of 5, it is evident that the section of the rim of the wheel must not be less than about 8 inches. In order to attain the given weight, the section of the rim would have to be far greater than this quantity, hence the rim would possess ample strength.

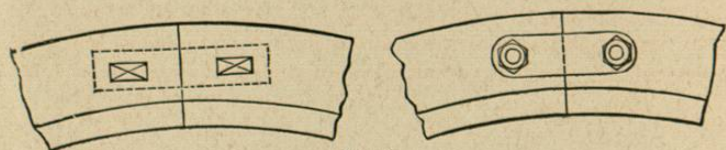
Fig. 83.

Small fly wheels are usually cast in one piece, but when so large that the weight would be unwieldy, the rim is cast in pieces which are afterwards put together by bars and cottars, or by bars and bolts, as shown in fig. 83.

The rim in very large wheels is generally fastened to the arms, as shown in fig. 84. The arms are fastened to the

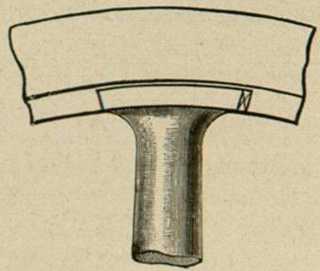

Fig. 84.

boss in a similar manner, and the latter, in the case of large wheels, is often cast in two halves, which are either bolted together or attached by wrought-iron rings shrunk on.

CHAPTER VII.

VALVES AND VALVE GEARS.

Action of the simplest form of D slide valve driven by single eccentric—Definitions of 'lap' and 'lead'—Position of eccentric as affected by the lap and lead of the valve—Effect on the steam distribution of the lap and lead of the valve—Effect of ratio of length of connecting rod to length of crank in modifying steam distribution—Means of varying the rate of expansion and of reversing—Stephenson's link motion—Effect of diminishing the throw of the eccentric—Reversing lever—Ramsbottom's reversing screw—Variations in the details of Stephenson's link motion—Other systems of link motion—Other means of varying the rate of expansion — Meyer's separate expansion valve —Corliss's valve gear — Varieties of valves—Valve gears in which eccentrics are dispensed with—Joy's gear—Geometrical representations of the action of slide valves—Zeuner's valve diagrams—Case of valve without lap or lead—Case of valve with lap and lead—Problems on simple valve setting—Zeuner's diagrams applied to valves driven by link motions—Analytical method of fixing centres of valve circles—Graphical method—Problems in link motion—The method of suspending link motions—Zeuner's diagrams applied to Meyer's valve gear—Reversing by Meyer's gear—Problems on valve setting with Meyer's gear.

THE successful and economical working of a steam engine depends in a very large degree upon the design and adjustment of the valve or valves which regulate the distribution of the steam in the cylinders. The subject is, perhaps, more complicated in its nature than any other question affecting the design of the engine. In order to treat it simply and, at the same time, systematically, it is intended in this chapter, first to explain the simplest examples, and then to proceed to the description of the cases which more frequently occur in practice.

Action of the simplest form of slide valve driven by a single eccentric.—As the motion of a slide valve is modified by the length of the connecting and eccentric rods relatively to the