and so acts on the regulator in its effort to increase the speed of *b.*

Another example of the differential type is the Allen@@1 governor, which has a fan directly geared to the engine, revolving in a case containing a fluid. The case is also free to turn, except that it is held back by a weight or spring and is connected to the regulator. So long as the speed of the fan is constant, the moment required to keep the case from turning does not vary, and consequently the position of the regulator remains unchanged. When the fan turns faster the moment increases, and the case has to follow it (acting on the regulator) until the spring which holds the case from turn­ing is sufficiently extended, or the weight raised. The term “ dy­namometric governor” is equally applicable to this form ; the power required to drive the fan is regulated by an absorption-dynamometer in the case instead of by a transmission-dynamometer between the engine and the fan. In Napier’s governor the case is fixed, and the reaction takes place between one turbine-fan which revolves with the engine and another close to it which is held from turning by a spring and is connected with the regulator.

180. Pump governors form another group closely related to the differential or dynamometric type. An engine may have its speed regulated by working a small pump which supplies a chamber from which water is allowed to escape by an orifice of constant size. When the engine quickens its speed water is pumped in faster than it can escape, and the accumulation of water in the chamber may be made to act on the regulator through a piston controlled by a spring or in other ways. This device has an obvious analogy to the cataract of the Cornish pumping-engine (§ 163), whieh has, however, the somewhat different purpose of introducing a regulated pause at the end of each stroke. The “differential valve-gear” invented by Mr H. Davey, and successfully applied by him to modern pumping-engines, combines the functions of the Cornish cataract with that of a hydraulic governor for regulating the expansion.@@2 In this gear, which is shown dia-

gramatically in fig. 107, the valve-

rod of the engine (*a*) receives its

motion from a lever *b,* one end of

which (c) copies, on a reduced

scale, the motion of the engine

piston, while the other (*d*), which

forms (so to speak) the fulcrum,

has its position regulated by at­

tachment to a subsidiary piston-

rod, which is driven by steam in a

cylinder *e,* and is forced to travel

uniformly by a cataract *f*. The point of cut-off is determined by the rate at which the main piston overtakes the cataract piston, and consequently comes early with light loads and late with heavy loads.

181. The government of marine engines is peculiarly difficult on account of the sudden and violent fluctuations of load to which they are subjected by the alternate uncovering and submersion of the screw in a heavy sea. However rapidly the governor responds to increase of speed by closing the throttle-valve, an excess of work is still done by the steam in the valve-chest and in the high-pres­sure cylinder. To check the racing which results from this, it has been proposed to supplement the control which the throttle-valve on the steam-pipe exercises by throttling the exhaust or by spoiling the vacuum. Probably a better plan is that of Messrs Jenkins and Lee, who give supplementary regulation by causing the governor to open a shunt-valve which connects the top and bottom of the low- pressure cylinder, thus allowing a portion of the steam in it to pass the piston without doing work. In Dunlop’s pneumatic governor@@3 an attempt is made to anticipate the racing of the screw by caus­ing the regulator to be acted on by the changes of pressure on a diaphragm which is connected by an air-pipe with an open vessel fixed under the stern of the ship. A plan has recently been intro­duced by Mr W. B. Thompson to prevent the racing of marine engines by working the valves from a lay shaft which is driven at a uniform speed by an entirely independent engine. So long as this lay shaft is not driven too fast the main engine is obliged to follow it; if the lay shaft is driven faster than the main engine can follow the main engine pauses so as to miss a stroke, and then goes on. Reversing the motion of the lay shaft reverses the main engine.

182. In connexion with governors mention may be made of an apparatus introduced by Mr Moscrop to give a continuous record of fluctuations in the speed of engines.@@4 It resembles a small centrifugal governor, but the displacement of the balls actuates, not a regulator, but a pencil which moves transversely on a ribbon of paper that is moved continuously by clockwork. The recorder responds so rapidly to changes of speed as to show not only the comparatively slow changes which occur from stroke to stroke, but also those short-period fluctuations between a maximum and

minimum, within the limits of each single stroke, which will be discussed in the next chapter.

X. The Work on the Crank-Shaft.

183. Besides those variations of speed which occur from stroke to stroke, which it is the business of the governor to check, there are variations within each single stroke over which the governor has of course no control. These are due to the varying rate at which work is done on the crank-shaft during its revolution. To limit them is the function of the fly-wheel, which acts by forming a reservoir of energy to be drawn upon during those parts of the revolution in which the work done on the shaft is less than the work done by the shaft, and to take up the surplus in those parts of the revolution in which the work done on the shaft is greater than the work done by it. This alternate storing and restoring of energy is accomplished by slight fluctuations of speed, whose range depends on the ratio which the alternate excess and defect of energy bears to the whole stock the fly-wheel holds in virtue of its motion. The effect of the fly-wheel may be studied by drawing a diagram of crank-effort, which shows the work done on the crank in the same way that the indicator diagram shows the work done on the piston. The same diagram serves another useful purpose in determining the twisting and bending stress in the crank.

184. The diagram of crank-effort is best drawn by representing, in rectangular co-ordinates, the relation between the moment which the connecting-rod exerts to turn the crank and the angle turned through by the crank. When the angle is expressed in circular measure, the area of the diagram is the work done on the crank.

Neglecting friction, and supposing in the first place that the engine runs so slowly that the forces required for the acceleration of the moving masses are negligibly small, the moment of crank- effort is found by resolving the thrust P of the piston-rod into a

component Q along the connecting-rod and a component O normal to the surface of the guide (fig. 108). The moment of crank- effort is

(rcosα ∖

1 + ^^=^=τ)'

√i2- r2sιn⅛∕

where CN is drawn perpendicular to the centre line or travel of the piston, *r* is the crank, *l* the connecting rod, and *α* the angle ACB which the crank makes with the centre line. A graphic deter­mination of CN is the most convenient in practice, unless the con­necting rod is so long that its obliquity is negligible, when the second term in

the above expres­

sion vanishes.

Fig. 109 shows

the diagram of

crank-effort de­

termined in this

way for an engine

whose connect­

ing-rod is 31/2

times the length of its crank, and in which steam is cut off at half-stroke. The thrust P is determined from the indicator dia­grams of fig. 108 by taking the excess of the forward pressure on one side of the piston over the back pressure on the other side, and multiplying this effective pressure by the area of the piston. The area of the diagram of crank-effort is the work done per revolution.

185. The friction of the piston in the cylinder and the piston- rod in the stuffing-box is easily allowed for, when it is known, by making a suitable deduction from P. Friction at the guides, at the crosshead, and at the crank-pin has the effect of making the stress at each of these places be inclined to the rubbing surfaces at an angle *φ,* the angle of repose, whose tangent is the coefficient of friction. Hence O, instead of being normal to the guide, is inclined at the angle *φ* in the direction which resists the piston’s motion (fig. 110) ; and the thrust along the connecting-rod, instead of passing through the centre of each pin, is displaced far enough to make an angle *φ* with the radius at the point where it meets the pin’s surface. To satisfy this condition let a “friction-circle be drawn about the centre of each pin, with radius equal to *a* sin *φ,*

@@@1 *Proc. Inst. Mech Eng.,* 1873.

@@@2 *Proc. Inst. Mech. Eng,* 1874.

@@@3 *Ibid.,* 1879.

@@@4 *Ibid.,* 1884.