bottom of the cylinder, at the ends, and take their motion from a separate wrist-plate which oscillates on the same pin with the plate A.[1]

175. Fig. 103 shows a compact form of trip-gear by Dr Proëll. A rocking-lever *ab* is made to oscillate on a fixed pin through its centre by a connexion to the crosshead of the engine. When the end *a* rises, the bell-crank lever *c* engages the lever *d*, and when *a* is depressed the lever *l* is forced down and the valve *e* is opened to admit steam to one end of the cylinder. As *a* continues moving down a point is reached at which the edge of *c* slips past the edge of *d*, and the valve is then forced to its seat by a spring in the dash-pot *f*. This disengagement occurs early or late according to the position of the fulcrum piece *g*, on which the heel of the bell-crank *c* rests during the opening of the valve. The position of *g* is determined by the governor. A similar action, occurring at the other end of the rocking-bar *ab*, gives steam to the other end of the cylinder. In one form of Proëll's gear both ends of *ab* act on the same steam-valve, which is then a separate expansion-valve fixed on the back of a chest in which an ordinary slide-valve works.

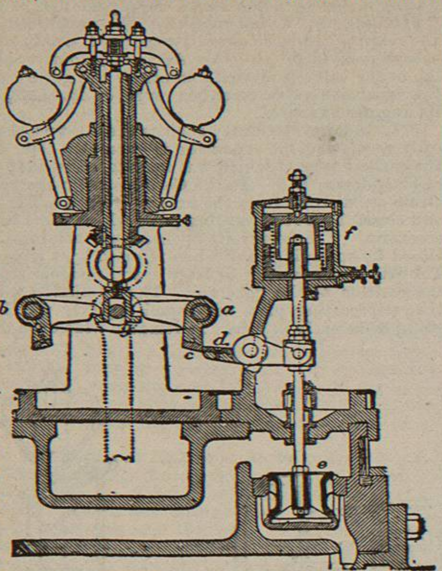

Fig. 103.—Proëll's Automatic Expansion Gear.

176. In the ordinary form of centrifugal governor the position of the throttle-valve, or the expansion-link, or the Corliss trigger depends on the configuration of the governor, and is definite for each position of the balls. In disengagement governors, of which the governor A shown on the right-hand side in fig. 104 is an example, any reduction of speed below a certain value sets the regulating mechanism in motion, and the adjustment continues until the speed has been restored. Similarly a rise of speed above a certain value sets the regulating mechanism in motion in the other direction. If the spindle *a* (fig. 104) is connected to the regulator so as to give more steam if it turns one way and less if it turns the other, the speed at which the engine will run in equilibrium must lie between narrow limits, since at any speed high enough to keep *b* in gear with *a* the supply of steam will go on being reduced, and at any speed low enough to bring *c* into gear with *a* the supply will go on being increased. This mode of governing, besides being sensibly isochronous, has the advantage that the power of the governor is not limited by the controlling force on the balls, since the governor acts by deflecting a portion of the power that is being developed by the engine to the work of moving the regulator. It is rarely applied to steam-engines, probably because its action is too slow. This defect has been ingeniously remedied in the supplementary governor of Mr W. Knowles, who has combined a disengagement governor with one of the ordinary type in the manner shown in fig. 104.[2] Here the spindle *a*, driven by the supplementary or disengagement governor A, acts by lengthening the rod *d* which connects the ordinary governor B with the regulator. It does this by turning a coupling nut *e* which unites two parts of *d*, on which right- and left-handed screws are cut. Any sudden fluctuation in speed is immediately responded to by the ordinary governor. Any more or less permanent change of load or of steam-pressure gives the supplementary governor time to act. It goes on adjusting the supply until the normal speed is restored, thereby converting the control of the ordinary governor, which is stable, and therefore not isochronous, into a control which is isochronous as regards all fluctuations of long period. The power of the combination is limited to that of the common governor B.

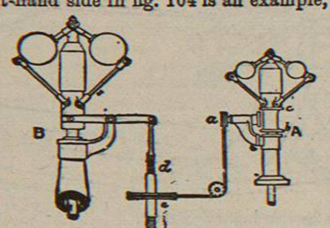

Fig. 104.—Knowles's Supplementary Governor.

177. Other governors which deserve to be classed as disengagement governors are those in which the displacement of the governor affects the regulator, not directly by a mechanical connexion, but by admitting steam or other fluid into what may be called a relay cylinder, whose piston acts on the regulator. In order that a governor of this class should work without hunting, the piston and valve of the relay cylinder should be connected by what is termed differential gear, the effect of which is that for each displacement of the valve by the governor the piston moves through a distance proportional to the displacement of the valve. An example of differential gear is shown in fig. 105. Suppose that the rod *a* is connected with the governor so that it is raised by an acceleration of the engine's speed. The rod *c* which leads from the relay piston *b* to the regulator serves as a fulcrum, and the valve-rod *d* is consequently raised. This admits steam to the upper side of the piston and depresses the piston, which pulls down *d* with it, since the end of *a* now serves as a fulcrum. Thus by the downward movement of the piston the valve is again restored to its middle position and the action of the regulator then ceases until a new change of speed occurs. A somewhat similar differential contrivance is used in steam-steering engines to make the position of the rudder follow, step by step, every movement of the hand-wheel;[3] also, in the steam reversing gear which is applied to large marine engines, to make the position of the drag-link follow that of the hand-lever; and also in certain electrical governors.[4] The effect of adding a differential gear such as this to a relay governor or other disengagement governor is to convert it from the isochronous to the stable type.

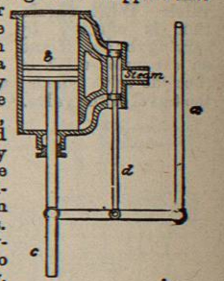

Fig. 105.—Differential Gear for Relay Governor.

Differential dynamometric governors.

178. Another group of governors is best exemplified by the "differential" governor of the late Sir W. Siemens[5] (fig. 106). A spindle *a* driven by the engine drives a piece *b* (whose rotation is resisted by a friction brake) through the dynamometer coupling *c*, consisting of a nest of bevel-wheels and a loaded lever *d*. So long as the speed remains constant the rate at which work is done on the brake is constant and the lever *d* is steady. If the speed accelerates, more power has to be communicated to *b*, partly to overcome the inertia and partly to meet the increased resistance of the brake, and the lever *d* is displaced. The lever *d* works the throttle-valve or other regulator, either directly or by a steam relay. The governor is isochronous when the force employed to hold *d* in position does not vary; if the force increases when *d* is displaced, the governor is stable. A governor of this class may properly be called a dynamometric governor, since it regulates by endeavouring to keep constant the rate at which energy is transmitted to the piece *b*. In one form of Siemens's governor the friction-brake is replaced by a sort of centrifugal pump, consisting of a paraboloidal cup, open at the top and bottom, whose rotation causes a fluid to rise in it and escape over the rim when the speed is sufficiently great. Any increase in the cup's speed augments largely the power required to turn it, and consequently affects the position of the piece which corresponds to *d*.[6] Siemens's governor is not itself used to any important extent, but the principle it embodies finds application in a number of other forms.

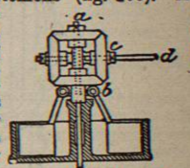

Fig. 106.—Siemens's Governor.

179. The "velometer" or marine-engine regulator of Messrs Durham and Churchill[7] is a governor of the same type. In it the rotation of a piece corresponding to *b* is resisted by means of a fan revolving in a case containing a fluid, and the coupling piece which is the mechanical equivalent of *d* in fig. 106 acts on the throttle-valve, not directly but through a steam relay. In Silver's marine governor[8] the only friction-brake that is provided to resist the rotation of the piece which corresponds to *b* is a set of air-vanes. The inertia is, however, very great, and any acceleration of the engine's speed consequently displaces the dynamometer coupling, and so acts on the regulator in its effort to increase the speed of *b*.

[1] Numerous forms of Corliss gears are illustrated in W. H. Uhland's work on Corliss engines, translated by A. Tolhausen (London, 1879). A more recent form of gear by Mr Inglis is described in *Engineering*, vol. xl. p. 251.
[2] *Proc. Inst. Mech. Eng.*, 1884.
[3] See a paper by Mr J. MacFarlane Gray, *Proc. Inst. Mech. Eng.*, 1867.
[4] Willans, *Min. Proc. Inst. C.E.*, vol. lxxxi. p. 166.
[5] *Proc. Inst. Mech. Eng.*, 1853.
[6] *Proc. Inst. Mech. Eng.*, 1866; or *Phil. Trans.*, 1866.
[7] *Proc. Inst. Mech. Eng.*, 1879.
[8] *Brit. Ass. Rep.*, 1859, p. 123.

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), which 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 diagramatically 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 attachment 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.

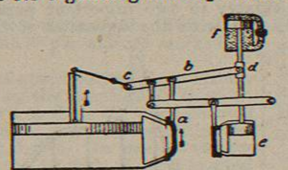

Fig. 107.—Davey's Differential Valve-Gear.

Government of marine engines.

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

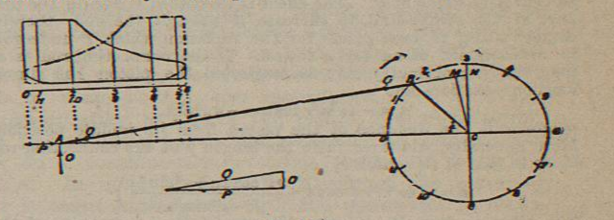

Fig. 108.

$$Q \cdot CM = P \cdot CN = Pr \sin \alpha \left(1 + \frac{r \cos \alpha}{\sqrt{l^2 - r^2 \sin^2 \alpha}}\right),$$

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 expression vanishes. Fig. 109 shows the diagram of crank-effort determined in this way for an engine whose connecting-rod is $3\frac{1}{2}$ 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.

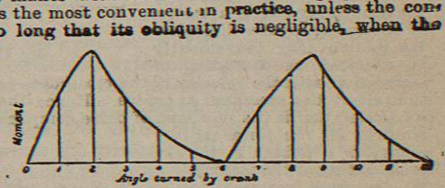

Fig. 109.—Diagram of Crank-Effort.

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.

where a is the actual radius of the pin. Any line drawn tangent to this circle will make the angle ϕ with the radius of the pin at the surface of the pin. The thrust of the connecting-rod must be tangent to both circles; it is drawn as in fig. 110, so that it resists the rotation of the pins relatively to the rod. The direction of rotation of the pins is shown by curved arrows in the figure, where the friction-circles are drawn to a greatly exaggerated scale. Finally, P (after allowing for the friction of piston-packing and stuffing-box) is resolved into O and Q, and Q·CM, the moment of Q on the shaft, is determined. This gives a diagram of crank-effort, correct so far as friction affects it, whose area is no longer equal to that of the indicator diagram. The difference, however, does not represent the whole work lost through friction of the mechanism, since the friction of the shaft itself, and of those parts of the engine which it drives, has still to be allowed for if the frictional efficiency of the engine as a whole is in question.

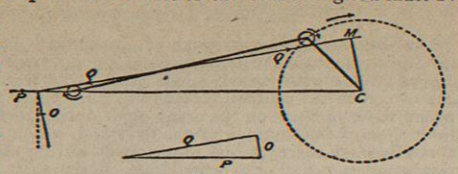

Fig. 110.

186. The diagram of crank-effort is further modified when we take account of the inertia of the piston and connecting-rod. For the purpose of investigating the effects of inertia, we may assume that the crank is revolving at a sensibly uniform rate of n turns per second. Let M be the mass of the piston, piston-rod, and crosshead in pounds, and a its acceleration at any instant in feet per second per second. The force required to accelerate it is Ma/g, in pounds-weight, and this is to be deducted in estimating the effective value of P. The effect is to reduce P during the first part of the stroke and to increase it towards the end, thereby compensating to some extent for the variation which P undergoes in consequence of an early cut-off. If the connecting-rod is so long that its obliquity may be neglected the piston has simple harmonic motion, and

$$a = -4\pi^2n^2r\cos\alpha = -4\pi^2n^2x,$$

where x is the distance of the piston from its middle position. More generally, whatever ratio the length l of the connecting-rod bears to that of the crank r,

$$a = -4\pi^2n^2r\left(\cos\alpha + \frac{rl^2\cos2\alpha + r^3\sin^4\alpha}{(l^2-r^2\sin^2\alpha)^{\frac{3}{2}}}\right).$$

The effect is to make, on the diagram of P, a correction of the character shown in fig. 111, where the broken line cd refers to the case of an indefinitely long connecting-rod and the full line ab to the case of a connecting-rod $3\frac{1}{2}$ times the length of the crank. In a vertical engine the weight of the piston and piston-rod is to be added to or subtracted from P.

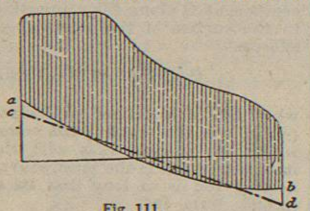

Fig. 111.

To allow for the inertia of the connecting-rod is a matter of somewhat greater difficulty. Its motion may conveniently be analysed as consisting of translation with the velocity of the crosshead, combined with rotation about the crosshead as centre. Hence the force required for its acceleration is the resultant of three components—F_1, the force required for the linear acceleration a (which is the same as that of the piston); F_2, the force required to cause angular acceleration about the crosshead; and F_3, the force towards the centre of rotation, which depends on the angular velocity, and is equal and opposite to the so-called centrifugal force. Let θ be the angle BAC (fig. 112), $\dot\theta$ the angular velocity of the rod about A, and $\ddot\theta$ its angular acceleration, and let M′ be the mass of the rod. Then

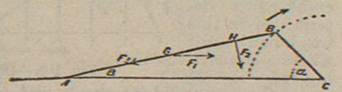

Fig. 112.

$$F_1 = M'a/g,$$

and acts through the centre of gravity G, parallel to AC;

$$F_2 = M'.\,AG.\,\ddot\theta/g,$$

and acts at right angles to the rod through the centre of percussion H;

$$F_3 = M'.\,AG.\,\dot\theta^2/g,$$

and acts along the rod towards A. Also,

$$\dot\theta = \frac{2\pi nr\cos\alpha}{\sqrt{l^2-r^2\sin^2\alpha}};$$

and

$$\ddot\theta = \frac{-4\pi^2n^2r\sin\alpha(l^2-r^2)}{(l^2-r^2\sin^2\alpha)^{\frac{3}{2}}}.$$

The moments of these forces about C are next to be found, and to be deducted from the moment of the thrust in the connecting-rod (and, if the weight of the rod is to be considered, its moment about C is to be added) in finding the resultant moment of crank-effort.

187. If, however, the friction at the crosshead and crank-pin is to be taken account of, the whole group of forces acting on the rod must be considered as follows. Compound forces equal and opposite to F_1, F_2, and F_3 into a single force R (fig. 113), which may be called the resultant resistance to acceleration of the connecting-rod. If the weight of the rod is to be considered, let it also be taken

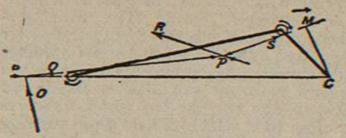

Fig. 113.

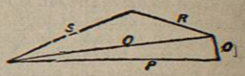

Fig. 114.

as a component in reckoning R. Then the rod may in any position be regarded as in equilibrium under the action of the forces Q, R, and S, where Q and S are the forces exerted on it by the crosshead and crank-pin respectively. These three forces meet in a point p in R, which is to be found by trial, the condition being that in the diagram of forces, fig. 114, after the triangle POQ has been drawn, and the force R set out, the force-line S shall be parallel to a line drawn from p tangent to the friction-circle of the crank-pin, as in fig. 113. When this condition has been satisfied by trial, the value of S, which is the thrust on the crank-pin, is determined, and S.CM is the moment of crank-effort. This method is due to the late Prof. Fleeming Jenkin, who has applied it with great generality to the determination of the frictional efficiency of machinery in two important papers,[1] the second of which deals in detail with the dynamics of the steam-engine. Fig. 115, taken

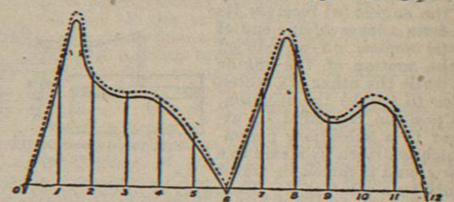

Fig. 115.

from that paper, shows the diagram of crank-effort in a horizontal direct-acting engine,—the full line with friction, and the dotted line without friction,—the inertia of the piston and connecting rod being taken account of, as well as the weight of the latter. It exhibits well the influence which the inertia of the reciprocating parts has in equalizing the crank-effort in the case of an early cut-off. The cut-off is supposed to occur pretty sharply at about one-sixth of the stroke. The engine considered is of practical proportions, and makes four turns per second; and the initial steam pressure is 50 lb per square inch. It appears from the diagram that, with a slightly higher speed, or with heavier rods, a better balance of crank-effort might be secured, especially as regards the stroke towards the crank, which comes first in the diagram; on the other hand, by unduly increasing the mass of the reciprocating pieces or their speed the inequality due to expansion would be over-corrected and a new inequality would come in.

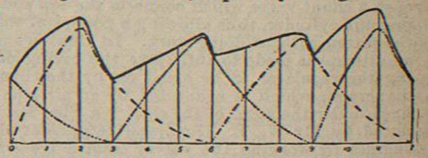

Fig. 116.—Crank-Effort Diagram for Two Cranks.

188. When two or more cranks act on the same shaft, the joint moment of crank-effort is found by combining the diagrams for the separate cranks, in the manner illustrated by fig. 116, which refers to the case of two cranks at right angles.

Another graphic method of exhibiting the variations of moment exerted on the crank-shaft during a revolution is to draw a circular diagram of crank-effort, in which lines proportional to the moment are set off radially from a circular line which represents the zero of moment. An example of this plan is given in fig. 117, which shows the resultant moment determined by Mr A. C. Kirk for one of his triple-expansion engines with three cranks set at 120° from each other. Curves are drawn for various speeds, giving in each case the resultant moment due to the steam pressure (as

[1] *Trans. Roy. Soc. Edin.*, vol. xxviii. p. 1 and p. 703.

determined from actual indicator diagrams) combined with the moments due to the inertia of the reciprocating parts. The line marked 0 is the steam line without inertia—or, in other words,

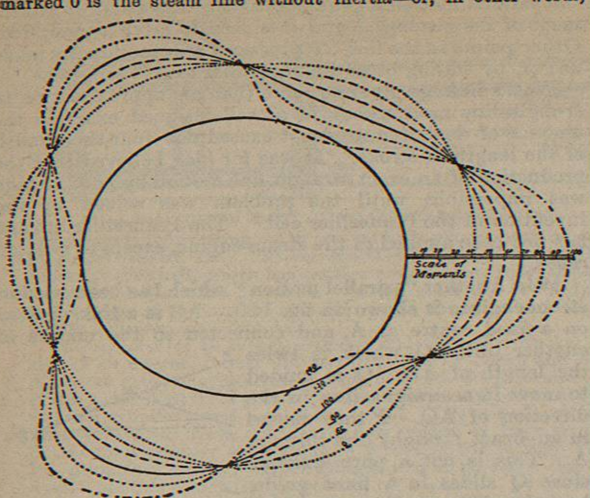

Fig. 117.—Circular Diagram of Crank-Effort for a Three-Cylinder Engine.

the curve corresponding to an indefinitely slow speed. The other curves refer to the number of revolutions per minute marked on them.

Fluctuations of speed. 189. To determine the fluctuations of speed during a revolution, the resultant diagram of work done *on* the crank-shaft is to be compared with a similar diagram drawn to show the work done *by* the shaft in overcoming its own friction, and in overcoming the resistance of the mechanism which it drives. In general the resistance may be taken as constant, and the diagram of effort exerted by the crank-shaft is then a straight line, as EFGHIJKL in fig. 118. At F, G, H, I, J, and K the rate at which work is being done on and by the shaft is the same; hence at these points the fly-wheel is neither gaining nor losing speed. The shaded area above FG is an excess of work done on the crank, and raises the speed of the fly-wheel from a minimum at F to a maximum at G. From G to H the fly-wheel supplies the defect of energy shown by the shaded area below GH, by which the demand for work exceeds the supply; its speed again reaches a minimum at H, and again a maximum at I. The excesses and defects balance in each revolution if the engine is making a constant number of turns per second. In what follows it is assumed that they are only a small fraction of the whole energy held by the fly-wheel.

Function of fly-wheel.

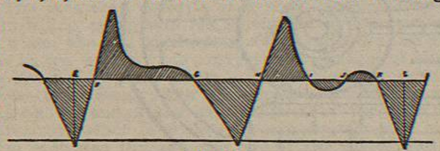

Fig. 118.

Let ΔE be the greatest single amount of energy which the fly-wheel has to give out or absorb, as determined by measuring the shaded areas of the diagram; and let ω_1 and ω_2 be the maximum and minimum values of the wheel's angular velocity, which occur at the extremes of the period during which it is storing or supplying the energy ΔE. The mean angular velocity of the wheel ω_0 will be sensibly equal to $\frac{1}{2}(\omega_1+\omega_2)$ if the range through which the speed varies is moderate. Let E_0 be the energy of the fly-wheel at this mean speed. Then

$$E_0 = \tfrac{1}{2}I\omega_0^2,$$

where I is the moment of inertia of the fly-wheel.

$$\Delta E = \frac{I(\omega_1^2-\omega_2^2)}{2} = I\omega_0(\omega_1-\omega_2) = 2E_0.\,\frac{(\omega_1-\omega_2)}{\omega_0}.$$

The quantity $\frac{\omega_1-\omega_2}{\omega_0}$, which we may write q, is the ratio of the extreme range of speed to the mean speed, and measures the degree of unsteadiness which the fly-wheel leaves uncorrected. If the problem be to design a fly-wheel which will keep q down to an assigned limit, the energy of the wheel must be such that

$$E_0 = \frac{\Delta E}{2q}.$$

The Moscrop recorder, alluded to in § 182, exhibits the degree of unsteadiness during a single revolution by the width of the line which it draws. On the other hand, any bending of the line implies the quite independent characteristic of unsteadiness from one revolution to another. The former is due to insufficient fly-wheel energy, the latter to imperfect governing.

190. An interesting consequence of the periodic alternations in crank-effort which occur in each revolution has been pointed out by Mr M. Longridge.[1] The fly-wheel receives its alternate acceleration and retardation through changes of the torsional stress in the shaft. If these occur at intervals nearly equal to the period of free torsional vibration which the fly-wheel possesses in virtue of the torsional elasticity of the shaft between it and the crank, strains of great amplitude will arise; and Mr Longridge has suggested that this may account for the observed fact that engine-shafts have been ruptured when running so that the fluctuations of crank-effort occurred with one particular frequency, although the greatest effort was itself much less than the shaft would safely bear.

XI. Examples of Steam-Engines.

Stationary Engines.

191. In classifying engines with regard to their general arrangement of parts and mode of working, account has to be taken of a considerable number of independent characteristics. We have, first, a general division into *condensing* and *non-condensing engines*, with a subdivision of the condensing class into those which act by surface condensation and those which use injection. Next there is the division into *compound* and *non-compound*, with a further classification of the former as double-, triple-, or quadruple-expansion engines. Again, engines may be classed as *single* or *double-acting*, according as the steam acts on one or alternately on both sides of the piston. Again, a few engines—such as steam-hammers and certain kinds of steam-pumps—are *non-rotative*, that is to say, the reciprocating motion of the piston does work simply on a reciprocating piece; but generally an engine does work on a continuously revolving shaft, and is termed *rotative*. In most cases the crank-pin of the revolving shaft is connected directly with the piston-rod by a connecting-rod, and the engine is then said to be *direct-acting*; in other cases, of which the ordinary beam-engine is the most important example, a lever is interposed between the piston and the connecting-rod. The same distinction applies to non-rotative pumping engines, in some of which the piston acts directly on the pump-rod, while in others it acts through a beam. The position of the cylinder is another element of classification, giving *horizontal*, *vertical*, and *inclined cylinder* engines. Many vertical engines are further distinguished as belonging to the *inverted cylinder* class; that is to say, the cylinder is above the connecting-rod and crank. In *oscillating cylinder engines* the connecting-rod is dispensed with; the piston-rod works on the crank-pin, and the cylinder oscillates on trunnions to allow the piston-rod to follow the crank-pin round its circular path. In *trunk engines* the piston-rod is dispensed with; the connecting-rod extends as far as the piston, to which it is jointed, and a trunk or tubular extension of the piston, through the cylinder cover, gives room for the rod to oscillate. In *rotary* engines there is no piston in the ordinary sense; the steam does work on a revolving piece, and the necessity is thus avoided of afterwards converting reciprocating into rotary motion.

192. In the single-acting atmospheric engine of Newcomen the beam was a necessary feature; the use of water-packing for the piston required that the piston should move down in the working stroke, and a beam was needed to let the counterpoise pull the piston up. Watt's improvements made the beam no longer necessary; and in one of the forms he designed it was discarded—namely, in the form of pumping-engine known as the Bull engine, in which a vertical inverted cylinder stands over and acts directly on the pump-rod. But the beam type was generally

[1] *Proc. Inst. Mech. Eng.*, May 1884, p. 163.

retained by Watt, and for many years it remained a favourite with builders of engines of the larger class. The beam formed a convenient driver for pump-rods and valve-rods; and the parallel motion invented by Watt as a means of guiding the piston-rod, which could easily be applied to a beam-engine, was, in the early days of engine-building, an easier thing to construct than the plane surfaces which are the natural guides of the piston-rod in a direct-acting engine. In modern practice the direct-acting type has to a very great extent displaced the beam type. For mill-driving and the general purposes of a rotative engine the beam type is now rarely chosen. In pumping engines it is more common, but even there the tendency is to use direct-acting forms.

193. The only distinctive feature of beam-engines requiring special notice here is the "parallel motion," an ordinary form of which is shown diagrammatically in fig. 119. There MN is the path in which the piston-rod head, or crosshead, as it is often called, is to be guided. ABC is the middle line of half the beam, C being the fixed centre about which the beam oscillates.

FIG. 119.—Watt's Parallel Motion.

A link BD connects a point in the beam with a radius link ED, which oscillates about a fixed centre at E. A point P in BD, taken so that BP : DP :: EN : CM, moves in a path which coincides very closely with the straight line MPN. Any other point F in the line CP or CP produced is made to copy this motion by means of the links AF and FG, parallel to BD and AC. In the ordinary application of the parallel motion a point such as F is the point of attachment of the piston-rod, and P is used to drive a pump-rod. Other points in the line CP produced are occasionally made use of, by adding other links parallel to AC and BD.[1]

Watt's linkage gives no more than an approximation to straight-line motion, but in a well-designed example the amount of deviation need not exceed one four-thousandth of the length of stroke. It was for long believed that the production of an exact straight-line motion by pure linkage was impossible, until the problem was solved by the invention of the Peaucellier cell.[2] The Peaucellier linkage has not been applied to the steam-engine, except in isolated cases.

194. Another "parallel motion" which has been used in steam-engines is shown in fig. 120. AB is a link pivoted on a fixed centre at A, and connected to the middle of another link PQ, which is twice the length of AB. Q is guided to move in a straight line in the direction of AQ. P then moves in an exact straight line through A. This is not a pure linkage, since Q slides in a fixed guide, but the distance through which

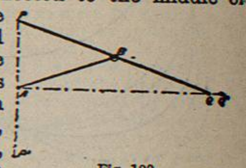

Fig. 120.

Q has to be guided is small compared with the stroke of P. If Q is guided to move in the arc of a circle of large radius, by using a radius rod from a fixed centre above or

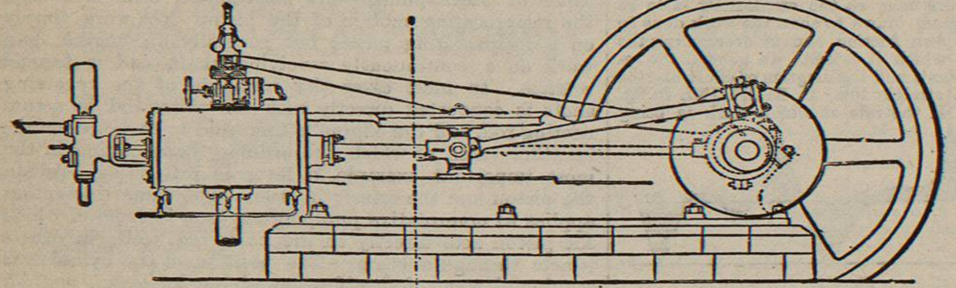

Fig. 121.

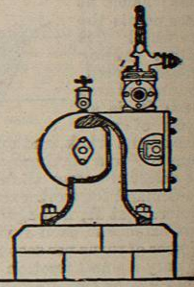

Fig. 123.

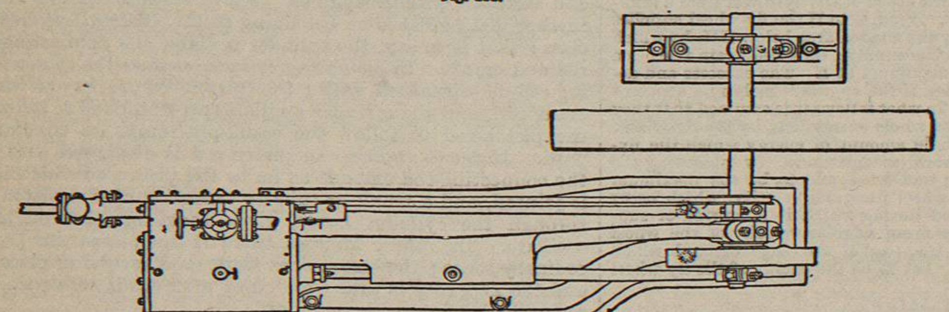

Fig. 122.

FIG. 121.—Small Horizontal Direct-Acting Steam Engine: Side Elevation. FIG. 122.—Plan. FIG. 123.—Section on AB in fig. 121.

below it, the guiding surfaces at Q are avoided, but the path of P is then only very nearly straight. An example of the linkage in this form, with the further modification that A is shifted out, and B is brought nearer to P, occurs in the pumping engine of fig. 130 below.

In by far the greater number of modern steam-engines the crosshead is guided by a block sliding on planed surfaces. In many beam-engines, even, this plan of guiding the piston has taken the place of the parallel motion.

195. No type of steam-engine is so common as the horizontal direct-acting. A small engine of this type, made by Messrs Tangye, and rated as a 10-horse-power engine, is illustrated in figs. 121 to 124. It furnishes a good example of a very numerous class, and serves to illustrate the principal parts of a complete engine. Fig. 121 is a side elevation, fig. 122 a plan, fig. 123 a transverse section through the bedplate in front of the cylinder, on the line

[1] The kinematics of the parallel motion are discussed in Rankine's *Machinery and Mill Work*, p. 275, and rules are given for the proportions and positions of the parts.

[2] See Kempe's *How to Draw a Straight Line* ("Nature Series,"), 1877.

AB; and fig. 124 is a horizontal section through the cylinder, valve-chest, valve, stuffing-boxes, piston, and crosshead. The bedplate

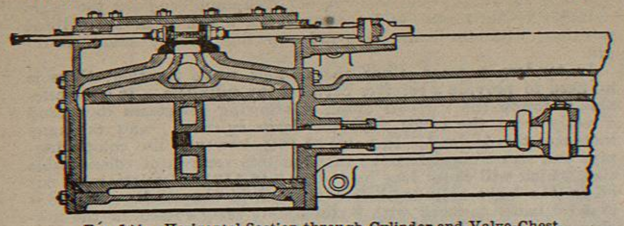

FIG. 124.—Horizontal Section through Cylinder and Valve-Chest.

is a single hollow casting, with two surfaces planed on it to serve as guides (see fig. 123). At one end the bedplate forms a pillow-block for the shaft, which has another main bearing independently supported beyond the fly-wheel. At the other end the bedplate is shaped so as to form the cylinder cover; the cylinder is bolted to this and overhangs the bed.

The cylinder (of 10 inches diameter and 20 inches stroke) consists of an internal "liner" of cast-iron, exactly bored, and fitted within an external cylindrical casting, of which the ports and sides of the valve-chest form part. The space between the liner and the external cylinder serves as a steam-jacket. The use of a separate liner within the main cylinder is now general in large engines. In the front cylinder cover there is a stuffing-box through which the piston-rod passes. The stuffing-box is kept steam-tight by a soft packing which is pressed in by a gland. In some instances the packing consists of metallic rings. The cylinder cover and gland are lined with a brass ring in the hole through which the piston-rod passes. The valve-rod is brought out of the valve-chest in the same way. The piston is a hollow casting into which the piston-rod is screwed and riveted over. It is packed by two split rings of cast-iron, which are sprung into recesses turned in the circumference of the piston. This mode of packing is used in locomotives and small engines. For large pistons the usual plan is to employ wider split rings, called floating rings, pressed against the sides of the cylinder, not by their own elasticity, but by separate springs behind them in the body of the piston; they are held in place by a movable flange called a junk-ring on one face of the piston. One example of the packing of a large piston is shown in fig. 134. The crosshead consists of a steel centre-piece with a round boss, in which the piston-rod is secured by a cotter, and a forked front, where the end of the connecting-rod works on a pin. A pair of pins at top and bottom carry the steel shoes or sliding-blocks, whose distance from the centre is adjustable by nuts to take up wear. There is no crank; the connecting-rod works on a pin fixed in a disk on the end of the shaft in front of the main bearing. The valve-rod, which is worked by an eccentric just behind the bearing, is extended through the end of the valve-chest, and forms the plunger of a feed-pump which is bolted to the end of the chest. Frequently the feed-pump is fixed at any convenient part of the bedplate, and is driven by a separate eccentric, and in some cases its plunger is connected directly to the crosshead. In the main bearing the shaft turns in gun-metal or phosphor-bronze blocks called brasses. In heavy engines these are generally lined with Babbit's anti-friction metal or other soft alloy, and in many modern engines the brasses are entirely dispensed with, a lining of Babbit's metal being let into the cast-iron surface of the bearing. When the brasses are in two pieces, the plane of division between them is chosen to be that in which the wear is likely to be least. A more satisfactory adjustment is possible when the brasses are in three or more pieces.

Condenser and air-pump. 196. When a condenser is used with a small horizontal engine it is usually placed behind the cylinder; and the air-pump, which is

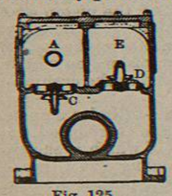

Fig. 125.

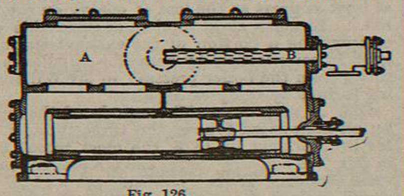

Fig. 126.

Condenser and Air-Pump.

within the condenser, is a horizontal plunger or piston-pump worked by a "tail-rod"—that is, a continuation of the piston-rod past the piston and through the back cover of the cylinder. Figs. 125 and 126 show in section one of Messrs Tangye's small condensers fitted with a double-acting air-pump to be driven by a tail-rod. The condenser proper is the chamber A, and into it the injection-water streams continuously through perforations in the pipe B, which has a cock outside to regulate the supply. The pump draws condensed water down to the lower part of the vessel at either end alternately through the valves C, and forces it up thence through the valves D to a chamber E, from which the delivery-pipe leads out. The pump is a gun-metal piston working in a cylinder fitted with a gun-metal liner. The valves are flat india-rubber rings held down in the centre by a spring, which allows them to open by rising bodily, as well as by bending.

197. The engine of figs. 121–4 makes 85 revolutions per minute, and its mean piston speed is consequently about 280 feet per minute. In some special forms of small horizontal engine the design is adapted to a much more rapid reciprocation of the moving masses, and the piston speed is raised to a value seldom exceeded in the largest land engines, although still higher values are now common in marine practice. Experience shows that the weight of engines of any one type varies roughly as the piston area. Their power depends on the product of piston area, piston speed, and pressure; and hence, so long as the pressures are similar, the ratio of power to weight is nearly proportional to piston speed. Cases present themselves in which it is desirable to make this ratio as great as possible; and, apart from this, an engine making a large number of revolutions per minute is a convenient motor for certain high-speed machines.

A good example of a small horizontal engine, specially designed by the symmetry and balance of its parts, by largeness of the bearing surfaces, and by very perfect lubrication, to stand the strains which are caused by high speed, is the Armington & Sims engine, made in America by the patentees and in England by Messrs Greenwood & Batley. The bedplate is symmetrical about the line of motion of the crosshead; it supplies two very long main bearings for the shaft, at each end of which there is an overhung fly-wheel. The bearings have an adjustable side-block to take up wear. They are formed entirely of white-metal, cast on to the cast-iron pillow-blocks. In the middle are two disks, forming crank-cheeks, which are weighted opposite the crank-pin, so that they balance the pin and that part of the connecting-rod which may be treated as having its mass applied there. The crank-pin and the crosshead-pin are wide enough to give a large bearing area. The crosshead-block is a hollow bronze casting, giving an exceptionally large surface of contact with the guides. The valve is a piston-valve of the Trick type, which works sufficiently tight without packing. The valve-rod and eccentric-rod are connected through a block which slides on a fixed guide. The governor, which has been already illustrated in fig. 100, is contained within one of the fly-wheels. An engine of this type, with a cylinder 12 inches in diameter and a stroke of 12 inches, makes 275 revolutions per minute, has a piston speed of 550 feet per minute, and indicates about 80 horse-power. Other good examples of high speed combined with double action are furnished by the Porter-Allen engine[1] and by the very light engines which Mr Thorneycroft and others have introduced for driving fans to supply air to the closed stokeholes of torpedo-boats. In these a speed of 1000 revolutions per minute is made possible by the use of light reciprocating parts and large bearing surfaces.

198. Fig. 127 shows a large non-compound horizontal Corliss engine for mill-driving, by Messrs Hick, Hargreaves, & Co. The cylinder is 34 inches in diameter, the stroke 8 feet, and the speed 45 revolutions per minute, giving a mean piston speed of 720 feet per minute. The cylinder is steam-jacketed round the barrel in the space between the liner and the outer cylinder, and also at the ends, which are cast hollow for this purpose. In large horizontal engines the weight of the piston tends to cause excessive wear on the lower side of the cylinder. In the example shown a part of the weight is borne by a tail-rod, ending in a block, which slides on a fixed guide behind the cylinder. To further diminish wear the piston is sometimes made much wider from front to back than the one shown here; and the device is sometimes resorted to of giving the piston-rod "camber"—that is to say, an upward curvature in the middle portion, which the weight of the piston reduces to straightness. Fig. 127 illustrates a common method of attaching the air-pump and condenser in large horizontal engines. The condenser is placed in a well in front of the cylinder, and the air-pump, which is a vertical bucket-pump, is worked by a bell-crank lever, connected with the crosshead by a link. The fly-wheel of this engine is grooved for rope-gearing; it is cast in segments, which are bolted to one another and to the spokes, and the spokes are secured by cotters in tapered sockets in the nave. It is large and heavy, to suit the inequality of driving effort which is caused by the use of a single cylinder and a very early cut-off in engines of this class. To facilitate starting and valve-setting, mill engines are often provided with an auxiliary called a "barring" engine. The barring engine turns a toothed pinion, which gears into a toothed rim in the fly-wheel, and is contrived to fall automatically out of gear as soon as the main engine starts. Large horizontal Corliss engine.

[1] *Proc. Inst. Mech. Eng.*, 1868.