the trip-gear acts. Messrs Sulzer have developed this type with much success. Many forms of trip-gear have been invented by Corliss himself and by others. One of these, the Spencer Inglis trip-gear, by Messrs Hick, Hargreaves & Co., is shown in figs. 45 and 46. A wrist-plate A, which turns on a pin on the outside of the cylinder, receives a motion of oscillation from an eccentric. It opens the cylindrical rocking-valve B by pulling the link C, which consists of two parts, connected to each other by a pair of spring clips *a, a.* Between the clips there is a rocking-cam *b,* and as the link is pulled down this cam places itself more and more athwart the link, until at a certain point it forces the clips open. Then the upper part of the link springs back and allows the valve B to close by the action of a spring in the dash-pot D. When the wrist-plate makes its return stroke the clips re-engage the upper portion of the link C, and things are ready for the next stroke. The rocking-cam *b* has its position controlled by the governor through the link E in such a way that when the speed of the engine increases it stands more athwart the link C, and therefore causes the clips to be released at an earlier point in the stroke. A precisely similar arrangement governs the admission of steam to the other end of the cylinder. The exhaust-valves are situated at the bottom of the cylinder, and receive an oscillating motion from a separate wrist-plate, behind A.

82. *Use of Flywheel.—*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 due to the varying rate at which work is done on the crank-shaft during its revolution. To limit these is the function of the flywheel, 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 flywheel holds in virtue of its motion. The effect of the flywheel 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.

The diagram of crank-effort is 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. The moment exerted to turn the crank is readily found when the direction and magnitude of the thrust exerted by the connecting-rod on the crank-pin is known for successive points in the revolution.

83. *Influence of the Inertia of the Reciprocating Masses.*—This thrust depends not only on the resultant pressure of the steam on the piston but also on the inertia of the reciprocating masses. The mass of the piston, piston-rod, cross-head, and to some extent that of the connecting-rod also, has to be started and stopped in each half revolution, and in high-speed engines the forces concerned in this action are so large as to affect most materially not only the distribu­tion of crank-effort, but also the stresses which the various parts have to be proportioned to bear. The calculation of the stresses due to inertia in high-speed engines consequently forms an essential part of engine design. Taking M to represent the whole reciprocat­ing mass, and *a* its acceleration at any instant in the direction of the piston’s motion, the force required to produce this acceleration is M*a*/*g*, and this quantity has to be deducted from the resultant pressure of the steam in finding the effective thrust. The effect is to reduce the effective thrust at the beginning of the stroke and to increase it at the end. The greatest acceleration *a* occurs in the extreme position of the piston, most distant from the crank-shaft centre, and its value there is *4π2n2r*(1 + *r*/*l*) where *r* is the radius of the crank, *l* the length of the connecting-rod and *n* the number of turns per second. When the piston is in the other extreme position, nearest to the shaft, the value of *a* is *4π2n2r(*1 *- r/l).* The exact calculation of inertia effects for the connecting-rod is com­plicated, but its influence on the thrust is approximately found by treating the mass of the rod as divided into two parts, one of which moves with the cross-head and is therefore an addition to the recipro­cating system, while the other moves with the crank-pin and is therefore an addition to the revolving system. The mass may be divided for this purpose into parts which are inversely proportional to the distances of the centre of gravity from the cross-head and crank-pin respectively. By com­bining diagrams of the steam thrust and of the forces due to inertia a diagram is obtained showing the true thrust through­out the stroke. Fig. 47 gives an example: there the line *ab* is drawn to show the inertia forces for an engine in which the con­necting-rod has 31/2 times the length of the crank. The straight line *cd* shows what the inertia force would be if the connecting-rod were treated as being so long that the deviation from simple-harmonic motion might be neglected.

The inertia of the reciprocating parts imposes a limit on the light­ness of engines of the piston and cylinder type. The proportion of weight to power is reduced by increasing mean piston speeds, but this process cannot be carried beyond a point at which the forces due to inertia become so great as to produce unsafely high alternating stresses in the piston-rods and other parts. In some torpedo-boat destroyers, where the reduction of weight has been carried as far as is practicable, the mean piston speed approaches 1200 ft. per minute with nearly 400 revolutions per minute and an 18-in. stroke. These engines develop 6000 h.p., and the weight of engines and boiler together is only 50 lb per indicated h.p. Such a figure is, however, to be regarded as exceptional; weights of 150 to 200 lb per h.p. are more usual even in conditions like those of high-speed cruisers where saving of weight is specially desirable..

84. *Balancing.—*Another aspect in which the inertia of the reciprocating parts is important is in regard to the balancing of the engine as a whole. Any forces required to accelerate the piston and its attached parts produce reaction on the frame and bed-plate of the engine, which will set up vibrational disturbances in the foundations and ground or the supporting structure. The object of balancing is to group the masses in such a manner that their inertia effects more or less neutralize one another. This is especially important in marine engines, where massive foundations are absent and where it may happen that the periodic impulses due to want of balance find some portion of the hull free to respond synchronously with vibrations so violent as to be inconvenient and even dangerous. Even in land engines a want of balance causes enough vibration to constitute a serious nuisance in the neighbourhood.

85. In considering the question of balance, the system of eccentric­ally revolving masses and the system of reciprocating masses have to be considered separately. A reciprocating mass such as a piston cannot be balanced by the use of revolving masses, for the forces which are due to the inertia of the piston necessarily act along the line of its stroke, while those due. to revolving masses are continually changing their direction. The inertia of each connecting-rod may be approximately treated by resolving its mass into two constituents, one of which moves with the crank-pin, and is therefore an addition to the revolving system, while the other moves with the cross-head, and is therefore an addition to the reciprocating system. The mass of the rod may be divided for this purpose into parts which are inversely proportional to the distances of its centre of gravity from the crank-pin and the cross-head respectively. Let M1, M2, M3, &c., represent the various revolving masses *r1*, r2, *r3*, &c., their effective radii of rotation, and *a1, a2 a3,* &c., their distances from any assumed plane of reference taken perpendicular to the shaft. Then the con­ditions necessary for balance amongst them are that the vector sum of M *r* shall vanish, and also that the vector sum of M *a r* shall vanish, this latter quantity being the resultant of the moments of the centrifugal forces with respect to the plane of reference. In a four-crank engine there is no serious difficulty in arranging the revolving masses in such a manner that these conditions shall be satisfied, so far as those masses are concerned. The problem, as Professor W. E. Dalby has shown, lends itself readily to graphical treatment (see his treatise on *Balancing of Engines).* With respect to the reciprocating masses, a first approximation towards balance is attained by satisfying the conditions which would secure balance if the motions were simply harmonic. These conditions are identical with those which have just been stated for the revolving masses, when *r* is interpreted as the semi-amplitude of the harmonic motion. When the conditions in question are satisfied, the only remaining source of disturbance is that which comes from the fact that the reciprocating masses are connected to the cranks by rods of finite length; in other words, that the motions are not simply harmonic. For this reason the force required to accelerate each piston is greater when the piston is at the end of the stroke farthest from the shaft than when it is at the other end, and consequently the balance, which would be perfect if the connecting-rods were