INSTITUTE OF MARINE ENGINEERS

INCORPORATED.

SESSION 1902-1903.

President—D. J. Dunlop, Esq.
Local President (B. C. Centre)—Sir Thos. Morel.

VOLUME XIV.

LECTURE

ON

THE BALANCING OF MARINE ENGINES.

BY

Mr. H. M. ROUNTHWAITE.

DELIVERED ON

MONDAY, MARCH 24th, 1902,

AT

58 ROMFORD ROAD, STRATFORD.

PREFACE.

58 Romford Road,

Stratford,

March 24th, 1902.

A meeting of the Institute of Marine Engineers was held here this evening, presided over by Mr. J. Macfarlane Gray (Vice-President), when a lecture on the Balancing of Marine Engines was delivered by Mr. H. M. Rounthwaite. The lecture and the figures used in illustrating it will be found in the following pages. The lecture occupied the whole evening, and Mr. Rounthwaite kindly agreed to attend the next meeting and further elucidate the subject.

JAS. ADAMSON,

Hon. Secretary.

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THE BALANCING OF MARINE ENGINES

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Chairman:

Mr. J. MACFARLANE GRAY, *Vice-President.*

About the year 1835, or shortly afterwards, locomotive engineers began to have trouble with their engines, especially with those that had outside cylinders. The engines slewed from side to side, and sometimes left the rails in consequence.

A study of the question showed that this was due to the inertia of, and want of balance in, the moving parts, and a remedy was found in the addition of balance weights to the wheels.

My attention has been drawn to an article in the *Mechanics' Magazine* for 1839 which deals with the question.

Then, in the late forties, marine engineers began to find trouble with their horizontal engines. They were beginning to couple them direct to the tunnel shafting instead of through gear wheels, and to run them faster in consequence. The wooden hulls were shaken violently, and seams were frequently started and leakages set up; so they, in turn, had to look for a remedy, and found it in the addition of balance weights to the crank arms.

As regards land engines there is a book, *On the Richards Indicator* — written by the American engineer, Chas. T. Porter, and published in this country in the early seventies—which deals somewhat fully with the question. But, so far as I know, the first English engineer to take up and insist on the importance of the question was Mr. Arthur Rigg. His book on *The Steam Engine* was published in 1878 and dealt with the matter very fully, and he also read several papers on the subject about that time.

I first had to deal with the question about 1888, when designs were got out for the engines of the first torpedo gunboats for H.M. Navy, and I found considerable difficulty in getting hold of the principles involved, though there were then several books which treated the question more or less fully. And during the last ten years much greater attention has been given to the subject, and many able mathematicians have investigated it and have read papers on the subject. In fact, as is often the case when a new thing is taken up, there is a tendency to run to extremes.

But the engineer thinks mostly in sketches, and a piece of chalk or pencil is a necessity to him; whilst the mathematician thinks in symbols, the use of which takes years to master, even when the learner has the mathematical brain. The result is that many engineers can make no use of these mathematical investigations and are still anxious to get a clear understanding of the question.

I have, myself, so little of the mathematical brain that I have sympathised strongly with the many engineers referred to, and have at length presumed to try and express the various aspects of the question almost without the use of mathematical symbols, using only "King's English" and a few simple blackboard diagrams. Hence these notes.

The difficulty of the methods suggested does not lie so much in the difficulty of any one step, but in the number of successive steps which must be taken. Therefore I propose rather to try and make the principles clear than to work out any case of an actual engine, for it is only by getting a sure hold of the principles that a man can follow all the windings of the path without straying. The variations and peculiarities of the individual cases with which he will have to deal will never confuse him if he really understands the why and wherefore of each step in the method.

I do not claim to have originated any method of procedure mentioned in these notes; I claim only to have acted as editor, to have put the facts, etc., into this shape for presentation to you. There are doubtless other methods which are neater from the mathematician's point of view, but these I propose have the virtue of simplicity.

In an unbalanced engine there are two perfectly distinct sets of forces which give trouble; first, those set up by unbalanced rotating parts, and second, those caused by unbalanced reciprocating parts. The effect of an unbalanced rotating mass, such as a crank, is to produce a force acting radially from the shaft axis through the centre of gravity of the crank mass, which causes the shaft to spring and to describe a small circle round the true axis at each revolution, the shaft thus moving eccentrically and grinding out the bearing round its whole circumference. This may result in undue wear and tear and heating of bearings; or, if the bearing surface is ample, it may slightly spring the engine seatings and cause the whole engine to move with the eccentrically-moving shaft; or, if the seatings are rigid, the whole hull of the ship may also move with the engines and thus be made to vibrate.

If the unbalanced mass is at mid-length of the engines the motion set up is simple eccentric, but if the mass is towards one end of the crank shaft the motion is more complex, the shaft axis tending to sweep out a cone at each end, whilst the part near the vertical centre line through the engines will scarcely move at all.

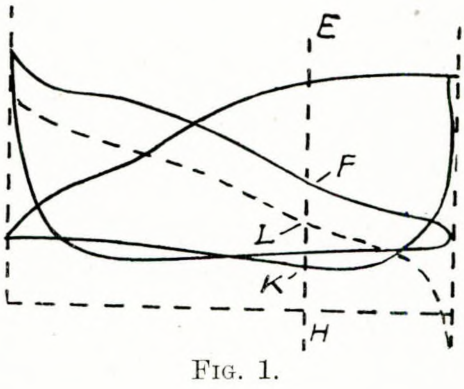

FIG. 1.

The forces set up by vertical reciprocating parts are also vertical, and, if central, tend to move the whole engine and ship up and down, but if acting away from the vertical centre line through the c. of g. of the engines they tend to tilt the engines endwise. The result in either case, if the vibrations are severe or the hull is lightly built, is to cause deterioration of the ship's structure, with more or less leakage and also discomfort to passengers and crew.

Before coming to the balancing question proper it will be necessary to say a further word or two about the unbalanced engine. An ordinary indicator diagram refers to one side of the piston only, but to ascertain the effective driving pressure, or the difference between the pressures on opposite sides of the piston, it is necessary to have a pair of diagrams, and to construct a third from the pair.

Let Fig. 1 represent such a pair of diagrams, and draw below them the line of zero pressure. Then draw any ordinate EH and set off HL = HF − HK; then L is a point in the required effective pressure diagram, and the other points may be found in a similar manner. The effective pressure at each

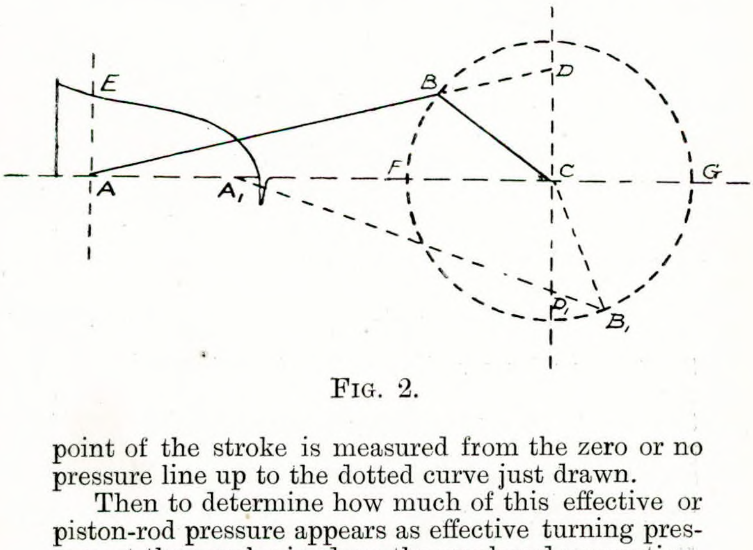

FIG. 2.

point of the stroke is measured from the zero or no pressure line up to the dotted curve just drawn.

Then to determine how much of this effective or piston-rod pressure appears as effective turning pressure at the crank pin, draw the crank and connecting-rod in any position as in Fig. 2 and produce the connecting-rod (if necessary) to cut CD in D.

Then the velocities of points A and B are to one another as CD is to CB: B is supposed to travel round uniformly whilst the speed of A varies as CD or CD_1 from zero at F and G to maxima at the points where the crank and connecting-rod are at right angles. And the pressures at these points vary inversely as the velocities, so that if AE represents the pressure at A, the pressure at B is $AE \times \frac{CD}{CB}$.

Now take a line (Fig. 3) equal in length to the half crank-path unrolled, and divide it into, say, six equal parts; each of which will then represent 30° angular movement of crank. At each of the dividing points set up an ordinate representing to some scale the turning force on the crank-pin at this point, and draw a fair curve through the extremities of the ordinates. Then this curve shows at a glance how

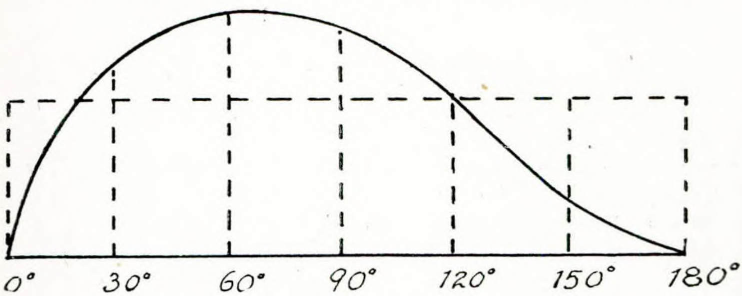

FIG. 3.

the turning force varies from uniformity, which is represented by the rectangular figure of area equal to that contained by the curve and the base line; or, if preferred, the diagram may be constructed by setting out the ordinates radially from a circle representing the crank-pin path.

Later on it will be necessary to return to these diagrams in order to note the effect on them of the inertia of the piston, etc.

Inertia is simply the resistance which any body offers to being put in motion, or, being in motion, to more rapid or less rapid motion. A heavy iron ball suspended by a long cord requires a certain push to set it in motion, and if moving slowly it will require another similar push to make it move twice as fast.

And, if the ball be attached to a rod for transmitting power to some mechanism beyond, it can only pass on that portion of the energy communicated to it which is in excess of its own demands. But though it thus takes toll, it begins to disgorge the moment it is checked, and by the time it is brought to rest again has restored practically the whole of the absorbed energy.

For definitions of "moment" and "couple" see p. 27.

Similarly, in a vertical steam engine, at the commencement of the down stroke say, although there is the same pressure on the piston as on the cylinder cover, part of that applied to the piston at the commencement of the stroke is absorbed in setting the piston in motion, and never reaches the crank-pin or main bearing, so that, for the time, the pressure on cover exceeds that on the main bearing brass, and there is a tendency to lift the whole engine; and towards the end of the stroke the pressures are reversed, the piston giving out energy, and so causing a pressure on the main bearing which is in excess of that on the cover, and so on.

The calculation of the pressures required thus to accelerate or retard the pistons, etc., of an engine must now be considered.

For this purpose the reciprocating weights may be considered as located at the crank-pin, as in a slot-crosshead donkey engine, and we may use the formula which expresses the amount of a centrifugal force, viz.: $F = WrN^2 \times {\cdot}00034$ where W = weight of body considered in pounds, r = radius in feet, and N = number of revolutions per minute. From which it is seen that centrifugal force varies directly as the weight of the body considered, and directly as the radius, but as the square of N, the number of revolutions per minute. Therefore it is not so much the circumferential velocity that counts as the angular velocity, or speed of turning, and balancing becomes rapidly more important as the engine approaches the "quick revolution" type.

Effects of Inertia of Reciprocating Parts.

As already pointed out the consideration of this question is simplified by supposing a single mass, equal in weight to that of the reciprocating parts (piston, piston-rod, crosshead and part of connecting-rod), to be concentrated at the crank-pin centre, and to revolve with it. To determine the proportion of the connecting-rod weight which is to be treated as reciprocating, find the *c* of *g* of the rod, and, if it be found to be, say, 70 per cent. of the rod's length from the top end, the reciprocating weight may be taken as 100—70=30 per cent. of the total weight of rod, and so for other percentages.

The radial force set up by the revolution of this

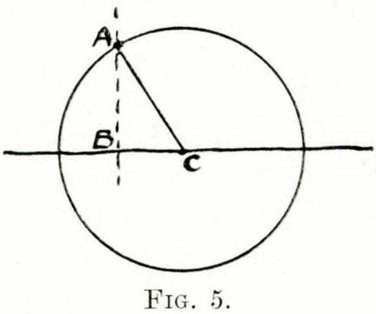

FIG. 5.

mass is equal to $WrN^2 \times \cdot 00034$, and in a horizontal engine the horizontal component of this force measures the acceleration or retardation force due to the reciprocating parts.

If a circle be drawn with radius $WrN^2 \times \cdot 00034$, to some convenient scale, the horizontal component for any position of crank is equal to radius × cosine of angle between crank and engine centre line, or, referring to Fig. 5, is equal to

$$WrN^2 \times \cdot 00034 \times \frac{BC}{AC}$$

And it follows from this, that, at the ends of the stroke, this force (required to accelerate or retard the reciprocating parts) has the full value $WrN^2 \times \cdot 00034$ pounds, because BC is then equal to AC and $\frac{BC}{AC}=1$; or, if it be computed in pounds pressure per square inch of piston,

$$\frac{WrN^2 \times \cdot 00034}{\text{Piston area in square inches}} \quad . \ (a)$$

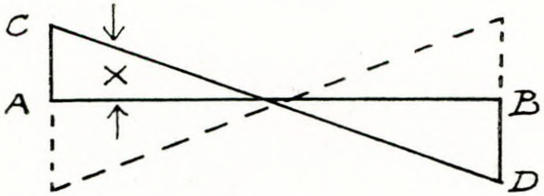

FIG. 6.

Draw a straight line AB (Fig. 6) of length equal to the piston stroke (to some convenient scale), and erect a perpendicular, of length given by expression (a), at each end, but on opposite sides of the line, as AC, BD, and join CD; then any ordinate

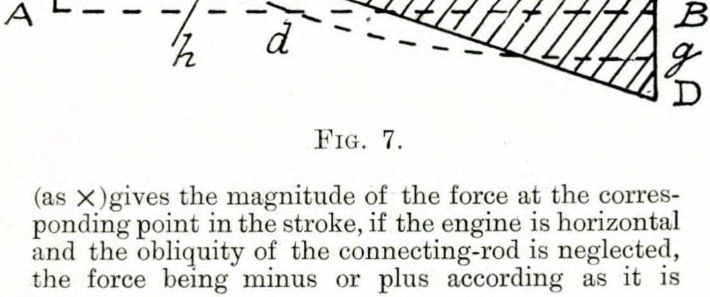

FIG. 7.

(as ×)gives the magnitude of the force at the corresponding point in the stroke, if the engine is horizontal and the obliquity of the connecting-rod is neglected, the force being minus or plus according as it is measured on one side of the line or the other. For the return stroke the diagram is as shown dotted.

For a vertical engine, as the weights assist acceleration on the down stroke and oppose it on the up stroke, the line AB (Fig. 7) must be replaced by a new one, above AB for the down stroke, and below it for the up stroke, and at a distance from it equal to, — *Weight of reciprocating parts ÷ area of piston in square inches*, laid down to the same scale as AC and BD.

The effect of this alteration is to move the position of zero force (E) further up the stroke, and to alter the relative proportions of the shaded triangles, which represent the work to be done in accelerating or retarding the reciprocating parts.

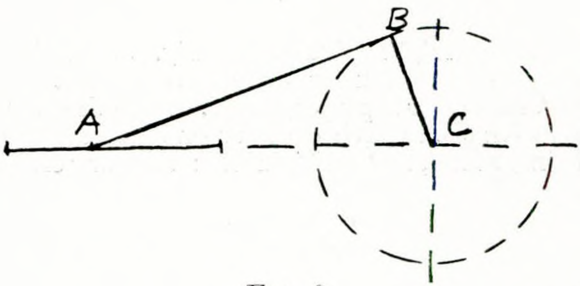

FIG. 8.

But the effect of the obliquity of connecting-rod is generally too great to be neglected, and may be determined as follows:

(1st) If great accuracy is not required. Find the point of maximum velocity of piston, A (which is also the point of zero acceleration force), by the method $AC = \sqrt{AB^2 + BC^2}$ (Fig. 8), and mark it on AB (Fig. 7) at *d*.

Then, if the connecting-rod be, say, four cranks in length, add one fourth to AC (top centre), obtaining point *f*, and subtract one fourth from BD (bottom centre), obtaining point *g*, and draw a fair curve through the three points f, d and g. The ordinates from ab to this curve will give the magnitudes of the forces at each point of the stroke.

(2nd) If great accuracy is desired. Use Rigg's method for determining the length of a number of ordinates to the curve, as follows:

Describe a circle (Fig. 9) with radius equal to $WrN^2 \times \cdot 00034$ to any convenient scale. Divide the diameter AB into ten equal parts, and re-divide each end part into two.

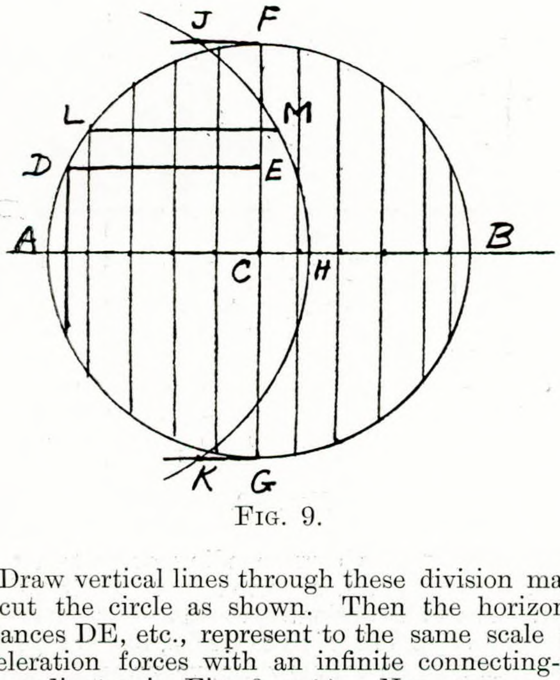

FIG. 9.

Draw vertical lines through these division marks to cut the circle as shown. Then the horizontal distances DE, etc., represent to the same scale the acceleration forces with an infinite connecting-rod (the ordinates in Fig. 6 *ante*). Now suppose the connecting-rod to be four cranks in length. From C set off CH equal to one-fourth CB; and from F and G set off, in the opposite direction, FJ and GK, each equal to CH. Through J, H, and K strike a circular arc (in this case from point A). Then the distances LM, etc., will be the ordinates required, and a parabolic curve similar to *fdg* in Fig. 7 may be drawn by setting them off from a base line corresponding to AB.

If now the net effective pressure diagram (the dotted line in Fig. 1) be taken and placed as shown in Fig. 10 above the newly determined acceleration diagram) both being drawn to the same scale), and each of its ordinates be increased or diminished by

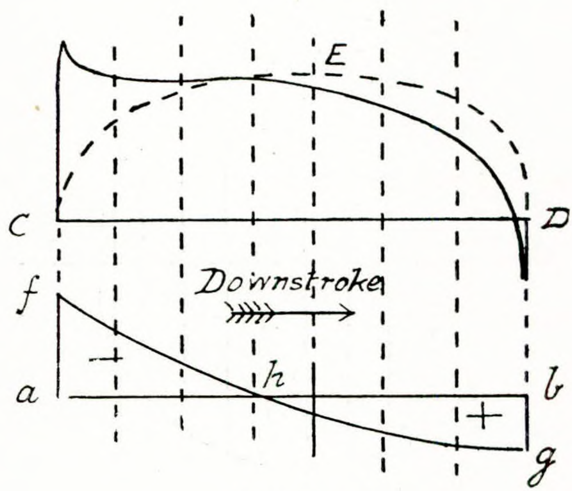

FIG. 10.

the amount of the corresponding ordinate in the acceleration diagram, a dotted curve CED will be obtained, and the ordinates to this new curve will give, for each position in the stroke, the net (except as regards friction losses) effective pressures acting through the piston-rod. And if these piston-rod pressures be resolved as shown in Fig. 2, and Fig. 3 be then re-drawn with the new ordinates thus derived, a new twisting moment curve will be obtained which will give (except as regards friction losses and the small fly-wheel effect of the propeller, cranks, etc.) the actual twisting moments in the crank-shaft at any point in the stroke—the rotating weights being supposed already in balance. Of course, if preferred, either the pressure (on crank-pin) or the twisting moment ordinates may be set off radially outside a circle representing the crank-pin path, thus producing a circular diagram, and showing the pressures at the crank-pin, or the twisting moments in the shaft through one complete revolution.

In the case of the rectilinear diagram, a second portion, equal in length to the first, must be added to one end to show the variations during the return stroke.

The Balancing of Engines.

The forces set up by rotating weights can only be truly balanced or neutralised by the similar forces set up by other rotating weights; and the forces set up by reciprocating weights can only be neutralised by the similar forces set up by other reciprocating weights. Thus the unbalanced parts of crank-arms, the crank-pin, and that part of the connecting-rod weight which may be considered as located at and revolving with the crank-pin centre (for which see p. 10) can be truly balanced by fixing rotating balance weights at the opposite side of the shaft axis, and in the same plane of revolution. But to properly balance the piston, piston-rod, crosshead, and remaining portion of connecting-rod, two cranks or two eccentrics, placed symmetrically, one on either side of the engine crank, must be employed; and these must be coupled to suitable balancing, or neutralising weights, and must cause them to reciprocate in the same plane with the piston-rod, etc., but in an opposite sense.

The commonly used arrangement, in which the rotating balance weights are made heavy enough to balance not only the rotating weights, but also a part (usually half to two-thirds) of the reciprocating weights, does not in any way balance (in the sense of neutralising) the forces set up by the reciprocating parts, but simply transfers the vertical forces into the horizontal plane (or *vice versâ* in a horizontal engine).

Thus, if a vertical marine engine be fitted in accordance with this system, the vertical forces tending alternately to pull upward and force downward the main bearings and the ship's bottom (and so to set up longitudinal vibrations of the hull in the vertical plane through the shaft axis) are not neutralised, but a portion of them is transferred into the horizontal plane through the shaft axis and then tends to shear off the holding-down bolts, to rack the seatings sideways, and to set up horizontal vibrations of the hull. For example, suppose the case of a single-crank engine carrying rotating balance weights which are intended to balance all rotating weights and, in addition, a part of the reciprocating weights. Then (Fig. 10A) the stress set up by the excess balance weight is always radial and outwards from the shaft axis. On top centre and on bottom centre it acts vertically and is equal to $wrN^2 \times \cdot 00034$ $\pm$ its own weight, where w represents the weight in excess of that necessary to balance the rotating parts. This effect of gravity is sometimes neglected, but should not be, as it may amount, even on top or bottom centre, to 25 per cent. of the whole.

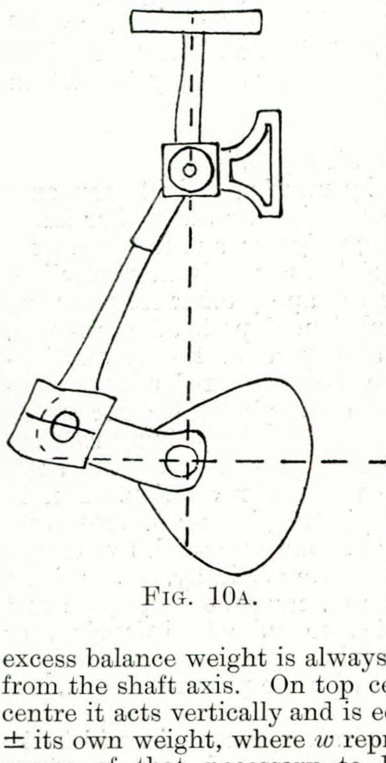

FIG. 10A.

In these positions the surplus weight (over that required to balance the revolving parts) produces just the desired effect and counteracts the inertia effect of the pre-determined fraction of the reciprocating parts. But in any other position, at half-stroke, say, the inertia effect is, as we have seen, almost nil, and yet the centrifugal effect of the balance weight remains unaffected, and is, in this position, horizontal, so that a new and useless (possibly harmful) force is set up. But the seatings are usually amply strong to transmit these forces to the hull, and the water, pressing against both sides of the ship, takes up and practically neutralises the horizontal vibrations, and the system is therefore fairly satisfactory in its results, and is frequently employed.

For the generality of auxiliary engines it serves admirably, and has the great recommendations of simplicity and cheapness. A very small weight attached to the rim of a fan, or placed in the crank-disc of a centrifugal pump engine, though not really in the proper transverse plane (viz., that through the mid-length of the crank-pin), yet produces a most marked improvement in the running of the usual single-crank engine, and should never be omitted. Balance weights forged with the crank arms are, of course, better, but they are very much more expensive.

But the theoretically correct method of balancing reciprocating parts by the addition of reciprocating balance, or "bob," weights to an engine means so much cost, weight, and complication that other methods of balancing have been devised.

The object of these methods is, not to balance the moving parts in connection with each cylinder independently, but so to space the centres of the various cylinders, so to arrange the various crank angles, and so to proportion the weights of the moving parts, as to cause the whole of the disturbing forces to cancel out. It will, of course, be understood that these methods of what may be termed "collective" balancing cannot be applied without passing considerable stresses through the framings, or bedplates, of the engines, and that these stresses must be carefully considered in settling dimensions and strengths of those parts.

With some types of engine collective balancing can be carried out more or less completely, but with others it is impossible, and, in these latter cases, recourse must be had to independent balancing, or partial balancing, of the moving parts by rotating weights as described above.

In the following remarks vertical engines only are referred to unless otherwise stated.

(*a*) *Single Crank Engines.* — These are best balanced by rotating weights attached symmetrically as regards the transverse plane through the mid-length of the crank-pin. The weight of the balance weights × the radius of their *c* of *g* should equal the following weights × crank radius :

$$\text{Unbalanced parts of crank arms} \times \frac{\text{Rads. of their } c \text{ of } g}{\text{Rads. of crank}}.$$

Crank pin.
Rotating part of connecting rod.
Half* remaining part of connecting-rod.
Half* crosshead, piston-rod, and piston.

(*b*) *Two Opposite Cranks.*—Here, after the revolving parts, or these + half* reciprocating parts, are balanced by suitable revolving balance weights, there still remains a couple—whose moment is equal to $(WrN^2 \times \cdot 00034) \times$ distance between centres of two cranks—where W stands for the weight of the unbalanced reciprocating parts connected with one crank—which tends to rock the engine like a see-saw about some point in the shaft axis, situated between the two cranks and varying in position according to the relative weights of the two sets of reciprocating parts. This couple can only be neutralised by the addition of two supplementary cranks,

* Or more if thought fit.

or eccentrics, driving reciprocating "bob" weights, as previously described.

(*c*) *Two Cranks at Right Angles*.—This case is similar to the previous one (*b*), but the couple is considerably more powerful, since, when the cranks are both above, or both below the shaft axis, the vertical forces set up by the parts attached to them combine in tending to tip up the engine first on one end of the bed-plate and then on the other. Hence proper rotating balance weights are very necessary with this type of engine.

(*d*) *Three Cranks at* 120°.—In this arrangement, if the pistons, etc., are of equal weight, the vertical forces balance one another exactly, i.e. — their algebraical sum is zero—no matter what the length of connecting-rod may be, but the couples which these forces set up cannot be balanced except by additional cranks, or eccentrics, and "bob" weights. If the pistons, etc., are not of equal weight the balance can be improved by slightly varying the crank angles. The arrangement in which a central heavy crank on one side of the shaft balances right and left light cranks on the opposite side of the shaft, does not come within the sphere of practical marine engineering.

(*e*) *Four Cranks at Right Angles*.—With this type of engine it is possible, by adjusting the distances between the cylinder centres and the weights of pistons, etc., and also, sometimes, by slightly departing from the 90° angle of crank to obtain a fairly perfect balance, both of the vertical forces and of the couples, without the addition of any special balance weights.

It must, of course, be understood that if a good balance is required, each valve with its eccentrics and gear must be treated as a separate set of revolving and reciprocating parts and its effects allowed for. Usually these effects tend to partly neutralise those of the pistons, etc., and thus reduce the size of any balance weights that may be required.

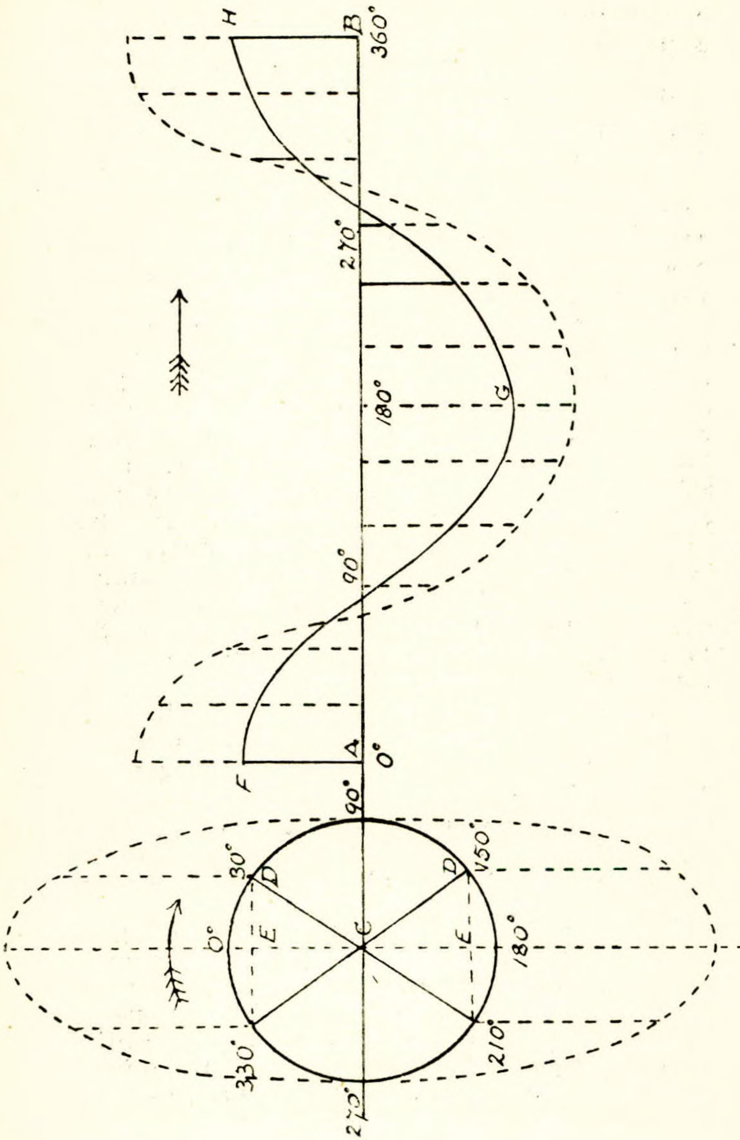

FIG. 13.

The most convenient form of diagram for displaying the manner in which the unbalanced forces in any engine vary in magnitude during a complete revolution is constructed as follows. Draw the crank-path circle (Fig. 13) with a radius representing to some convenient scale the centrifugal force set up by the rotating parts $WrN^2 \times \cdot 00034$. Also draw a line (AB) equal in length to the circumference of the crank-pin path, and divide it into twelve equal parts, each of which will then correspond to 30° angular movement of the crank. Then suppose the case to be that of a single-crank vertical engine without any balancing arrangements, and that it is desired to see the variations in the resultant vertical force set up in the main bearings by the unbalanced parts during a complete revolution.

To arrive at this the value of the vertical component of the force set up by the rotating parts must first be determined for each of the twelve positions of the crank; and an ordinate representing this component, to scale, must then be set up, or down, from AB, and a fair curve drawn through the extremities of the ordinates as indicated. This curve will show at a glance how the vertical forces due to the rotating parts vary during one revolution.

At 0° (or 360°) the vertical force acts upwards, and has the value $WrN^2 \times \cdot 00034 - W$, since gravity acts against the force at this point. At 180° its value is $WrN^2 \times \cdot 00034 + W$, as gravity here acts with the vertical force. At 90° and at 270° the vertical component of the centrifugal force is nil, so the ordinates must be drawn downwards and must represent W, the weight of the rotating parts, only. At any intermediate point in the upper half of the crank path (as at 30°) the vertical component will have the value $WrN^2 \times \cdot 00034 \times \frac{CE}{CD} - W$; and at any point in the lower half (as at 150°) the value will be $WrN^2 \times \cdot 00034 \times \frac{CE}{CD} + W$.

Then, to represent the vertical forces due to the reciprocating parts, as modified by the action of the connecting-rod and by gravity, the ordinates already determined and recorded in the lower part of Fig. 10 may be used, after multiplying them by the piston area. (In the figure they represent pressures *per square inch* of piston.)

These ordinates must be set off outside the curve already drawn and a new curve drawn through their extremities. This new curve (dotted in the figure) will then represent the variations in the resultant vertical force as required; that is to say, at each position of the crank during the revolution the total

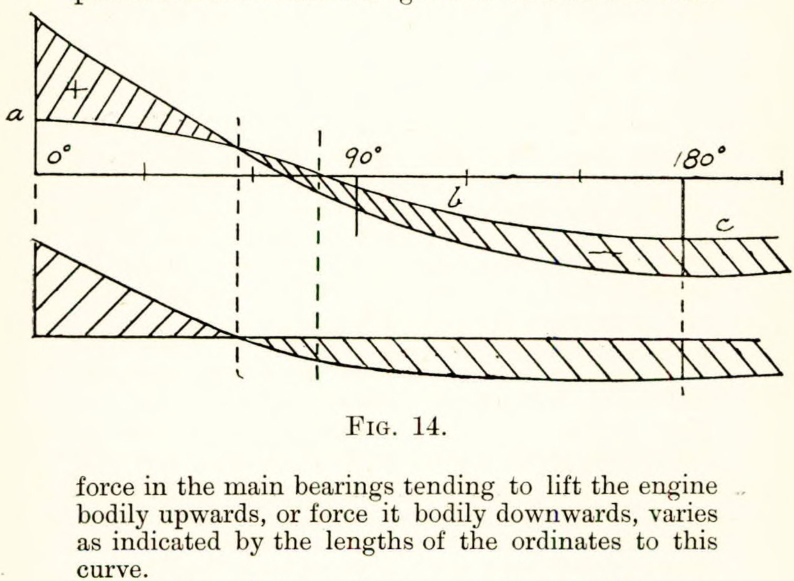

FIG. 14.

force in the main bearings tending to lift the engine bodily upwards, or force it bodily downwards, varies as indicated by the lengths of the ordinates to this curve.

If preferred, the ordinates may be set off from a circle representing the crank path, thus producing a circular diagram, as indicated at the left hand (Fig. 13), which is a typical one for all such cases. But, in order that this diagram may scale correctly, a new horizontal centre line should be drawn across it at a height above C, representing, to scale, the weight of the revolving parts, and the forces be measured from this new centre line; or, the line AB may be drawn through at this distance above C to commence with.

If the rotating parts are balanced by revolving weights, the inner curve (FGH) disappears, and the parts of the ordinates beyond it should then be set off direct from the base line and a curve drawn through their extremities as indicated in Fig. 14, which is similar in form to Fig. 7 previously obtained.

If the revolving weights be made so heavy as to balance a portion of the reciprocating weights, in addition to the rotating ones, call w the weight of the portion so balanced, and, by the method already described for obtaining curve FGH (in Fig. 13) draw the curve *abc* on Fig. 14, using $wrN^2 \times {\cdot}00034$ for the radius of the circle in this case. Then the ordinates of the shaded areas give the magnitudes of the remaining unbalanced vertical forces, and a final curve may be constructed by setting them off from a base line as shown in the lower figure of the pair. The reduction in the length of the ordinates indicates a nearer approach to a perfect balance, the curve for which is a straight line coinciding with the base line.

In a similar manner a curve may be constructed to show the variations in the resultant horizontal force, but here gravity has no influence, and the value of the horizontal component of the force set up by the rotating parts at each point is given by $WrN^2 \times {\cdot}00034 \times \frac{DE}{CD}$ (Fig. 13). And, as the forces due to the acceleration and retardation of the reciprocating parts are purely vertical, they have no place in this diagram.

In quick revolution engines the forces set up by the unbalanced eccentrics and by the inertia of the slide valves, etc., are too important to be neglected, and each valve with its gear must be treated as though it were a separate piston, crank, etc., the curve for it being set off outside those of the cranks and pistons, but shifted right or left to suit the

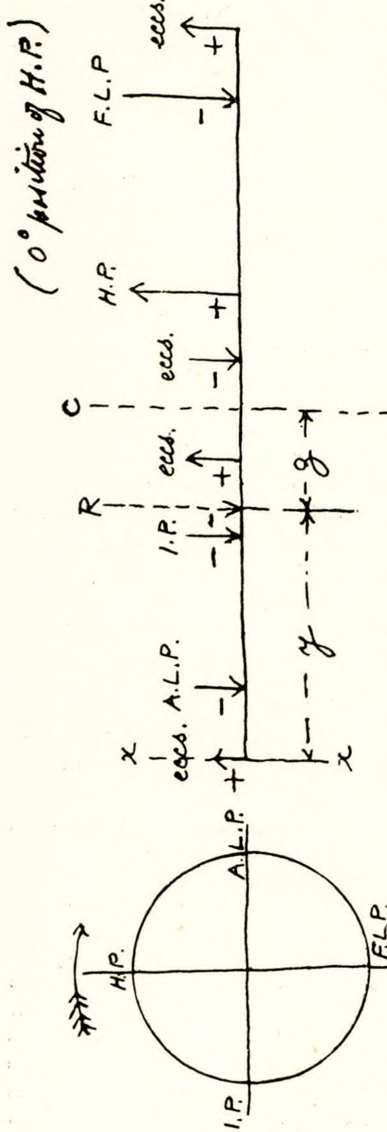

FIG. 15.

angle at which the eccentric is fixed on the shaft. But so far as the vertical force components due to the eccentrics, straps, etc., are concerned—as the pair of eccentrics is placed exactly opposite the crank—they can be dealt with by simply deducting them from those due to the crank.

Having now sufficiently examined the methods for obtaining the resultant forces which arise in an engine from the action of unbalanced parts, and which set up stresses in the bearings and framings and modify the twisting moments, etc., we are in position to consider the principal effects which these forces produce outside of the engine, viz., their tendency to move the engine as a whole, either vertically or horizontally, or to tilt it in one direction or another. And here it may be remarked that when we come to these external or vibration-causing effects it is no longer necessary to allow for gravity in estimating vertical forces, and this simplifies matters somewhat. The reason is that as the weights of the moving parts act constantly downwards, they may be considered as producing a slight initial depression of the engine seatings, in addition to that caused by the weight of the stationary parts, and the vertical vibrations may then be considered as taking place equally above and below this lowered position, instead of unequally above and below the higher position.

To determine these actions, take a line (Fig. 15) representing the axis of the crank-shaft, and lay off from it, at their proper fore and aft positions, ordinates representing in magnitude and direction the vertical forces due to each reciprocating part at, say, 0° position of high-pressure crank. Then collect all the pluses and all the minuses and subtract one from the other to find the resultant R. Then assume a transverse reference plane at XX (say) and take the moments of each force about it by multiplying the force (in pounds or tons) by its distance in feet from XX, taking care to distinguish the products by their proper signs. Now collect all the pluses and all the minuses, and subtract for resultant moment, which, being divided by the resultant force R, will give the distance Y, and by a further subtraction, the distance Z.* Then the conclusion is that at 0° H.P. crank angle there exists a couple of moment RZ tending to tilt up the forward end of the engines on the after end of the bedplate as a fulcrum ; and, a vertical force R, acting at C (the centre line of the engines) and tending to press the engines as a whole down upon their seatings.

The moment RZ may now be represented by an ordinate set up at 0° as in Fig. 16; and when similar ordinates have been obtained for the other crank positions, a curve *abc* may be drawn, which shows at a glance the variations in the vertical tilting moments during one complete revolution.

* NOTE.—The centre line **C** should be drawn mid-way between the outer cranks.

The forces may be shown by a separate diagram or by a second curve on this one. Similarly, if desired, curves may be drawn to show the horizontal slewing moment and force at each crank position.

The construction of these vertical force and

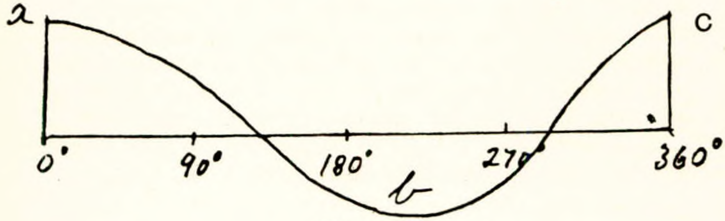

FIG. 16.

couple curves is dealt with at length because it is these curves which are required by the Admiralty to be sent in with every tender now.

Now, it is evident that the plotting of these

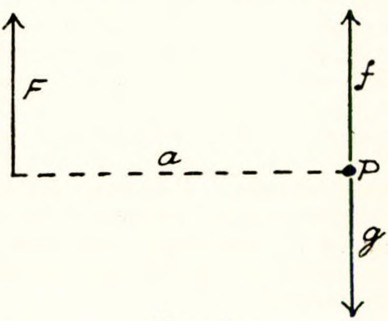

FIG. 12.

curves is a somewhat laborious business, and that before alternative proposals can be compared curves must be plotted for each. But there is a quicker method of arriving at an approximation to the best arrangement of weights, centres, and angles, which may now be described. It is approximate because it takes no account of connecting-rod disturbances.

First, it may also be pointed out, for guidance in considering the action of the vertical forces in engines, that when a force F (Fig. 12) acts at any distance, a, from a point P, the effect of said force is equivalent to those of an equal and parallel force, f, acting at P, and a couple whose moment is Fa. This is demonstrated by adding at P a force g, equal and opposite to f, when it is seen that the three forces F, f, and g are equivalent to the original force F, and that F and g constitute a couple, whilst f is a free force acting in accordance with construction.

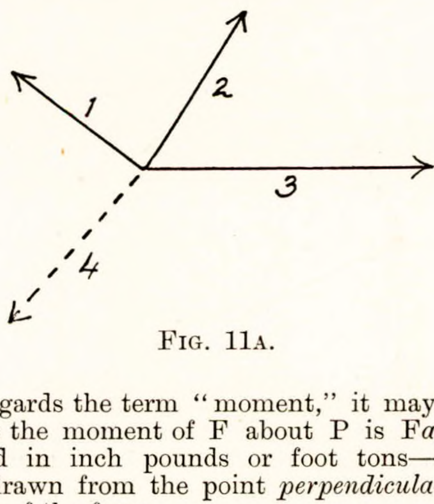

FIG. 11A.

As regards the term "moment," it may here be said that the moment of F about P is Fa, and is expressed in inch pounds or foot tons—a being always drawn from the point *perpendicular to* the direction of the force.

And, second, that the principle of the polygon of forces may be very readily applied to the system of forces existing in any engine, or combination of engines, and simple diagrams can be constructed to show at a glance whether the engine is in balance or not, and, if not, what additional force must be applied to produce a balance.

For example, when any number of forces act on a point as indicated by Fig. 11A, and the direction and magnitude of the resultant or balancing force is required, number the force lines in rotation 1, 2 and 3, and construct a polygon as shown in Fig. 11B drawing the first line parallel and equal to 1, and *in the direction* shown by the arrow on that line, the second line parallel and equal to 2, and in the direction shown by its arrow, and so on. Care must be taken to draw each successive line from the termination of the previous one, because sometimes a line must be drawn back over upon one previously drawn. Then the force required to produce equilibrium is indicated, in magnitude and direction, by the dotted line 4, which closes or completes the polygon, and it

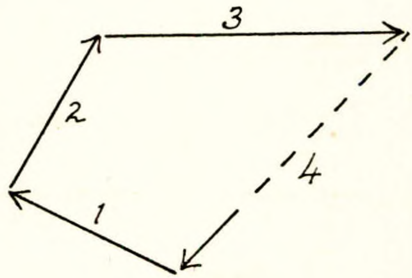

FIG. 11B.

will act in the same "sense" as the other forces, i.e., in this case, "clockwise."

If the three forces 1, 2 and 3 be taken of equal magnitude and be placed at right angles to one another, they, with their resultant, give a polygon which is a square—thus giving a proof at sight of the correctness of the method.

It also follows, as corollary to the above, that, if the polygon representing any system of forces does not close, the system is not in equilibrium.

A polygon should first be drawn for the forces, and then another separate one for the couple moments; and, if a perfect balance does not then appear to be practicable, or if it be considered that the modifications necessary to secure it are undesirable, an adjustment by way of compromise may be effected, and the one improved at the expense of the other.

The moments of the couples relating to the cranks for which the weights are assumed fixed to begin with should be taken about a transverse plane passing through the mid-length of the remaining crank-pin for which the necessary weight is to be

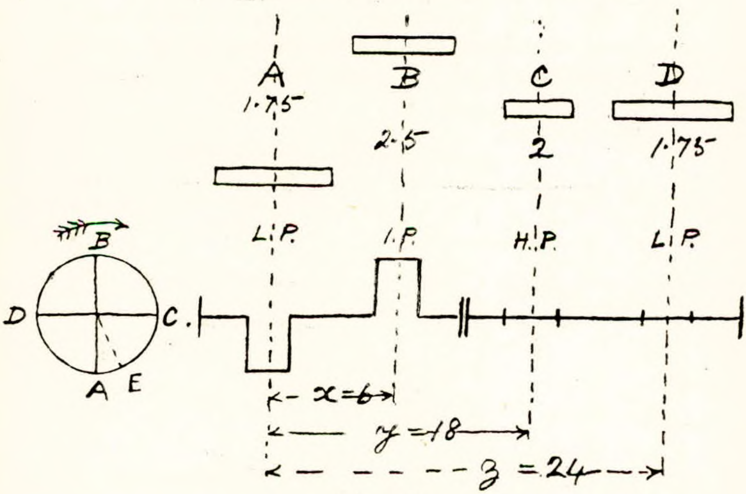

FIG. 17.

ascertained (so that this weight may have no moment), and this crank should, preferably, be that at one end of the engine.

For the purpose of ascertaining if the reciprocating parts of a proposed engine, such as that shown diagramatically in Fig. 17, are in balance, it is sufficient to use the proposed weights of the parts attached to each crank, since, the cranks being of equal radius, any forces set up by the parts will be proportional to their weights.

Assume a transverse plane of reference through the after L.P. centre and take moments about it, these being:

$$B = 15$$
$$C = 36$$
$$D = 42$$

Crank A has, of course, no moment about this plane.

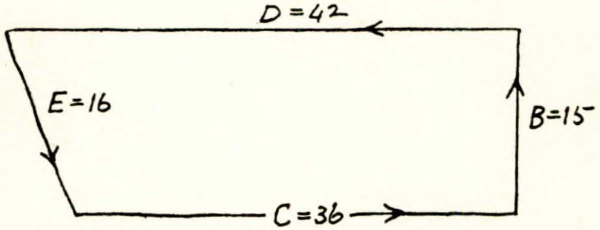

FIG. 18.

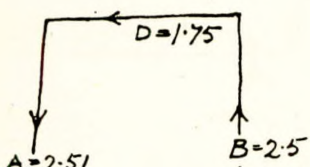

FIG. 19.

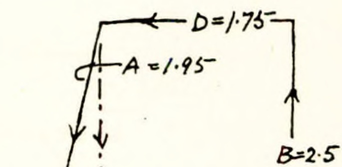

FIG. 20.

Then construct polygons of the moments and of the forces as in Fig. 18 and Fig. 19.

If the moments were in equilibrium Fig. 18 would be a triangle, but, evidently, they are not, as it does not close. Fig. 19 shows that to balance the forces crank A should be set at a small angle with its present position, and that the weight of its parts should be practically 2·5; and, as it has no moment, both these may be altered without disturbing the moment diagram.

But to balance the moments, either the angles of

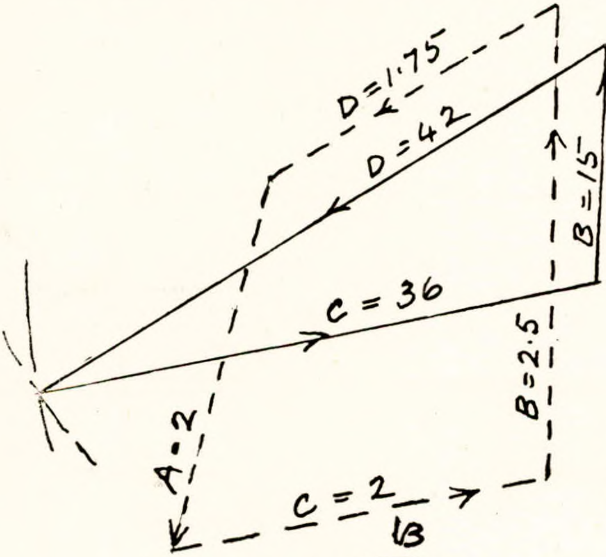

FIG. 21.

FIG. 22.

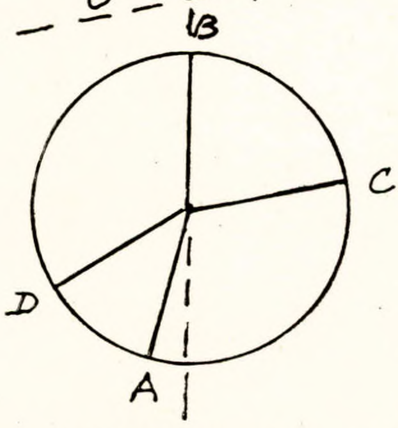

FIG. 23.

cranks C and D must be altered, or a balance weight with moment represented by E (about 16) must be added. A "bob" weight of ·57, with same stroke as pistons added at the forward coupling, would effect the desired balance; but any such addition alters the force polygon, and its effect must be examined. Strike an arc (Fig. 20) with radius ·57 and draw a radius parallel to E in the moment polygon. Then it appears that, to complete the balance, crank A must be angled a little more and its weight increased to 1·95. If the crank A be retained in its original

MOMENTS.

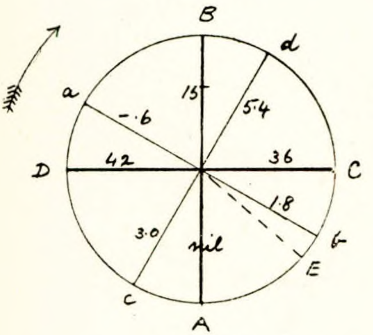

FIG. 24.

FORCES.

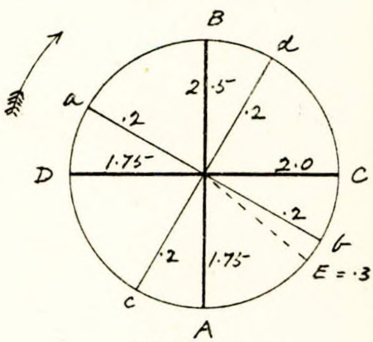

FIG. 25.

$$\begin{array}{l}
a = \cdot 2 \times -3 = -\cdot 6 \\
b = \cdot 2 \times 9 = 1\cdot 8 \\
c = \cdot 2 \times 15 = 3\cdot 0 \\
d = \cdot 2 \times 27 = 5\cdot 4
\end{array}$$

position, and with its original weights, the disturbance of balance, shown by the gap between the points F and G, is very small, and if the matter be compromised by altering the angle of E in the moment polygon so as to split the difference, the engine would give very excellent results as regards vibration. But such "bob" weights are objectionable, and a revolving weight (cast with turning wheel, say), would be better, though, of course, it would give rise to unbalanced horizontal forces.

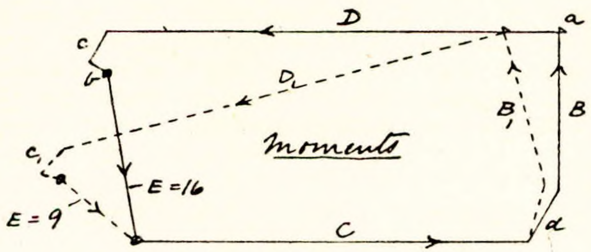

FIG. 26.

The effect of altering the angles of cranks C and D, instead of fitting the balance weight, is shown

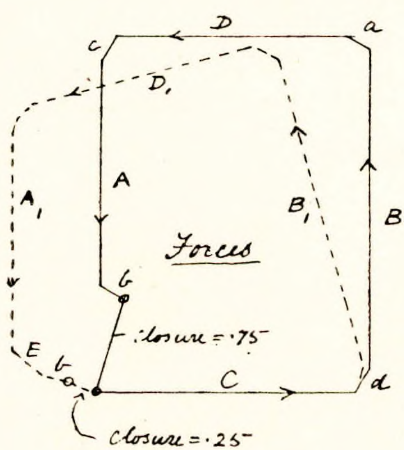

FIG. 27.

by Figs. 21, 22, and 23, which give the new angles for A, C, and D, and show that A's weights must be increased to 2 to give a perfect balance of forces.* But since there are four weights, three distances, and four crank angles (11 variables), all of which may be altered in moderation, it is evident that the possible arrangements are many; and there is no better way of getting a mental grip of the methods than by trying how many different solutions with complete balance can be found.

It is also evident that this type of engine lends itself to this system of "collective" balancing in a special degree, and that single-crank, two-crank, or three-crank engines cannot be similarly balanced.

The modifications produced in Figs. 21, 22 and 23, by taking the valves and gear into account, are shown by Figs. 24 to 28.

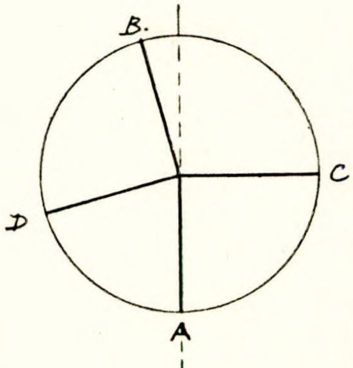

FIG. 28.

Similar diagrams should be constructed for the rotating weights, if these are not to be balanced by rotating balance weights, and, when the most suitable crank angles have been determined for these—if the angles differ from those most suitable

* This solution has the advantage that the crank-shafts are duplicates, and interchangeable when reversed.

for the reciprocating parts—some compromise or splitting of differences must be resorted to.

Before carrying out any scheme of balancing or of partial balancing, care should be taken to ascertain its effect upon the curves of turning moment, by sketching out the probable indicator diagrams and proceeding as previously directed when describing Figs. 2 and 3.

The questions of interchangeability of crank-shafts, and of ease in starting, should also be borne in mind when settling crank angles and distances between centres of cylinders.

DISCUSSION,

MONDAY, MARCH 24th, 1902.

58 ROMFORD ROAD, STRATFORD.

CHAIRMAN:

MR. J. MACFARLANE GRAY.

THE CHAIRMAN said he had been very much interested in the paper that had just been read, but it was one of those papers that required some study before they could exactly get into the groove the author was moving in. They could not fully appreciate a thinking paper without time to think. He had not followed the paper so closely as to be able to grapple with all that it contained, although the subject was a very familiar one to him, as most of them knew. The mention of the name of Porter carried his memory back to the year 1862, for one of the most interesting objects in the exhibition of 1862 was the Porter engine. A remarkable feature of this engine was its astonishing piston speed, and what particularly puzzled them at the time was that in obtaining this speed the inventor used an enormously heavy piston. The explanation was that he set to work with very high pressure steam with a view to its subsequent expansion, and he made a very heavy piston which absorbed energy at the beginning and gave it up at the end of the stroke, so as to bring about a more uniform pressure. Or perhaps he should rather say that the energy was invested in the piston at the beginning of the stroke and paid out at the end. At that time there was little said about balancing, the object was to get a more uniform rotation of the engine, which was intimately connected with it, although *per se* quite a different thing, and Mr. Rounthwaite had very properly connected the two. Balancing, itself, was a problem quite distinct from the twisting moment. He proposed a hearty vote of thanks to Mr. Rounthwaite for his paper. It was a thoroughly practical paper by a practical man who had not thinned out his subject by too much theory. If they would read this paper, along with that which he himself read on the subject some time ago, one would assist the other. One did not contradict the other in any way, and each paper would help to a proper understanding of the other.

Mr. G. W. Newall (Member of Council) in seconding the motion, said he was not versed in the balancing of engines, and, having come in late, he had not heard the whole of the paper, but he thought it a subject to which engineers, and particularly an Institute like this, should give special attention. In listening to Mr. Rounthwaite it struck him that in order to make this matter quite plain to him, and perhaps several others, it would be a great benefit if the author would be kind enough to work out one of the problems given in the paper. They had various diagrams in the paper, and they saw certain lines and curves, but to him, and possibly others also, they did not mean very much. If Mr. Rounthwaite could spare the time and would kindly take the trouble to demonstrate any one of these problems—setting out the figures and showing their application to the sketch—they would then gain a great deal. He was concerned a little while ago in the balancing of a pump, but he had previously had no experience in balancing an engine of any kind. This pump was driven at ten revolutions a minute, but it was a most erratic sort of machine—sometimes it took out too big a bucketful, and at others missed fire altogether. It was not a large pump, but what he did was this—he took the diameter of the pump, multiplied the area by the pressure, and then put on the crank shaft a weight just equal to that. The pump had previously refused to work at all, but directly the balance weight was put on the crank it worked very smoothly. This was far too important a paper to be dismissed in one evening, and although another paper was fixed for reading at the next meeting on 14th April, he hoped that on that occasion they would also have time for some further discussion on Mr. Rounthwaite's paper. None of them had seen the paper until that evening, and it was not fair to expect that they could take in this sort of science right away. Mr. Macfarlane Gray had given them a similar paper, but it was a maze of mathematics, and although he (Mr. Newall) tried to understand it he confessed there was too much Greek in it for him. The paper now before them appealed to them more, but if they could get one step further and induce Mr. Rounthwaite or Mr. Gray to be good enough to demonstrate even one single diagram on the black-board, he, and possibly many others, would be only too pleased to come and listen.

The motion was carried unanimously, and with reference to Mr. Newall's suggestion, Mr. Rounthwaite said that if at the meeting on the 14th prox. he could be of any help or assistance in any way he should be only too happy to do so.

The CHAIRMAN announced that at the next meeting on April 14 there would be a paper by Mr.

Northcott on "The Steam Turbine," and on April 21 there would be a lecture by Mr. Houghton on the microscopic structure of iron and steel. Mr. Gray added that what Mr. Newall did to the pump was not the balancing which formed the subject matter of their lecture that evening. Mr. Newall tried to equalise the twisting moment—the turning effort of the engine ; he put on a weight which played an important part in the turning effort.

Mr. NEWALL said he quite realised that it was not balancing in the sense that they had been discussing the matter that evening.

A vote of thanks to the Chairman, proposed by Mr. Bales, and seconded by Mr. Johnston, concluded the meeting, and the Chairman, in reply, said on the 14th April he hoped to satisfy Mr. Newall and other members by producing for their edification a number of examples of actual engines, their balancing fully worked out, without mathematics and without Greek.

DISCUSSION CONTINUED,

MONDAY, APRIL 14th, 1902.

58 ROMFORD ROAD, STRATFORD.

CHAIRMAN:

MR. J. MACFARLANE GRAY (VICE-PRESIDENT).

MR. ROUNTHWAITE re-opened the subject by further explaining one or two diagrams referred to at the previous meeting.

Mr. FARENDEN (Ass. Member) said he thought that Mr. Rounthwaite had succeeded admirably in placing this important matter before them very clearly and without the introduction of any intricate mathematical problems. With regard to the forces set up in a vertical engine, it was well known that an engine would run at a certain speed with very little vibration, but if the revolutions were increased or diminished the vibrations were much greater and more severe, and in time would cause deterioration and cracking of the ship's plates, frames, etc., besides which the vibration was most disagreeable and annoying to the passengers on board. If after careful calculation they knew that the engine which they had to deal with was a well-balanced one, and vibration still continued, he took it that this must be put down to the ship's structure or the pitch at which the propeller was set. In the paper which he read on this subject last year Mr. Macfarlane Gray mentioned the case of a recent battleship in which considerable vibration existed, and which was successfully remedied by altering the position of the cranks. Perhaps Mr. Gray could inform them how this was done, as in the ordinary marine engine, where crank shafts were made interchangeable, the number and position of the bolt holes in the coupling fixed the limit in which alteration could be made. A four-crank engine usually had eight bolts, and a three-crank engine nine. The pitch of the bolt holes being, say, at 45° where there are eight bolts, to move the angle of the crank round from 90° to the next bolt hole would increase the angle to 135°, so that in this case the angle of cranks could not be between the angles of 90° and 135°. There appeared to be a great divergence of opinion amongst engine designers as to the proper angles the cranks should be set at, the latter varying very much, and yet this hardly bore out the author's statement that for a good balance for this class of four-crank engine the angles should be as near 90° as possible. Consequently it seemed that in order to do away with the necessity for balance-weights the angle of the cranks must be altered considerably from 90° with this class of engine in order to ensure a good balance.

There was no doubt of the importance of having a well-balanced engine on board ship, especially in the case of fast running jobs, but the question of balancing appeared to him to come more within the scope of the designer than the marine engineer. The engineer at sea had to get the best results he could out of his engine, whether it was properly balanced or otherwise, and it did not come within his province to alter angles of cranks, or weights of parts, even if it was possible to do so.

After some remarks from Mr. Bales,

Mr. ROUNTHWAITE said he thought the explanation of vibration in some steamers and not in others was that no engine, however carefully balanced, was absolutely balanced when they were done with it. There always remained residual forces that were unbalanced, and they set up vibration when the periods of those vibrations happened to synchronise with the natural periods of the ship herself. The natural periods of the ship would depend on the way she was subdivided. An ordinary tramp steamer, for instance, would be in a very different condition to a modern passenger steamer—say a P. and O. boat—of the same dimensions, and whenever the small vibrations set up by the engines or the propeller happened to coincide with the natural period of the ship, or to be some multiple of the natural period of the ship, then the vibrations of the ship suddenly increased. Supposing very great vibrations were observed at forty-five revolutions, these might almost disappear until ninety revolutions were reached, when the vibrations would again become very bad. But at almost any speed between the two there might be practical freedom from vibration.

The CHAIRMAN explained in some detail, and with the aid of blackboard sketches, certain phenomena, a proper understanding of which would, he said, greatly help a correct appreciation of this subject of the balancing of engines.

Mr. W. LAWRIE (Member of Council) observed that for a long time—in fact, until very recently—this subject had been entirely in the hands of the experts, who had worked it out very fully and in a very scientific manner, so that only those who had a knowledge of the higher mathematics could follow them on the subject. In the present paper Mr. Rounthwaite had divested the subject of all that was irksome in the way of mathematical formulæ, and had brought it before them from an engineering point of view, so that they could all understand it. The author was therefore entitled to their deepest gratitude, and although he disclaimed all title to originality in the matter, the manner of the paper, the sequence and treatment generally were quite different from anything that they had hitherto been accustomed to. With such a paper as that contributed by Mr. Rounthwaite to assist them, he thought the subject would no longer remain in the hands of the few.

A vote of thanks to the Chairman, proposed by Mr. BALES and seconded by Mr. HOWIE, concluded the meeting.