Modern Marine Engineering

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Chapter IX.

CHAPTER IX.

THE PRINCIPLES OF THE MARINE ENGINE.

To set aside the principles of any subject is similar to building a structure without knowing the status of the foundation; it is also apparent that the effect of any cause is due to its origin; the main question, at present, however, is the best method of arriving at the truth of certain results, and at the same time to develop the means for further improvement.

The marine engine, as it is at present, consists of a sliding action imparting motion to a pin which revolves around the axis of suspension or support, and therefore our first subject refers to the actual difference in the speed of the piston and crank pin at equal positions on the centre line of motion.

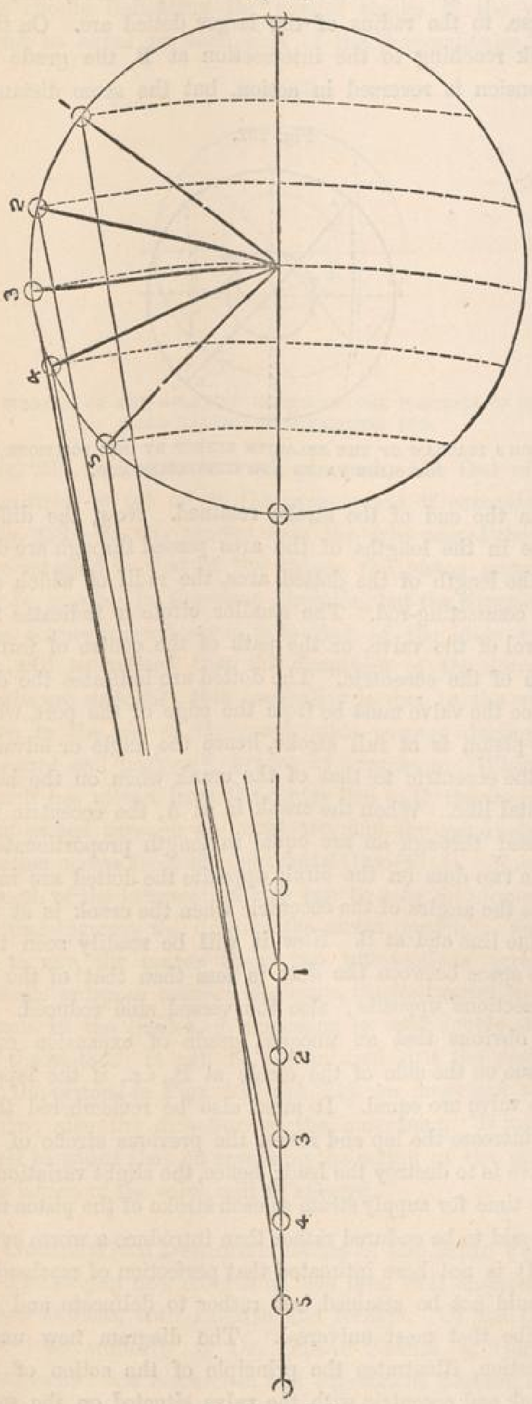
Variation in the Speed of the Piston and Crank Pin.—The diagram, Fig. 230—on page 247—represents the paths of the cross-head or piston, and crank pin; the stroke is 3 ft. 6 in., and the length of the connecting-rod between centres is 9 ft. 6 in., being less than three times the stroke of the piston. The plane line of motion is divided equally into six parts, thus producing five points of cut-off, or grades of expansion. Now it is obvious that the piston, on arriving at 1, caused the crank pin to be at 1 on its path, but the distance passed through is in the proportion of about 1 to 3. On the piston arriving at 2, the crank pin has advanced to the same number on its circuit, but the length passed is about as 7 is to 9; from 2 to 3, and 3 to 4, the passages in each case are more equal; but from 5, to the completion of the stroke, an inequality of importance again occurs in the proportion of 7 to 12·35. With a shorter connecting-rod the variation between all the points of motion on the circular path will be effected in due proportion to the radii of the dotted curves intersecting with the circle in question.

From this diagram, or any other of the same principle, the variation in the speeds of the piston and crank pin can be easily known, also the time of admission of the steam in proportion to the amount of duty effected. For example, if the piston is moved forth and back from the end of the stroke to 5, sixty times in each direction in a minute, the distance traversed $= 14 \times 60 = 840$ in.; but the crank pin's passage $= 42 \times 60 = 2520$ in., and thus the speed of the piston for one-sixth of the stroke is one-third of the crank pin's velocity. With this inequality, however, there is a certain gain, for the steam admitted is more powerful before the cut-off than after, or during expansion, and therefore when the greatest effect is requisite it occurs.

Delineation of the Path of the Crank Pin.—The motion imparted to the slide valve is generally derived from two principles of action—vibratory and rotary. Now, when the former is the prime mover, the speed of the valve is the same throughout the stroke, or rather, if the motion is imparted by the piston, the motion of it and the valve would be equal. Rotary motion is more often adopted than any other for the transmission of power and action, and to the present day the small cranks and circular eccentrics are the prevailing means employed to impart the motion required for the slide valve. The relative speeds of the crank and the eccentric are proportionately the same in theory and practice. The length of the connecting-rod in all examples of rotary motion regulates the inequality of the speed of the sliding body. It must be remembered, however, that a reciprocity of motion can be attained by a correct proportion of the details.

Before proceeding further with the definition of the better arrangement, it will not be out of place to define the relation of the path of the crank to the sliding motion imparted to the valve, also the controlment of the steam. It is doubtless universally known that, virtually, the crank path is divided into four distinct parts, also that for the eccentric. The proportions of these divisions are practically regulated by the grade of expansion agreed on to be maintained. Fig. 231 represents a crank pin's path with the chords indicating the divisional points, the

Fig. 230.

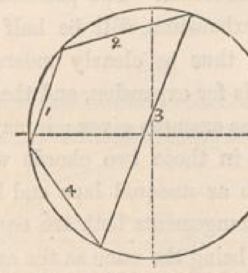


Scale ¾ inch = 1 foot.

BURGH'S DIAGRAM OF THE PATHS OF THE PISTON AND CRANK PIN.

relative proportion is, therefore, readily understood. No. 1 is the chord of supply, No. 2 that for expansion, No. 3 relates to exhaustion, and No. 4 represents neutrality, or that portion of the stroke of the piston when the port on the exhaust side is covered, often termed compression; consequently, the piston for a period is

Fig. 231.



BURGH'S DIAGRAM OF THE PATH OF THE CRANK PIN IN RELATION TO THE SLIDE VALVE'S ACTION.

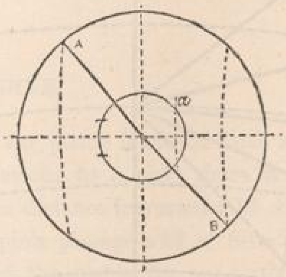
devoid of pressure or vacuum. The length of the chord 1 is due to two causes, which are the grade of expansion and the length of the connecting-rod. It will be noticed that the chord at the plane line intersects with the circle slightly below the same. This last intersection is the angle that the crank assumes when the slide valve commences to open the port, and the vertical distance from the intersection to the plane line is due to the lead required. The upper point of intersection, as before explained, is subject to the curve assumed by the connecting-rod from a point, or distance, on the plane line to the circle of the crank path. The length of chord No. 2 is regulated by the inside and outside laps of the slide valve. The expansion of the steam is now in full operation, and is released by the opening of the port on the exhaust side, hence the intersection of the chord at No. 3. The length of the last-mentioned chord is more than any other, due to the traverse of the valve, or the time occupied in opening and closing the port for exhaustion. Now in the case of an increase of supply steam, the time for expansion and exhaustion would be lessened in proportion, it being remembered that the circle described by the crank pin cannot be increased or decreased for a given length of stroke of piston. The circle, as before stated, is divided into four divisions, and the alteration in the grade of expansion or length of connecting-rod affects the whole proportionately. The concluding chord, No. 4, represents neutrality, or, as before stated, that portion of the stroke of the piston where the vacuum and steam is cut off for a given period, commonly known as compression. It will be remembered that the chord of expansion is due to the outside and inside laps, *i.e.*, when the valve is at the edge of the supply side of the port, the valve has to travel forth until the inside edge permits exhaustion or destroys expansion. The valve is now at half stroke, plus inside lap; exhaustion of steam, therefore, must ensue until the valve is in the same position, but travelling in a reverse direction. The position of the valve when terminating exhaustion will be half stroke, minus inside lap. It can thus be clearly understood that the length of the chords for expansion, and that for compression, are equal in the example given; it may also be added that any variation in these two chords will depend on various causes, such as unequal laps and leads, &c., &c., but with certain arrangements both are equal. The fact of the compression being the same as the expansion, is of no vital importance. It is certain there would be a gain in maintaining expansion longer, and exhausting, till the supply commenced, and thus dispensing with compression; but the present motion of the slide valve would have to be altered, as an extreme unequal action would have to be attained.

The proportions of the path of the crank pin can thus be clearly understood. As at present arranged—compression, supply, and expansion, form one portion of the circle, and the remainder is occupied by exhaustion, which is the most. The diagram given relates to one revolution of the crank only, it being well known that all in principle are alike. There is, however, a slight variation in practice, due to the versed sines of the rods, and inequality of the length of arcs passed through, in proportion to the propelling or sliding motion attained.

The arrangement of the slide valves, in relation to that of the connecting-rod, should be considered as to the attainment of equal action. The versed sines of the chords of the arc passed through by the crank pin, at each angle of the stroke, for a given supply of steam, are unequal, also the versed sines of the eccentric, when the slide is at the same side of the crank as the connecting-rod; but this can be counteracted to a certain extent by short eccentric rods, and levers, reverse in action, and arranged to compensate for the inequality of the speed of the circular and sliding motions, the attainment of which being of great importance.

Fig. 232 represents the circle described by the crank pin for a given stroke of piston. The horizontal line is presumed to be the centre of the engine, and the larger dotted arcs represent an even grade of expansion, on each side of the piston or at each stroke. Now it will be seen that on the crank—represented by the thick lines—moving from the plane to the intersection at A, a given length of stroke of piston is produced, due, of course, to the radius of the larger dotted arc. On the crank reaching to the intersection at B, the grade of expansion is reversed in action, but the same distance

Fig. 232.



BURGH'S DIAGRAM OF THE RELATIVE EFFECT BY THE POSITIONS OF THE SLIDE VALVE AND CONNECTING ROD.

from the end of the stroke retained. Now, the difference in the lengths of the arcs passed through are due to the length of the dotted arcs, the radii of which are the connecting-rod. The smaller circle *a* indicates the travel of the valve, or the path of the centre of formation of the eccentric. The dotted arc indicates the distance the valve must be from the edge of the port, when the piston is at full stroke, hence the angle or advance of the eccentric to that of the crank when on the horizontal line. When the crank is at A, the eccentric has passed through an arc equal in length proportionately. The two dots on the circle opposite the dotted arc indicate the angles of the eccentric when the crank is at the plane line and at B. Now it will be readily seen that the space between the dots is less than that of the intersections opposite; also the versed sine reduced. It is obvious that an unequal grade of expansion must ensue on the side of the circle at B., *i.e.*, if the laps of the valve are equal. It must also be remembered that, to increase the lap and retain the previous stroke of the valve is to destroy the lead; hence, the slight variation in the time for supply steam at each stroke of the piston may be said to be endured rather than introduce a worse evil.

It is not here intimated that perfection of mechanism should not be attained, but rather to delineate and describe that most universal. The diagram now under question, illustrates the principle of the action of the crank and eccentric with the valve situated on the same side of the crank shaft as the connecting-rod, direct action, in each case being maintained. It will be seen that the arcs are all struck by the radii on the same side of the perpendicular line, hence the variation above alluded to. Now in order to counteract, or rather obviate the imperfections now under notice, the lengths of the chords indicating the partial stroke of the valve must be equal on each side of the perpendicular line.

Fig. 233.

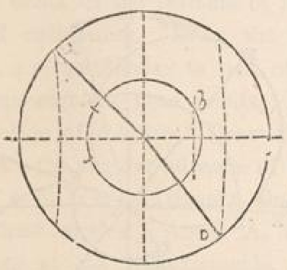


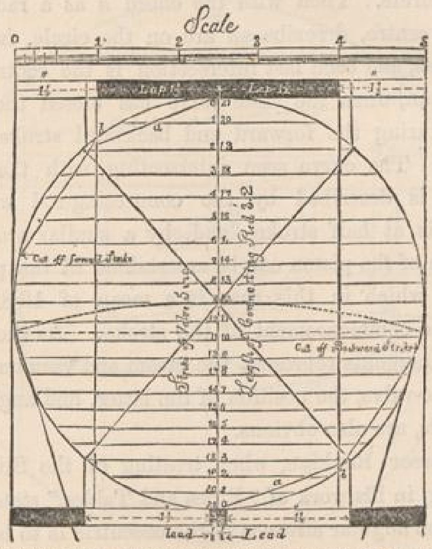
DIAGRAM OF THE RELATIVE EFFECT BY THE POSITIONS OF THE SLIDE VALVE AND CONNECTING ROD.

Fig. 233 represents a crank pin path, and that of the eccentric, to cut off at the same grade of expansion as Fig. 232. In the present case the arcs passed through are reversed to that of the former, the piston presumed to be moving in the same direction, but the situation of the connecting-rod opposite to that of the slide valve. It will be noticed that the diameters of the eccentric paths are unequal; this inequality is due to the variation in the arc of the crank pin's passage during the forward and backward grades of expansion. When the crank has moved from the centre line to C, the eccentric has passed through an equal arc, and the same relative motion occurs from the horizontal line to D. Now the length of the intersection at *b* can be seen, in proportion to that between the dots in the circle opposite. Suffice it to say, the nearer these two intersections agree in length or space between the same, the less variation will ensue in the grades of expansion at each double stroke of the piston. It will be remembered that the position of the pistons in Figs. 232 and 233 are presumed to be alike; also the direction of the movement. It may as well be added that on reversing the action of the piston, the action of the valve will be affected.

Geometrical Demonstrations of the Slide Valve.—The subject now entered on has been well digested by many authors, both English and foreign. Of the latter the best authority is Dr. Gustav Zeuner, a German gentleman: his researches and expositions pertain, however, more to the locomotive than the marine type; but, in the main, his views are correct for either, under relative circumstances. The views of our American friends are represented in a work termed the "Cadet Engineer." At home, Messrs. Watt, Professor Rankine, Mr. D. K. Clarke, C.E., and Mr. M'F. Gray, claim attention for their productions; and we have also done something to solve the problem.

Messrs. Watt's mode of treating the matter under notice is illustrated by Fig. 234, which represents the

Fig. 234.



MESSRS. WATT'S MODE OF PRODUCING THE POINTS OF CUT-OFF BY A KNOWN LAP.

path of a crank pin 21 inches in diameter; length of connecting-rod between centres 3 feet 2 inches, and the stroke of the slide value 5 inches. The method of utilizing the diagram is thus: "Divide the path of the cross-head pin into inches; with the connecting-rod's length as a radius, and each inch as centres, describe arcs, intersecting with the path of the crank pin; join the intersections by chords or lines, as depicted, and the result is a correct illustration of the relative positions or progress of the crank pin and piston. Next draw vertical lines parallel to the centre line as tangents, and above the circle, between these tangents, form a scale of as many equal divisions as the number of inches in the stroke of the valve, which is virtually assuming that the stroke of the piston is that of the valve. Now, having previously settled the outside lap of the slide valve, next draw two tangents parallel with the centre horizontal line, and on the top tangent on each side of the vertical centre line, set off the lap according to the scale: the lap being $1\frac{1}{2}$ inches actually, $1\frac{1}{2}$ divisions in the scale is the length required, each main division in the scale being virtually inches. Draw vertical lines from the laps cutting the circle, and prolong them to the bottom tangent. On each side of the centre line set off the lead of the slide valve, at the same scale as the lap, on the lower tangent, and from the laps set off the leads also. From these last points draw lines parallel with the vertical centre line, to intersect with the circle above and below; connect these intersections by angles crossing each other at the centre of the circle. Then with the chord *a* as a radius, and *b* as a centre, describe an arc on the circle, in reverse localities, and each last intersection is the centre of the crank pin, when the slide valve has closed the supply ports, during the forward and backward strokes of the piston. The curve seen intersecting with the circle's centre is described by the connecting-rod when the piston is at half stroke; and, by a similar process, the position of the piston can be ascertained at the points of cut-off, which in this case is a mean of 12·82 inches from the commencement of the stroke. As the top and bottom tangents represent the steam and vacuum sides of the slide-valve, the position of the latter, and angle of the eccentric, are also obvious.

Professor Rankine, when treating of the Slide-valve Gearing, in his work of "Rules and Tables," states:—

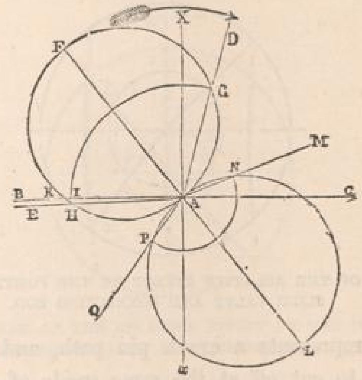
By the angular advance of the eccentric is to be understood the angle at which the eccentric radius stands in advance of that position which would bring the slide valve to mid-stroke when the crank is at its dead points.

Rule I.—Given, the positions of the crank at the instants of admission and cut-off; to find the proper angular advance of the eccentric, and the proportion of the lap on the induction-side to the half-travel of the slide.*

In Fig. 235 let A B and A C be the positions of the crank at the beginning and end of the forward stroke; let the arrow show the direction of rotation; let X *x* be perpendicular to B C; let A D be the position of the crank at the instant of cut-off, and A E its position at the instant of admission. Draw A F, bisecting the angle E A D; A F will represent the position of the crank at the instant when the slide is at the forward end of its stroke; and F A X will be the angular advance of the eccentric.

Lay off the distance A F to represent the half-travel; and on A F as a diameter describe the circle A H F G, cutting A D in G and A E in H; then $\frac{AG}{AF} = \frac{AH}{AF}$ will be the required ratio of lap at the induction-side to half-travel; and A G = A H will represent that lap, on the same scale on which A F represents the half-travel.

Fig. 235.



DR. ZEUNER'S GEOMETRICAL DIAGRAM OF THE SLIDE VALVE AND CRANK.

On the same scale, I K represents the width of opening of the valve at the beginning of the stroke, sometimes called the "lead of the slide." Strictly speaking, this is the lead of the induction-edge of the slide only; the lead of the centre of the slide being A K; that is, its distance from its middle position at the beginning of the forward stroke.

Given the data and results of the preceding rule, and the position A M, of the crank at the instant of release, to find the ratio of lap on the eduction-side to half-travel, and the position of the crank when cushioning begins. Produce F A to L, making A L = A F; on A L as a diameter draw a circle, cutting A M in N; then $\frac{AN}{AL}$ will be the required ratio of lap at eduction-side to half-travel.

About A draw the circular arc N P, cutting the circle A L again in P; join A P; then A P will be the required position of the crank at the instant when cushioning begins.

Rule III.—Given, the data and results of Rule I., and the position, A Q, of the crank at the instant of cushioning; to find the ratio of lap at the eduction-side to half-travel, and the position of the crank at the instant of release—produce F A as before; on A L = F A as a diameter draw a circle cutting A Q in P: $\frac{AP}{AL}$ will be the required ratio of lap at the eduction-side to half-travel.

About A draw the circular arc P N, cutting the circle

* The method used in this and the following rules is that of Professor Dr. Zeuner of the Swiss Federal Polytechnic School at Zürich, published in his treatise on Slide Valve Gearing, entitled *Die Schiebersteuerungen.*

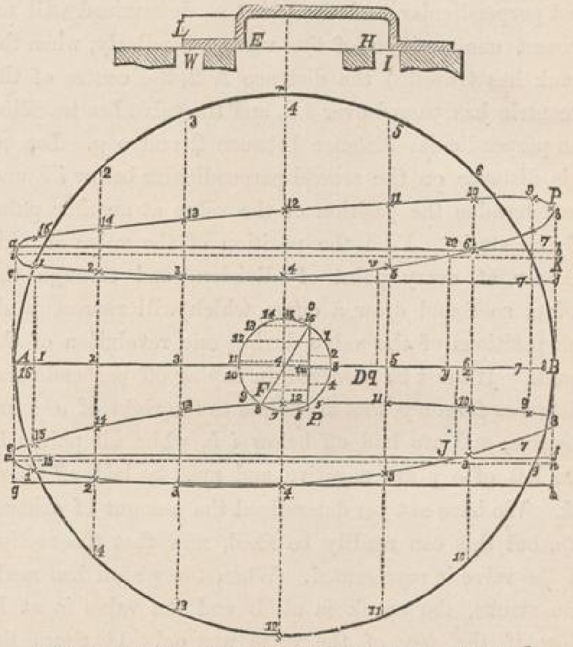
A L again in N; join A N: A N will be the position of the crank at the instant of release.

Given, the angular advance of the eccentric, the half-travel of the slide, and the lap at both sides; to find the positions of the crank at the instants of admission, cut-off, release, and cushioning. Draw the straight lines B A C and X A *x* perpendicular to each other, and take B and C to represent the dead points. Let the arrow denote the direction of rotation. Draw F A L, making the angle F A X = the angular advance of the eccentric; and make A F = A L = half-travel. On A F and A L as diameters, draw circles. About A, with a radius equal to the lap at the induction-side, draw an arc cutting the circle on A F in H and G; also, with a radius equal to the lap at the eduction-side, draw an arc cutting the circle on A L in N and P. Draw the straight lines A H E, A G D, A N M, A P Q. These will represent respectively the positions of the crank at the instants of admission, cut-off, release, and cushioning.

The information alluded to in the "Cadet Engineer" is described and illustrated as follows:—

Now, if we wish the port to be closed before the termination of the stroke, we make the face of the valve longer, or put on lap. In this case the throw of the valve must be increased by an amount equal to twice the lap. But if excessive lap be put on, it is evident that the travel will be so much increased as to permit the steam to exhaust at an early part of the stroke. To obviate this, we must put lap on the exhaust side of the valve. This has a bad effect in causing the exhaust valve to close too soon. This will be seen clearly in the illustration of the geometrical action of the slide valve, Fig. 236. Let A B equal the length of stroke of the engine drawn to any scale. We will give the valve an amount of lap on the steam side equal to half the breadth of the steam port. The travel of the valve will then be equal to three times the breadth of the steam port. On A B, as a diameter, describe a circle which will represent the path described by the centre of the crank pin, while the piston is travelling twice the distance A B. Divide this circle into any number of equal parts, and draw perpendiculars to A B from every point of division. We shall thus determine the position of the piston corresponding to that of the crank at various points. With the same centre *t*, as that of the circle A *r* B *s*, describe a circle C *o* D *p*, having the travel of the valve for its diameter. This will represent the path of the centre of the eccentric during a revolution of the crank. When the crank is on the centre, the line connecting the centre of the crank pin and the centre of the shaft will be A *t*; so that if the valve had

Fig. 236.



AMERICAN GEOMETRICAL DIAGRAM OF THE SLIDE VALVE.*

neither lap nor lead, the line connecting the centre of the eccentric and the centre of the shaft should take the direction *t r*, perpendicular to A *t*. But in the present case, when we have both lap and lead, we make *t u* equal to the sum of the lap and lead, and through *u* draw a line parallel to *t r*. Connect the point *o* where this line cuts the circle with the centre, and *o t* will be the proper position for the line connecting the centre of the eccentric and the centre of the shaft, when the crank is on one centre. When the crank is on the other centre, this line will appear at *t* F. Divide the circle C *o* D *p* into the same number of equal parts as we divided the circle A *r* B *s*, and draw perpendiculars to *o p* from every point of division. The lengths of these perpendiculars show the distances travelled by the valve at various points. Now, let A B represent the centre of the exhaust port. Then draw *a b* and *c d* to represent the width of one steam port, and *e f* and *g h* for the other. Make *a i* equal to L W, the steam lead, and draw a line *i k* parallel to *a b*. This is the line to which all the measurements must be referred, since the valve commences to move from this position. Thus, when the crank has moved the distance A 1, the centre of the eccentric has moved

* The curve from *a* to P, should be more as that below, *e* to B.

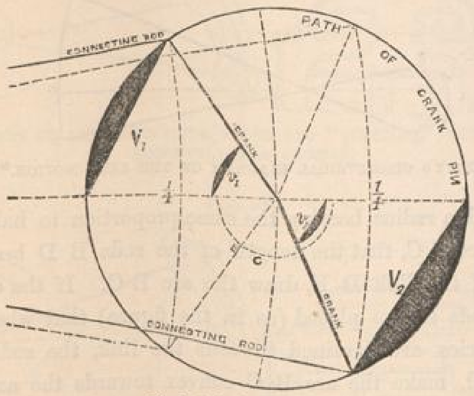
the distance *o* 1, and the perpendicular distance of this point 1 from *o p* will be the distance the valve has travelled. Lay off this distance below the line *i k* on the first perpendicular, and the point so determined will represent one position of the valve. Similarly, when the crank has travelled the distance A 2, the centre of the eccentric has passed over *o* 2, and the valve has travelled the perpendicular distance between 2 and *o p*. Lay off this distance on the second perpendicular below *i k*, and we determine the position of the valve at another point of the stroke. Find the position of the valve in this manner at every point of division, and through the points so found draw a curve which will represent all the positions of the valve during one revolution of the crank. It must be observed, in laying off perpendicular distances from *o p*, that all points to the right of *o p* are positive, and are laid off below *i k*, while all points to the left of *o p* are negative, and must be laid off above *i k*. We have not yet determined the amount of exhaust lap, but this can readily be fixed, now that the motion of the valve is represented. When the piston has made one stroke, the crank is at B and the valve is at F. Now, if the face of the valve was only 1½ times the width of the port, the whole port would be open for the steam to exhaust. So we must put lap on the exhaust side of the valve, and we put on enough to have the exhaust lead equal to H I. This gives us the width W E, of the valve face.

From the curve *a i* P, we can readily find the position of the valve, corresponding to any position of the piston. Thus, when the piston has travelled the distance A *q*, the valve is at *v*, and the distance the port is open is equal to the perpendicular distance between *v* and the top of the port *a b*. At *x*, where the curve cuts *a b*, the port is closed, and the steam expands during the remainder of the stroke from *y* to B.

We can readily lay down the curve described by the lower extremity of the valve. The distance *e m* between *e f*, the top of the port, and *m n*, is the exhaust lead, and we have only to transfer the distances of the various points in the first curve from *i k* to their respective distances from *m n*, on the same perpendicular on which they were first laid down. By this curve we see that the exhaust opening is closed at *j*, when the piston has travelled the distance A *z*.

During some portion of the year 1865, the Author wrote a series of articles over and under his name, which were inserted in the pages of the scientific journal, the "Engineer." He there stated, when alluding to the lap of the valve, thus:—"The rules given by many authors for defining by calculation the amount of lap due to the grade—expansion—and stroke of the piston are useful, and no doubt are of value to a certain extent. Rules, however, if complicated, cause mistrust, and in some instances create confusion of ideas or perception where none should exist. Now, in most if not in all these rules, the travel of the valve is mentioned, which, in fact, is the most important feature in the formulæ. This is, of course, correct, when the width of the ports—or openings—are not known, and the travel of the valve is assumed. Now, as the ports are generally determined before the valve and gear is proportioned, the rules should therefore be *seriatim*." It is, of course, apparent from this that the grade of expansion, being settled, and the width of the opening caused by the slide valve known, the *outside lap should define the stroke of the valve*. On returning to page 347, and consulting Fig. 230, it will be seen that the angles of the crank, at the points 1 and 5, are unequal, although the positions of the piston from each end of the stroke are at equal distances. Now, as the eccentric's point of formation, and the centre of the crank pin both rotate on the same axis, and the angles of each, in relation to each other, are unalterable, *the paths of the crank pin and eccentric are alike in principle of action*. Neither must it be forgotten that all *versed sines* bear a relative proportion to the radii or diameters of the respective circles. The question may be raised, What *can* the versed sine have to do with the lap of the valve? The best answer will be a reference again to Fig. 230 —page 347—it will be seen that, as the *arcs* passed through by the crank pin, from the plane line to 1, and from the plane line to 5, are *unequal* in length, the versed sines must be similarly effected, because the radius is unalterable. Now then for the application of this fact in practice. The illustration, Fig. 237—page 353—represents the path of a crank pin 2 feet in diameter, and the length of the connecting-rod between centres is 5 feet, or 5 cranks; the grade of expansion ¼ from the commencement of the stroke. When the crank pin has passed through the *arc* which produces the chord V_1, the crank is at a given angle, due to the length of the connecting-rod, and the piston has advanced 6 inches. On the completion of the stroke the crank descends, and when the chord V_2, has been produced, the piston has returned 6 inches, but the angle of the crank more obtuse than the former alluded to. It may be added that the *length* of the connecting-rod, in proportion to the stroke of this piston, determines this inequality in all cases. Now, the versed sines, V_1, and V_2, are in proportion to the angles of the cranks, or the lengths of the

Fig. 237.



Scale 1 *inch* = 1 *foot.*

BURGH'S MODE OF DETERMINING THE LAP OF THE SLIDE VALVE, POINTS OF CUT-OFF, AND POSITIONS OF THE ECCENTRICS.

chords of supply, to which attention has been directed by Fig. 231, in page 347. The ratio of the versed sine V_1 is $\frac{1}{16}$th of the diameter of the crank-pin's path, and V_2 $\frac{1}{12}$th of the same.

The proportions of the steam ports in the cylinder must next be noticed: Width of opening caused by slide ¾ inch, lead ¼ inch: so that when the piston is at each end of its path or the crank on the plane line on either side, the slide valve has opened the port ¼ inch, and therefore ½ inch more travel is the completion of the stroke of the valve. To return to the diagram, Fig. 237, is now necessary: it will be seen that near the centre of the circle are two lesser shaded portions, v_1 and v_2, corresponding in form and proportion with V_1 and V_2 respectively; these are termed the versed sines of the eccentric, while the larger are known as the versed sines of the crank. The author has stated in his work on the Slide Valve, that the true versed sine of the eccentric arc of supply steam equals *width of the opening caused by the slide valve minus the lead.* The certainty of this conclusion is evident from

Fig. 238.

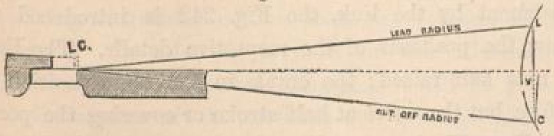


DIAGRAM OF BURGH'S MODE OF PRODUCING THE "VERSED SINE" OF THE ECCENTRIC.

the Fig. 238. This represents a steam port and end of the lap portion of the valve at full travel; the dotted line L is the position of the valve for the lead, and at C when it is closed. It has been clearly proved that the lengths of the connecting-rods affect the travel of the piston, and it is therefore obvious that the lengths of the eccentric rods affect the travel of the valve. The lines L L and C C are the length of the distance between the centres of the crank-shaft and block-pin in the link; the centre of the pin being virtually the ends of the slide valve. Then with L at the dotted line as a centre, describe the arc cutting the plane line and the eccentric circle at L, and with C as a centre describe an arc cutting the circle at C; the distance between the points of intersection on the plane line is the amount of the lead, or as the space between the dotted lines L C. Join L C on the circle by a chord or a straight line, and the intersection at V is the *versed sine of the eccentric.*

Now if the versed sines of the crank equal respectively $\frac{1}{16}$th and $\frac{1}{12}$th of the stroke of the piston, the mean ratio will be $\frac{1}{14}$th. Then if the versed sine of the eccentric equals ½ inch, $\cdot 5 \times 14 = 7$ inches or the stroke of the slide valve, as shown in dotted lines—E C—inside and outside the shaded portions v_1 and v_2 in Fig. 237. Alluding further to this diagram, it is obvious by its use that the lap of the slide can be known; for *any* grade of cut-off, *any* lengths of eccentric and connecting rods, *any* amount of lead, *any* width of opening caused by the slide, *any* length of stroke for the piston, and for *any* deviation; in general practice by the following simple rule: *divide the radius of the crank circle by the versed sine of the crank; multiply the quotient by the versed sine of the eccentric; the product, minus the width of the opening*, equals the outside lap.

For example, the travel of the valve in the diagram is stated to be 7 inches, and the versed sine of the crank $\frac{1}{14}$th of the stroke of the piston; then $\frac{24}{14} = 1 \cdot 7142$, or $\frac{12}{7} = 1 \cdot 7142$, $\frac{12}{1 \cdot 7142} = 7 \cdot 0 \times \cdot 5 = 3 \cdot 5 - \cdot 75 = 2 \cdot 75$ the outside lap of the slide valve.

The positions or angles of the eccentrics outside the crank's centre line are known by the chord of the eccentric's versed sine, intersecting with the eccentric circle.

Width of Ports and Bars in the Slide Valve and Cylinder.—Having settled the question of lap and travel of the valve, it is next requisite to know the arrangements of the ports and the solid spaces between them, termed bars,—which are represented fully in pages 284 and 285—and next, the proportions of the component parts, taking the form of valves illustrated as examples.

Area of Opening caused by the Slide Valve = horse-power nominal × 1 to ·75 square inch (area of supply port = area of opening × 2).

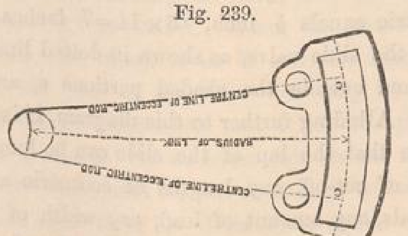
Width of Exhaust Space in Valve = width of two ports supply × 1·5 or 2 + half travel of valve + width of small bar in cylinder − inside lap.

Width of Exhaust Port in Cylinder = width of small bars in cylinder − inside laps of valve, deducted from the width of exhaust space in valve.

Width of large Bar in Cylinder = travel of valve + width of small bar in valve.*

Link Motion—Geometrical Demonstrations.—The main question with this detail is the radius; and the illustration, Fig. 239, shows the general practice of

Fig. 239.



GENERAL MODE OF DETERMINING FORM OF LINK.

determining the same and form of the link, as follows. Describe the circle denoting the path of the eccentric; join C C as tangents; with the centres of the eyes as centres, and the angular length to the centre of the circles as radii, form arcs cutting the circle, and the intersections are the centres of the eccentrics when loose on the shaft, from which position the correct length of the rods can be ascertained.

The radius of the link is the length from C to the centre of the eccentric, or the horizontal distance from the centre of the shaft to the centre of the link.

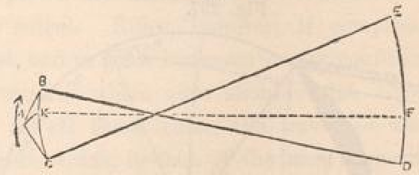
Professor Rankine, in allusion to this matter, states:—

In Fig. 240 let A be the axis of the shaft; A B, the forward eccentric radius; A C, the backward eccentric radius; B D, the forward, and C E, the backward eccentric rods; D E, the link; F, the slider or stud. Radius of curvature of link = length of rods or nearly so.

* When the width of the small port in the slide valve exceeds the width of the opening caused by the slide, this rule must be:—Travel of valve + width of small bar + the excess alluded to.

To find the motion of the slide valve produced by any intermediate position of the stud, such as F.

Fig. 240.

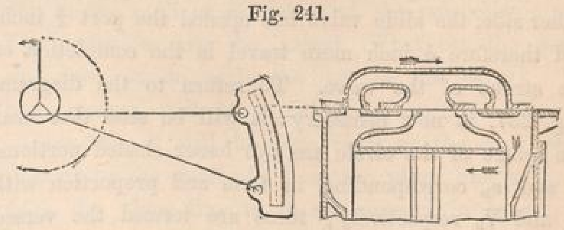


GRAY'S GEOMETRICAL DIAGRAM OF THE LINK MOTION.*

With a radius bearing the same proportion to half the distance B C, that the length of the rods B D bears to that of the link D E, draw the arc B C. If the eccentric rods are so placed (as in the figure) that when the eccentrics are inclined towards the link, the rods are crossed, make the arc B C convex towards the axis A. If the eccentric rods are so placed as not to be crossed when the eccentrics are inclined towards the link, make the arc B C concave towards A. In that arc take a point, K, dividing it in the same proportion in which the stud F divides the link D E. Then the motion of the stud, F, will be very nearly the same as if it were directly connected by a rod K F with a crank A K.

The position of the link when the eccentrics are correctly fixed on the cranked shaft affects the position of the slide valve, as illustrated by Fig. 241. In

Fig. 241.



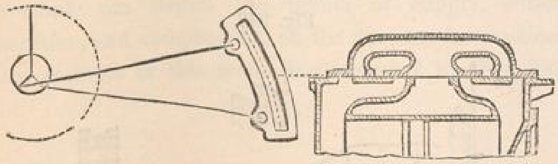
POSITION OF LINK AND SLIDE VALVE FOR STARTING, WITH THE ECCENTRIC RODS CROSSED.

this example the link is down, the crank perpendicular, and the valve cutting off, the different arrows indicate the direction of the moving parts and the flow of the supply steam—this diagram shows the position of the valve for starting. As stopping is the next accomplishment by the link, the Fig. 242 is introduced to show the positions of the respective details. The link is now half raised; the crank in the same position as before, but the valve at half stroke or covering the ports equally. Having started and stopped the piston, the

* This construction is due to Mr. M'Farlane Gray (see his Geometry of the Slide Valve).

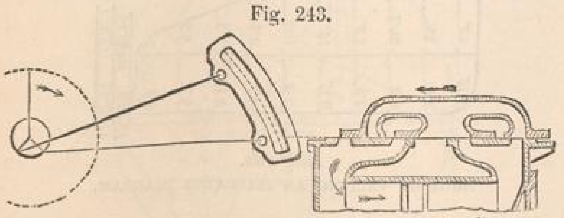
process of reversing next follows in succession, and the position of the link, crank, and valve is depicted

Fig. 242.



POSITION OF LINK AND SLIDE VALVE FOR "STOPPING," WITH THE ECCENTRIC RODS CROSSED.

Fig. 243.



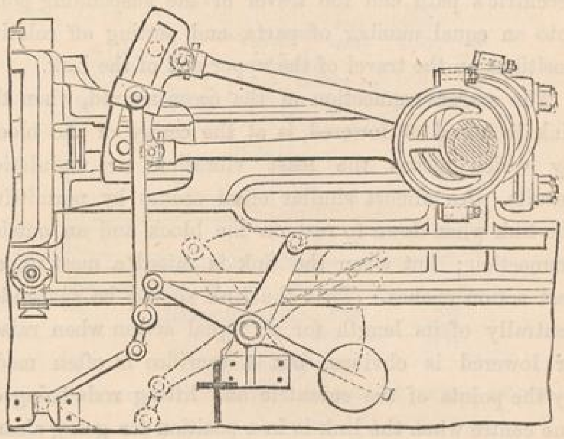
POSITION OF LINK AND SLIDE VALVE FOR "REVERSING," WITH THE ECCENTRIC RODS CROSSED.

by Fig. 243. Here it is shown that a reverse direction for the crank, piston, and valve ensues to that in Fig. 241; and also that the link is up or fully raised, while the crank is assumed to be in the same position in each instance.

Now, the position of the crank for the purposes under notice is favourable; but it may not be out of place to add that in certain positions the valve will not be affected sufficiently to start the engine, unless a slight momentum or movement is in force. With two cranks or coupled engines, in no position whatever can this disadvantage occur, simply because the cranks, being at right angles to each other, the eccentrics are similarly situated. Presume the cranks to be both at an angle of 45°, the pistons will be progressing in opposite directions; and when both links are raised or lowered, the valves will likewise affect the motion of the forward or aft engine, as the case may be.

The means adopted for suspending or supporting the link has been illustrated and described in pages 288 to 293 inclusive: the principles of the arrangement are, therefore, almost evident. The main consideration in this matter is, doubtless, the best means of acquiring a truthful motion from the rotation of the eccentric. By referring to most of the illustrations alluded to and the last three diagrams, it will be noticed that the centres of the eccentric and sliding-block are not on the horizontal line, which will be further obvious by alluding to the Fig. 239—page 354. Not only, however, is there this difference of position to consider, but also the motion described by the point of suspension. Suffice it to say, that the less the versed sine produced by the motion of the point of suspension, the more accurate will be the motion of the slide valve in relation to the action of the eccentric. Independently of the screw and sliding-block arrangement for shifting the link, Messrs. Watt have lately adopted the mode illustrated by Fig. 244—being the parallel motion originated by that firm for steam engines. The attainment in this

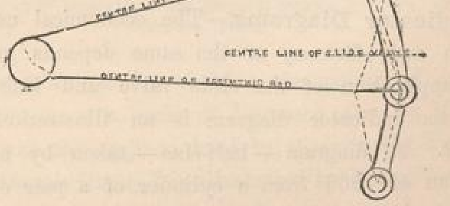
Fig. 244.



MESSRS. WATT'S LINK MOTION.

case is that the link is raised and lowered without affecting the vertical motion of the lower end of the lifting rod, or similar in effect as the screwed rod and sliding block adopted by other firms. To the present it has been stated that the stroke of the eccentric and travel of the valve should be alike, and that the point of suspension should be arranged not to affect that, if practicable. Messrs. Humphrys, however, are doubtless

Fig. 245.



RESULT OF CONNECTING THE ECCENTRIC RODS TO THE ENDS OF THE LINK BEYOND THE BLOCK PIN.*

not of this opinion, as the illustration, Fig. 245, illustrates—being a diagram of the link motion shown in

* See Burgh's Link Motion and Expansion Gear.

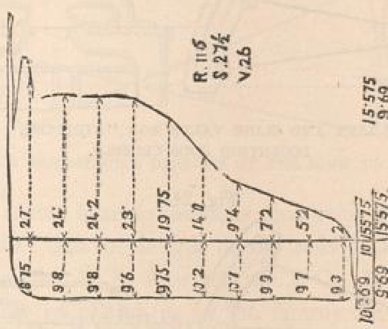
page 290. The eccentric rods being connected at the extremities of the link; the centre line of the slide valve rod between the same when the link is raised or lowered; and the point of suspension at the lower eccentric rod's connection; the travel of the valve must be less than the stroke of the eccentric. Another result also ensues; the top end of the link describes a figure, nearly approaching to an ∞; and to accomplish this, a certain amount of slip must have resulted—the form of this figure can always be determined by dividing the circle of the eccentric's path and the travel of the suspending point into an equal number of parts, and setting off relative positions on the travel of the upper end of the link.

The correct connection of the eccentric rod, when the link is raised or lowered, is at the centre of the block, by which means the least vibration or undulation results. An almost similar effect occurs by permitting the link when down to rest on the block and an outside connection; but when the link is raised a most imperfect action ensues. That the link should be suspended centrally of its length for an equal action when raised or lowered is obvious, but a sacrifice is often made, by the points of the eccentric and lifting rods being on one centre when the link is in a position for going ahead. As twin screw propulsion is now general, the link should be suspended, so that an equal action ensues when the engines are working in either direction, or the hull propelled ahead or astern.

Vertical Motion for shifting the Link.—This is the distance or chord between C C, shown in Fig. 239 —page 354—the general practice = stroke of eccentric × 2·5 to 3; using the latter numeral for short travels of the slide valve. The remaining proportions of the link, rods and eccentrics are obvious on consulting the plates and illustrations.

Indicator Diagrams.—The economical use of the steam and efficiency of the same depends greatly on the application of the slide valve and link motion, and the indicator diagram is an illustration of the result. A diagram — half-size — taken by us in the autumn of 1866 from a cylinder of a pair of marine engines is shown by Fig. 246, being introduced as an exposition. The paper being fixed on the vibrating barrel of the indicator, the steam passages freed from the condensed steam, the motion cord connected to the barrel, and the pointer lightly pressed on the paper, the atmospheric line was formed; the communication between the engine cylinder and the indicator was again opened; the motion being allowed for a moment or two without indication: all being now ready, the pointer

Fig. 246.



Half size.
MODE OF VALUING AN INDICATOR DIAGRAM.

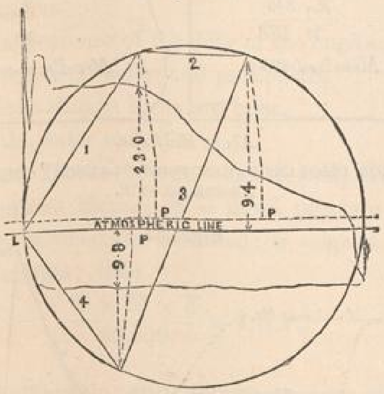
was again pressed against the paper, and the diagram described as illustrated.

At the left hand the poincer commenced from the atmospheric line; it rose instantly to the highest point, and as quickly fell, and performed the undulation to 27·. It followed from thence in an uneven progress to 23·, when the steam was cut off. Expansion now commenced, and as a decrease in the pressure of the steam in the cylinder ensued, the pointer gradually fell from 23· to 14·, when the indented line was traced to 9·4. Expansion now ceased, and as at the same instant exhaustion commenced, the pointer gradually fell to the atmospheric line before the end of the stroke, describing below it the undulated line until the completion. On turning the corner—the return action here commenced —the pointer marked its path in an uneven line until reaching 9·8. Here compression commenced, and continued until the steam was readmitted into the cylinder termed the lead, which produced the round corner at the right hand. The pointer next ascended and joined the line it commenced. By the figure in question the action of the steam on entering and leaving the cylinder is faithfully portrayed.

Now, the veracity of a diagram is sometimes doubted; a belief perhaps circulates in the mind of the sceptic that the figure has been cooked to please the palate of the eye. The genuineness of a figure can always be known in the following manner:—Describe a circle whose diameter is equal to the length of the indicator diagram; describe at the same scale the circle of the eccentric's path; find next the angles of the eccentric, when supply, expansion, exhaustion, and compression commence: from these angles the angles of the crank respectively are easily obtained, and vertical lines projected from the intersections with the main circle or path of the crank pin depict the points of supply, cut-off, exhaustion, and compression, on the diagram in question.

An example of this mode is illustrated by Fig. 247.

Fig. 247.



BURGH'S MODE OF TESTING THE TRUTH OF AN INDICATOR DIAGRAM.*

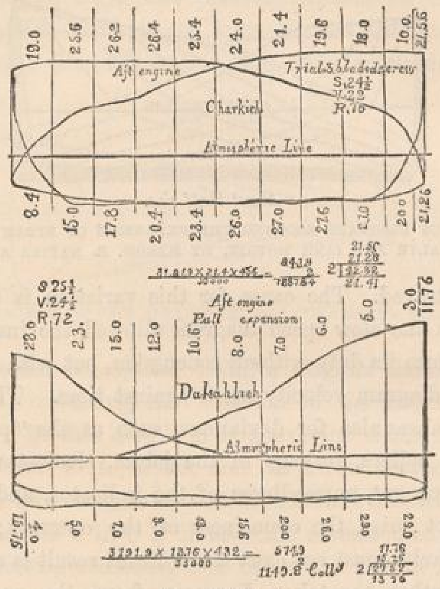
The circle denotes the path of the crank pin, and the chords 1, 2, 3, 4, relate to supply, expansion, exhaustion, and compression respectively; also the intersection with the circle depicts the positions of the crank pin, and the dotted arcs P P P show the positions of the piston, due to the length of the connecting rod. The indicator diagram is presumed to be laid on the circle, the atmospheric line intersecting with the lead point L; by which position the accuracy of the diagram, Fig. 246, is established.

It is the usual practice to indicate from both ends of the cylinder, and thus learn the action of the steam on each side of the piston. Four examples are depicted by Fig. 248, showing the cut-off by the ordinary slide valve, and the result of adopting the expansion valve, the top diagrams being the former, and those below the latter. These diagrams were taken from the engines shown in plate 22 by the Messrs. Rennie, and, as the calculations are given in detail about the figures, their relation will be obvious without description.

Another pair of diagrams is illustrated by Fig. 249, taken by Messrs. Penn from the engines of H.M.S. "Arethusa," illustrated by plate 27. It will be noticed that these examples show an almost equal cut-off on each side of the piston, a result not always produced with some engines.

It has often been argued, and with some cause too, that the adoption of the expansion valve is more whimsical than essential, and that the link motion and slide

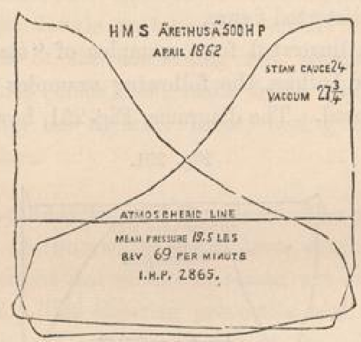
Fig. 248.



About Half size.

DIAGRAMS FROM RETURN ACTION ENGINES, BY MESSRS. RENNIE.

Fig. 249.



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INDICATOR DIAGRAM FROM DOUBLE TRUNK ENGINES, BY MESSRS. PENN.

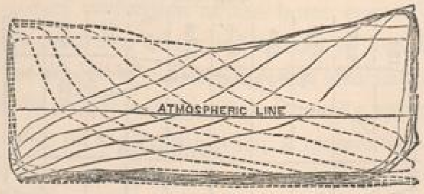
valve are sufficient for altering the points of cut-off. An illustration of a series of diagrams taken by Messrs. Napier from the engines fitted by them in the Turkish frigate "Osman Ghazy," is illustrated by Fig. 250—page 358—the respective diagrams from each end of the cylinder are depicted by plain and dotted lines, which renders obvious their relation without further notice.

Those who have taken diagrams from engines under various speeds and circumstances are aware of the results. Two diagrams, for example, taken from the same cylinder

* See Burgh's Indicator Diagram.

at different speeds of piston will often be of entirely different aspect, although the same points of cut-off may

Fig. 250.



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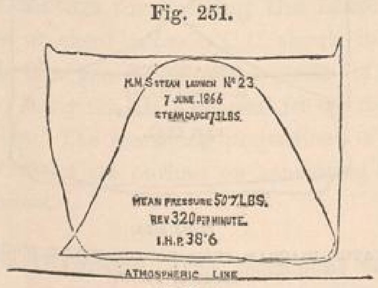
SERIES OF DIAGRAMS SHOWING THE EXPANSION OF STEAM BY THE SLIDE VALVE AND LINK MOTION, BY MESSRS. R. NAPIER AND SONS.

be portrayed. The cause for this variation is that the time in the slow speed diagram allowed the instrument to perform its duty without concussion, but with the fast speed diagram velocity acted against time. There are other causes also for deviations, such as the "priming" of the boilers, leakage of the joints, slackness of the string or gut, unsteadiness of the indicator, and, lastly, but not least, the clumsiness of the operator; indeed, these faults must not exist if a truthful result is required. The author has taken diagrams from the engines of a steam yacht belonging to a gallant admiral, when the piston was making 400 strokes per minute, and by properly arranging the gear a correct result was attained without a distorted figure.

Having illustrated four examples of "steam launch" engines and boilers, the following examples of diagrams are introduced. The diagrams, Fig. 251, have been taken

Fig. 251.



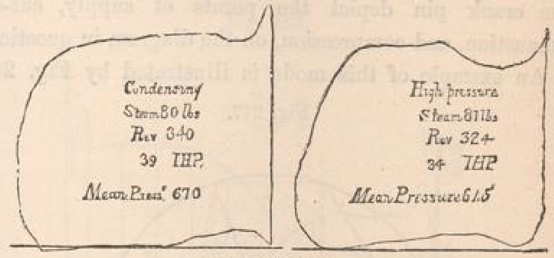
Half size.

INDICATOR DIAGRAMS TAKEN FROM "LAUNCH" ENGINES, BY MESSRS. PENN.

by Messrs. Penn; the engines and boiler being shown by plate 32. The engines and boiler in plate 29 are by Messrs. Rennie, and the diagrams—condensing and non-condensing—are illustrated by Fig. 252 from one end of the cylinder.

Diagrams taken from pumps are of equal value as those from steam cylinders, and three examples from as many pumps are represented in Fig. 253, taken by Mr. Spencer from the pumps belonging to the engines of I.S.S. "Frankfort," the arrangement and details of

Fig. 252.



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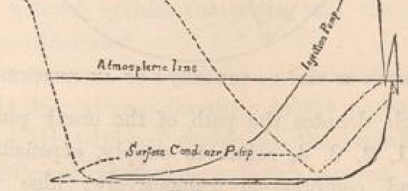
INDICATOR DIAGRAMS TAKEN FROM "LAUNCH" ENGINES, BY MESSRS. RENNIE.

Fig. 253.



Full size.

PUMP DIAGRAMS BY MR. SPENCER.

the condensers and engines of which are illustrated by numerous plates in this work.

The injection and circulating pump diagrams depict the pressure exerted when discharging by the form of the figure above the atmospheric line, and the traverse of the pointer below indicates the amount of exhaustion or vacuum attained, or the suction properties of the pumps.

The notes to be taken when indicating an engine are generally as follows:—

Name of ship.
Type of engines.
Nominal horse-power.
Effective area of piston in square inches.
Length of stroke of piston in feet.
Name and locality of trial.
Time and date of ditto.
End of cylinder (top or bottom, back or front).
Number of strokes of piston per minute.
Grade of expansion in gear.
Pressure of steam indicated by the gauges in the engine and boiler rooms.
Exhaustion indicated by the vacuum gauge.
Amount of injection water passage opened.

Feed water, off or on.

Temperature of the engine and stoking-rooms.

State of fires in the boilers.

Height of water in ditto.

Priming, if any symptoms.

State of the working, bearings of the engine, and shafting from the forward main bearing to the stern tube stuffing box.

General appearance of the duty of the engine.

Type of screw-propeller and particulars.

Speed of the ship in knots per hour.

State of the water and tide.

Indicated Horse-power.—Let E = effective area of piston in square inches; M = mean pressure of steam and vacuum attained, as indicated; S = speed of piston in feet per minute; then

$$\frac{E \times M \times S}{33,000} = \text{I.H.P.}$$

Strains—Tensile and Compression.—The connecting rod of an engine is subject to certain strains—direct and angular being the most prominent, and these act with contrary effects, *i.e.*, compression and tensile. The direction of the circuit of the crank pin also affects the lines of the strains, inasmuch that if the direction is reversed, so are the strains. To exemplify—presume the diagram, Fig. 254, to illustrate

Fig. 254.

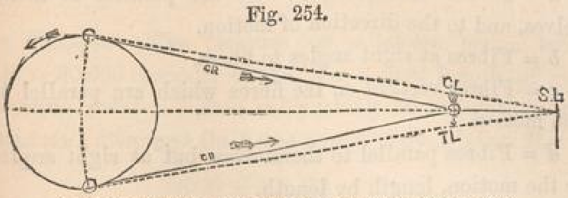


DIAGRAM OF THE STRAINS OF THE CONNECTING ROD.

the circle of the path of the crank pin, angles of the connecting rod at the half stroke of the piston, and the position of the stuffing box of the piston rod. Now, lines drawn from the stuffing box S B to the centre of the crank pin indicate the lines of strain; C L being above the centre line of motion, and T L below. When the connecting rod C R is forcing the crank pin, the effect is to disturb the inertia, but the virtual resistance may be said to be in a reverse direction, as depicted by the arrow above the centre line; the connecting and piston rods being attached at C L—compression line—the strain is in the direction of the downward arrow, or to descend from C L. When the crank returns on the lower half of the circle, the steam acting also reversely, the strain is reversed; here also is no actual resistance, but a decided pull or tensile strain occurs as shown by T L. The point of the rod's connection inclines towards the line of strain T L, and thus the friction on the face of the guide block is the same at each half stroke of the piston. The illustration, Fig. 255, is introduced to

Fig. 255.

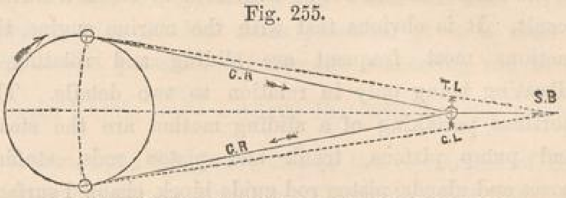


DIAGRAM OF THE STRAINS OF THE CONNECTING ROD.

further assist conception—being a diagram showing the result of reversing the direction of the crank pin's motion. In this case T L is above the centre line of motion, and C L below, and the arrow between C L and T L ascending from and to the lines of strain, the arrows behind C R, C R indicate the reverse direction of the connecting rod above and below the line of motion, and S B is the stuffing box as before.

Not only is the connecting rod exposed to the strains alluded to, but also the pump, slide valve, eccentric, piston, and other rods, receiving an alternate motion. The details partaking of tensile strain only are the bolts of the connecting rod, eccentric, trunk gudgeon, guide block, and main frame cap, also the studs securing the cylinder cover, boring hole cover, and pump doors.

Strains—Shearing and Torsion.—The crank shaft, screw shaft, starting wheel shaft, lever weigh-shaft, and all rods or shafts that vibrate or rotate, are subject to a torsion strain. The shearing properties bear a relation to the crank pin, coupling bolts of the screw shaft, trunk gudgeon, guide block crosshead, link pins, sliding block pin, and all details, receiving a lateral strain at right angles to the longitudinal line of repose or support. It may be added that the crank shaft partakes of the shearing strain also when the cranks are on the horizontal line, or in a line with the centre line of motion. The strain imposed and to be resisted can always be known by remembering the requirement and locality of the detail.

Frictional Surfaces.—The moving friction of an engine has been estimated in some instances as 10 per cent. of the actual horse-power, but the only means of knowing the actual resistance or power requisite to overcome the inertia is to calculate the area of the frictional surfaces and define the nature and effect of the strains exerted on them. Now the nature of friction is due to the direction of the moving surface, in relation to the supporting surface; also the grain or fibre of the metals should not be neglected to attain a correct result. It is obvious that with the marine engine, the motions most frequent are sliding and rotating,—vibrating being only in relation to two details. The portions partaking of a sliding motion are the steam and pump pistons, trunk and piston rods, stuffing boxes and glands, piston rod guide block, channel surface, slide valve and expansion valve rods, stuffing box and gland, slide valve rod guide, stuffing box and gland of feed and bilge pump, plungers, gravity of water, and passage through valves.

The rotating details are the crank shaft bearings, crank pins, eccentrics, expansion gear, thrust block, plummer block, stuffing box, stern tubing, and propeller bearings—the friction of the latter is increased when the propeller overhangs. The vibrating motion is confined to the link motion and suspending pins.

Exceptional to the quantity of the vibrating surface, the preceding remarks apply to oscillating engines also; but as with this type the cylinders and lever blocks vibrate, the result of that movement exceeds that for screw engines.

Table of co-efficients of the friction, with unguents of the ordinary kind, of sliding surfaces :—

Surfaces in Contact.	Co-efficients.		Unguents.
	Starting.	Motion.	
Cast iron upon cast iron . .	0·1000	0·100	Tallow.
Ditto	0·064	0·064	Olive oil.
Cast iron upon brass	..	0·103	Tallow.
Ditto	..	0·078	Olive oil.
Brass upon cast iron	0·106	0·086	Tallow.
Ditto	..	0·077	Olive oil,
Brass upon wrought iron . .	..	0·081	Tallow.
Ditto	..	0·072	Olive oil.
Brass upon brass	..	0·058	Olive oil.
Wrought iron upon brass . .	..	0·078	Olive oil.
Steel upon brass	..	0·056	Tallow.
Ditto	..	0·053	Olive oil.
Wood upon iron	..	0·085	Tallow.
Wood upon wood	0·254	0·083	Tallow.

With reference to the friction of shafts in motion, M. Morin has deduced as follows :—

FRICTION OF SHAFTS IN MOTION.

Designation of surface in contact.	Dry or slightly greasy, or wet.	Oil, Tallow, or Hog's Lard.	
		Supplied in the ordinary manner.	The grease continually running.
Brass on brass	..	0·079	..
Brass on cast iron	..	0·072	0·049
Iron on brass	0·251	0·075	0·054
Iron on cast iron	..	0·075	0·054
Cast iron on cast iron	0·137	0·075	0·054
Cast iron on brass	0·194	0·075	0·054
Iron on lignum vitæ	0·188	0·125	..
Cast iron on lignum vitæ . .	0·185	0·100	0·092
Lignum vitæ on cast iron . .	..	0·116	0·170

Professor Rankine, in allusion to this subject, states: "To calculate the moment of friction of an axle, multiply the resultant load by the radius of the axle, and by the sine of the angle of repose."

This subject has been treated briefly also by Mr. Nystrom in his Pocket-Book, thus :—

Letters denote.

a = Fibres of the woods that are parallel to themselves, and to the direction of motion.

b = Fibres at right angles to fibres.

c = Fibres vertical on the fibres which are parallel to the motion.

d = Fibres parallel to themselves, but at right angles to the motion, length by length.

e = Fibres vertical, end to end.

Example. A vessel of 800 tons is to be hauled up an inclined plane, which inclines 9° 40′ from the horizon; the plane is of oak, and greased with tallow. What power is required to haul her up?

The co-efficient for oak on oak with continued motion is $f = 0 \cdot 097$, say $0 \cdot 1$, then,

$$800 \times \sin. 9° 40' = 800 \times 0 \cdot 16791 = 124 \cdot 328 \text{ tons},$$

the force required if there were no friction, and

$800 \times \cos. 9° 40' \times 0 \cdot 1 = 800 \times 0 \cdot 9858 \times 0 \cdot 1 = 78 \cdot 864$ tons, the force required for the friction only, and

124·328
78·864

203·192 tons, the force required to haul her up.

The effect lost by friction in axle and bearing is expressed simply by the formula

$$P = \frac{\pi d W n f}{12 \cdot 60} = \frac{W d n f}{230},$$

in which W = the weight of pressure in the bearing, d = diameter on which the friction acts in inches, n = number of revolutions per minute, and f = co-efficient of friction. In common machinery kept in good order, the co-efficient of friction can be assumed to $f = 0 \cdot 065$, then

$$P = \frac{W d n}{3530}, \qquad H = \frac{W d n}{1941500}$$

Example. The pressure on a steam piston is 20000 pounds, and makes $n = 40$ double strokes per minute. Required the friction in the shaft of $d = 8$ inches.

$$H = \frac{20000 \times 8 \times 40}{1941500} = 3 \cdot 3 \text{ horses, the loss by friction.}$$

Friction in Guides.

W = pressure on the steam piston in pounds.
S = stroke of piston in feet.
l = length of connecting rod in feet.
H = horse-power of the friction.

$$H = \frac{W S n}{350000 \sqrt{5l^2 - S^2}}$$

Example. The pressure on a steam piston being $W = 30{,}000$ pounds, stroke $S = 4$ feet, length of connecting rod $l = 7$ feet, and making 50 revolutions per minute. Required the horse-power of the friction H.

$$H = \frac{30000 \times 4 \times 50}{350000 \sqrt{5 \times 7^2 = 4^2}} = 1 \cdot 13 \text{ horses.}$$

R. Mallet, Esq., C.E., when alluding to the result of adopting a *short* connecting rod, states: "The *mechanical* disadvantages of the shorter connecting rod are presumed to be comprehended under—1st, Increased friction, produced by the increased rubbing pressure of the guide piece, that secures the movement parallel to itself of the piston rod head; 2nd, A certain amount of increased friction at the joint pin between the piston and connecting rods; and 3rd, A certain amount of increased friction at the crank shaft bearings, due to the alternate lift and pressure down upon the shaft."

The two last sources of increased friction may be passed over as *very* small.

The *increase* of friction due to the first, at its maximum, viz., when the connecting rod makes its largest angle with the path of the piston rod, will be very nearly in the ratio of the length of the connecting rod, divided by that of the crank.

Let us consider what this would amount to in an engine delivering 300 horse-power.

The area of cylinder being taken at 4536 square inches, with 20 lbs. mean pressure, gives a total of 90720 lbs. pressure. If now the connecting rod be six cranks length, $\frac{90720}{6} = 15120$ lbs. will be the greatest pressure upon the guide; and as the co-efficient of friction of brass on iron when lubricated is $0 \cdot 0909$ the pressure; then the increased resistance to the piston's motion, due to this cause, over and above that of an engine with an *infinite length* of connecting rod, will be $\frac{15120}{11} = 1374$ lbs. In the same way, with a connecting rod of only three and a half times the length of the crank, $\frac{90720}{3 \cdot 5 \times \cdot 0909} = 2356$ lbs. is the corresponding increased resistance to the piston. The difference, or $2356 - 1374 = 982$ lbs., is the disadvantage incurred by the shorter rod; but this is the maximum disadvantage, which is zero, when the crank is on either centre; so that the mean resistance is only $\frac{982}{2} = 491$ lbs., over and above that of the connecting rod of even *six cranks* in length, which is longer than employed in any horizontal marine engine; and if this be diffused upon the above area of piston, it will be about 1-9th of a pound per square inch, or 1-180th of the total pressure per inch of piston. This quantity is extremely small—so small that, in the same example, if nine inches diameter were sufficient for the connecting rod, if always in line with the piston—*i. e.*, infinite in length, it would require, when reduced to three and a half cranks in length, to be made about $9 \cdot 1874$ inches diameter for equal pressure on the unit of section. This very simple consideration appears to render it extremely doubtful whether, in the case of the trunk type of engine, the increased length of connecting rod be not achieved at the expense of an appreciable *loss*, in place of any gain in power; from the friction of the trunk, gripped by its packing, over a very extensive surface, the degree of pressure upon which to produce stanchness, can only be at any time conjectured. And this friction is, in relation to the length of the rod, increased by the oblique pressure, just as is that of the piston guide in the former case; so that in our judgment simplicity and come-at-ability of all the working parts should be held paramount to the question of whether the connecting rod be three and a half, or be four or five cranks in length.

Strength of Materials.—The table devoted to this important subject is a *bonâ fide* compilation from the mean of the results of experiments by Rennie, Fairbairn, Napier, Maudslay, Mallet, and Telford.

Name.	Breaking *Tensile* Strain in lbs. per square inch.	Disturbing *Compression* Strain in lbs. per square inch.	Dividing *Shearing* Strain in lbs. per sq. inch.	Twisting *Torsion* Strain in lbs. per sq. inch.	Ditto for bar, one inch diameter.
Wrought iron.	56,000	36,000	50,000	15,360	12,063
Steel	100,000	..	..	25,497	20,025
Cast iron. . .	20,000	110,000	27,700	15,206	11,943
Gun metal . .	35,000	10,350	..	8,000	6,000
British oak. .	10,000	10,000	2,300	2,350	..
Red pine . . .	13,000	6,000	600	1,540	..

The author personally received from G. B. Rennie, Esq., C.E., the following particulars of experiments by the late G. Rennie, Esq., C.E., and also some note of American Practice on the resistance of Metals to Torsion.

"Mr. George Rennie, found by experiment, 1 inch square bar bore 112 lbs. at 3 feet radius, or 336 lbs. at one foot; wrought iron bar is taken 14 : 9; then, 1 inch square bar wrought iron at one foot radius = 984 lbs. American experiments—good castings—1 inch diameter at 1 foot radius will break at 583 lbs.

"Wrought iron 1 inch diameter at 1 foot radius distorts without breaking, 642 lbs., but begins to yield at —permanent load—300 lbs."

Application of Materials.—Before a correct formula can be deduced the *permanent load* must be known, *i.e.*, the weight of the moving material, and the nature of the motion. Now, all rods subject to compression and tensile strains are connected at the extremities; and the load on them is the weight, or inertia of the connecting portions. For example, the piston rods are subject to the strains alluded to, and the load consists of the piston at one end and the crosshead at the other; obviously the length between these details affects the sectional area of the rod. The connecting rod is hung between a sliding and rotating motion; consequently, the load is in some measure due to the length of the rod in proportion to the radius of the circle described. In the first case, the sliding point has a load on it due to the weight of the piston rods beyond the stuffing boxes, crosshead, guide block, bolts and nuts, and a portion of the weight of the connecting rod—the latter mostly when it is ascending from the full stroke; in the second instance, the rotating surface is affected by the weight of the rod, the weight of the cranks, or centrifugal force and inertia of the shafting throughout the bearings. It is for these reasons that makers differ in their proportion; and when an engineer correctly defines the least proportions, he is justified in adopting them due to the reduction of the permanent load. Reverse to this, it is possible to design an engine with such maximum proportions that the inertia exceeds the pressure of the steam on the piston, or similar to constructing a girder that cannot bear its weight, or the permanent load exceeding the resistance to fracture.

It is obvious from these conclusions, that when the sectional area of the piston rod is required, the pressure of the steam must be noticed first; next, the effective area of the cylinder; thirdly, the load and the length of its travel; fourthly, the compression strain on the rod; and, lastly, the factor of safety. The sectional area of the connecting rod is exposed to nearly the same strain as the former detail, but the load is obviated, and the inertia of the crank shafting and propeller takes its place.

Professor Rankine treats inertia, when requiring to know the power to disturb it, as follows:—

To reduce the inertia or mass of the machine to the driving point. Multiply the weight of each moving portion of the machine by the square of the ratio of its velocity to the velocity of the driving point; and add together the products; the sum will be the weight of the mass which, if concentrated at the driving point, would require the same force to produce a given change in its speed, in the course of a given time or of a given motion, that is required by the actual machine.

The crank shaft's diameter is determinable from the length of the crank as a lever, and the total pressure on the piston as a weight on the end of the same, the torsion strain being the divisor. Now, as the strength of shafts are in proportion to the cubes of their diameters, the sectional area, therefore, is not often alluded to in the formula.

In allusion to oscillating paddle engines, the proportions are widely different to screw engines, simply because the permanent load and its travel settles the proportion of the piston rod, as much as the pressure of the steam.

Sectional Area of Piston Rod = *effective area of cylinder in square inches* × *pressure of steam in pounds per square inch* × [*permanent load in pounds* × *stroke of the piston in feet*] ÷ *compression strain in pounds* × *factor of safety*—(*area for two rods* = *preceding area*).

Sectional Area of Connecting Rod.—The correct mode of deducing this section is to consider the angle of the rod, inertia of the shafting and propeller, and number of revolutions per minute. In practice, however, the diameter of the piston rod and the connecting rod are often the same, and universally so when the length of the latter between the centres of connection = *throw of crank* × 3·5. Should this length be exceeded, which it is with return-action engines, the maximum area of the rod in question = *area of the piston rod* + 3 *square inches per foot of the length of the turned portion of the connecting rod*. The ends are often reduced to the diameter of the piston rod; the increased area being preserved only at the centre of the length. It may also be added that symmetry has much to do with this detail—the piston rod being the main determination.

Sectional Area of Connecting Pin.—This detail is supported by the guide block, and secured in each branch or fork of the connecting rod. The strain from the piston rod is transmitted to the centres of the forks, and from thence to the centre of the connecting rod. The pin, therefore, receives shearing and deflecting strains—the former being at the ends of the bearing, and the latter beyond the same. Now, the certainty that the deflection of the pin must disturb the securing bolts of the guide block before the rupture occurs, prompts the idea that the shearing strain is the main consideration in this example when the length of the bearing of the pin equals its diameter. The formula is, therefore, as follows:—*Effective area of the piston in square inches* × *pressure of the steam in pounds per square inch* + [*permanent load in pounds* × *stroke of the piston in feet*] ÷ *shearing strain in pounds* × *factor of safety*. This produces a diameter less than the single piston rod; the practice of many engineers, however, is to make the diameter of both these details alike, and the length of the bearing of the pin in the block = *diameter* × 1·5.

Sectional Area of Crosshead for double Piston Rods.—In this case the detail in question can be treated as a beam supported at the extremities, and the load centrally situated for a certain length. The piston rods form the supports, and the force from the piston rods transmitted to the centre, and met by the resistance, is the load. Now, the strains imposed are shearing at the ends of the bearing, deflection beyond, and compression and tensile combined when a rupture occurs.

It is well known that when a bar of wrought iron is broken across a certain portion is torn asunder, while the remainder is disturbed only, or compressed, without a perfect division. Now, the crosshead is partially exposed to the same result of fracture, *i.e.*, if the detail does not bend, it will be sheared, and if not sheared, the side receiving the force will be compressed, and that opposite extended, or torn asunder; from which result it is evident that the shearing strain must be resisted and the distance between the supports observed to prevent deflection. The correct formula, then, is resolved thus:—*Effective area of the steam piston in square inches* × *pressure of the steam in pounds per square inch* + [*permanent load in pounds* × *stroke of the piston in feet*] ÷ *shearing strains in pounds* × *length in feet of crosshead between centres of piston rods* × *factor of safety*.

The permanent load is considered the same throughout these formulæ, as the weight in motion is unalterable. The factor of safety also is considered alike in each instance. Now, the sums of these two functions are dependent on the weight of the material and velocity of the piston. With the load the travel must be noticed, due to the fact, that the amount of matter shifted or lifted a certain distance in a given time requires a proportionate power, and the material employed a relative resistance to prevent fracture. Obviously, then, if the load moves through one foot in a second, the power and resistance are proportionate to the speed; and if the same weight moves through 3 feet in a second, the power and resistance are also increased. It is, therefore, for this reason that the *load* × *travel* is introduced in the formula. Next, as to the factor of safety,—this multiplier determines the increase of sectional area proportionate to the maximum speed and strains. It also bears a relation to the load and its travel, inasmuch that if the speed is increased, the load will affect the area of the rods in question. The weight of the material comprising the load differs considerably in each type of engine by various makers. The following table of the load and factor of safety is the result of the mean of several examples :—

Type of Engine	Nominal H.P. collectively.	Load = lbs. per sq. inch × area of cylinder.	Maximum speed of piston in feet per minute.	Factor.
Double Trunk	200 to 1500	4·5 to 2·5	1,000 to 1,800	6 to 10
Direct acting	200 to 1500	6 to 4	1,000 to 1,800	9 to 14
Return acting	200 to 1500	8 to 5	1,000 to 1,800	10 to 15

The sectional areas of the pump, slide valve, and eccentric rods, are deduced from the same formula as the piston rods, remembering the permanent load in connection. The link pins and all portions having a lateral stress must be considered as the crosshead, being virtually beams with a central load. The securing or cap-bolts of the connecting rod and main frame are subject to a tensile strain; but when the cap-bolts of the main frame are at right angles with the central line of motion, a shearing strain is imposed. It is obvious, therefore, that any detail of an engine can be correctly proportioned without doubt, by considering the direction of the strains, the permanent load, and the material employed.

Crank Pin, Cranks, and Shafts.—The area of the crank pin = *effective area of the steam piston in square inches* × *pressure of the steam in pounds per square inch* + [*permanent load* × *length of piston's stroke in feet*] ÷ *shearing strain in pounds* × *factor of safety* × *proportion of bearing's length to the virtual diameter.* The length of the bearing or distance between the cranks = *virtual diameter* × 2. For screw engines the diameter of the crank pin and shaft are the same, due to construction rather than correct proportion.

The sectional area of the crank depends on the form and length between the centres of the pin and shaft. With screw engines, construction is observed rather than proportion; the rule generally is, when the *width of the crank equals the diameter of the shaft, the thickness is about three-fourths of the diameter.* The *correct* proportion is known by considering the cranks as beams supported at one end, and the load at the other extremity; the factor of safety depending on the form of the section. The length = throw of the crank; the load = pressure of the steam on the piston in pounds + permanent load and its momentum effect, and the sectional area should be proportioned to the length. With paddle engines the area of each crank is much less than for screw engines, on account of the cranks being forged separately and keyed on the shaft. The average proportion is; *width of the web = diameter of crank shaft,* and the *thickness = half the diameter;* the width being tapered from the main boss.

The shaft of an engine is the transmitter of the power developed and expended at its extremities. The force exerted by the steam is thrown on the crank pin, from thence to the crank, and follows on through the shaft to the opposite end, where the propeller receives it. Now, evidently the strains imposed are not alike; the pin is subject to a shearing strain, the crank deflecting or bending, and the shaft to torsion and shearing strains; and when the latter is exerted, the crank is subject to a compression and tensile strain. The portion between the outer crank and the propeller is exposed to an elongated twisting strain, rather than a direct torsion. With the screw, compression and tensile strains are in force beyond the thrust block; and, therefore, to deduce a correct formula, the type of propeller must be noticed. Another cause for the last consideration is that the area of the propelling agent when acted on by the sea, tends to hold the engine; and thus two forces are straining the shaft in opposite directions simultaneously.

In most cases it is preferred to represent the length of the shaft, and all contingent circumstances, by a constant number or factor of safety. The practice of T. B. Winter, Esq., C.E., has kindly been put at the author's disposal, being as follows. Let

x = *diameter of crank shaft.*
A = *total pressure in pounds on the piston.*
L = *length of crank in inches.*
C = ·000004 *as constant number.*
F = *factor of safety.*

Then—

$$x = F \times \sqrt[3]{\cdot 000004 \times A \times L}.$$

Other engineers adopt the following; when

D = *diameter of shaft in inches.*
P = *total pressure on piston in pounds.*
L = *length of crank in inches.*
T = *torsion strain in pounds for round shafts.*
F = *factor of safety.*

Then—

$$D \sqrt[3]{\frac{P \times L}{T}} \times F$$

Now, when considering the sum of the total pressure, it must be remembered that both pistons, at certain positions, are affecting the crank pins simultaneously in opposite directions; and, therefore, it is not erroneous to recognise the united areas of the pistons. When the propeller overhangs the bearing of the shaft, the area of the latter should be enlarged, as the weight of the propeller acts as a load in constant motion.

The screw shaft should be supported in proportion to its length and diameter, or the distance between the "plummer blocks" should not exceed 12 diameters; the general practice being 8 to 10 diameters. By supporting the shaft correctly, and the adoption of two thrust-blocks—one at each end of the "screw alley," the contingent and constant strains on the shaft are more effectually resisted.

Condensers and Pumps.*—The proportions of these portions are obvious from the various examples quoted in this work. The principles to be noticed are the pressure of the steam in the cylinder, and the lowest temperature available; the cubical contents of the steam employed at each stroke of the piston, which amount is actually what must be condensed at the same time; the temperature of the circulating water, and the amount requisite for each volume of steam, remembering that the *water absorbs* the *caloric*.

The general practice is to consider the superficial area for the tubes and cubical contents for the pumps in relation to the cubical contents of the cylinder. It will be remembered that the circulating water is sometimes forced through or amongst the tubes by the centrifugal pump, separate motive power being often adopted; the theory of the centrifugal pump has been treated at some length, by Joseph Glynn, Esq., F.R.S., in his work on the "Power of Water." He states:—

Of the total work employed in producing rotation, that portion which represents the force in the normal or the centrifugal force is that alone which under any circumstances can become duty in the centrifugal pump.

Generally—

Let G = the weight of a revolving body; and hence its mass $M = \frac{G}{g}$, g having the usual relation to gravity.

r = the radius of revolution.

v = the velocity of revolution in the circumference.

P = the force in the normal, or the centrifugal force.

* See Burgh's "Condensation of Steam."

$$\text{Then } P = \frac{M v^2}{r} = \frac{G v^2}{g r} = 2 \frac{v^2 G}{2 g r}; \text{ hence}$$

$$P : G :: 2\frac{v^2}{2g} : r.$$

That is, the centrifugal force is to the weight of the body in revolution as twice the height due to its velocity is to the radius of revolution.

The resistance being uniform, we may express v in terms of the time of revolution T with the radius r, and

$$P = \left(\frac{2 \pi r}{T}\right)^2 \frac{M}{r} = \frac{4 \pi^2}{T^2} M r = \frac{4}{g T^2} G r;$$

and as the constant $4 \pi^2 = 33 \cdot 4784$,

$$P = \frac{39 \cdot 4784}{T^2} M r = 1 \cdot 224 \times \frac{G r}{T^2};$$

or if n = the number of revolutions per minute, so that $T = \frac{60''}{n}$, then

$$P = \frac{39 \cdot 4784}{3600} \times n^2 \times M r = \cdot 010966\, n^2 \times M r,$$

$$\text{or } P = \cdot 000331 \times n^2 \times G r.$$

Lastly, as $\frac{2 \pi}{T} = \omega$, the *angular* velocity,

$$P = \omega_2 \times M r = \omega^2 \times \frac{G}{g} r.$$

These various expressions for P become convenient for all calculations in which centrifugal force enters.

Where the revolving body is a fluid, and a particle whose weight is G is transferred by the normal force from the axis of rotation to the extremity of the radius r, then the work done, L, is—

$$L = \frac{\omega^2 r^2}{2 g} \times G = \frac{v^2}{2 g} G,$$

v being the velocity of rotation at the *extremity* of the radius r. Or if the particle start not from the axis, but from some intermediate point in a radius, then

$$L = \left(\frac{v_2 - v_{12}}{2 g}\right) G.$$

In the case of a properly proportioned centrifugal pump—

Let $C R = r_1$.

v_1 = the surface velocity of the vanes, which must be proportioned to

H = maximum dynamic head of water to be overcome, and which consists of

z = the elevation to which the water is to be delivered from the lower level.
h = the height due to the velocity of delivery.
h_1 = the head lost in overcoming resistances in the machine.

$$\text{Then } \frac{v_1{}^2}{g} = H = z + \frac{V^2}{2\,g}(1 + \Sigma f),$$

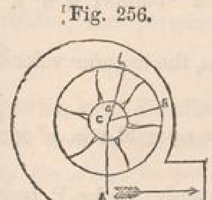
V being the velocity in the ascending main of the pump, and Σf the sum of the several resistances; and the surface velocity of the blades is—

$$v_1 = \sqrt{\left(g\,z + \frac{V^2}{2}\left(1 + \cdot 025\frac{d}{z}\right)\right)}$$

d being the diameter of the pipe when it is wholly vertical, and therefore its length $l = z$; but when otherwise for $\cdot 025\frac{z}{d}$ we must substitute $\cdot 025\frac{l}{d}$.

Let d, = the diameter of the ascending main, be taken as unity. Then in proportioning the pump, let the external radius of the blades, C R (fig. 256), = $\frac{7}{4}d$;

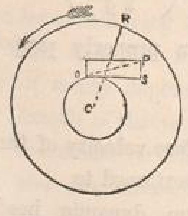
Fig. 256.



the radius of the ears of the pump = $\frac{3}{4}d$; and the diameter of each of the indraught passages = d. The breadth of the blades = $\frac{1}{4}d$ nearly, and the mean radius of the casing of the pump = $\frac{7}{4}d \times \frac{v_1}{V}$ = C A, fig. 256.

The fan-blades should be perfectly radial at the outer extremity, and for at least one-half their length. The inner portion should be curved, as in fig. 257, forwards

Fig. 257.



in the direction of revolution of the fan, and should so reach the inner edge of the revolving disc of blades, that the angle $p\,o\,s$, = β, should be—

$$\beta = \frac{V_0}{V_2}$$

V_0 being the radial velocity of the water, and V_2 being that of the inner edge of the fan-blade.

As regards the power required to drive a centrifugal pump, and to raise per minute a given weight of water, W, it may be taken at

$$2\,W\left(z + \frac{V^2}{2\,g}\left(1 + \cdot 025\frac{l}{d}\right)\right)$$

for very few of such pumps in reality return in duty more than fifty per cent., and the great majority far less.

Propellers — Paddle and Screw.— Commencing with the paddle-wheel, attention must be directed to the illustration in page 240. The mode of setting out the floats, or the pitch, is as follows:—Determine the number of floats to be immersed at the same time; which conclusion is due to the area of each float, in proportion to the friction of the vessel to be propelled. This can be more readily appreciated by remembering that the weight of the vessel and its form defines the friction to be overcome; and also that momentum assists the vessel in her progress when the inertia is fully overcome, or what is often termed the way in action. It is obvious, then that the areas of the floats immersed simultaneously are the propelling agents of the vessel, and therefore the starting power must be the main consideration. For tug steamers a broad float is preferred, because speed is sacrificed to power; but with steamers with fine lines, narrow floats are adopted, because the momentum shall not be retarded by the width of the propelling surface. Another mode of settling this question is, to recognize the maximum area of the displacement, proportionate to the length of the immersed portion in front of the paddle-wheel, by which the resistance to be overcome is known. But while acknowledging these facts, it must be remembered that, if the section of the displacement is enlarged behind the paddle-wheel, a drag results, and thus the momentum is impeded.

Professor Rankine's views on the subjects of displacements of hulls, friction, and paddle and screw propulsion, are expressed in his "Rules and Tables," thus:—

"Given the intended greatest speed of a ship in knots;

to find the least length of the *after body* necessary, in order that the resistance may not increase faster than the square of the speed; take *three-eighths* of the square of the speed in knots for the length in feet.

To fulfil the same condition, the *fore body* should not be shorter than the length for the after body given by the preceding rule, and may with advantage be $1\frac{1}{2}$ times as long.

To find the greatest speed in knots suited to a given length of after body in feet; take the square root of $2\frac{2}{3}$ times that length.

When the speed does not exceed the limit given by the above, to find the probable resistance in lbs.; measure the *mean immersed girth* of the ship on her body plan; multiply it by her length on the water line; then multiply by 1 + 4—mean square of sines of angles of obliquity of stream lines.—The product is called the *augmented surface.* Then multiply the augmented surface in square feet by the square of the speed in knots, and by a constant co-efficient; the product will be the probable resistance in lbs.

Additional Resistance of Ship, due to short after body.—Let v be the speed in knots; l, the proper least length of after body, in feet $= \frac{3}{8}v^2$; l', the actual length of after body; S, the area of midship section, in square feet: $\sin^2\gamma$, the mean of the squares of the sines of the angles of obliquity of the stream lines of the after body; then, additional resistance in lbs.—

$$= 5\cdot 66 \sin 2\gamma \cdot S \sqrt{\left(1 - \frac{l'^2}{l^2}\right)}, \text{ nearly.}$$

Co-efficient for clean painted iron vessels, ·01:
„ for clean coppered vessels ·009 to ·008;
„ for moderately rough iron vessels, ·011 and upwards.

For an approximate value of the resistance in well-designed steamers, with clean painted bottoms; multiply the square of the speed in knots by the square of the cube root of the displacement in tons. For different types of steamers, the resistance ranges from ·8 to 1·5 of that given by the preceding calculation.

To estimate the *net* or *effective horse-power* expended in propelling the vessel; multiply the resistance by the speed in knots, and divide the product by 326.

To estimate the *gross* or *indicated horse-power* required; divide the same product by 326, and by the combined *efficiency* of engine and propeller. In ordinary cases that efficiency is from ·6 to ·625—average, say ·613: therefore in such cases the preceding product is to be divided by 200.

Thrust of Propellers.—To calculate the thrust of a propelling instrument—jet, paddle, or screw—in lbs.; multiply together the transverse sectional area, in square feet, of the stream driven astern by the propeller; the speed of that stream, *relatively to the ship*, in knots; the *real slip*, or part of that speed which is impressed on that stream by the propeller, also in knots; and the constant 5·66 for sea-water, or 5·5 for fresh water.

Given, the product of the velocity of advance, in knots, of a screw propeller as if through a solid (= pitch in knots × revolutions per hour) into the slip of that screw relatively to the water in which it works (also in knots); required the product of speed and slip of the stream from the screw, for use in the above Rule. Multiply the first product by $1 - \dfrac{\cdot 8 \text{ pitch of screw}}{\text{circumference}}$. This is a good rough approximation when the circumference is between $1\frac{1}{2}$ and $3\frac{1}{2}$ times the pitch.

The speed of the stream driven astern by feathering paddles is sensibly equal to that of their centres; by radial paddles, to that of their outer edges. The gross power required to drive a radial paddle-wheel is greater than that required to drive a feathering paddle-wheel of equal thrust, in the ratio of

$$\sqrt{\left(\frac{\text{outer radius of wheel}}{\text{height of axis above water}}\right)}, \text{ nearly.}$$

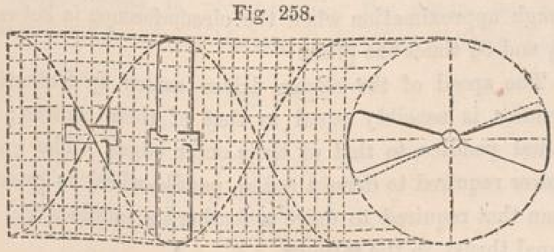
Geometry of the Feathering Paddle Wheel.—An illustration of the relative positions of the floats is depicted by Fig. 70—page 240. To define the throw of the eccentric is the first consideration, which, in general practice, is thus:—

Produce a perpendicular line through the centre of the paddle shaft; describe thereon the centre of the "polygon, or the centre line of the lever shafts; determine the position and length of the lever of the lowest float; connect the top intersection of the perpendicular line and the polygon, by an angular line passing through the centre of the small eye of the lever in question; next draw a line at right angles with the angular line passing through the centre of the shaft; and the intersection with the angular line is the centre of the eccentric.

The position of each float can be readily ascertained by describing a circle equal to the diameter of the polygon on the centre of the eccentric; and as the pitch of the floats and lever shafts is equal, and the length of the levers alike, the points of intersection on the eccentric circle are obvious; also the angles of the floats. The pitch of the centres of the lever rods on the eccentric is equal in this case; but in some instances the pitch is unequal, being proportioned from the inequality of the pitches of the lever's centres on the eccentric circle.

Geometrical Delineation of the "Common Screw Propeller."—The screw is simply a raised surface around the circumference of a cylinder, the ends of the surrounding portion being unconnected, due to the spiral form or longitudinal advance of the curve. Now the screw propeller, in its ordinary application, is the same in principle; or the pitch of the thread, its depth and length, determine the form of the blades. As an evidence of this law, the illustration, Fig. 258, is introduced, being an example in actual practice: the pitch is

Fig. 258.



DELINEATION OF THE COMMON SCREW.

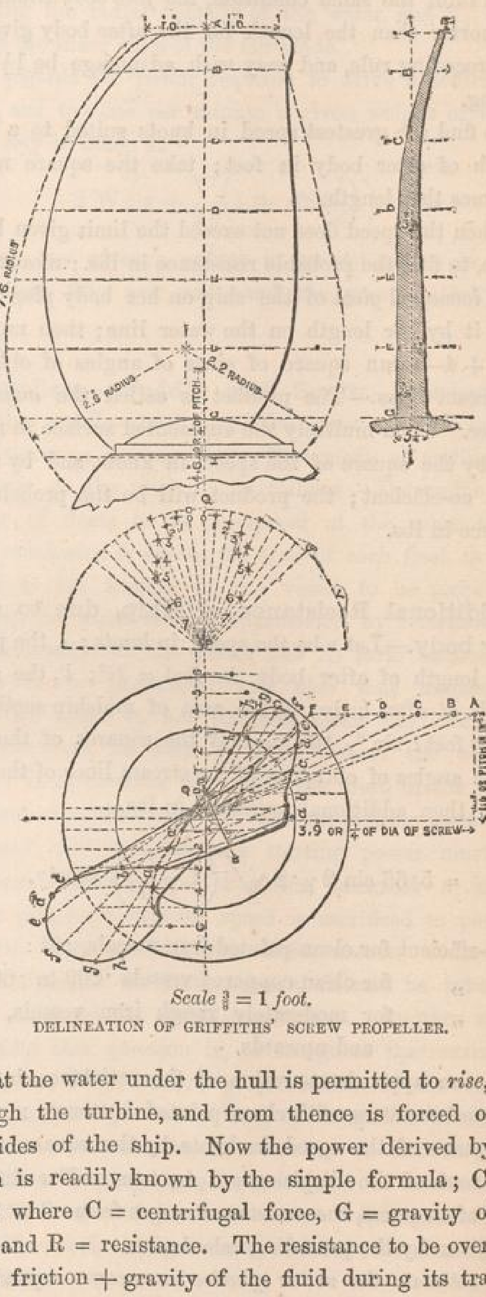
28 feet, the diameter 16 feet, and the length 3 feet. The correct form of the *edge* of the blade is known by *dividing the pitch and circumference into equal divisions*, or by dividing *half* the pitch and circumference into equal parts; and a *continuous curve drawn through the intersections of the horizontal and vertical lines*, denotes the requisition. In the illustrated example, the side, and end elevations, and plan are shown in full lines, to more clearly portray the application of the diagram. The mangin screw propeller is similarly deduced also, by recognizing the variation in the pitch of the leading and following half of the blade.

*** Geometrical Delineation of Griffiths' Screw Propeller.**—The mode of determining the form of this example is fully shown by Fig. 259. As description in this case could not do justice to the principle, it has been considered better to introduce letters and figures of reference corresponding with each view.

* Burgh's "Screw Propulsion."

Water Propulsion.—In page 326, the "propeller" suitable for the mode of propulsion now under notice is illustrated by Fig. 201. The principle of this system

Fig. 259.



Scale $\frac{3}{8}$ = 1 *foot.*

DELINEATION OF GRIFFITHS' SCREW PROPELLER.

is, that the water under the hull is permitted to *rise*, pass through the turbine, and from thence is forced out at the sides of the ship. Now the power derived by this action is readily known by the simple formula; C + G − R; where C = centrifugal force, G = gravity of the fluid, and R = resistance. The resistance to be overcome is the friction + gravity of the fluid during its traverse through the turbine and discharge-pipes; and obviously, the higher the ascent of water, the greater the power requisite. Now the fall of the water outside the hull is the main effect, simply because the friction is almost neutral, and gravity, therefore, in full power; which, to a certain extent, compensates for the friction incurred previously.

The next consideration is the height the water should be raised, to produce the best *effect*, which is only evident by knowing the exact length of the unbroken volume from the outlet at the ship's side proportionate to its sectional area. That this water is of vital importance is obvious to all who consider that a spray cannot be as powerful in its impact as a solid volume of similar contents, due to the interspersion of the *air* between the currents.

Now, according to Professor Rankine, the "gross power of a fall of water" = "the weight of water discharged in a given unity of time into the total head; that is, the vertical elevation of the upper surface of the water at the points where the fall in question begins and ends." The Professor's mode of expressing this is thus:—Let Q = the flow or volume of water discharged in cubic feet per second; D = weight of a cubic foot of water in lbs.; H = the total head: then D Q H = the gross power in foot lbs. per second, which being divided by 550 = the gross horse-powe

As, however, there is a certain loss of *head* power, from the energy being expended at that point, the consideration of the loss is requisite, and the formula by Professor Rankine for this purpose is arranged thus:—$(l-K)$ D Q H = the effective power when K = the loss of the "head" power. The efficient power = $(l-K)$ when KH = the loss of head, and $(l-K)$ H = the effective head.

Proportions of Paddle Wheels.—The rules here given are approximately correct for practical purposes, having been deduced from practical results.

Depth of central immersion for the lowest float from the line of flotation = *draft of hull* ÷ 2 *to* 3.

Diameter of polygon = *depth of the central immersion of the lowest float from the line of flotation* × 5 *to* 7.

Width of float = *depth of the central immersion alluded to* × 1 *to* ·75.

Area of each float = *area of the immersed section of the wheel parallel with the keel of the hull, from the line of flotation to the centre of the float* × ·75 *to* 1.

Pitch of the floats = *width* × 1·5 *to* 2.

Nominal horse-power for each paddle-wheel = *total area of the floats immersed in feet* × 1·3 *to* 2.

These proportions are proportionate to the speed of the wheel and the draft of the hull, being further explained in the Author's "Pocket-book of Rules."

Proportions of the Common Screw Propeller.— Depth of the top of the blade's immersion from the line of flotation = *diameter of screw* ÷ 10 *to* 12.

Pitch of blade's edge = *diameter* × 2 *to* 1·25.

Length of screw = *diameter* ÷ 5 *to* 6.

Proportions of Griffiths' Screw Propeller.—

Diameter of boss = $\frac{\textit{diameter of screw}}{3 \textit{ to } 4}$

Length of boss = *diameter of flange* + ½ *diameter of shank, in some cases diameter of boss.*

Diameter of flange of blade = *diameter of boss* × ·5.

Thickness of flange at edge = $\frac{\textit{diameter}}{20}$

Lap of blade on boss beyond flange = ¾ *of an inch per foot of diameter of screw.*

Width of blade at widest part = $\frac{\textit{diameter of screw}}{3}$

Width of blade at point = $\frac{\textit{diameter of screw}}{7}$

Thickness of blade at root = ⅓ *of an inch to each foot's diameter.*

Thickness at point = ⅙ *of that at root.*

Diameter of shank = *diameter of boss* × ·25.

Metal around shank = $\frac{\textit{diameter of boss}}{23 \textit{ to } 24}$

Metal beyond flange and cotter = $\frac{7}{12}$ *of depth of cotter.*

Width of main cotter = *d ameter of shank* × ·5.

Thickness of main cotter = $\frac{\textit{diameter of shank}}{6}$

Thickness of feathers in boss = $\frac{\textit{diameter of boss}}{40}$

Width of small cotter = $\frac{\textit{diameter of boss}}{20}$

Thickness of small cotter = $\frac{\textit{width}}{2}$

Angle in side of wedge box = 7½ *degrees.*

Metal in cheeks where cotters enter = $\frac{\textit{diameter of boss}}{40}$

Thickness of plate over wedges = $\frac{\textit{diameter of boss}}{48}$

Blades of screw to curve forwards ½ inch, to each foot of diameter of screw, from face at root, curve to commence at centre of blades.

Positive Speed and Slip of the Paddle Wheel. —This matter resolves itself into simple calculation in the following manner:—

Speed of floats = *circumference of central line of immersion or polygon* × *unit of time.*

Slip of wheel or floats = *result of the preceding formula* − *actual speed of ship.*

Positive Speed and Slip of the Screw Propeller.—

Speed of screw per minute = *pitch of screw* × *number of revolutions per minute.*

Theoretical speed of ship in knots per hour =

$$\frac{\textit{speed of screw in feet per hour}}{6080 = \textit{Admiralty knot in feet}}$$

Loss of speed or slip of screw = *theoretical speed of ship minus actual speed of ship.*

Actual speed of ship = *speed of screw minus slip.*

To ascertain the actual pitch, required at a given speed of the screw, to produce a given speed of the ship, the rule will be as follows:—

Pitch of screw in feet =

$$\frac{\textit{actual speed of ship in feet per hour}}{\textit{number of revolutions of screw per hour} \times \cdot 9 \textit{ to } \cdot 75}$$

This rule allows a slip or loss of speed of 10 to 25 per cent., 20 per cent. being the average for war ships.

Negative Slip of the Screw Propeller.—The pitch of the propeller multiplied by the number of revolutions in a certain time = its relative advance. Now if the progress of the hull to which the screw is fitted falls short of this sum, the result is termed propeller or positive slip; but if matters are reversed, the application of hull or negative slip results. Now the cause of this latter effect has not been rendered obvious, although various and talented authorities have investigated the subject. The author's ideas on this matter are—that the advance of the speed of the ship over that of the screw is due to the pressure of the water on the after-body of the hull directly in advance of the screw, and that the form of the hull and pitch of the screw determine this result.

In the case of the after lines of the hull being fine, and those forward full, the water displaced forward will incline inwards when the maximum section of the displacement is passed. The principle of the screw's action is, then, to drag the water immediately in front of it; and doubtless with fine lines aft, and a propeller with a coarse pitch, a backward vortex is also caused, and thus the surrounding volume presses against the stern portion of the hull, thereby producing negative slip for the propeller and *positive* slip of the hull. This theory has also been proved by the author not to be without foundation in practice. By dropping a piece of wood in front of a revolving propeller at full speed, the wood was forced directly in a line with the ship's progress against the hull; thus proving that a current of water was in advance of the propeller, proceeding at a greater speed than the hull.

A Paper on "Some Remarks on Apparent Negative Slip," by Professor Rankine, was read at the Institution of Naval Architects on the 11th day of April, 1867, and is here reproduced with the sanction of its author.

"1. When the attempt has been made to account for the apparent negative slip of a screw propeller by the fact of its laying hold of a current of water that is following the ship, this objection has been raised: that the forward momentum impressed on that current in a second is equivalent to the resistance of the ship; that the backward momentum impressed by the screw on the propeller-race in a second is equivalent to the thrust of the screw, which is equal and opposite to the resistance of the ship; and that, consequently, even if the screw were to take hold of every particle of the following current, that fact would account for a diminution of positive slip only, but not for negative slip.

"2. If the velocity of the following current in which the screw worked were simply the mean forward velocity of the ship's wake, the objection in question would be unanswerable; for it is the momentum per second due to that mean velocity which is equivalent to the resistance of the ship, and to which the reasoning just mentioned applies.

"3. But the water, affected by the passage of the ship through it, has various reciprocating or wave-like motions combined with the mean velocity of the wake; and, in particular, there is forward motion under every crest, and backward motion under every hollow, of the waves that accompany the ship. The velocity of those reciprocating motions is not connected directly with the resistance of the vessel—in fact, their resultant momentum is equal to nothing; and it is only the momentum of the uniform current, which remains after the wave-motions have died out, that is equivalent to the ship's resistance.

"4. Hence, if there happens to be, as there generally is, the crest of a following or filling wave under the ship's counter, the water of which the screw lays hold has a temporary forward velocity over and above the permanent velocity of the wake; that temporary forward velocity, indeed, may be many times greater than the permanent velocity of that current whose momentum is equivalent to the resistance of the ship; and thus any extent of apparent negative slip may be accounted for.

"5. The existence of a following wave explains also the fact that any considerable apparent negative slip is always accompanied by waste of motive power, the resistance to the motion of the engine increasing in a greater proportion than its speed is diminished. For amongst the laws of wave motion are the following: that all forward motion of the particles in a wave is accompanied by an elevation of level, and that the pressure against a body in front of the wave, due to that elevation of level, is exactly equal to the pressure required to impress the forward motion upon the particles of water. Such is the pressure exerted upon the stern of a ship by the wave which follows under her counter, when that wave is undisturbed by the action of the screw. But the screw, by checking or reversing the motion of the particles of water, lowers the level of the crest of the following wave, and diminishes the forward pressure which that wave exerts on the vessel. That diminution of pressure is virtually equivalent to an increase of the ship's resistance; so that the thrust of the screw must be equal not merely to the resistance properly due to the dimensions and figure of the ship, but to that resistance increased by a force equal to the diminution which the action of the screw produces in the pressure exerted on the ship by the following wave. Thus the total thrust of the screw is increased above its effective thrust—that is, above the proper resistance of the ship, in a proportion greater than the proportion in which the speed of the screw is diminished through apparent negative slip, so that the result is an increased expenditure of motive power above what would be required if the screw acted in water not affected by wave motion.

"6. The principles of the preceding paragraph do not apply to uniform forward motion of the particles of water produced by friction, because such motion is not accompanied by the production of a swell, and hence the permanent following current in the ship's wake due to frictional resistance does not give rise to a loss of thrust. as the wave motion of the particles of water does."

THE PRINCIPLES OF THE MARINE BOILER.*

Priming.—This phenomenon is the terror of the engineer and stoker, inasmuch that it occurs without warning, and as suddenly ceases. The stoker may turn his back for a moment or two, and on viewing the water gauge on his return, may find it empty, instead of half full as before. The work of an instant is to open the furnace doors, close the dampers, and either damp or draw the fires, while the engineer has put on the feed from the donkey. Apart from the danger of explosion and burning the boiler, "priming" impedes the progress of the engines, and in some instances bursts the covers and ends of the cylinders when the relief valves are insufficient to release the water in time to prevent fracture.

Now, as to the causes for priming, they are mostly, change of the feed water, smallness of the steam space in the boiler, lowness of the steam space above the water line in the boiler, the non-congregation of the steam, or the exit of the steam exceeding the supply—being the result of opening the stop valves too much, incorrect proportion of the heating surfaces, and the arrangement internally. Obviously, then, if these causes are known, the engineers, designing on land and in charge at sea, should not fail to obviate them as far as practicable.

Feed Pump.—The cubical contents of this detail bears a relation to the pressure of the steam and capacity of the cylinder and steam passages formed therewith. The formula is readily attained by recognising these relations as follows:—*Cubic contents of cylinder and steam passage in feet* × 3 *times the number of cubic inches of water to produce* 1 *cubic foot of steam according to the maximum pressure required* = *cubic contents of the feed pump in inches for one engine.*

The table of the relative volumes of the steam and water being of importance, it is therefore introduced:—

Pressure of Steam in lbs. per square inch.	Cubic inches of Water to 1 cubic foot of Steam.
10	1·7
15	2·0
20	2·3
25	2·6
30	2·9
35	3·2
40	3·5
45	3·8

* See Burgh's "Boilers and Boiler-making."

Pressure of Steam in lbs. per square inch.	Cubic inches of Water to 1 cubic foot of Steam.
50	4·0
60	4·6
70	5·1
80	5·65
90	6·2
100	6·68

Cubic Contents of Steam Space = *four to six cubic feet per superficial foot of grate surface,*

Area of Grate Surface in Square Feet.—This proportion depends on the quality of the fuel employed, and amount of steam required proportionate to the *time* for combustion; also the pressure of the steam must not be neglected, and its traverse from the boiler to the engine. As in practice the combustion is retarded by the slag or clinker forming, it is essential to retain a maximum area, which under adverse circumstances will be sufficient. An exposition of the principles of combustion is afforded in detail in Chapter III.; and, therefore, attention must be directed to it before the matter in question can be thoroughly appreciated. The proportion of the indicated to the nominal horse power must also be noticed, so that if an engineer desires to attain a high ratio, the means of development must be produced in the form of fuel and water to cause the steam.

The practice of the engineers of the present day is to make the area in question = *horse power nominal collectively* × ·68 to ·75 when the indicated power = nominal power × 5, and the consumption of the fuel about 2½ to 4 lbs. per indicated horse-power per hour on the voyage.

Area of Tubular Passage = *area of fire grate* ÷ 6·5 to 7: this is the general practice for construction, but doubtless the effective area is not more than ⅕th when the lower tubes are choked.

Area of Chimney = *area of tubular passage* × ·6 to ·75 when the height = 6·5 diameters, which is the usual proportion.

Area of Safety Valve in Square Inches.—This area must bear a strict proportion to the fire grate, as the utility of the former is due to the effect of the latter: a good and universal rule = *grate surface in feet* ÷ 3, which is the practice in the present day, recognising all the essentialities.

Superficial Areas of Heating Surfaces.—These proportions are fully investigated in Chapter III., both explanatorially and tabular; from which the best practice is evident, and the proportions can be deduced by simple division.

Construction and Strength of Boilers.—The strains imposed are tensile, compression, shearing, and torsion; the latter in an indirect form. Mr. Fairbairn, in his work already alluded to, states, when alluding to the "strength of plates":—

Comparative results of rolled iron as derived from experiment, the Yorkshire plates being unity.

Names of Iron.	No. of Experiments.	Mean Breaking Weight in Tons per square inch.	Mean Breaking Weight in Tons per square inch.	Ratio of the Strength of Plates drawn in the direction of the Fibre, and across it. Also of rolled and faggoted Bars drawn in the direction of the Fibre.
Yorkshire plates .	8	25·514	..	..
Derbyshire plates .	4	..	20·160	1 : 0·7882
Shropshire plates .	4	..	22·413	1 : 0·8789
Staffordshire plates	4	..	20·264	1 : 0·7946
Mean . . .	..	25·514	20·945	1 : 0·8209
From Mr. Telford and Capt. Brown's experiments on bars	..	..	26·41	1 : 1·0351

When treating of the strength of "riveted joints," Mr. Fairbairn's conclusions are:—

Cohesive strength of Plates, Breaking Weight in lbs. per square inch.	Strength of Single riveted Joints of equal Section to the Plates, taken through the Line of Rivets. Breaking Weight in lbs. per square inch.	Strength of Double-riveted Joints of equal Section to the Plates, taken through the Line of Rivets. Breaking Weight in lbs. per square inch.
57,724	45,743	52,352
61,579	36,606	48,821
58,322	43,141	58,286
50,983	43,515	54,594
51,130	40,249	53,879
49,281	44,715	53,879
43,805	37,161	..
47,062	..	..
Mean 52,486	41,590	53,635

The relative strengths will therefore be:—

For the plate . . .	1000
Double-riveted joint . .	1021
Single-riveted joint . .	791

From the preceding results, it will be seen that the single-riveted joints have lost one-fifth of the actual strength of the plates, whilst the double-riveted have retained their resisting powers unimpaired. These are important and convincing proofs of the superior value of the double joint; and in all cases where strength is required this description of joint should never be omitted.

The best proportions, according to the same authority, are :—

Table exhibiting the strongest forms and best proportions of riveted joints as deduced from the experiments and actual practice.

Thickness of Plates in Inches.	Diameter of Rivets in Inches.	Length of Rivets from the Head in Inches.	Distance of Rivets from Centre to Centre in Inches.	Quantity of Lap in Single Joints in Inches.	Quantity of Lap in Double-riveted Joints in Inches.
·19 = $\frac{3}{16}$	·38	·88	1·25	1·25	
·25 = $\frac{4}{16}$	·50	1·13	1·50 } 6	1·50 } 6	
·31 = $\frac{5}{16}$	·63 } 2	1·38	1·63	1·88	
·38 = $\frac{6}{16}$	·75	1·63 } 4·5	1·75 } 5	2·00 5·5	For the double-riveted joint, add two-thirds of the depth of the single lap.
·50 = $\frac{8}{16}$	·81	2·25	2·00	2·25	
·63 = $\frac{10}{16}$	·94 } 1·5	2·75	2·50 } 4	2·75 } 4·5	
·75 = $\frac{12}{16}$	1·13	3·25	3·00	3·25	

The figures 2, 1·5, 4·5, 6, 5, &c., in the preceding table are multipliers for the diameter, length, and distance of rivets, also for the quantity of lap allowed for the single and double joints. These multipliers may be considered as proportionals of the thicknesses of the plates to the diameter, length, distance of rivets, &c. For example, suppose we take three-eighth plates, and required the proportionate parts of the strongest form of joint, it will be—

·375 × 2 = ·750 diameter of rivet, $\frac{3}{4}$ inch.
·375 × $4\frac{1}{2}$ = 1·688 length of rivet, $1\frac{3}{4}$ inch.
·375 × 5 = 1·875 distance between rivets, $1\frac{3}{4}$ inch.
·375 × $5\frac{1}{2}$ = 2·063 quantity of lap, 2 inches.
·375 × $5\frac{1}{2}$ = 3·438 quantity of lap for double joints, $3\frac{1}{2}$ in.

·75, 1·68, 1·87, 2·06, and 3·43 are, therefore, the proportionate quantities necessary to form the strongest steam- or water-tight joints on plates three-eighths of an inch thick.

The formula for "stays" is a simple matter, and can be deduced as follows. Let

x = area of stay in square inches.
A = area of surface in square inches supported by the stay.
P = pressure of steam in pounds per square inch.
S = the strain to be resisted.
F = factor of safety in proportion to the strain.

Then

$$x = \frac{A \times P}{S} \times F.$$

Mr. Fairbairn's conclusion as to the material to be employed for "stays" are—

"It will be found that the iron stay and copper plate (not riveted) have little more than one-half the strength of those where both are of iron; that iron stays screwed and riveted into iron plates are to iron stays screwed and riveted into copper plates as 1000 : 856; and that copper stays screwed and riveted into copper plates of the same dimensions have only about one-half the strength of those where both the stays and plates are of iron. These are facts in connection with the construction of locomotive, marine, and other descriptions of boilers having flat surfaces, which may safely be relied upon, and that more particularly when exposed to severe strain, or the elastic force of high-pressure steam."

In arranging the "stays" retain a distance of sixteen inches between them above the fire-boxes, and twelve inches pitch, below the tubes: a fair specimen of this matter is represented in plate 35.

Superheater.—This apparatus has been illustrated in detail, and as the type determines the proportion, a fixed ratio is not available. For tubular superheaters the superficial area in feet = *nominal horse power* × 1·5 *to* 2; with steam at 80 to 120 lbs. on the square inch, while in some instances 2·5 to 3 square feet per horse-power are adopted for lower pressures

STATISTICAL PROPORTIONS of the WEIGHTS of ENGINES, BOILERS, WATER, SPARE GEAR, &c. &c.

TYPE OF ENGINE.	Weight in Cwts. per Nominal Horse-power.								Cubical Contents of Bunkers in feet.	Number of Days' Steaming.
	Engines.	Boilers.	Water in Boilers.	Propeller and Shafting.	Spare Gear.	Coal Bunkers.	Coal.	Total, including Fittings.		
Trunk	3 to 2·5	4 to 4·3	2·5 to 2·7	1·75 to 2	·7 to ·75	·4 to ·5	13·5	12 to 16	31	4·5 to 6·5 Full Steam. 6 to 8 days expansively.
Direct Acting . . .	4·3 to 4·8	4·5 to 5	2·5 to 2·8	2 to 2·1	·8 to 1	·56 to ·7	13·6	15·6 to 18	32·125	
Return Action . . .	4·5 to 5	5 to 5·6	2·5 to 3	2·25 to 2·3	·98 to 1·1	·68 to ·8	14	17 to 20	33 to 35	

RETURN of TONNAGE, POWER (Nominal and Indicated), DESCRIPTION of ENGINES, and DISPLACEMENT of the Ships "OCTAVIA," "CONSTANCE," and "ARETHUSA," setting forth the RESULTS of the TRIAL from PLYMOUTH to MADEIRA, in a Tabulated Form, showing the Time of Departure and Arrival; when previously Docked; the Total and Daily Number of Hours under Steam; Total and Daily Distance Steamed; Total and Daily Consumption of Coal; Description and Quantity of Coal put on Board; the greatest Number of Revolutions on any Day; Distance run, with the Average Pressure on that Day; and if prevented at any Time from Steaming, the Cause and Period of Stoppage.

"OCTAVIA."					"CONSTANCE."					"ARETHUSA."				
Tonnage.	Horse Power. Nominal.	Horse Power. Indicated.	Description of Engines.	Displacement.	Tonnage.	Horse Power. Nominal.	Horse Power. Indicated.	Description of Engines.	Displacement.	Tonnage.	Horse Power. Nominal.	Horse Power. Indicated.	Description of Engines.	Displacement.
3,161	500	1,839·1 greatest on passage. 1,399·8 mean on passage.	Horizontal direct with double piston-rods, three cylinders, and surface condensers.	3,747	3,213	500	2,322·3 greatest on passage. 1,747 mean on passage.	Engines on Woolf's principle, direct acting, with inclined cylinders, and surface condensers.	3,669	3,141	500	1,881·9 greatest on passage. 1,052·2 mean on passage.	Direct horizontal trunk, with surface condensers.	3,598

RESULTS of the TRIAL from PLYMOUTH to MADEIRA.—(Taken from Reports received from the Ships.)

NAME OF SHIP.	Time of Departure from Plymouth.	Time of Arrival at Funchal.	When previously Docked.	Day of Month.	Number of Hours under Steam Daily.	Distance Steamed Daily.	Coal. Consumption Daily.	Coal. Description.*	Coal. Quantity put on Board.	Greatest Number of Revolutions on any Day.	Distance run on that Day.	Average Pressure on that Day.	Cause and Period of Stoppage.
				1865.	Hours.	Knots.	Tons.		Tons.		Knots.	Lbs.	
"OCTAVIA"	6 p.m. 30 Sept. 1865.	6·45 a.m. 9 Oct. 1865.	12 June 1865.	30 Sept.	6	58·1	9·15	Llangennech and Dunraven mixed, good. Carr's Hartley and Welsh mixed, good. Dunraven Merthyr and Thomas Merthyr mixed, inferior.	300	55·75 on 2 Oct. 1865.	211·6	12·78	30th Sept., stopped for 10 minutes; cause, priming of boilers. 3rd Oct., stopped 2 hrs. 52 min.; cause, heating of centre crank bearing, and to shift brasses. 4th Oct., stopped 5 hrs. 30 min.; cause, shifting brasses of centre crank bearing. 6th Oct., engines stopped at 10·5 p.m., only 15 tons of coal remaining.
				1 Oct.	24	226·5	43·05						
				2 "	24	211·6	51·64						
				3 "	21	152·7	51·15						
				4 "	19	119·2	40·89						
				5 "	24	178·6	47·47						
				6 "	22	105·0	33·39						
				Total . .	140	1051·7	276·74						
"CONSTANCE"	6 p.m. 30 Sept. 1865.	3 p.m. 7 Oct. 1865.	13 June, 1865.	30 Sept.	6	68·85	12·2	Mixture of Dunraven Merthyr, Cameron, Coalbrook, and Gellia Cadoxton, good.	277	53·9 on 1 Oct. 1865.	255·95	21·13 high pressure. 4·15 low pressure.	4th Oct., stopped 10 hrs.; cause, heavy sea and head swell. 6th Oct., 8·50 a.m., eased engines, in consequence of a heavy head sea and westerly gale.
				1 Oct.	24	255·95	50·3						
				2 "	24	219·65	47·2						
				3 "	24	188·40	48·0						
				4 "	14	103·85	26·7						
				5 "	24	192·6	44·2						
				6 "	8	61·4	13·9						
				Total . .	124	1090·7	242·5						
"ARETHUSA"	6 p.m. 30 Sept. 1865.	5·35 p.m. 10 Oct. 1865.	20 Sept. 1865.	30 Sept.	6	67·0	20·925	Llanelly, Llangennech, Lambton's Wallsend, fair.	260	60·5 on 1 Oct. 1865.	255·6	12·76	1st Oct., stopped 10 min.; cause, gudgeon-end of after connecting rod heated. 6th October, engines stopped at 7·15 a.m., only 31 tons of coal remaining.
				1 Oct.	24	255·6	59·675						
				2 "	24	223·8	47·7						
				3 "	24	163·28	50·5						
				4 "	24	95·76	22·55						
				5 "	24	164·08	21·85						
				6 "	8	61·0	5·65						
				Total . .	134	1030·52	228·85						

* The coals used were mixed, in the proportion of one-third North Country to two-thirds Welsh.

Department of the Controller of the Navy, 20th March, 1866.

ROBERT SPENCER ROBINSON.

THE DUTIES OF ENGINEERS IN CHARGE AFLOAT.

Before Starting.—Over-haul condensers, pumps, and valves, test nuts of bearings; while the stokers have filled the boilers and lighted the fires; allow the safety valves to be open until steam blows.

Before Leaving Harbour.—Steam being up, before moving engines, open all the stop valves, feed, &c., fill lubricators, attend to worsteds, see all clear, blow through cylinder and condensers, try engines ahead and astern, before stating ready to start.

Order in Stoke-hole.—The amount of steam pressure required without variation; stoke freely when under steam, but not too heavily, so that when stopping suddenly, the combustion can be lessened by opening the flue or smoke box door and closing the damper doors, and thus the rate of combustion is reduced.

Before Engineer takes Charge.—The one leaving must report all right; and in the event of heated bearings, the one in charge should stop till all is cool.

On Change of Watch.—Engineer in charge examines all bearings, fills lubricators, and inspects fires; in case of cleaning fires, see that the fires are burnt enough for cleaning, and complete his log, which is done hourly.

Engineer on Duty.—Learn pressure of steam in general required; see that the level of water is correct as indicated by the gauge glass and cocks, blowing through to test them. Examine all bearings before taking charge, sound the bilges, or take depth of the water in the engine room and stoking-hole bilges; test the density of the water in the boiler by the salinometer.

In the Case of Stopping Suddenly.—Open safety valves, particularly when the fires are brisk; when captain gives notice when going in harbour, fire light when he requires no more steam. The relief valves should be opened after stopping a quarter of an hour.

In the Case of Heated Crank Pins and Shaft Bearings.—When the symptoms occur, reduce the supply of the steam, turn on the "spray" and "jet" gently, and if the temperature increases, increase the flow of the water: should the latter not be adequate to the occasion, slacken the nuts half a turn at the most, for if the "caps" are loose, a "play" results, injurious to the motion of the engine. Slacken the speed of the engine or stop.

In the Case of the Boilers Priming.—Partially close the supply steam valve; open fire doors, close dampers, turn on feed; and if these emergencies are not sufficient, damp or draw fires; shut steam valves; which latter operations are the last resource.

Regulation of the Feed Water.—Having arranged the proportion of the sea to the condensed steam—if surface condensation—control the supply into the boilers to be proportionate to the formation and expenditure of the steam as far as practicable; but with injection condensers supply the boilers from the discharge-pipe, or hot-well; and (surface) "blow-out" *continuously* to obviate excessive incrustation.

Before Finally Drawing Fire or Banking.—See that there is sufficient water in the boiler for a short stop in port. When a long stay is contemplated, or at the end of the voyage, draw fires and blow out water; finally, take out worsteds, open relief valves, close sea cocks after all is clear, blow through all boiler cocks and gauge, and wipe engines throughout.