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The Steam Engine Indicator and its Diagram.

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Chairman: Mr. J. LANG, R.N.R. (Member of Council).

The Indicator, as its name implies, is an instrument for indicating and recording the action of steam in the cylinder of an engine : like so many steam engine devices its conception was due to the fertile brain of James Watt. It consists of a miniature cylinder to which steam is admitted and acts on a piston simultaneously with that in the engine cylinder ; a helical spring pressing on the top of the piston causes it to return as the pressure beneath it falls ; the strength of this spring is proportioned to the pressure of steam. A drum, revolving synchronously with the stroke of the engine, has wrapped round it a sheet of paper, technically called a " card " ; the turning of the drum is effected by means of a string attached to a reciprocating part of the engine ; the return rotation is accomplished by the reaction of a coiled spring within the barrel. If now a pencil be fitted to the projecting end of the piston rod, and this, while rising and falling be brought into contact with the revolving drum, a shoe-like figure will be traced on the card: to increase the vertical height of the figure beyond that due to the stroke of the indicator piston, which is short in order to make the instrument compact; the pencil is carried on the end of a lever, parallel motion or other device being fitted to eliminate the error which the versed sine would produce, were the pencil to be permitted to describe an arc.

The steam pipes to the Indicator should be as short and direct as possible, and the string for rotating the drum flexible, and led by easy running grooved pulleys through as few and least acute angles as circumstances will permit. A special flexible wire is mostly now used for this purpose; but when whipcord is the medium of motion, before being applied the first time, its elasticity should be "killed" by suspending it with weights attached to the ends for some time before being put into use.

The Indicator should be kept clean, but not necessarily polished: scouring brasswork is apt to leave particles of grit in the delicate places of the mechanism: more especially in the cylinder is this detrimental. The piston is a perfect sliding fit in the cylinder, with grooves in its periphery for steam packing; a grain of emery or bathbrick therein may ruin the instrument. The external motions also, if free from rust, work just as well when dull as bright; the pressure laid on by a heavy hand when polishing delicate parts may slightly distort them, and the next set of cards come out different to what has been usual: and the engineer wonders what has altered in the engines. When applying the Indicator; before admitting steam to it, the pipes should be blown through; then gradually open the steam cock, and permit the piston to work up and down a few times: the presence of a little water is very apt to break springs by "water hammer" projecting the spring too violently. Next, hook on the cord to the drum attachment and bring the pencil point gently into contact with the paper while the piston is at rest, this will trace the atmospheric line. Open the steam cock to one end of the cylinder and again apply the pencil, when the diagram for that end will be drawn. Then turn the cock open to the other end, and similarly describe its card. I hope I may be excused for making these simple preliminary remarks for the benefit of the junior section of our membership.

Fig. 1 was taken from a vertical engine, *F* being the upper, and *B* the lower card. The spring used is equivalent to a scale of $\frac{1}{32}$ in. per lb. pressure, measuring the height, from the atmospheric line on this scale, we get 74 lbs.; the steam gauge indicated 78 lbs., showing a loss of 4 lbs. between the boiler and the engine. Following the perimeter of the top diagram, starting from *a*; the slanting admission line suggests that the valve is opening late, at *b* the valve is full open, and cut-off occurs at *c*. Steam is expanding from *c* to *d*, where release begins to take place; and exhaust is fully open at *e*. It closes at *f*, where the steam remaining in the cylinder is compressed up to the point *a*, where it is met by the incoming steam.

Measure 14·7 lb. below the atmospheric line and draw a line parallel to it, this represents perfect vacuum. The vertical distance *P.H.* is the gross pressure: *A.H.* the pressure as shown by gauge: and *E.H.* the effective pressure.

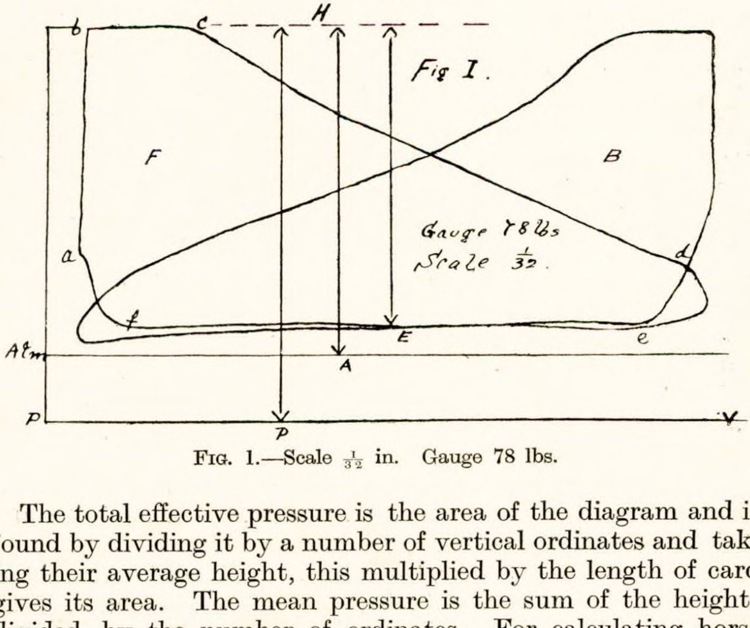


FIG. 1.—Scale $\frac{1}{32}$ in. Gauge 78 lbs.

The total effective pressure is the area of the diagram and is found by dividing it by a number of vertical ordinates and taking their average height, this multiplied by the length of card gives its area. The mean pressure is the sum of the heights divided by the number of ordinates. For calculating horse power, where both ends of the cylinder are taken into account, the area as above found, or by means of a planimeter, is sufficiently accurate: but for comparison of work done at either end of the cylinder, or where great exactitude is required, the heights of the ordinates should be measured from a fixed base, as the atmospheric line ; and the back pressure measured from the *other* card. This is evident when it is considered that the back pressure affecting the work while the steam line is being drawn, is that caused by the steam on the opposite side of the piston, and consequently should be measured from the diagram of that side. In compound engines the receiver pressure often varies and makes considerable fluctuations in the exhaust lines of the cards.

On Fig. 1 it will be seen that on the top card exhaust opens late as well as the admission : the bottom card shows exhaust opening earlier and more cushioning ; lowering the valve a little on the spindle will tend to equalize the diagrams. The cut-off is effected by a separate expansion valve and independently adjusted.

Fig. 2 is from a horizontal engine fitted with Corliss valve gear. The admission and exhaust valves are actuated by separate eccentrics, and the operations of opening and closing are almost instantaneous.

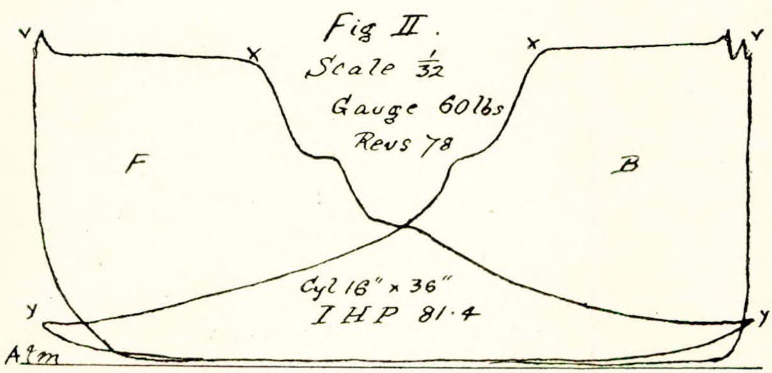


FIG. 2.—Scale $\frac{1}{32}$ in. Gauge 60 lbs. Cyl. 16 in × 36. Revs. 78. I.H.P. 81·4.

The cut-off occurs at *xx*, the curves are of small radius showing that little time was occupied in closing the ports. Release *yy*, was accomplished still more suddenly. The compression at the back end is very small ; it is greater at the other as the engine was found to run easier by admitting it. The humps on the expansion curves, and the vibration at *vv* were due to a weak spring in the indicator.

In reversing engines ; variations of cut-off within limits, can be procured by "linking up". To illustrate this on ideal indicator cards, construct a Zeuner's diagram, viz :—

Draw the line AB, Fig. 3, to scale, say equal to travel of valve; on it describe a circle which represents the path of crank. About centre O describe a circle with radius OL equal outside lap; and a smaller circle $O\ l$ with radius equal inside lap.

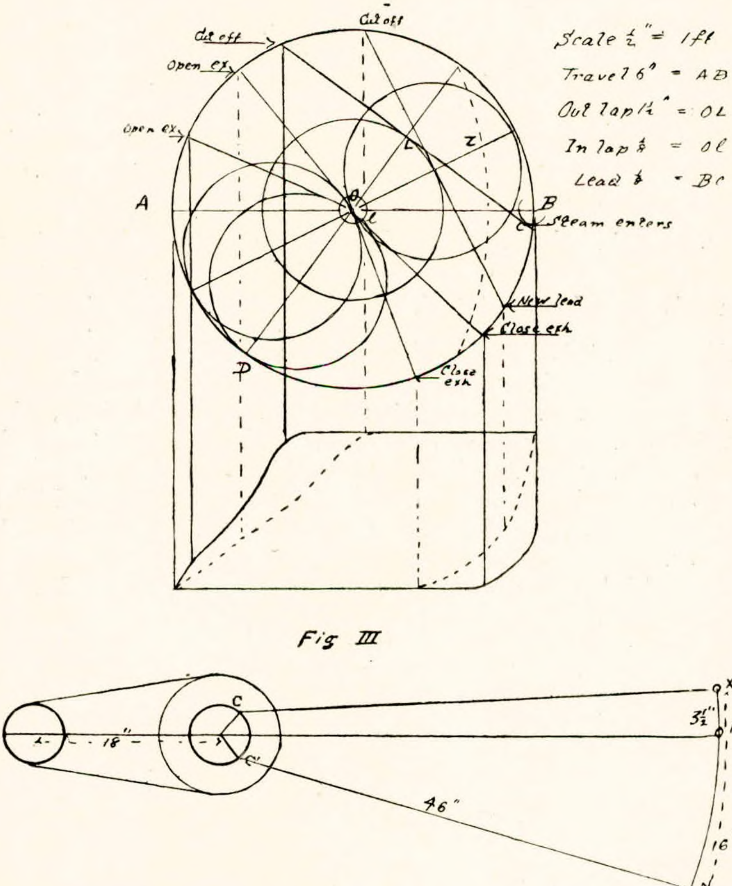


FIG. 3.—Scale $\frac{1}{2}$ in. = 1 ft. Travel 6 in. = $A\ B$. Lap $1\frac{1}{2}$ in. = $O\ L$. Inside lap $\frac{1}{8} = O\ l$. Lead $\frac{1}{8} = Be$.

With B as centre, and the lead for radius, describe the arc Be. A line drawn tangential to this arc, touching the lap circle, and continued to outer circle, indicates the position of crank at cut-off. Bisect this and draw the line and circle *OD*.

Lines from the centre, through the intersections of this and the inside lap circle, indicate the points of opening and closing exhaust. Lines dropped from these several positions enable an approximate diagram to be constructed.

To demonstrate the effect of linking up : let the line sketch represent the crank, eccentric, and link. When in "full gear" the end of the eccentric rod *X* is in line with the valve spindle : suppose the engine is linked in to the position *M*.

On the valve diagram describe the dotted arc shown, the radius being the length of eccentric rod from centre to centre, multiplied by half the distance between the centres of eccentric sheaves, divided by the distance between the centres of pins on the link. Referring to sketch ; radius is $\frac{C.X. \times C.\ C'.}{2\ X.\ N.}$

On this arc take a point *Z*, so that $\frac{C.Z.}{C.C.}$ equals $\frac{X.M.}{X.N.}$ Draw *O.Z.* and on this diameter describe a new valve circle. A perpendicular bisecting this new axis gives the new points of cut-off and admission on the one side ; and an antipodal circle, with lines drawn through its intersections with the inside lap circle gives the new points of exhaust opening and closing. From these a new diagram, shown dotted, may be constructed illustrating the theoretical effect of "linking up," approximately correct.

In multiple cylinder engines, the losses between the cylinders

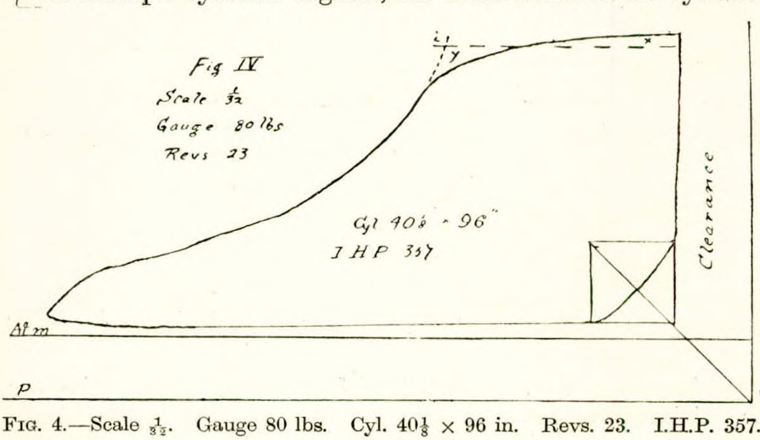


FIG. 4.—Scale $\frac{1}{32}$. Gauge 80 lbs. Cyl. $40\frac{1}{8} \times 96$ in. Revs. 23. I.H.P. 357.

are shown graphically by combining the diagrams on to one common card. To effect this, the clearances must be known, which can be roughly estimated from the diagram by the following method.

On the compression curve describe a rectangle, having two of its corners resting on the extremities of the curve ; a line drawn across the opposite diagonal, continued to the *P.V.* line, will mark the point whence to erect the perpendicular clearance line. Similarly the point of cut-off may be approximated where steam is wire drawn. Draw a horizontal line cutting the curve so that the area *X* equals that enclosed by a continuation of the expansion curve *Y* : the point of intersection *i* is the cut-off.

The above diagram was taken from an engine making twenty-three revolutions per minute, 8 feet stroke, cutting off at 3 feet ;

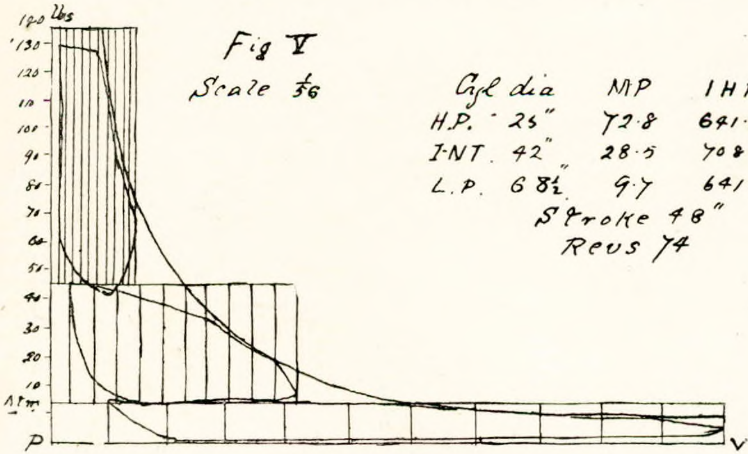


FIG. 5.—Scale $\frac{1}{56}$.

the exact point of cut-off cannot be read from the diagram on account of the slow motion of the valve (Fig. 4).

A practical method of ascertaining the clearance is, to place the piston at end of the stroke, close up the steam port, then fill up through the indicator cock a measured quantity of water, from which can be computed the cubic capacity of clearance of cylinder and steam port.

The following is the combination of the diagrams from a triple expansion engine. The diameters are 25 in., 42 in, and

$68\frac{1}{2}$ in., and these squared give the ratios, reduced to the *L.P.* cylinder, as 1 : 2·82 : 7·5. Draw a line *P.V.* to represent the *L.P.* cylinder, plus its clearance to a suitable scale : in this particular engine the clearances were about one-tenth the volume of the cylinders (Fig. 5).

Erect a perpendicular from *P* and graduate it for pressures, starting from perfect vacuum ; at *Atm* draw the atmospheric line.

Divide *P.V.* into eleven equal parts, viz. one-tenth for clearance, and the remainder for the card. Erect ordinates as shown.

P.V. is the volume of the *L.P.* cylinder : above it, at corresponding pressure, divide a space 2·82 long, to same scale, for the intermediate cylinder : and superimposed on this another space 1·1 long, representing the *H.P.* cylinder plus its clearance. These spaces to be divided by ordinates similar to that of the *L.P.*

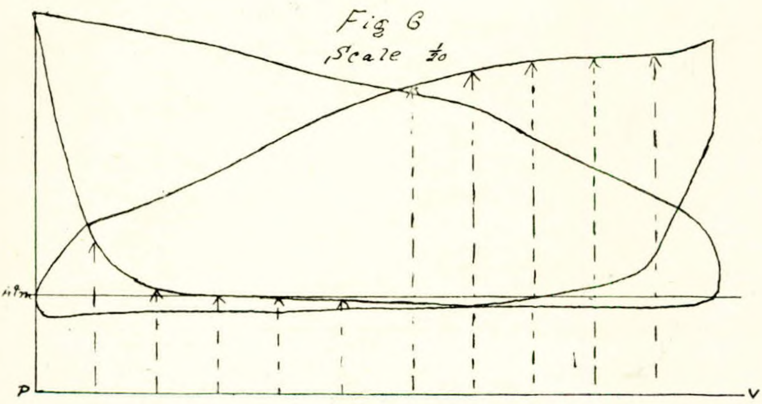


FIG. 6.—Scale $\frac{1}{20}$. Gauge 58 lbs. Cyl. 42 in. × 48 in. Revs. 74. I.H.P. 641·27.

The diagrams from each of the cylinders having been divided into ten parts, it is only necessary to measure the height of each of the ordinates from a fixed base line and transfer them to the combination ; we thus get the *H.P.* and *I.P.* figures shortened in proportion to their volume in relation to the L.P.

The back pressures must be plotted from the opposite cards as already explained, in order to get a correct combination :

the spaces between the figures show the losses between the cylinders ; and those on the left, that due to clearances.

The intermediate cards, Fig. 6, exhibit fluctuations of back pressure, and are sufficient to illustrate the method of plotting, so the cards from the other cylinders are not reproduced.

For comparison with an ideal expansion curve, the hyperbola, being a near approximation to the true curve, is often used : in the set of diagrams, Fig. 5, the actual expansion is almost identical with the hyperbolic curve. The effects of re-evaporation are shown where the figures slightly overlap the preceding ones, showing a higher pressure than that due to the cylinder just exhausted.

The set of cards, Fig. 7, is interesting as being those of the pioneer triple expansion marine engines, and are probably the last record extant of Dr. Kirk's original design.

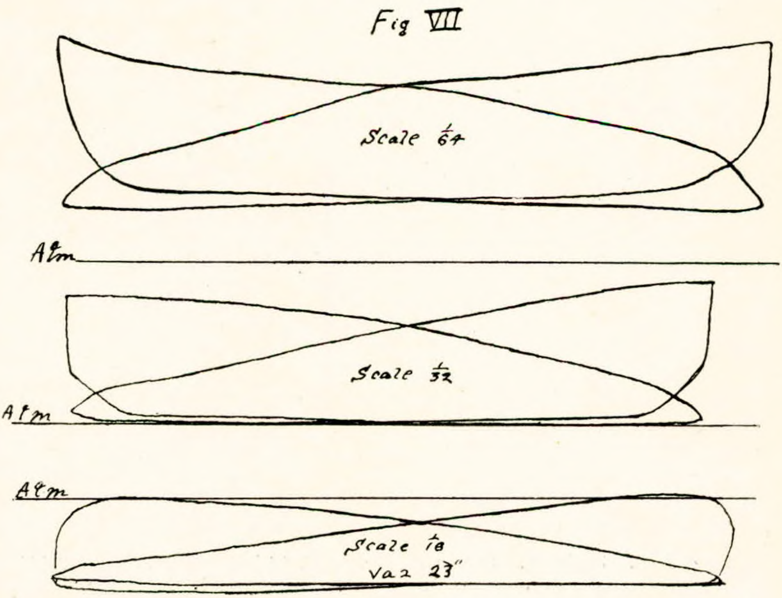


Fig. 7.—H.P. Scale $\frac{1}{64}$. Gauge 140 lbs. I.P. Scale $\frac{1}{32}$. Gauge 31 lbs. L.P. Scale $\frac{1}{16}$. Vac. 23 in. Gauge 4 lbs. Revs. 52.

The ss. *Propontis* marked an epoch in marine engineering, being the first departure from the two cylinder compound of 80 lbs. pressure or thereabouts, which had held sway for twenty years and had apparently reached finality in efficiency; to the modern multi-cylinder high pressure engines.

Here let it be remarked that the terms "compound, triple, and quadruple expansion" are employed in this paper as popularly understood: strictly speaking, they do not in all cases represent the number of cylinders or cranks, and in no case the number of expansions.

Those engines were built in 1874, but at that period boilers were not so perfect as they are now; a serious accident in the stokehold, coupled with constant trouble experienced in the boilers and connections, resulted in one engine being uncoupled, and the machinery ran compound for the next ten

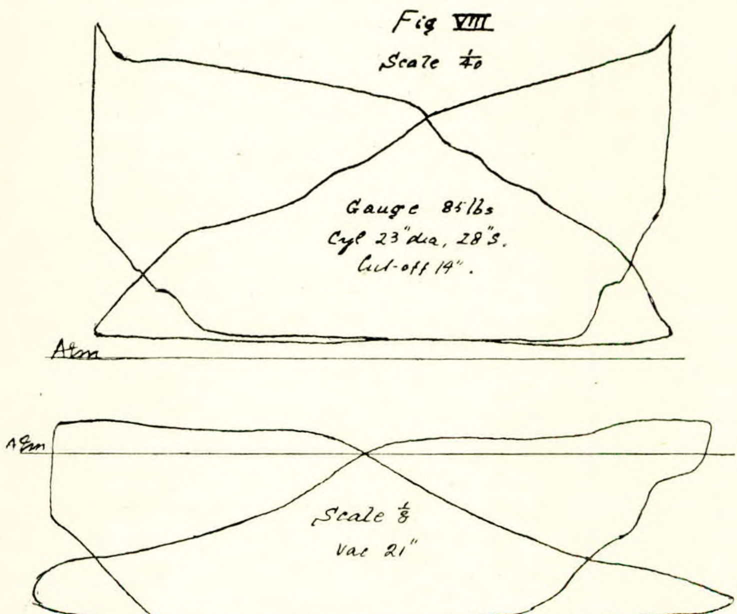


FIG. 8.—H.P. Scale $\frac{1}{40}$. Gauge 85 lbs. Cyl. 23 in. × 28 in. Cut-off 14 in.
L.P. Scale $\frac{1}{8}$. Vac. 21 in. Revs. 106.

years, the development of higher expansion being retarded until the quality of boiler construction caught up with the improvements in the motor.

These diagrams were taken in September, 1894. The engines had been thoroughly overhauled and altered, but the only changes which would affect the diagrams were, the ratio of the cylinders differed ; and the *H.P.* steam jacket, which orginally consisted of its own exhaust was closed up, and the space left as a dead air jacket.

The original cylinder diameters were 23 in., 42 in., and 62 in. : the high and intermediate cylinders having been lined, and the *L.P.* bored out, left the diameters respectively $21\frac{3}{4}$ in., 38 in., and $62\frac{3}{8}$ in. Stroke 3 ft. 6 in. Gauge pressure 140 lbs., receivers 31 lbs. and 4 lbs. Revolutions 52. These engines were scrapped at Shanghai some years ago ; and the handsome hulk of the *Propontis* now serves as a floating warehouse on the Yang-tse Kiang.

Fig. 8 shows an uneven set of cards due to defective valve motion. The gear is of the radial type, where the movement is taken from the connecting rod, there being no eccentrics. The multiplicity of pins and bushes is a detrimental feature in this class of gear ; a slight slackness is intensified by the system of levers, producing erratic diagrams. In illustration, the *L.P.* valve shows considerable leakage at one end.

Another example of leaking valve is depicted by Fig. 9. It is from the *L.P.* cylinder of a Sternwheeler engine ; 38 in. diameter, 6 feet stroke. The valve was large and heavy, of the *D.* type, situated on the side of the cylinder, and the pressure on the back was insufficient to keep it up to the port face.

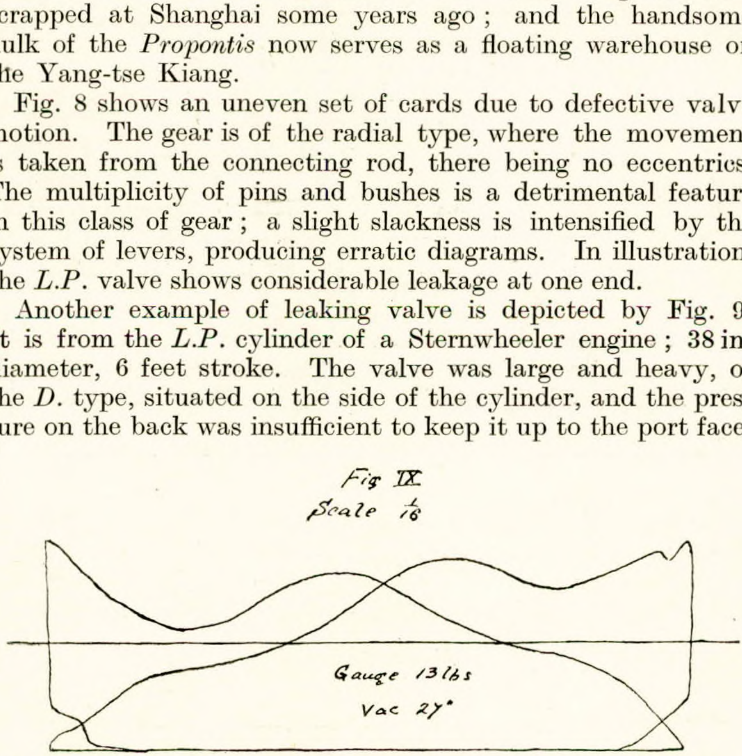


FIG. 9.—Scale $\frac{1}{16}$. Gauge 13 lbs. Vac. 27 in. Revs. $19\frac{1}{2}$.

Indicator diagrams can sometimes deceive. Fig. 10 shows serious throttling on the top card : yet the engines were running well, and the consumption normal. The upper pipe to the indicator was shorter and more direct than the lower, and was of full bore.

The unbalanced cards remained a mystery until the discovery was made that the spigot of the cylinder cover partly closed the steam passage when in place, thus wire-drawing the

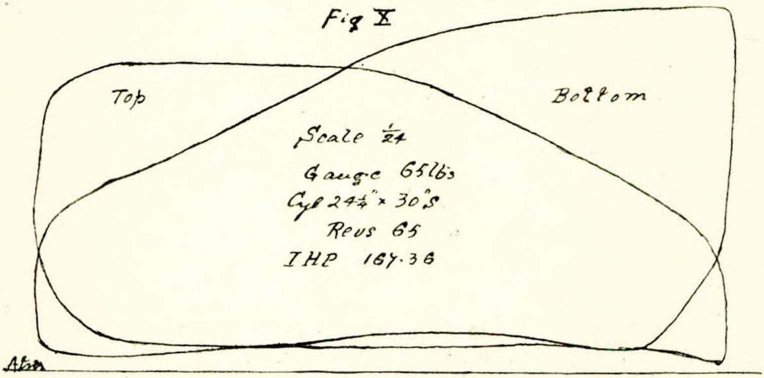


FIG. 10.—Scale $\frac{1}{24}$. Gauge 65 lbs. Cyl. $24\frac{1}{4}$ in. × 30 in. Revs. 65. I.H.P. 167·36.

steam as it entered the indicator. A notch cut out of the spigot opposite the orifice remedied the defect.

Fig. 11 shows the result of nuts being slack on the slide spindle.

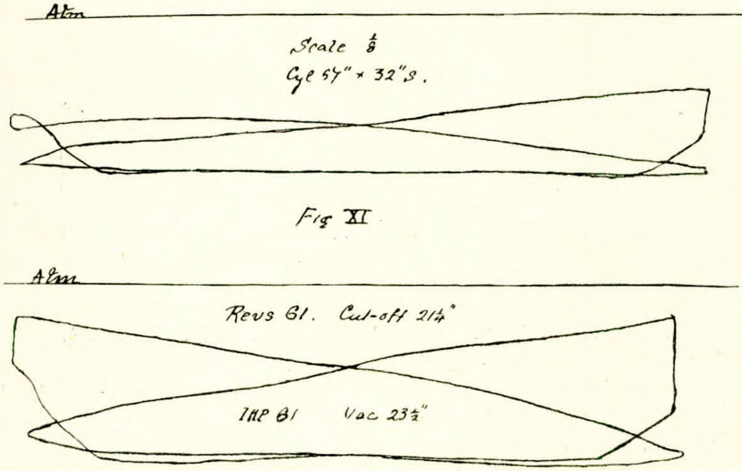


FIG. 11.—Scale $\frac{1}{8}$. Gauge O. Cyl. 57 in. × 32 in. stroke. Revs. 61. Cut-off $21\frac{1}{4}$ in. I.H.P. 61. Vac. $23\frac{1}{2}$ in.

The valve being held motionless on the face during the time the spindle traversed the loose distance. Without removing the indicator the valve cover was lifted and the nuts tightened, when the cards depicted by the lower figure was the result.

Mention has already been made of the effect produced on the diagrams by fluctuating back pressures : this not only is governed by the capacity of steam receiver, but also by the rela-

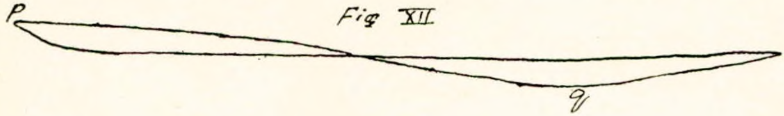


FIG. 12.—Scale $\frac{1}{32}$. Gauge 65 lbs.

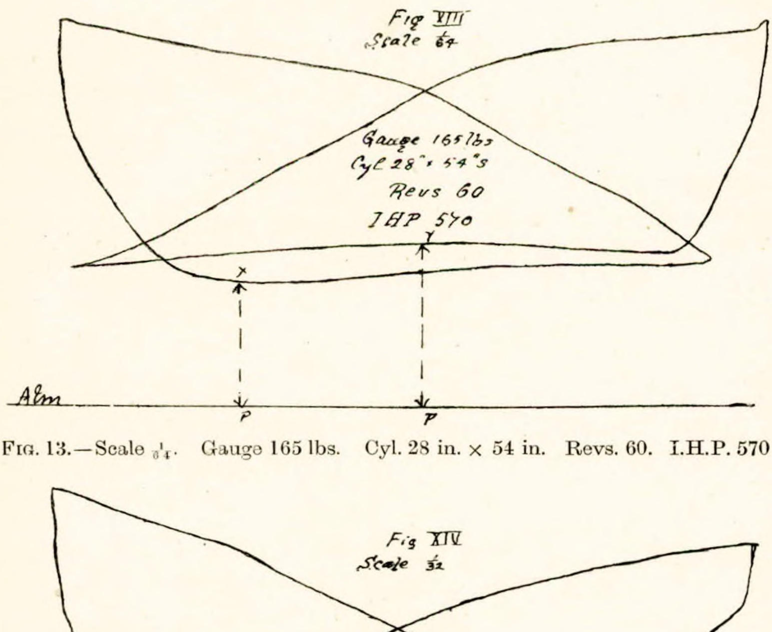


FIG. 13.—Scale $\frac{1}{64}$. Gauge 165 lbs. Cyl. 28 in. × 54 in. Revs. 60. I.H.P. 570

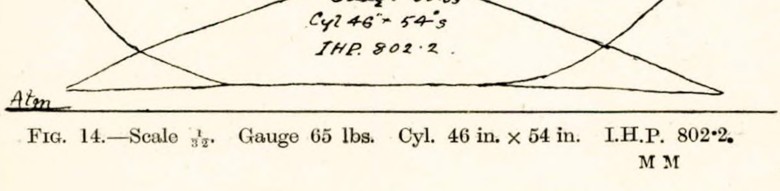


FIG. 14.—Scale $\frac{1}{32}$. Gauge 65 lbs. Cyl. 46 in. × 54 in. I.H.P. 802·2.

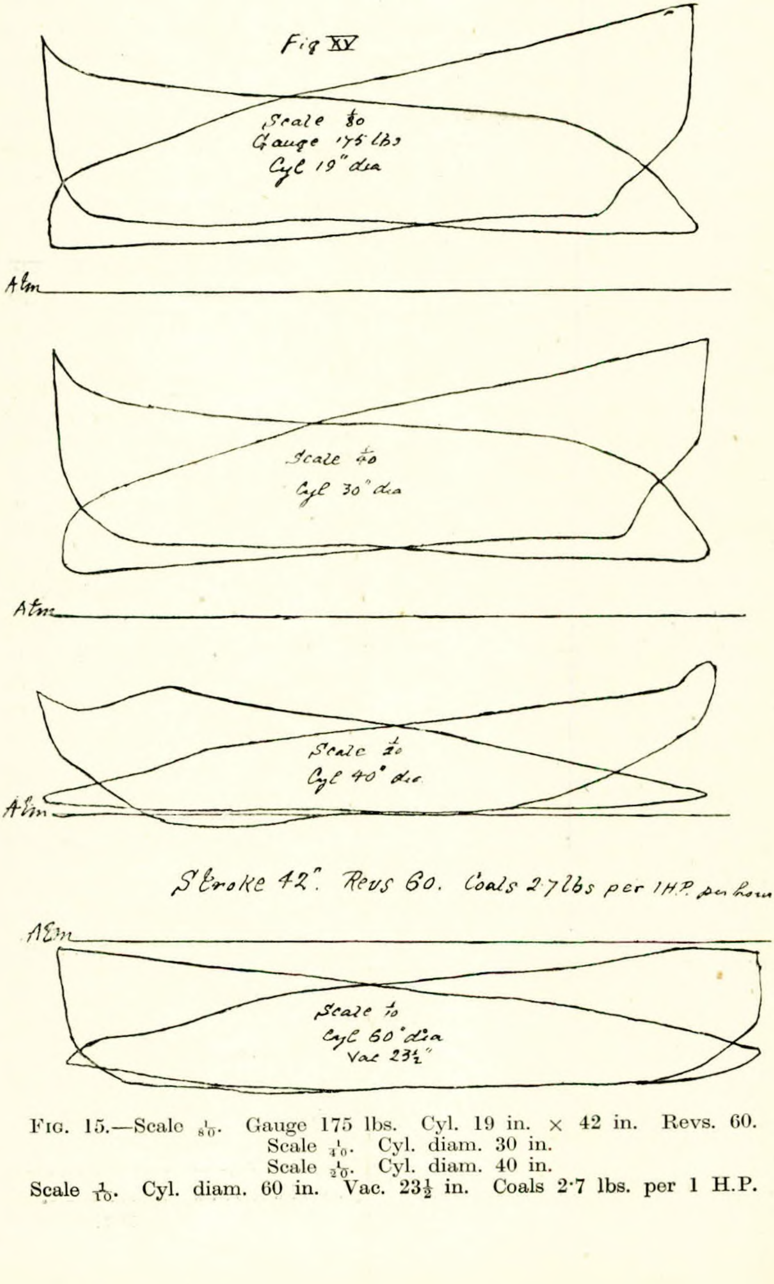


FIG. 15.—Scale $\frac{1}{80}$. Gauge 175 lbs. Cyl. 19 in. × 42 in. Revs. 60.
Scale $\frac{1}{40}$. Cyl. diam. 30 in.
Scale $\frac{1}{20}$. Cyl. diam. 40 in.
Scale $\frac{1}{10}$. Cyl. diam. 60 in. Vac. $23\frac{1}{2}$ in. Coals 2·7 lbs. per 1 H.P.

tion of steam admittance of one cylinder to the closing of exhaust of another. In multiple crank engines equality of steam distribution is most nearly approached, though much may be attained by a correct sequence of cylinders.

The fluctuation of pressure in the receiver of an expansive engine is graphically depicted by Fig. 12. The motion of the drum was derived from the *I.P.* connecting rod ; the difference in pressure from the highest point p, to the lowest q, measured by scale is 13 lbs.

The greatest difference in back pressure of the *H.P.* viz. *P.x.* and *P.y.* is 15 lbs. Measuring the initial pressures on the *I.P.* cards. Fig. 14 we find a difference of 11 lbs. The discrepancies are due to the diagrams not being traced simultaneously, which would have necessitated the use of a number of indicators ; consequently slight difference of conditions might prevail, sufficient to prevent the results appearing identical. Another factor producing difference of reading is that of condensation. The gauge pressure in above receiver was 65 lbs., this steam being admitted into a cylinder whose walls have been cooled down to a temperature corresponding to say, 5 lbs. above atmosphere, loses a portion by condensation, which rapidly reduces the pressure. The highest pressure measured on above *I.P.* cards is 65 lbs. ; but this immediately commences to fall, showing that the receiver is not receiving an influx of steam during the time the port is open. The receiver is evidently too small, and this accounts for the fluctuation of pressure as delineated.

To avoid excessive size of receivers and yet maintain constant pressures, variations from the ordinary sequence of cylinders, and angles of cranks, have been devised : in triple engines, the sequence *H.L.I.* enables a smaller receiver to be employed than the ordinary *H.I.L.* as the *H.P.* steam is exhausting during part of the time the *I.P.* cylinder is taking steam ; and similarly the *L.P.* thus adding a portion of cylinder capacity on each occasion to that of the receiver. Variations from the regular crank angles are objectionable for practical reasons.

The set of cards, Fig. 15, are from Quadruple engines, the four pistons operating on two cranks 180° apart : each pair of piston rods being attached to the upper apices of a triangular connecting rod, the lower apex of which contains the crank pin bushes. The pairs of pistons do not traverse synchronously but work independently as in the ordinary fore-and-aft arrangement ; a peculiarity of this design is that, while the stroke of the engine is 42 in., the diameter of crank circle is 36 in.

Scientifically designed as these engines were, they did not prove in practice to be particularly economical, the consump-

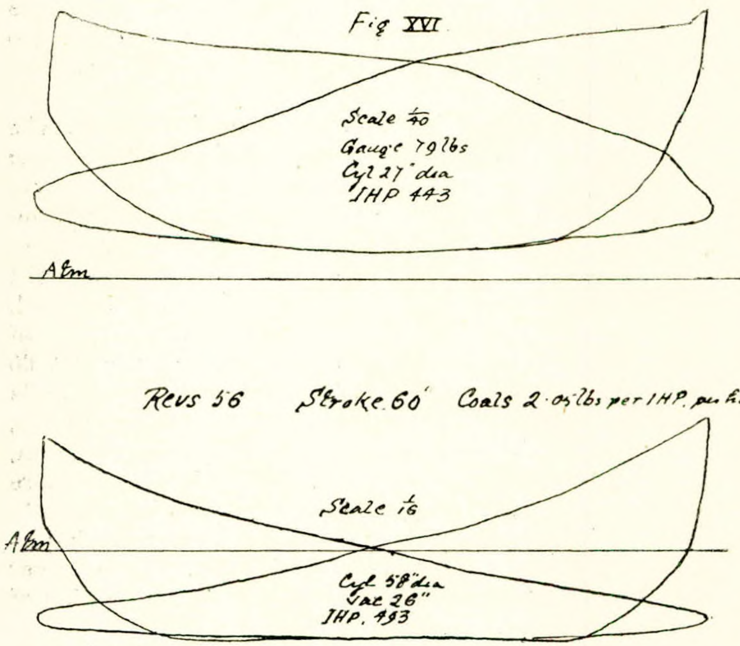


FIG. 16.—Scale $\frac{1}{40}$. Gauge 79 lbs. Cyl. 27 in. × 60 in. Revs. 56 in. I.H.P. 443. Scale $\frac{1}{16}$. Cyl. 58 in. × 60 in. Vac. 26 in. I.H.P. 493. Coals 2·05 lbs. per 1 H.P. per hour.

tion of coal per *I.H.P.* being $2\frac{3}{4}$ lbs. per hour. A modern compound tandem engine, whose cards are depicted by Fig. 16, only consumed a trifle over 2 lbs. per hour per *I.H.P.* It will be noticed that the compression curves are extra large, more cushioning being required in single crank engines than where the turning moment is spread over a number of cranks : the loss however proved to be little as the working results demonstrated.

The application of the Indicator is not necessarily confined to steam cylinders ; it records the work done by any gas, or

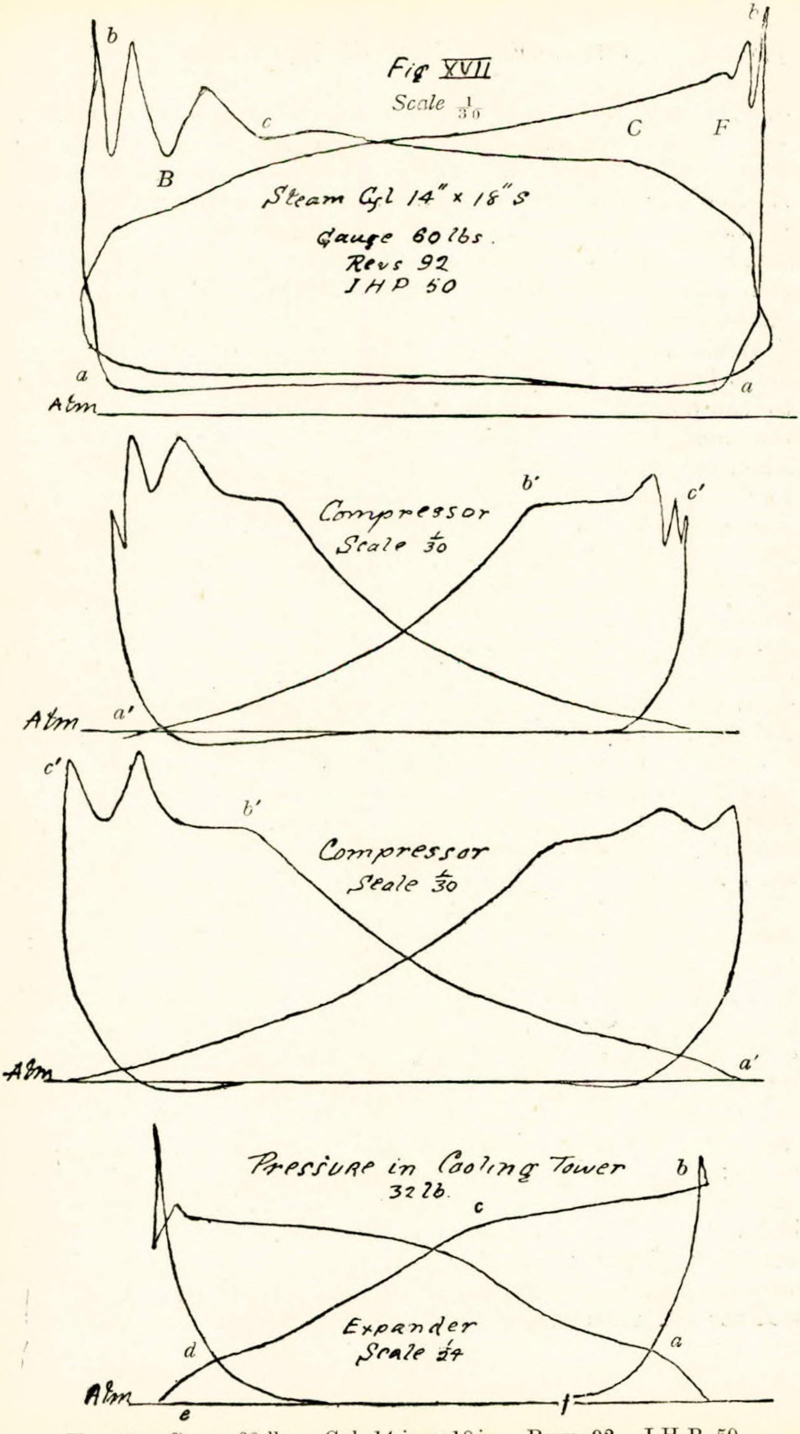


Fig. 17.—Gauge 60 lbs. Cyl. 14 in. × 18 in. Revs. 92. I.H.P. 50.

even liquids, such as diagrams taken from pumps. Figs. 17 and 18 are from the steam and air cylinders of two dry-air Refrigerating machines. Referring to Fig. 17 the first diagram is from the steam cylinder of a machine driving two compressers and one expander. It is a double crank machine, the steam cylinder and one compression piston being in line on one rod, and the expansion cylinder and the other compression piston on the other.

Whilst steam is admitted at *a*, full open at *b* and cut-off at *c*, air is being drawn into the compressors at *a′* compressed up to the point *b′* and released at *c′*. The release valves are of the mushroom type with springs on the back; these vibrate when pressure is suddenly relieved, thus causing the "waves" on the diagram, the shocks of compression, transmitted by the continuous piston rod, account for similar vibrations on the steam diagram.

After compression, the air is conducted into a cooling tower, which may be regarded as equivalent to the receiver of a steam engine; whence it flows to the Expander where it performs identically the same duty as steam; being admitted at *a*, fully open at *b*, cut-off at *c*, expanding and doing work to *d*, exhaust full open at *e*, and closed at *f*, leaving a portion of air in the cylinder to create a cushion. In this cycle of operations the compressed air is doing precisely the same work as though it were steam; returning about 40 per cent. of the power to the machine which was expended in compression, and exhausting at a very low temperature.

Fig. 18 is a set of cards from another machine, fitted with one compressor and expander. The same remarks apply: the air is here discharging from the expander at 50° below zero, Fahr. In both sets the compressor exhaust falls below the atmospheric line; this may be attributable to the momentum of the spring carrying the pencil a trifle further than its correct distance.

For convenience of illustration and description, the cards in this paper where necessary, have been divided into ten equal parts: any other number of divisions would do as well; for plotting, or computing mean pressures this is near enough; but for calculating stresses on the crank pin, owing to the obliquity of the connecting rod, the piston moves through varying distances for similar angular movements of crank; and the ordinates should be spaced accordingly. See Fig. 19

The curves followed by expanding gases which are used in comparing indicator diagrams, are based on Boyle's Law, viz.

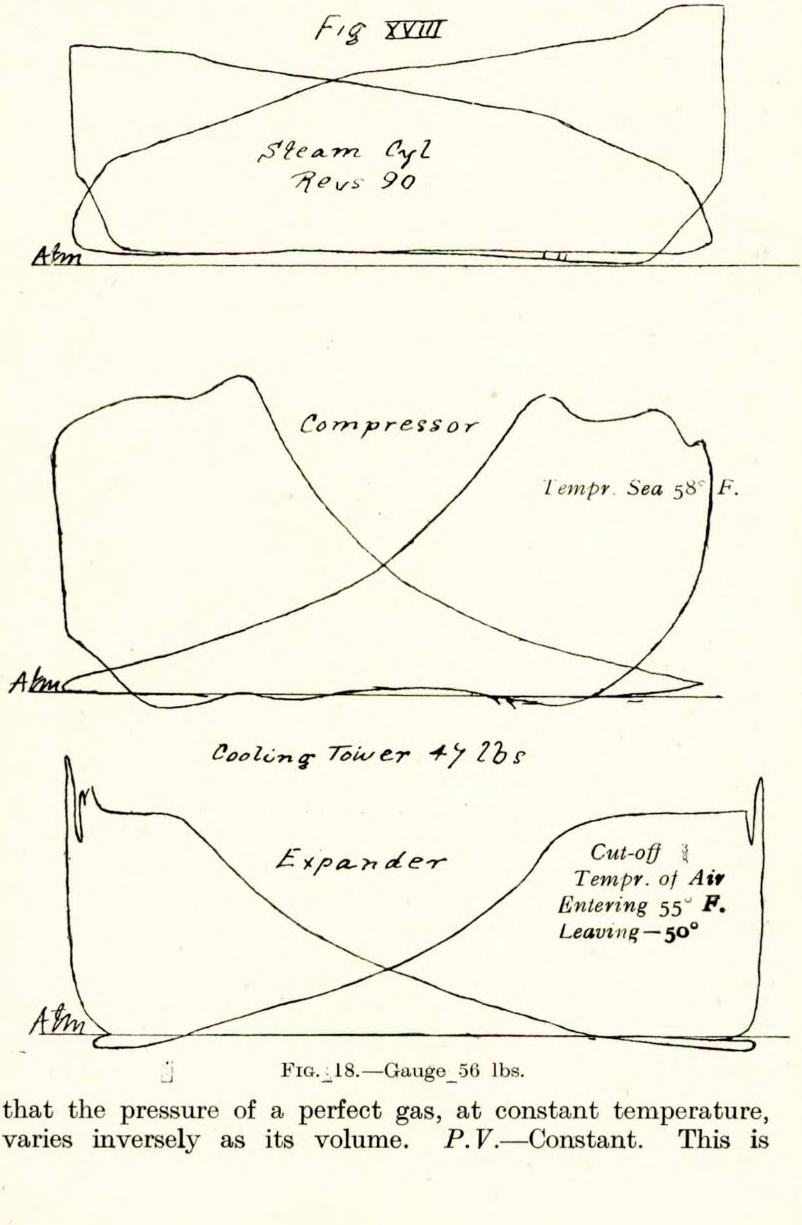


FIG. 18.—Gauge 56 lbs.

that the pressure of a perfect gas, at constant temperature, varies inversely as its volume. *P.V.*—Constant. This is

Isothermal Expansion, and the curve is an hyperbola, which is the one ordinarily employed for comparison. In steam

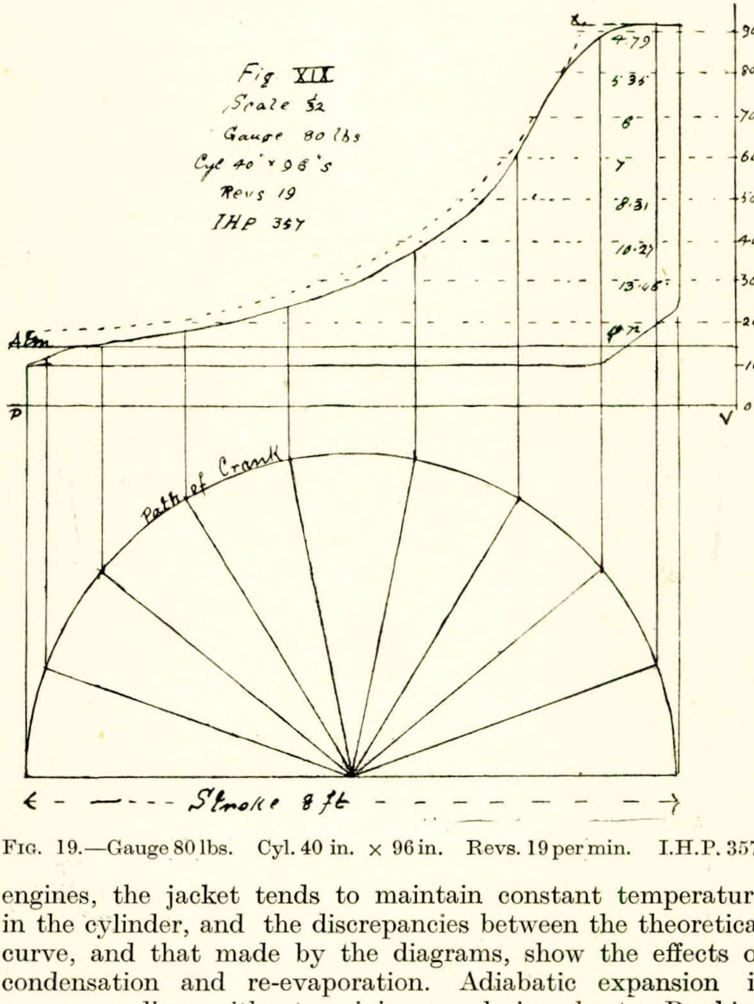


FIG. 19.—Gauge 80 lbs. Cyl. 40 in. × 96 in. Revs. 19 per min. I.H.P. 357.

engines, the jacket tends to maintain constant temperature in the cylinder, and the discrepancies between the theoretical curve, and that made by the diagrams, show the effects of condensation and re-evaporation. Adiabatic expansion is gas expanding without gaining or losing heat. Rankine deduced this curve, for a perfect gas, to follow the law $P.V.\ \frac{10}{9}$ $= C.$ and for saturated steam $P.V.\ \frac{17}{16} = C.$

The construction of these curves involves the use of logarithms: in the equation for saturated steam, the pressure P at any volume V may be found thus. $P = \log c \left(\frac{\log V \times 17}{16}\right)$. c being the absolute pressure multiplied by the volume of *H.P.* cylinder, plus its clearance, to point of cut-off. A simpler method is, to pick out the volume of one lb. of steam corresponding to the different pressures, from tables published in engineering pocket books; and on the points thus obtained construct the curve. For example. Fig. 19. From zero line erect a perpendicular at a distance to the right of the figure to embody clearance, say $\frac{1}{10}$, and on it mark pressures to scale. At point of cut-off x, tick off a horizontal distance representing the volume of steam at that pressure; 94 lbs. = 4·6 cubic feet.

The next volume, corresponding to 90 lbs. = 4·79; 80 lbs. = 5·35; 70 lbs. = 6, and so on until all the points are obtained; through these draw the curve. The rapid fall below the theoretical curve observed in the latter half of the stroke is due to the very early cut-off: condensation must always result from expansion, and as this condensed steam cannot exert any effort on the piston the result is loss.

In conclusion, it may be well to call attention to the difference between *real* and *apparent* cut-off. The apparent cut-off is that which is obtained by the ratio between the portion of stroke swept through by the piston at the moment when the steam port closes, and the total length of stroke. The real cut-off is the ratio between the volume of the portion of the cylinder where cut-off occurs, plus the clearance; and the whole volume of the cylinder including the clearance: the real cut-off is taken when calculating the number of expansions.

DISCUSSION.

MR. J. G. HAWTHORN: There is not a great deal of original matter in the diagrams to illustrate the paper, but there are one or two points about them which give food for thought. With regard to Fig. 1, I do not think the author is quite correct, where he describes the exhaust being fully open at the point *e*, at least the indicator card does not seem to show it to be so. Looking at the diagram we presume the engine is full open for exhaust at one end of the cylinder and full open for admission at the other end. We are shown in this diagram, at the point *b*, the most extreme left-hand point of the card, that the engine is full open for steam; I cannot realize that, as being the case, I should take it to be somewhere between the points *b* and *c*. Then again, to be fully open for exhaust at *e*, right at the extreme end of the card, shows the valve to be extremely low down. If the card is brought before our notice as an illustration of the distribution of the steam when the valve is low, it is perhaps a very good example; but the author does not describe it as being for this purpose; he simply says "the slanting admission line suggests that the valve is opening late." I will sketch on the blackboard a diagram to give an idea of what I consider are the points in a typical card.

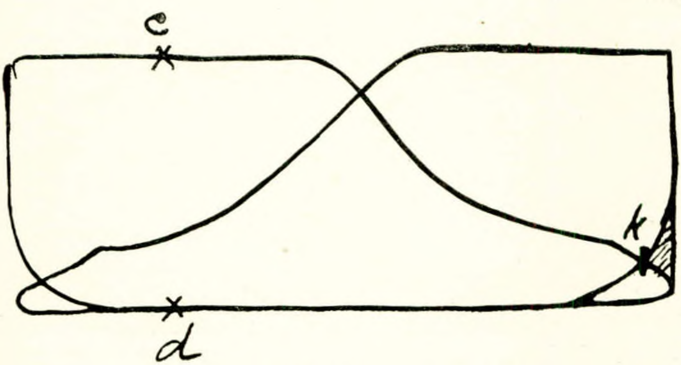


FIG. 1.

In reading the indicator diagram it is very interesting to see the relative positions of the points. It is usually divided into eight points. The extreme left-hand point of the card must be the top centre of the engine, because the barrel is then at the extreme pull of the cord, independent of the shape of the card. This being so, it cannot be fully open for steam at the point *b*.[1] Presuming cut-off at half stroke, it will be full open for steam at the point *c*, and the point *d* immedi-

[1] Fig. 1, page 441.

ately under that in the other card must show full open for exhaust. Where it is full open for steam on the upstroke it must be full open for exhaust in the same position at the top side of the piston. Say the engine is open for exhaust at eight-tenths of the stroke; if you compress immediately under that point there is no inside lap on the valve, or if you close the exhaust on one side of the piston at the same moment you open on the opposite side the valve must cover the two steam ports and nothing else. The point *k*, about nine-tenths of the stroke, is where cushioning starts. The little triangle at *k* represents the slowing down at each end of the stroke, and shows the loss of work. We may take the indicator diagram as a graphic representation of the work done independent of the faults of the engine. I have had some experiences of that nature. I have had an indicator card from an engine which showed a leaky piston, and on overhauling the piston we found everything was perfectly tight. We found when turning the engine down from top centre to bottom with water on it, that not a drop was lost, proving it to be tight; and afterwards we discovered the fault shown on the diagram was caused by the piston rod being bent. That is an instance where no assistance was given by the indicator card. I might mention another case of a leaky piston. In this case the indicator diagram showed a fault in the steam pressure line after admission. This is a sketch of the card:

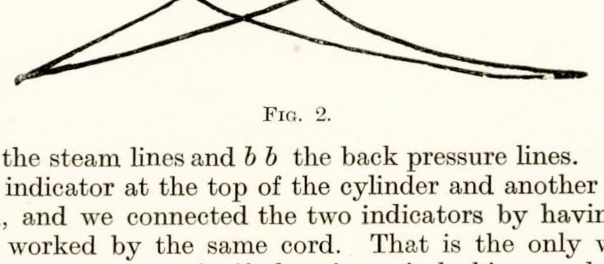


Fig. 2.

d d are the steam lines and *b b* the back pressure lines. There was an indicator at the top of the cylinder and another at the bottom, and we connected the two indicators by having two barrels worked by the same cord. That is the only way in which you can ascertain if the piston is leaking, as by that means you can see what is going on at both sides of the piston at the same instant. If you inspect the corresponding marks you can tell at once if the cylinder is leaking, but if you take it off the same side the whole of the time, how can you tell whether it is leaking or not ? You infer it is a leaky piston because you see a rise in the back pressure line, but the back pressure line is not a sign because it is on that side of the piston. Therefore you require to have both diagrams taken at once. This method of having an indicator at each end of the cylinder is adopted very largely in connexion with trial trips.

Among the interesting questions is that of ascertaining the cut-off, and of course many of us know that the method usually adopted is that described in Fig. 4, and is given in most of the good standard works on the subject. What we, as sea-going engineers, want more than anything else in connexion with the indicator diagram, are the inferences to be drawn from it under the actual conditions at sea. The paper is an excellent one, especially for the junior members, and I think it is worthy to be studied, as there is a great deal of information to be derived from it. On page 453, in reference to Fig. 12, the author says " the fluctuation of pressure in the receiver of an expansive engine is there graphically depicted." He goes on to say : " The discrepanies are due to the diagrams not being traced simultaneously, which would have necessitated the use of a number of indicators ; consequently slight difference of conditions might prevail, sufficient to prevent the results appearing identical. Another factor producing difference of reading is that of condensation." When we see an indicator diagram from the high pressure or intermediate cylinder, we always notice a rise in the back pressure line about mid-stroke. It is due to natural causes and cannot be avoided. If you do not find a " hump " in the line you will find the line sloping steadily towards the compression point. " The reason is as follows. Taking the intermediate engine, from the time you close the top steam port until you open the bottom port is about half the stroke of the engine. During two-thirds of that time the high-pressure engine is open to exhaust ; it is compressing steam up in the intermediate valve box and raising the pressure, and on the intermediate gauge you will find fluctuations of from 8 to 10 lb. The pressure is going up until you open the bottom steam port for admission again. It is absolutely impossible to maintain a perfect back pressure on a piston. On the low pressure piston you can by making the condenser temperature constant, but it is not possible on the high or intermediate pistons. Both steam ports are closed and there is no release for steam out of the valve box, consequently compression is going on. It always seems to me, when there is a straight back-pressure line, that there is something wrong—that is, in connexion with the intermediate or high pressure. I like to see the line rise slowly. Another point which is of more value to engineers is the balancing of the engine, the synchronization of the three parts. The cards themselves should be regarded more as a method of ascertaining the power of the engine, We are so used to making use of it in order to discover faults that we neglect the fact that the indicator is not so much for that purpose as for recording the power of the engine and for comparing it with respect to the displacement of the ship. Use it as Watt used it, for measuring the power put into the engine, and comparing that with the actual performances so as to see what the efficiency of the engine really is. The detection of faults is important, but it is a secondary matter. Let us take the case of a diagram in which a difference is made in the valve setting, say the valve is half an inch low. That would be shown plainly, as in Fig. 3.

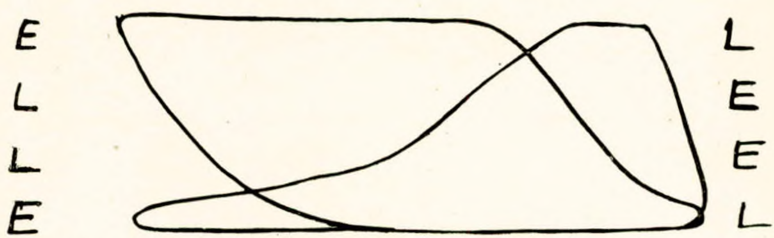


FIG. 3.

That would be shown very plainly in such a manner as this. You would get the indicator diagram representing the valve being very low. The sequence can be remembered ELLE, representing early lead, late cut-off, late exhaust, early compression, and for the other card LEEL, late lead, early cut-off, early exhaust and late compression. Looking at these diagrams you will observe that this one has about twice the area of the other, showing that most of the work has been done on the downstroke. That engine had a difficulty in getting over the top centre. When you have two indicator diagrams entirely dissimilar, it is always due to a fault in the valve, and it cannot be due to anything else. If you have two diagrams where all the faults are the same, it is the eccentric that is wrong. If you have what is early in the one late in the other, examine the valve and see what is wrong with it. Faulty valve setting is a point of the greatest importance in connexion with indicator diagrams. When the cards on being taken off show 10 per cent. more in the intermediate than in the high pressure and 10 per cent. more in the low pressure than in the intermediate, that is what is to be expected, and the cards are of very great importance under these circumstances in helping us to arrive at the correct results. Then again it will enable you to see the difference between the gauges and the pressure in the cylinders. Take another high-pressure card such as in Fig. 4. One measures $2\frac{1}{2}$, the other 2, giving a mean of $2\frac{1}{4}$; multiply by the scale,

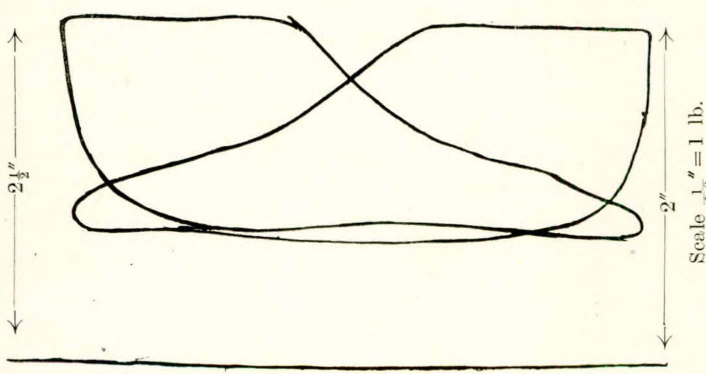


FIG. 4.

which is, say, $\frac{1}{64}$, and that gives us a pressure of 144 lb. The steam pressure gauge in the stokehold shows 160 lb. Where has that 16 lb. of steam gone to ? Again the high pressure card gives us, say, 60 lb., and on looking at the intermediate gauge we find it only shows 50 lb. Where has that 10 lb.

gone ? These are the points which are of great importance, brought to our notice by the indicator diagrams. We have a right to expect 155 lb. out of the 160 lb.; that is to say, there should not be a loss of more than 5 lb., due to initial condensation in the cylinders.

Mr. Wm. McLaren : Mr. Hawthorn has covered so much ground there is not much left for us to say, and it is something for the author to have produced such a paper and to have had such a critic. I do not think we have any one connected with our Institute who gives so much attention as he does to expound the principles and the uses of the indicator diagram. I have appreciated what he has said, and hope that it will be so grafted with the paper that it will be more useful to the older members as well as to the junior section. The paper is an educative one, and it brings home to me what I said at the last Annual Meeting, that we, as an Institute, should collect material for a pocket book for the benefit of those who follow. I think it would be very interesting if Mr. Hawthorn could give a lecture on this subject. I should also like to see an account of the various connexions in the way of gear for multiplying or shortening. I have two indicators, and now that we have taken out the steam plant and gone in for electrical driving I make use of them for testing the boiler gauges ; they can be used for steam gauges or pressure gauges of any description. I might say I use them for 42 in. to 7 in. reducing gear. That means that each stroke has to have its own drum, in the rotation of these indicators. They are rather difficult to apply, but after getting initiated into the use of them they seem to answer admirably. They are put right on to the crosshead, and there is no intermediate gear. I prefer them to the links or any other kind of reduction gear.

Mr. Hawthorn : I have in my possession a tracing of indicator diagrams, which illustrates one of the finest examples of their value and use. In a triple expansion engine the intermediate valve broke in heavy weather. The indicator cards are taken off the three cylinders of the engine previous to the breakdown. The valve was withdrawn and only the valve spindle was left in the valve box. The two eccentric rods were taken down, and the steam was led to exhaust from the high-pressure cylinder direct to the low pressure. The indicator cards were then taken off and the result was astonishing. The power which was developed by the intermediate cylinder was split up equally between the high and low pressure. The low-pressure engine had 5 H.P. more than high pressure after intermediate valve was taken out. The steam was kept at the same pressure in the boiler, exhausting direct from the high pressure to the low pressure, and the power was equally divided between the two. It is one of the best examples I know of, and I shall be pleased to give the tracings of these cards.

Mr. J. Clark: I understand Mr. Winterburn is one of our members abroad, and I think we are much indebted to him for the trouble he has taken in writing this paper to put before us. It is one thing to write a paper and to read it in person; it is quite another thing to write a paper and send it here to be discussed. I think the author deserves a special vote of thanks; it is a pity we do not get more papers in a similar manner. The value of a paper does not altogether depend upon what it contains; a good deal of its value may be derived by what it brings out in discussion. We have had some very interesting information about indicator diagrams that we should probably never have heard of if it were not for this paper. Of course it is somewhat easy to criticize a paper. All papers are open to criticism; most people write them with the idea of eliciting criticism, and if the criticism is severe at times I do not think they take any umbrage. Mr. Hawthorn told us some very interesting things about indicator diagrams; but one thing I noticed, he did not tell us where the loss in pressure went between that shown on the boiler gauge and the initial pressure on the indicator card. "The Steam Engine Indicator and its Diagram" is the title of the paper, but Mr. Winterburn might have told us something about steam engine indicators without diagrams, such as Professor Ripper's arrangement. I believe this has been used in connexion with quick speed engines—that is, in the days before the turbine—especially in destroyers, where the number of revolutions is so high and the diagrams so small that it would be comparatively useless to have the ordinary arrangement. The "Ripper" indicators give a mean pressure continuously. Mr. Winterburn remarks about the spring being proportioned to the pressure of steam, of course we all know that, but it is not necessarily only proportioned to the pressure of steam, there is a kind of double proportioning in the sense that the diagram should be symmetrical. A point also in connexion with the expansion of steam has not been touched upon, that is the value of and the work to be gained by expanding steam. Although there are inaccuracies in the wording of the paper, none of us need be taken unawares in connexion with them; such as the statement about the atmospheric line of 14·7 lb. Of course it may, or it may not, be 14·7 lb., but it seems to me there are more serious things in connexion with the paper, if I read it aright, as it appears to be a mistake in principle. Mr. Winterburn deals with it on page 442 but it is illustrated better on page 446 in connexion with the matter illustrated by diagram 6. I am not quite sure, but I understand Mr. Winterburn to say that when ascertaining the power on one side of the piston, back pressure readings from the other side must be taken into consideration. That I do not agree with. When the steam is doing work on one side of the piston—let it be the top side—it does work against the back pressure on the under side, plus the work of the engine. It has nothing whatever to do with the back pressure on the other side beyond the fact that the engine will take more steam for a given duty. At the same time I am open to correction if I am not reading it as he intended it. Then again, in connexion with the suggestion for estimating clearance and cut-off. So far as I am aware it certainly does answer as described on page 444, Fig. 4. In some particular diagrams it will give a correct percentage of clearance and cut-off, but it depends on the compression curve, and I have found diagrams that did not agree with the method at all. In this case it appears to be not very far out, but all diagrams have not that special compression curve. Mr. Winterburn tells us that "a practical method for ascertaining the clearance is to place the piston at the end of the stroke, close up the steam port, then fill up through the indicator cock a measured quantity of water." I never knew any one who could do this, and if it has been done how is it possible to know? As a rule, when engines are placed in position it is quite impossible to adopt the method suggested. I think the author might have enlarged upon this detail a little more. He has given a number of diagrams, and in one or two instances he has given the coal consumption; it would also have been instructive if he had given the steam consumption also, and the method of calculating the steam consumption, showing the difference between that indicated and the "missing" quantity. It is always useful to be able to refer to a paper to find out what you want, and to find a lot of things besides diagrams that you can apply to cases under your own observation. The author gives particulars about compressors, which are very interesting; he might also have shown a diagram from an air pump, they are not often published. As I have previously remarked, although some of the wording of the paper may not be absolutely correct, it is not likely to deceive any of us. Strictly speaking, it is not quite correct to say "Adiabatic expansion is gas expanding without gaining or losing heat." It is perfectly right in principle, but you cannot possibly do work without the expenditure of heat.

Mr. HAWTHORN: I think it would be better, if possible, to prevent the heat passing out. I have always defined adiabatic expansion as being "that expansion of steam where no heat passes in or out and the fall of temperature accounts for the work that is done."

Mr. CLARK: Another little point the author might have touched upon is with regard to indicators having external springs as against those with internal springs. I do not know that there is very much in it, but the internal spring seems to hold its own, and apparently both types give us equally good results.

Mr. HAWTHORN: I might give a sketch of the card one would expect to get from an ordinary reciprocating air pump.

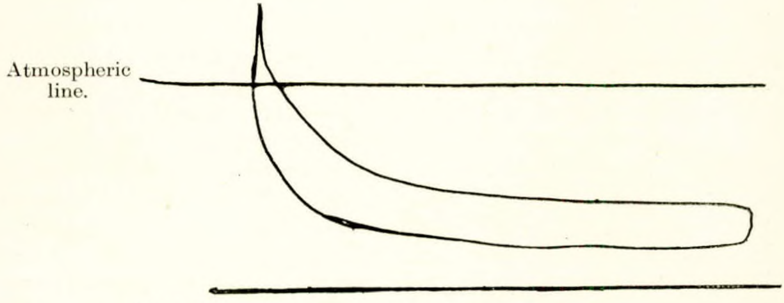


FIG. 5.

Let us assume the small air valve to be placed underneath the head valve, showing the effect of the admission of air on the under side of the air valve. Most of us have the impression with regard to the air valve that the air is admitted for cushioning purposes. The air pump is lifting 2 in. of water, the whole of the water formed from the steam cut-off at half stroke. A cubic foot of steam gives a cubic inch of water. You should get half the steam carried, plus the clearance and presuming the clearance to be one-twelfth of the volume that will give about another stroke of the piston. Condense the steam and you get $\frac{1}{8}$ in. water on the low-pressure piston. Seeing the air pump is single-acting you would get 2 in., representing the whole of the water for the two strokes, making the air pump bucket one-third of the diameter. You would get 18 to 1 as the volume ; that multiplied by $\frac{1}{4}$ in., gives $4\frac{1}{2}$ in. as the extreme amount of water in a two-foot stroke. As soon as it begins to come down a little air is admitted. What for ? Many think it is for cushioning, but that is not so. It is to assist the low-pressure engine up, for with the pressure on the air-pump bucket we should have had a true vacuum, or nearly so, say 28 in. or 29 in., on the top side if that air had not been admitted. That would be a pull against the low-pressure engine going up ; therefore by admitting the air you are assisting the low-pressure piston up. Here is a sketch of an indicator diagram, showing that valve shut.

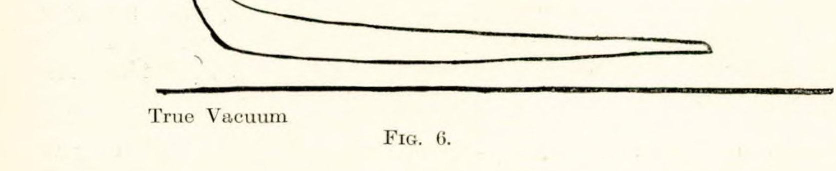


FIG. 6.

If the valve were open the diagram would be as Fig. 7. The maintenance of the vacuum is due to the temperature of the condenser, and the duty of the air pump is to make a better vacuum in the air-pump chamber. If there was a fall of from 3 ft. 0 in. to 3 ft. 6 in. from the bottom of the condenser to the bottom of the air-pump chamber, the condenser would

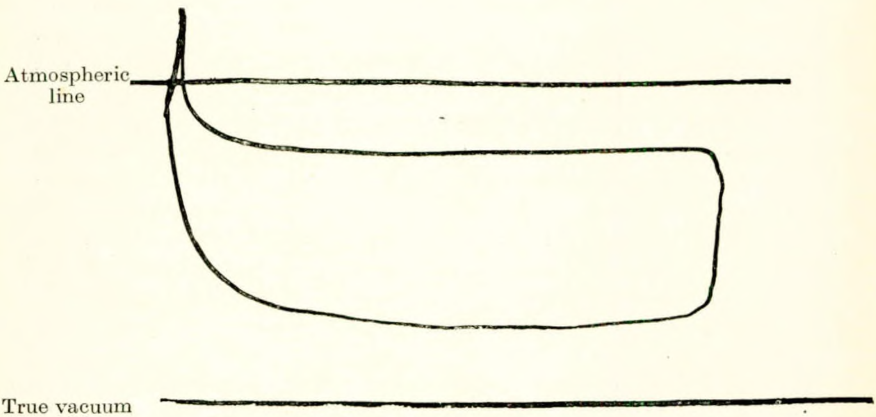


FIG. 7.

drain itself. After the overflow from the hot well had been closed the feedwater would find its way into the hot well without any help from the air pump. The admission of air is to prevent a drag on the low-pressure engine.

Mr. E. SHACKLETON : The paper to-night is entirely confined to the steam engine, and I do not think I can add anything to this highly interesting discussion. The steam engine diagrams are different from those of the internal combustion engine. I might mention that in some cases in connexion with the latter class of engines the indicator is made with an outside spring—that is, for gas engine work. I must say that the paper is of a non-controversial nature, but at any rate it has brought forward some very interesting points from Mr. Hawthorn.

CHAIRMAN : I think we might urge Mr. Hawthorn to give a lecture on the Indicator Diagram, and I have no doubt it will be of great interest if he consents to do so, as he is an authority on the subject.

Mr. Hawthorn expressed his willingness to give a lecture, more especially for the Junior Section of the Institute and it was decided that it should be given on Monday, December 12, 1910.

The Hon. Secretary then read the following communication from Mr. W. E. Farenden (Member) :—

Referring to cards taken from Cold Air Refrigerator Machines, shown on page 457, Figs. 17 and 18, it should be noted that cards to be relied upon should not be taken (especially from the compressor) until the machine has been running long enough to have attained its complete working condition. Until the compressor has been running continuously and at full air pressure for some time the full temperature of working parts has not been attained, and cards taken from a compressor that has only just been started will give a lower compression line, and a lower mean effective pressure, than those taken after the cylinders have been heated up to their mean working temperature. This also applies to the steam and expansion cylinders. An explanatory card from an air compressor is as shown in Fig. 8 :—

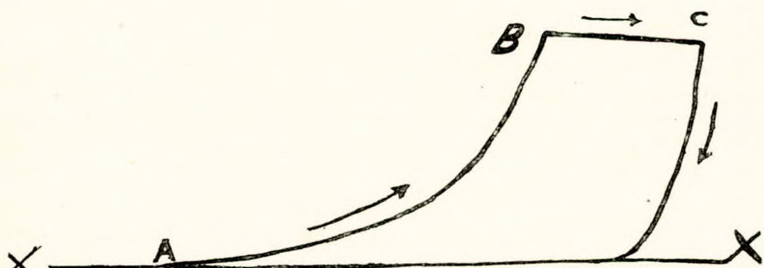


FIG. 8.

A B is the compression line, *B C* the delivery, *C D* re-expansion, *D A* the admission line and *X X* the atmospheric line. In Fig. 17 the expansion card does not show clearly the point of cut-off ; this is definitely shown in the expansion card of Fig. 18. In the compressor and also the expansion cards, the line falling below the atmospheric line is not good, showing a loss, due probably to leaky valves. This paper of Mr. Winterburn's is very instructive and will be much appreciated.

A vote of thanks was accorded to Mr. Winterburn on the proposal of Mr. Hawthorn, seconded by Mr. Clark.

MR. WINTERBURN: In reply to criticisms forwarded to me, I have to admit that it is difficult to say much that is really original on such a developed instrument as the indicator; and the opening paragraphs are, as stated in the paper, addressed to the junior section of the Institute.

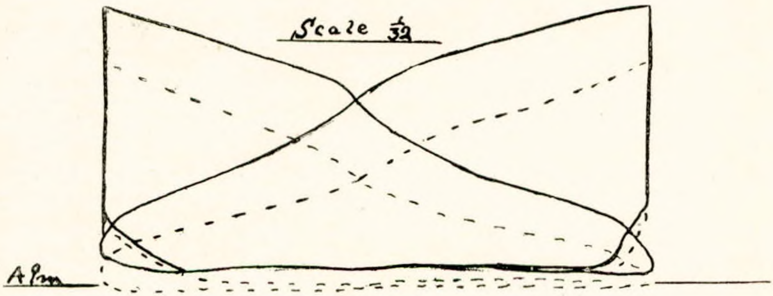
All the diagrams were taken a long time ago; many of them not by the writer, but procured from engineers with whom he was in contact: consequently only the data in many cases was recorded which at the time was considered necessary: the diagrams were not taken with any view of publication; and are only now utilized to illustrate the paper, and to prove text-book truisms. In the latter case, actual figures got from practice are considered preferable to ideal ones constructed to fit a theory.

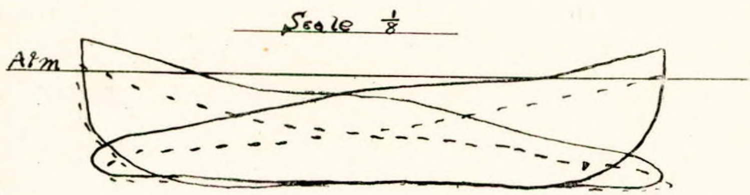
I do not quite concur with all that Mr. Hawthorn says, though it may be, through not having the benefit of seeing his sketches, that I sometimes mistake his meaning. My Fig. 1, he says, is a very good example of the valve being low: just the opposite was, I think, the fault in that instance.

His assertion that the port cannot be fully open to steam at the point *b*, is contrary to what is usually taught. Granting that the piston is on the top centre when the pencil is at the extreme end of the card *a*, there is an appreciable lapse of time before it begins to return, during which period the valve is travelling and opening wider the port, whilst the line *a b* is being traced.

The answer to the question, what has become of the steam where the gauge pressures are higher than those shown by the indicator, is that it has been condensed; and the blank spaces between the theoretical curve and the actual one shown on Fig. 5 exhibit this.

The interesting diagrams mentioned by Mr. Hawthorn, showing the same horse power being developed by the high and the low pressure engines which had formerly been produced by three engines, induces me to append the following set of cards.





DATA.

Four blades.	*One blade.*
I.H.P. 471·6 H.P. Engine.	414·8.
,, 360·2 L.P. ,,	283·8.
,, 831·8 Total.	698.
,, 111·4 Difference.	131.
Revolutions, 61.	80 per minute.
Coals, 1·9 lb.	2 lb. per I.H.P. per hour.
Speed, 10·37.	8 knots per hour.

The steamer had a four-bladed cast-iron propeller; three of them broke off close to the hub, and the vessel traversed the distance from Thursday Island to Sydney, N.S.W., by means of one blade only. The full lines are the normal cards before the accident, and the dotted diagrams were taken when running with the one blade.

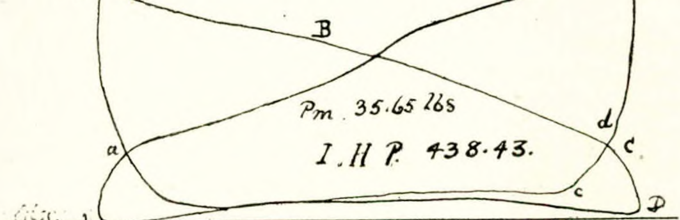
In reply to Mr. Clark's request for particulars of the "Ripper" or similar devices; and also indicators with external springs: the latter have the advantage of permitting the spring being changed while the intrument is in use, without burning the fingers: but I am not prepared to enter into details of modern innovations and improvements, this can be much better done by members nearer home.

In British Columbia, refinement of steam motive power is not asked for, and economy of fuel is of small account. Incinerators are at work in the heart of the large cities of Vancouver and Victoria consuming fuel to get it out of the way : saw mills will give you all the cuttings you want if only you will remove it : throughout the country, steam plants are devised with a view of eating up the wood as fast as it can be pushed through the furnace doors : steamboats gobble up a decent-sized plantation every trip : tarred wood blocks taken up from pavements are carted to the sea and dumped overboard : and miles of forest are always burning to waste during each dry season.

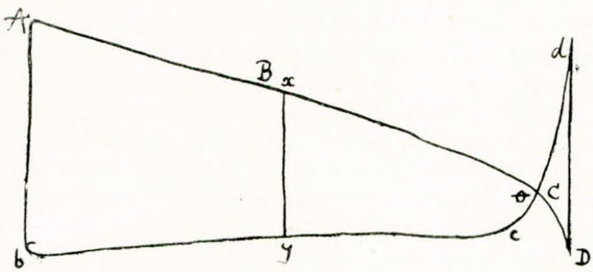
Indicators, therefore, are rare, and my paper was written round records which I had collected and preserved before coming to this country.

I do not exactly grasp what information is expected in regard to diagrams being symmetrical. The cord operating the revolving cylinder should, of course, be so attached to the engines as to produce figures of as nearly equal length as possible : the initial height of the diagrams should also approximate, and this is accomplished by using correctly proportioned springs. As for the "faulty statement" of the atmospheric line being 14·7 lb. above zero ; as it may or may not be so : this is merely a quibble. We all know that the pressure of the atmosphere varies, and generally 15 lb. is considered near enough, and a handier number than 14·7 to work with, though further from the truth.

Perhaps I was not sufficiently lucid regarding back pressure (Fig. 6). In calculating the work done by the engine, we have to get the net pressure exerted on the crank pin ; i.e. the steam pressure as indicated, less the pressure at the same moment on the other side of the piston resisting its advance. When working out stresses on the crank pin, take, for example, the H.P. pair of diagrams herewith :—



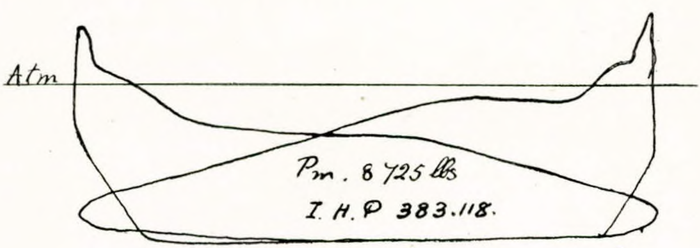
Combine the curves $A\,B\,C\,D$ and $a\,b\,c\,d$, and we get the following figure :—



Join $A\,a$ and $D\,d$ to complete the card.

Any ordinate, say $x\,y$, measured to the scale of the indicator spring is the net pressure on the piston urging it forward. The pressure is equal on both sides at θ, and beyond that, the space enclosed by $\theta\,D\,d$ is back pressure greater than that of the expanding steam on the other side of the piston, and the stroke is completed by the momentum of the moving parts, and the assistance of the L.P. engine.

These cards were taken from compound engines having "Trick" slide valves ; unfortunately I have no further data than that given ; but I append the L.P. diagrams, and would like members to discuss them ; where is the point of cut-off for instance ?



I do not appreciate the impossibility of ascertaining the clearance by filling the cylinder end with water. If the piston is tight, and the steam port plugged, a known quantity of water poured in through the indicator cock will give the cubic contents of that end of the cylinder and steam port.

To calculate the steam consumption: take a point on expansion line just before release, say d on Fig. 1 in paper. Measure the absolute pressure from $P\ V$ line, and from a table of densities find the weight of a cubic foot of steam at that pressure; this, multiplied by the cubic contents of cylinder and clearance, will give the weight of steam in the cylinder at that point. Then take a point midway between a and f on compression curve; measure its absolute pressure as before, and get the weight of a cubic foot from the table; multiply this by the volume of cylinder and clearance at that end: subtract from the former result, and the difference is the weight of steam used per stroke.

There must be a limit to the length of a paper (and also to the reply to criticism), consequently pump diagrams were not touched upon. I fear that the patience of the members will be exhausted, and beg to thank all who took so much interest in the subject.

MR. T. A. GALLOWAY (Member) has forwarded to the Institute a copy of a number of indicator diagrams taken from the s.s. *Benvoirlich*, which may be consulted by members interested.