SESSION 1910–1911

President: Sir David Gill, K.C.B., F.R.S., etc.

Lecture on the Steam Engine Indicator Diagram

By Mr. J. G. HAWTHORN (Hon. Minute Sec.).

Monday, December 12, 1910.

Chairman: Mr. William McLAREN (Member).

Mr. Hawthorn: I will preface my lecture to-night by asking, What is measurement? We may say it is only a term for comparison, or a means of comparison. In this sense the indicator diagram is a measure, a graphic representation by means of which one can ascertain the work done during one stroke of the engine. Measurement starts from the infinitesimal. A point has no magnitude. An infinite number of these gives us a line, length without breadth. Our unit of length is the exact distance between two marks representing one foot, on a platinum bar at a temperature of 62° F. and 30 in. barometric pressure, kept in South Kensington Museum, and the Parliament buildings at Westminster. Our unit of weight is a ball representing 1 lb., and the exertion required to move that weight a distance of 1 ft., gives us the ft.-lb. which is the unit of work. What is meant by "horse-power"? The measure which represents one-horse power is 33,000 ft.-lb.; it is more, as the element of time enters and it may further be termed the rate at which an engine is capable of doing its work. Therefore if 33,000 ft.-lb. are performed in 1 minute it is at the rate of 1 H.P. The indicator diagram is a means of measuring that work. Ft.-lb. is a measurement of two dimensions, therefore the indicator diagram is measured by its area. Let us now consider what is meant by the "mean effective pressure." If the difference of the pressures be taken between the forward and backward stroke of the piston at any particular moment of the stroke, that is the effective pressure of that moment. The back stroke of the engine is the "in" stroke, the "out" stroke being termed the forward one. We have retained the term "back-pressure," but have dropped the use of "forward-pressure." If we take the difference between these two pressures that will give the effective pressure at that moment. If ten of these observations be taken, at ten equal intervals of the stroke added together and divided by ten, this will give the mean effective pressure. In an indicator diagram, by dividing its length into ten equal spaces, by measuring the width of the diagram at each of those spaces, adding together and dividing by ten, you get the mean width in inches. Multiply the mean width by the scale and you get the mean effective pressure. The horse power is obtained by multiplying this by the constant for the cylinder, which is the area of the piston, multiplied by the stroke in feet, multiplied by the number of revolutions, and the whole multiplied by two. Divide this product by 33,000, and this will give the horse power.

Let us now see what inferences can be drawn from the indicator diagram, and how to read it. Each of the points is determined entirely by the position of the eccentric on the shaft relative to the crank. The position of the points on the diagram is never influenced by the slide valve, but the length of the lines between the points is determined by the valve setting. If the valve is too high or too low, it does not interfere with the points, it only affects the lines on the cards. We have eight points on the indicator diagram.

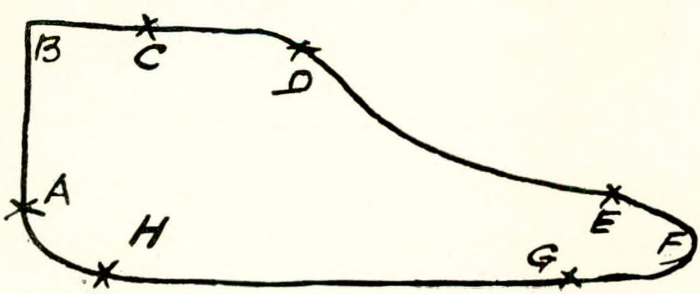

FIG. 1.

Referring to Fig. 1, starting at A we have what is called the lead point. A may be generally understood to be the point on the indicator where the curve of compression joins the straight line. Your engine is near the top centre, the piston or slide valve is beginning to open. At the most extreme point B on the left of the indicator diagram it may always be assumed that the engine is on the top centre. C is the point mid-way between the top centre and the point of cut-off. The engine is now one-third of the stroke down, cutting off steam at about ·6 of the stroke D. That point may usually be taken as about mid-way between B and F, the bottom centre. At E, the point of exhaust, the piston is probably eight-tenths of the stroke down ; with 600 ft.-piston speed it is usual to open the exhaust at eight-tenths of the stroke, so as to give ample opportunity for the steam to flow out. At G the engine is full open for exhaust. The point where it is full open for steam on the upstroke, it is full open for exhaust on the other side. It is closed at about eight-tenths of the stroke up, giving two-tenths for the compression of the steam. The clearance volume of the piston will be one-fifth of the volume of the steam in the cylinder at the moment we close that exhaust. Sixty lb. pressure absolute by 5 gives us 300 lb. pressure. That would be the case if the whole of the compression was in accordance with Boyle's law, but seeing a good deal of

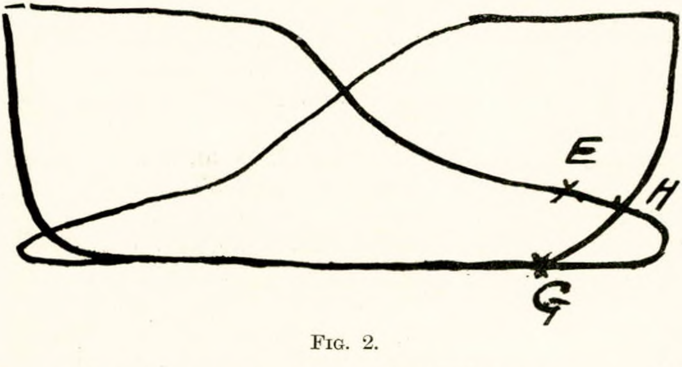

FIG. 2.

it is used in other ways we do not get 300 lb. pressure. What we want to get is the pressure that is necessary to absorb the momentum of the moving parts. If the weight of the piston rod, connecting rod, piston, etc., be all concentrated at the centre of gravity of the connecting rod, and the distance from centre of shaft to this point be taken as the radius about which this weight revolves, then the centrifugal force so created, divided by area of piston, is the pressure necessary to absorb the momentum.

Referring to Fig. 2, I would call your attention to the two points E and G, if they are immediately over one another, i.e. if exhaust takes place from top of piston simultaneously with compression on under side, when the valve is in mid position it must just cover the two steam on the inside and, therefore, there is no inside lap on the valve, but if G being to the left of E a compression taking place before exhaust there is positive exhaust lap. And if E be to the left of G there must be minus exhaust lap. The point H, where the two cards cut each other, is the point where cushioning starts, for we see at that particular moment about nine-tenths of the stroke; the pressure of the steam is the same on each side of the piston, the pressure falling from E to H and rising on the other side of the piston from G to H. Therefore as far as the pressures are concerned H is the point of least effective pressure and therefore from H to end of stroke the piston is doing work on the steam (cushioning). Such then are the points that require consideration from an indicator card.

The next point is the question of pressures as shown on the indicator diagram. It is very interesting to measure the pressures on the indicator cards and see how they correspond with the steam gauges. All that is necessary is to get the pressure as shown by the diagram to see if it is in accordance with the valve setting.

Referring to Fig. 3, supposing A is an indicator diagram from the high-pressure cylinder, the scale being $\frac{1}{64}$ to 1 lb. pressure, draw the atmospheric line about 1 in. below the card. Now draw B the intermediate diagram and make that atmospheric line about $\frac{1}{2}$ in. below the card, making the scale $\frac{1}{32}$ in. Draw also C the low pressure diagram to the scale $\frac{1}{16}$ in. and make the atmospheric line $\frac{1}{4}$ in. below the card. The first thing to do is to ascertain the pressures. X will measure the initial pressure of the steam in the high-pressure cylinder. T will measure the terminal expansion pressure. T divided into X will give the number of expan-

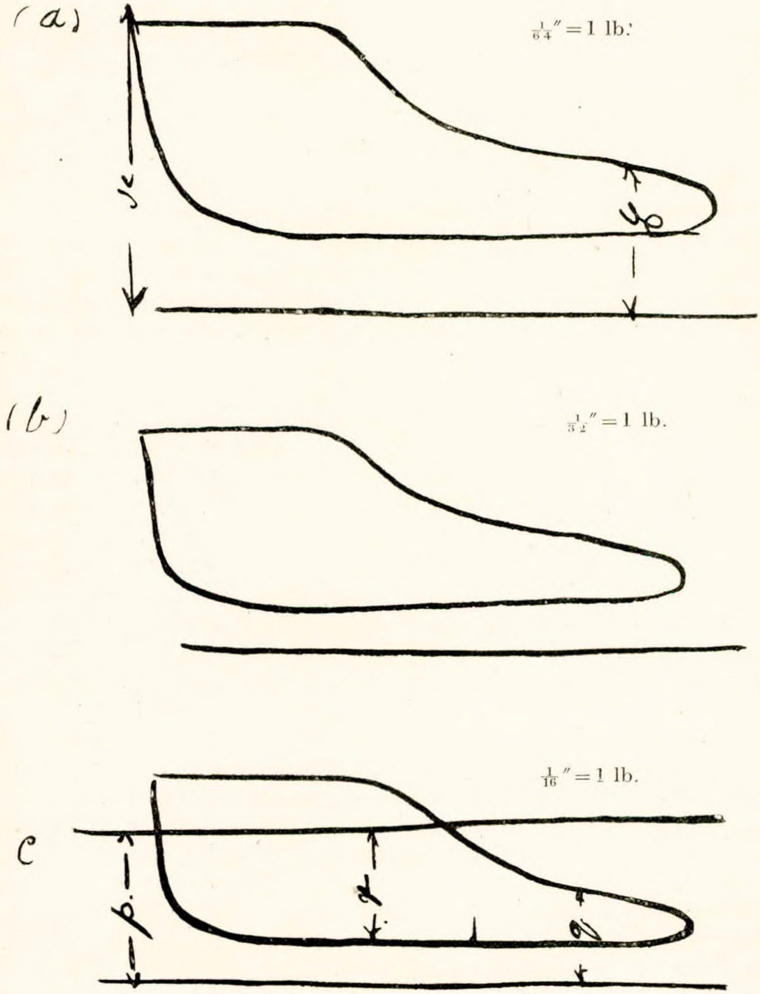

FIG. 3.

sions theoretically ; but A E (Fig. 1), including the clearance value, divided by B D, will give the expansions practically. The final volume divided by the initial volume will be the number of expansions. When a vessel has any pressure in it less than that of the atmosphere a vacuum is said to exist. Measuring down the atmospheric pressure P to scale on the card (Fig. 3) we come to the line of true vacuum, L P. If it is not measured down but upwards it would be the absolute pressure, Q is the absolute expansion pressure. So that everything measured upwards from the atmospheric line is pressure by gauge, everything measured down is vacuum. With 64 lb. as the pressure on the exhaust side of the high-pressure piston, that would give 58 to 60 lb. as initial pressure in the intermediate valve box. With the intermediate valve open to admission it would go down to 56 lb. when one of the steam ports was full open to steam and with the high pressure open to exhaust it, and the intermediate steam port closed, it would be about 66 lb. So that it fluctuates between 56 to 66 lb., or 61 lb. back pressure. The same thing refers to the low pressure engine; it depends on the size of the valve box, the length of the journey the steam has to make. So that we expect to find that 10 lb. is the mean back pressure on the intermediate piston. R is the vacuum as recorded on the vacuum gauge, while P is the atmospheric pressure as given by the barometer. Take the difference between this and what the vacuum gauge shows you, which will be probably about $4\frac{1}{2}$ in. or 5 in. with a vacuum of about 25 in. in the condenser. Such are the means of reading the different pressures. It is very interesting to read the pressures and to compare the diagrams for the three cylinders of an engine. I remember one case of a chief engineer who showed the initial pressure on the intermediate cylinder to be greater than the back pressure in the high pressure cylinder. He wanted to show a very good record, and to do so he kept the safety valve just on the blow; took the high pressure indicator diagram, and having taken it off, linked up the intermediate engine. The earlier closing of the intermediate valve throws up the back pressure in the high-pressure cylinder, probably getting 70 instead of 61, and that 70 is now showing as the initial pressure of the intermediate engine, and by working the high and intermediate engines in full link, he worked up in the same way 5 or 6 lb. in the low pressure valve box. The peculiar thing was he showed an increase of 200 in horse power for the same consumption of coal. Coal is a convenient form of stored up energy, and we know that 1 lb. of it is equal to 14,000 units of heat. That heat can only do a certain amount of ft.-lb. work, and when a man says he is getting 800 when the normal is 778, it shows there is something wrong. When the superintendent engineer combined the diagrams, he found they were overlapping. Part of the work had been charged up to both engines. The cards therefore enable you to detect if any wrong arrangement has been made with respect to the fixing up of the gauges.

So, if you wish to ascertain some of the faults in the distribution of the steam, you can make very good inferences by looking at the diagrams. For instance, if we advance the eccentric.

In Fig. 4, supposing the circle A B C represents the travel

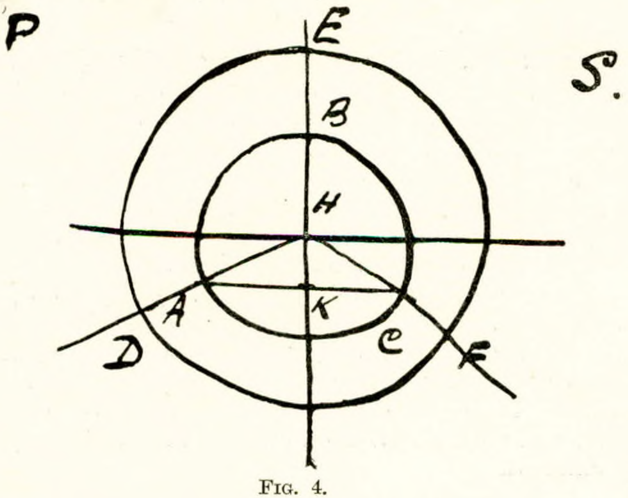

FIG. 4.

of the valve, D E F represents the section of the shaft. Draw two straight lines at right angles, passing through the centre and measure down the lap and lead of the valve. The engine runs from port to starboard, P is port and S is starboard. The angle of advance would be about 120°. If the eccentric be advanced we must increase that angle. Now the distance below the centre of the shaft is H K, therefore we can quite easily see the amount of the valve which overlaps the steam port is constant when in mid position. Seeing that the steam port must be opened more when the engine is on the top centre—that is, there is an early lead—she must close earlier and must open the exhaust earlier. Therefore every point in the distribution will be early if the eccentric is advanced from its true position (Fig. 5). If we get two indicator diagrams

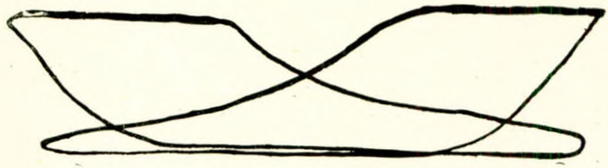

FIG. 5.

Eccentric advanced from true position. Every point early.

from both sides, both cards being exactly alike and each early, we would say the eccentric was too far advanced. There is no fault in the valve, it is always the eccentric that is wrong in such a case. Supposing the eccentric is back from its true position, making every point later. We would then get late cut-off, late exhaust, late compression. Here again both cards would be alike, everything is now later, therefore the eccentric is back from its true position. The angle of advance is not sufficient (Fig. 6). You can imagine a case where something has gone wrong with the liners or the eccentric rod; so that if you have a low valve, when the engine is on the top centre the lead will be early. On the downstroke the card will show early lead, late cut-off, late exhaust, and on the upstroke it will be late lead, early cut-off, early exhaust.

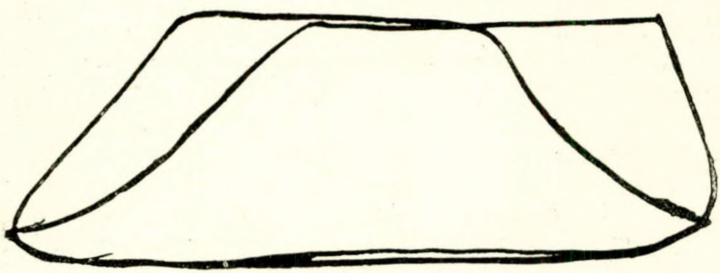

FIG. 6.

Eccentric back from true position. Every point late.

I have exaggerated them in the sketch (Fig. 6) so that you will be able to see more clearly the amount of work done on the downstroke. This would be easy to detect in a ship's engine working with the valve low. Most of the work would be done on the upstroke if the eccentric rod is too long, and the valve is too high.

I might point out one or two peculiarities of the card. First of all, let us assume we have broken down our high pressure engine, say there is a broken piston rod or the high pressure valve is broken. It would be necessary to work the intermediate and low pressure engines as a compound engine. The first thing to do is to take out the high-pressure valve, shut down the engine room stop valve and convert it into a reducing valve. The steam from the boilers giving 20 to 30 lb. more in the intermediate cylinder than if it were connected direct. You could not carry the full pressure. The probability is that the diameter of that cylinder would be about 1·6 times the diameter of the high-pressure cylinder, and under these circumstances the thickness of the metal ought to be 1·6 times the present thickness of the cylinder, so that you would run the risk of bursting the cylinder with the full pressure.

$$\text{Stress} = \frac{5{\cdot}1 \text{ W A.}}{d^3}$$

This formula gives the stress per square inch on the shaft. The stress usually allowed is from 9,000 to 10,000. If the full boiler pressure was admitted to the intermediate cylinder, say 150 lb. where previously we carried 60 lb., i.e. $2\frac{1}{2}$ times as much, then $2\frac{1}{2}$ times 10,000, or 25,000 lb., would be the stress on the shaft, leaving a factor of safety of 2, it is quite

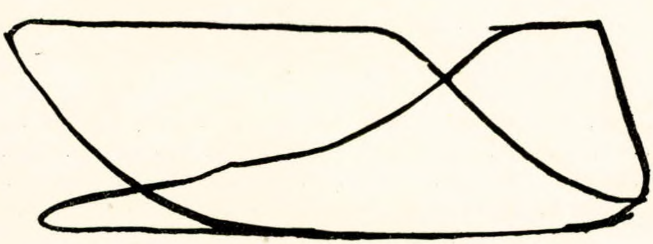

Fig. 7.
Low Valve.

probable that the shaft would give way. Therefore it could not work with the full boiler pressure. What would be the effect upon the condenser. The condenser is designed for a certain weight of steam and the pumps, etc., to contend with a certain weight of water. There would be $2\frac{1}{2}$ times the weight of steam in the condenser and you would not get the proper vacuum. The feed pumps and air pumps would not be large enough. There are other factors also which would prevent the full boiler pressure being used.

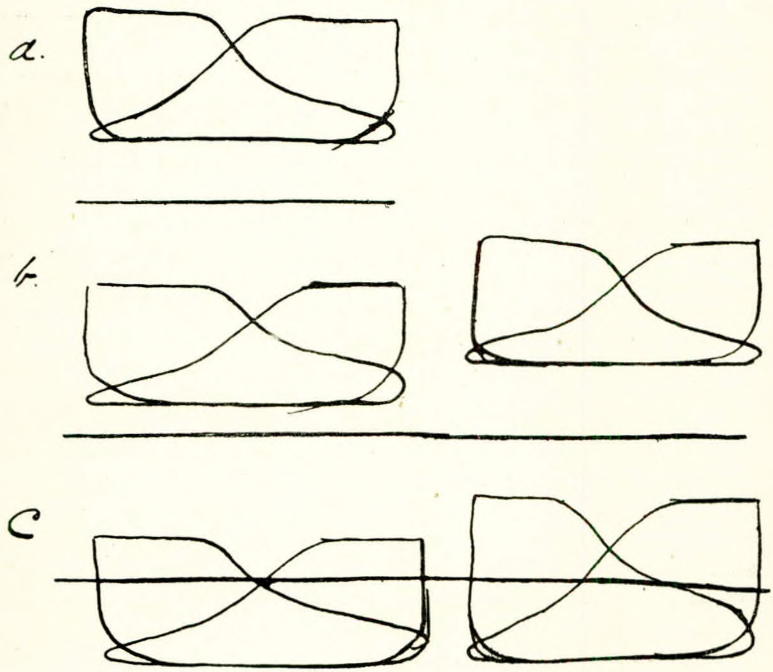

Fig. 8.

In Fig. 8, supposing that *a* is the high pressure card, under the conditions described, *b* the intermediate, and *c* the low pressure, *b* will come up higher, that is, the intermediate engine would be working between a higher range of pressure. The low pressure cylinder pressure will probably rise 8 to 10 lb., and if the same vacuum be maintained, the power will be considerably increased. As far as the power sent through the shaft is concerned this is the worst kind of breakdown you could have in regard to any of the three cylinders. You may have a breakdown of the intermediate or low pressure cylinders and yet be able to get the remaining two cylinders to give the same amount of power ; but you could not get the intermediate and low pressure to give the same power, for the reason that they could not carry the boiler pressure. It would be necessary to carry not more than about 75 per cent. ; so that with a breakdown of the high pressure cylinder you could compound so as to get about 60 to 70 per cent. of the power of the three engines.

With regard to breakdown of the intermediate engine, I know of one case where the intermediate valve was broken. All round the upper port the valve liner gave way. They took out the broken pieces, took out the valve, slung up the link to the stuffing box, took down the connecting rod and exhausted from the high pressure right round to the low, and the power developed was equally divided between the high and low pressure engines. I have the cards from this job, and it is one of the best cases I know of showing the distribution of the work of the engine.

Under normal conditions a leaky intermediate piston should give a similar result. A leaky intermediate valve should give the same thing. The pressure should equalize between the high and low pressure engines and the work lost in the intermediate engine should be seen on the high pressure and low pressure engine cards. That extra work in the high pressure is due to the reduction in the back pressure ; the low will get the extra work in the rise of initial pressure. If the low pressure engine leaks the work is lost altogether. A certain amount will be seen in the intermediate, but it will not be so much. A great deal will go into the condenser in the shape of increased temperature of the steam.

Fig. 9 is a very interesting card, which shows a case where the intermediate engine had to exhaust into the condenser against a pressure greater than the atmosphere. The case is that of an engine which took " a long stroke," when racing heavily, breaking one of the top end bolts of the L P engine. The next upstroke knocked the cover out, and the final result was the entire breakdown of the low pressure engine. The first thing they did was to exhaust into the skylight from the intermediate engine until they patched the eduction pipe. They then exhausted from the intermediate into the condenser; got their circulating pump going and worked

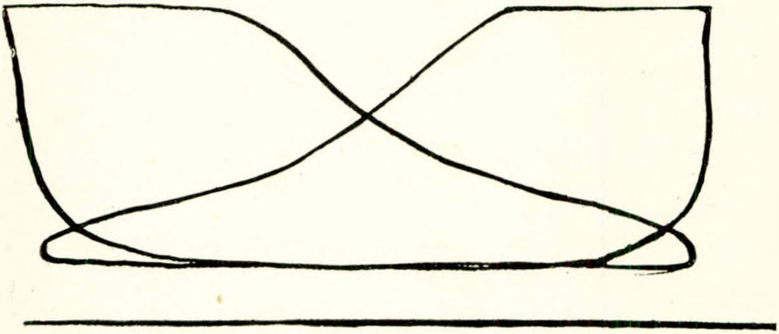

FIG. 9.

condensing. That is one thing which should be observed, if possible, to keep the engine condensing and surface condensing if it can be done. As soon as they had the high pressure and intermediate engines connected and the intermediate exhausting into the condenser, the next point was to get the feedwater into the boilers. They drilled a 4 in. hole in the bottom of the condenser, and connected to the feed donkey, drew the water out of the condenser and put it into the boilers. They were practically two high pressure engines, with no vacuum, the pressure was 4 or 5 lb. above the atmosphere in the condenser.

There was another case in which the low pressure engine had to exhaust against the atmosphere. This is a most interesting case.

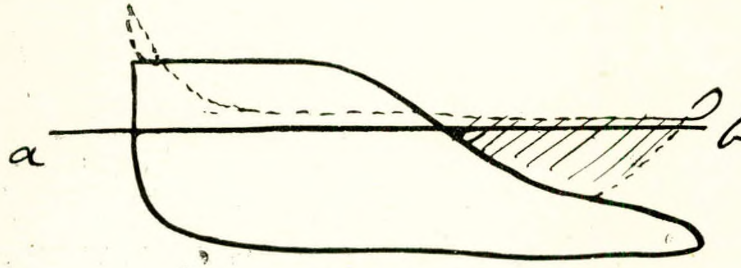

FIG. 10.

Fig. 10 is a low pressure card and *a b* is the atmospheric line. The steam is admitted above the pressure of the atmosphere. It cuts off at about ·6 of the stroke and forms a loop on the exhaust. That is what you would expect to get, in a case where the low pressure engine is exhausting into the condenser above the pressure of the atmosphere. The back pressure line, for part of the stroke, being greater than the steam pressure, it is only doing negative work. Although it may be exhausting into the condenser the low pressure engine is a drag on the other two, and under these circumstances it would have been better for the intermediate to have exhausted into the condenser, but the whole of the pumps were on the L P. All shaded part represents the drag on the other cylinders.

In bringing these matters more particularly before the younger members of the Institute, it is not so much my intention of throwing any new light upon the subject of the indicator diagram as to enlist your sympathies in the subject. Nothing sets your thoughts in the right direction in connexion with an engine, like the indicator diagram. It helps you to diagnose the faults of the engine, and it is that kind of study which I should like to foster.

Mr. J. LANG, R.N.R. : We are much indebted to Mr. Hawthorn for his very lucid and interesting lecture. He has cleared up a great many points in connexion with the subject, but I think it would be a help to the younger members if he would give a sketch of an air-pump card.

Mr. HAWTHORN : First of all I would ask, what is the duty of an air pump ? When Watt first introduced the jet condenser he called it an air pump, and perhaps it was more rightly named so than the modern air pump. The duty of an air pump is certainly to remove any air that may be in the condenser, but its main function to-day in mode is to lift the water of condensation from the condenser and deliver it to the hot well. One of the chief considerations is to get a good vacuum between the foot valve and the bucket valve. That is why so small a quantity of water is lifted in comparison with the stroke of the pump. Probably a 4 ft. stroke engine, with a 2 ft. stroke pump, only deals with $2\frac{1}{4}$ in. of water at one operation. A large amount of space is allowed so as to give every facility or freedom for air, reducing the pressure and thus forming a better vacuum in the air-pump chamber.

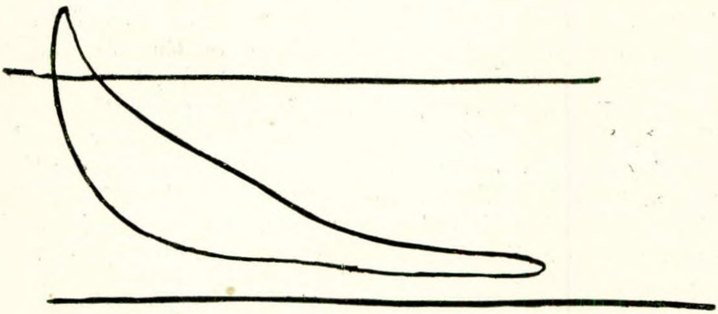

FIG. 11.

Starting at the left-hand side (Fig. 11) the pump goes down, forming a vacuum between the bucket and head valves. There has been no air admitted, and probably there is a vacuum of 25 in. or 23 in. on the top side of the bucket. The bucket goes right down through the water and displaces the water. When coming up you get a better vacuum under this side. Probably the vacuum is maintained until you get to the top centre and deliver the water there, into the hot well at 6 lb. above the atmosphere. That is the card I should expect to obtain. If air was admitted under the head valve I should get the type of card shown in Fig. 12. During the downstroke I do not get such a good vacuum,

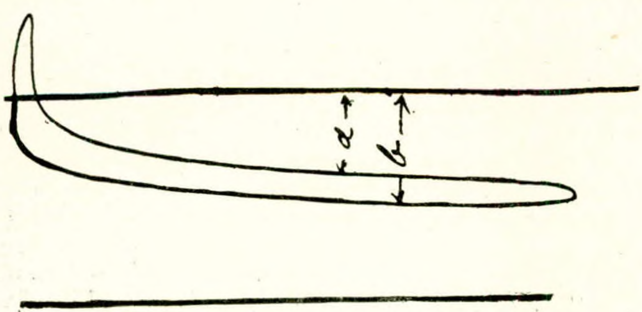

FIG. 12.

but do so during the upstroke on the other side. That would be the position of the line on the under side. That is where air is admitted on the top side of the bucket, making as good a vacuum as before on the other side. This is where the pump is working with practically the same amount of vacuum on each side. *a* measures the vacuum on the top side of the bucket on the downstroke. *b* measures the vacuum on the under side. These are the results I would expect to get.

CHAIRMAN: Perhaps Mr. Hawthorn might add a word or two as to the advantages or disadvantages of a $4\frac{1}{2}$ in. or $2\frac{1}{4}$ in. card as regards the length.

Mr. HAWTHORN: It has always been my impression that we shall never get a perfect diagram. However, if we reduce the length to 4 in. for a 4 ft. stroke, some of the faults are not seen. In my opinion the card should be at least 5 in., the longer the better. When taking into consideration the high-pressure engines of to-day, a slight difference in the line could easily put you out in measuring the pressures. Say I am $\frac{1}{16}$ in. out in measuring the mean effective pressure in an engine with a 5-ft. stroke and 60 revolutions. That very slight error in measuring the pressure will give a difference of 9 H.P. So that there is every reason for making the cards larger to ensure accuracy. The card should be at least 3 in. wide and the length 5 in.

Mr. A. ROBERTSON: Would Mr. Hawthorn give us his idea as to the degree of accuracy to be obtained in measuring the consumption of steam from the indicator diagram?

Mr. HAWTHORN: I will explain, first of all, the method of measuring the consumption of steam or water passing through the engines. The card is divided into ten equal spaces.

Measure the absolute pressure of steam at eight-tenths down. We will presume the point of exhaust is there and the same with the other card. Add the two together and call them p_1 and p_2. Then the following formulae

$$\text{Tons steam } (p_1+p_2)\,\frac{d^2 \times l' \times R}{140{,}000}$$

gives the tons of steam passing through the engine during twenty-four hours. If that is compared with the water passing from the hot well, you will see how much water is going through as steam.

Mr. Robertson: What degree of accuracy would you expect to get from cards taken off a first-class triple expansion engine?

Mr. Hawthorn: I should say about 90 per cent. It would vary about 10 per cent. either way. Your card may show this, but it presumes a dry gas. If there is any moisture in the steam you are using more water than you should. It would depend greatly upon the particular engine; you would require to work out the co-efficient for the engine first. The factor of dryness ought to be about 95 per cent. of the weight of steam. That would represent the true dry gas and the other 5 per cent. would be water.

Mr. Robertson: I am rather against the use of formulae too extensively. I think it is a great fault in the education of young engineers. They get formulae memorised and do not know how these formulae are arrived at. Personally I would sooner work a thing out by a longer method and comprehend it rather than use a formula which I did not understand.

Mr. Hawthorn: For a practical engineer I do not see much difference between a box of tools and a book of rules. Just as he uses different tools for different kinds of work, so he should be able to know which formula applies to a particular case. The formulae are worked out by scientists who have made a life-long study of the conditions; it is the duty of the engineer to apply them when he has to meet those conditions. The engineer has not the time, even if he had the ability, to work these things out for himself. They are there for him to use and the best engineer is the one who knows best how to use them.

Chairman: From my own experience I would say that it is a good thing to know how to use the various engineering formulae. For one thing, it makes you methodical. In some engine rooms and workshops, the men throw the things about here and there; they have no method; and when a job is finished they cannot find it when it is wanted. In one instance I remember being sent into a shop to find something in the dark, and under such conditions one finds the value of being methodical. Certainly mathematical formulae are used by the deepest thinkers. In this room we have seen Mr. Macfarlane Gray so wrapt up in formulae that he has made long pauses in his remarks in order to think them out. Mr. Hawthorn himself has shown us to-night how useful they are in solving difficulties, and I should say that the Institute as a whole and the Members of Council ought to take notice that Mr. Hawthorn has dealt fairly with the Junior Section. To my knowledge, ever since I have been a Member, he has always been willing to come forward and give assistance, and we should be deeply thankful for the many lectures he has given and the way he has illustrated and demonstrated them.

Mr. Robertson: I should like to remove the evident misapprehension of what I intended to convey. I was not deprecating the use of formulae. What I was trying to impress was that engineers, before using formulae, should be completely conversant with the way those formulae are obtained. I did not mean that they should originate them, but that they should understand them. There are many formulae which are quite compatible with an engineer's knowledge to be able to fathom; there are many that any one with an ordinary training should be able to understand. I think it behoves every engineer to be able to work out his own formulae if possible; he will then be far better able to apply it in practice. I have very great pleasure in moving a hearty vote of thanks to Mr. Hawthorn for his lecture this evening. It has been most instructive, and I do not know any other member of our Institute who could have dealt with the subject in a better manner.

Mr. T. A. Crompton seconded the motion, which was carried with applause.

The meeting closed with a vote of thanks to the Chairman on the proposal of Mr. Lang.

The following were elected at the meeting of Council held Thursday, February 9, 1911 :—

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John N. Cook, London.
G. Lloyd Jones, London.
Robt. G. Lapper, Hertford.
Wm. J. Lapper, Hertford.
Colin A. McPherson, Woolston.
P. M. Mistry, Bombay.
Engineer Lieut.-Colonel V. Sagowsky, I.R.N., London.
Harold Sparks, Rockferry.
T. W. Twaddle, Aden.
Wm. T. Watson, Lahore.

As Associate Members.

George H. Nunn, Aden.
Alexr. T. Spence, Abington.
P. H. Writer, Bombay.

As Associates.

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Thomas Jones, London.
Maurice N. Wyse, Karachi.

As Companions.

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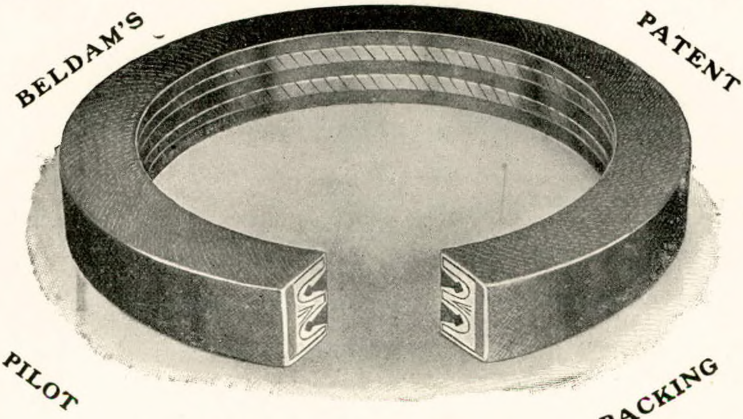
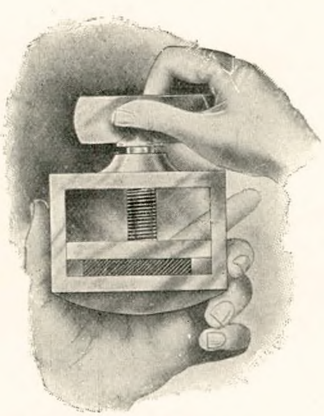


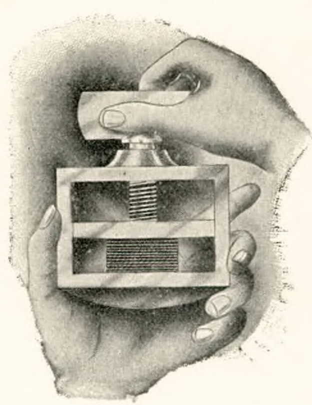
Awards :—Gold Medal, Milan International Exhibition, 1906.
2 Gold Medals, Franco-British Exhibition, 1908.

Sole Agents in Great Britain for

Dod's Diagonal Cross Expansion Packing for Ammonia, Low Pressure Steam, Compressed Air or Water.



Dod's Packing under Pressure.



Ordinary Packing under Pressure

Both pieces of Packing above illustrated were square before pressure was applied.
Note the difference in cross expansion.

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Vol. XXII.

LECTURE ON THE HISTORY AND PRACTICE OF LUBRICATION IN MARINE ENGINES

By Mr. J. VEITCH WILSON.

Delivered Monday, December, 19th, 1910.

Chairman : Mr. J. ADAMSON (Hon. Secretary).

DISCUSSION

Monday, January 23rd, 1911.

Chairman : Mr. J. ADAMSON (Hon. Secretary).

ELECTION OF PRESIDENT, OFFICE-BEARERS AND MEMBERS OF COUNCIL FOR SESSION 1911–1912

ELECTION OF MEMBERS.

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SESSION 1910-1911

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