INSTITUTE OF MARINE ENGINEERS.

SESSION 1891-2.

TWENTY-NINTH PAPER

(OF TRANSACTIONS)

(POSTHUMOUS PAPER).

Ratios & Distribution of Power

IN

Compound Engines.

BY

Mr. D. McMILLAN

(MEMBER).

Read Tuesday, 14th April, 1891.

Discussion continued Tuesday, 28th April, 1891,

IN

THE LANGTHORNE ROOMS.

STRATFORD,

April 30th, 1892.

PREFACE.

A Meeting of the INSTITUTE OF MARINE ENGINEERS was held in the Langthorne Rooms, Stratford, on Tuesday evening, April 14th, presided over by Mr. F. W. WYMER (Vice-President), when portions of a Paper prepared by Mr. D. McMILLAN (Member) were read and discussed.

The subject matter of the Paper was of such a character that it was considered desirable to have it condensed, and the leading features brought into prominence, with a view to the special points being brought more concisely before the Meeting for discussion. As the result of a resolution to this effect being passed, the condensed form of the Paper was read at a Meeting held in the Langthorne Rooms on Tuesday, April 28th, also presided over by Mr. F. W. WYMER.

The Paper as read at the Meeting on the 28th April, with a few additions from the original manuscript, is what is now printed, the original Paper being retained for reference in the Library.

JAS. ADAMSON,
Honorary Secretary.

*RATIOS AND DISTRIBUTION OF POWER IN COMPOUND ENGINES.

BY

MR. D. McMILLAN

(MEMBER).

The writer of this paper was induced to give his special attention to the subject matter treated of in the following pages, by experiencing considerable difficulty in discovering reliable rules or methods for finding the mean pressures under different conditions in the working of engines, proportions of cylinders, and receivers, best adapted for giving the best possible results.

It occurred to him that by studying carefully certain data, gathered from reliable sources, tables might be constructed which would give a set of constants, useful for various purposes, and interesting to the Members of this Institute. With these objects the ideas given in the following pages were placed on paper.

Taking first the question of capacities, it has doubtless been observed that makers differ greatly in respect to the ratios of cylinders and receivers. The proportions of the receivers depend relatively upon the angles of the cranks, thus, if the angle of the cranks is over 90°, the receiver may with advantage be reduced below the normal. An instance of a small receiver may be cited as in engines with an angle of crank, such as those built by Messrs. Jas. Howden & Co., where the steam is exhausted from the high pressure cylinder direct to the low pressure, the portways being large enough to carry the steam in its course from the one cylinder to the other. The greater the angle between the cranks, as a rule, the less the need for a receiver.

* Posthumous Paper.

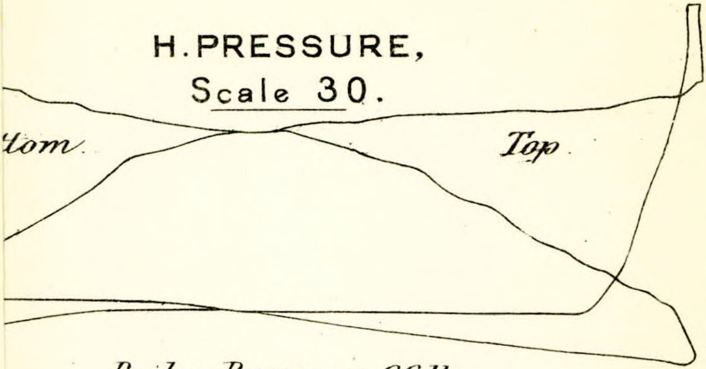
The ratios between the cylinders is a question which depends upon the pressure to be used; but, even with this element taken into consideration, there is a considerable difference in the practice of engine builders. It may be of interest to give figures to show the variations. Example I. = absolute pressure × ·0366. Example II. = absolute pressure × ·03455. Example III. = absolute pressure × ·0341. The differences in these examples do not amount to very much, and probably they are partly due to other details of arrangement, and founded upon the results gained by the different makers in their own individual practice.

In compound engines, the ratio of capacities of cylinders, the relative volume of the receiver, and the point of cut off in the low pressure cylinder, are each of great importance in determining the best distribution of power between the different cylinders.

The ratio of capacities in examples already referred to range in compound from ·0341 to ·0366 times the gross initial pressure of the steam. If the ratio be left to the discretion of the designer, he will be guided by what he has found satisfactory before for pressure and indicated power, and in similar vessels. For more or less horse power or for much higher pressure the ratio would be different. Mr. NORTHCOTT puts it that the ratio for compound engines should be the square root of the total ratio of expansion, and the variations met with are generally nearly in accordance with this. Thus, when there are N cylinders and R is the capacity, ratio R^n = the total ratio of expansion, or the number of times the final terminal pressure is contained in the initial gross pressure. This rule is one which aims at getting the same power from each piston.

In almost every instance more horse power is developed in the low pressure cylinder than in the high. This is because it is desirable to take all the power practicable out of the steam. A very slight alteration in the cut off in either cylinder will make a difference in this division of power. By linking up on the L.P., the receiver pressure, and practically the back pressure of

H. PRESSURE,
Scale 30.



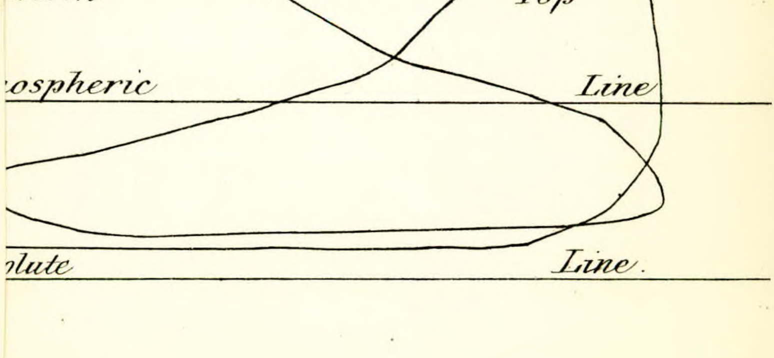
Boiler Pressure 66 lbs.

Mean Receiver Pressure 31·75 lbs.

heric. *Line.*

Vacuum in inches 26 3/4.

L. PRESSURE,
Scale 16.



the H.P. is increased almost inversely as the cut off fraction. In this way the power developed in the high is reduced and the initial pressure in the low is increased.

The careful working of the link contributes to a saving of fuel. By expansion on the link, to a certain extent, the heat is kept up and the motion not perceptibly retarded. This is because the expansion on the whole is not altered, and the same amount of steam passes through the engine, but the pressure and, therefore, also the temperature in the receiver is kept higher, which seems to be an advantage.

When linking up is carried to a greater degree it is useful to check racing in a sea way. The increased compression then retards the motion and considerably diminishes the vibration. This plan of linking up the L.P. engine along with a good governor would certainly reduce racing to a minimum, unless the vessel be over light. I believe that no governor, up to the present, can be depended on to stop racing entirely, when a light ship is labouring and dipping heavily.

The diagrams on annexed sheet show the effect of linking on the low pressure valve separately, the high pressure link being full out.

The low pressure diagram is not what would be expected from a valve motion set symmetrically. As usually set, the obliquity of connecting rod causes the steam to be carried farther down than up. In this diagram steam is carried farther up than down, showing that some unexplained distortion of action had been introduced. It is important to consider which crank leads. In regard to the engines whose diagrams are given in this paper, the high pressure crank leads. The diagram shown, speaking for itself, tells us that all the steam of one stroke of the high pressure and one-fifth of another carry the low pressure piston up, while only four-fifths of one high charge of steam is given to the low on the down stroke.

What is called the receiver pressure is the pressure just before the H.P. exhaust shuts—the mean of the ends. In these diagrams this is $31\frac{3}{4}$ lbs. above the atmospheric line. From pressure, compression begins, and it is carried up to 13 lbs. above the boiler pressure, and a loop is consequently formed.

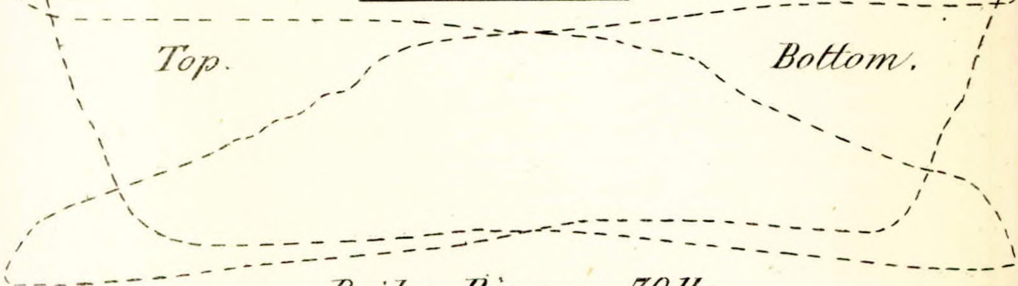
The excessive compression in the H.P. is due to the high pressure in the receiver when the exhaust closes. The excessive compression in the L.P. is due to early cut off, which makes earlier closing of the exhaust. There is inside cover on both slide valves, and this gives more compression. The inside cover declares itself in the relative positions of the points of opening and closing of the exhaust at the same end. Here the exhaust shuts at one edge before it opens on the other, because the closing point is farther from the end of the stroke than the opening point is. The reverse order would indicate that the exhaust was partly open when the valve is at half stroke. It may be matter for discussion to explain why there is a difference in the vacuum lines down and up. In these diagrams the power has been made excessive on the L.P. at the expense of the H.P., and there has been no gain in power; there is actually a dead loss in speed. Under these conditions the engine does not work smoothly, but with a disagreeable sound and much knocking and jarring.

Reverse this plan. Link up the high and let out the low pressure, and there may still be a loop at the admission corner; but that will not now be due to the excessive back pressure in the receiver, for that will be, perhaps, only half what it was in the other. It will be due to commencing compression earlier through the alteration in the action of the valve by linking up. The action of the high pressure valve, if the eccentric rods are, as they ought to be, crossed, crank on top, would be early at every point; there would be early cut off, early exhaust, early closing of exhaust, and also excessive lead—that is, early steam. With the eccentrics crossed the other way, all except the lead would have been early. The lead would then be less when linked up, not more, than it is in full gear. The rods are crossed the latter way in the case cited.

PLATE I.

H. PRESSURE,

Scale 30.



Boiler Pressure 70 lbs.

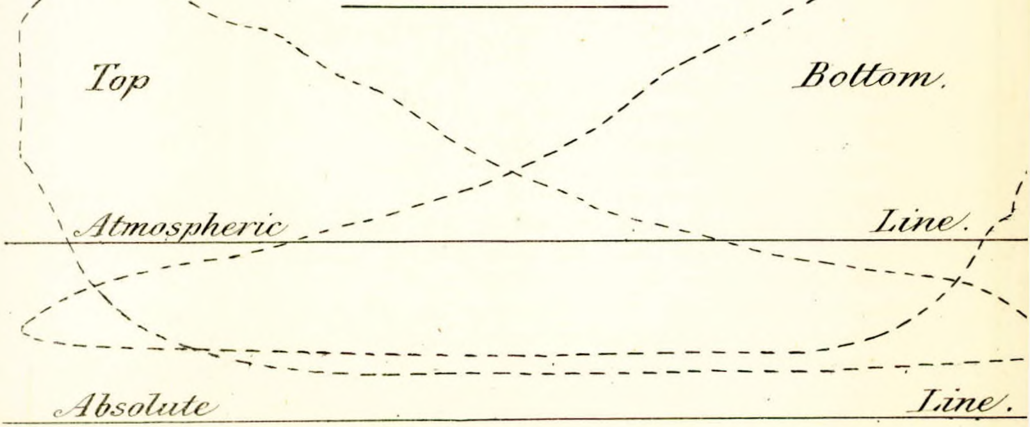
Mean Receiver Pressure 30·375 lbs.

Atmospheric. Line

Vacuum in inches 25.

L. PRESSURE,

Scale 16



The diagrams on Plate I. are another example of the linking up on the L.P. In the former set, the cut off on top was ·25 in the L.P., and in these it is ·425. The steady expansion of the receiver steam is represented on the bottom diagram up to the point of cut off. The horizontal initial admission denotes that the exhaust from the H.P. was keeping up the pressure. The motion of piston is slower from the bottom than from the top, and this is marked on the diagrams, where the horizontal admission part extends farther on the top diagram. The steam is carried farther in the down stroke in the H.P. than it is in the up stroke, and this heavier charge of steam is seen in the up stroke of the L.P., showing that the high pressure crank leads in Plate I. The low pressure slide valve seems to be set high, or to have too little lead, as the receiver pressure is really higher at the commencement of the low pressure down stroke than it is at the commencement of the low pressure up stroke; this is supposing the H.P. to lead. The high pressure exhaust maximum pressure occurs just before the L.P. takes steam, and that is higher on the high pressure diagram bottom, or when the low pressure crank is turning the top centre, if the high pressure crank leads. This, however, is due to the opening of the low pressure valve. In the author's opinion the admission is throttled, and the diagram does not represent the pressure in the receiver at the time.

It cannot be stated with certainty if the eccentric rods are crossed in Plate 1. The impression on the mind of the author is that they are open, crank-on-top; that is, the wrong way for working expansively on the link. Considering that the eccentric rods are crossed crank-on-top, under these circumstances steam would be early at every point, with excessive lead and an early cut off, open to exhaust soon and closed soon. It appears that the angle of the eccentric rods linked up as the low pressure valve is, in the engines these cards were taken from, causes a vastly different action at the opposite ends of the cylinder. In some engines this is not so great, as will be seen from the indicator cards in Plate 1, but then it must be stated that the cards on Page 7A were taken off when the low pressure valve was cutting off on top at ·25 of the stroke, the extreme of the link, and the cards in Plate 1 at ·425 of the stroke on top, the extreme of the link also, which doubtless accounts for the difference. I am aware that in the former case the eccentric rods are open when the crank is on the top centre, but I am not in a position to state at present whether the arrangement is the same in the latter case, but the cards lead me to conclude that it is the same.

I have for the sake of clearness, and so that anyone may understand my mode of proceeding, arranged part of the necessary data in the form of a tabular statement, so that should any experienced engineer take the trouble to study it, he can readily understand the action of the steam pent up within the cylinders, from the lowest rate of expansion to the highest, or at $\frac{2}{10}$ of the stroke in the high pressure cylinder. It may not be unreasonable to say that a young persevering studious engineer will have no difficulty whatever in understanding it when all is explained, and it may be the means of making him more eager to know more about the working of marine engines. Such subjects generally give impetus to the mind of the anxious persevering student. The following table then, contains the necessary data for finding the mean pressures in the cylinders of compound engines, and as there are several tables to follow, this will be table 1, and it is according to engines the cranks of which are set at right angles, and if 1 be taken to represent the capacity of the high pressure cylinder, neglecting clearance, that is, area multiplied by stroke, it being 42 inches, then the capacity of the receiver is 2·726 times greater, making all necessary deductions, and the results are as shown in the table appended.

The following tabular statement gives the variations of the receiver pressures and the mean pressures for various expansions from $\frac{2}{10}$ admission to $\frac{7 \cdot 25}{10}$.

These have been arrived at from actual diagrams taken from a compound engine, cranks at right angles, receiver capacity 2·72 times the stroke capacity of the H.P., stroke 42 inches, diameter of cylinders 34 inches and 60 inches, high pressure crank leading, ratio of cylinder capacities 3·114.

The table is for tenths, and for intermediate cuts off the numbers will be proprotionately intermediate.

EXPANSION TABLE I.

TENTHS.	$\frac{2}{10}$	$\frac{3}{10}$	$\frac{4}{10}$	$\frac{5}{10}$	$\frac{6}{10}$	$\frac{7}{10}$	$\frac{7\cdot25}{10}$
Rates of Expansion ..	5	3·333	2·5	2	1·666	1·428	1·379
Gain or Loss at terminals	+3·125	+2·25	−1	−4	−6·51	−10·147	−12
Factor for Receiver pressure ..	·647	·61	·593	·594	·584	·575	·575
Factors for H. pressure Cylinder mean pressure ..	·941	·96	·88	·856	·856	·835	·819
Factors for Low pressure Cylinder mean pressure ..	·693	·683	·692	·706	·716	·702	·708
Total back pressure in Low pressure Cy.	2·9	2·29	3·25	3·4	3·25	3·5	3·5
Absolute Receiver pressure ..	12·37	16	18·38	21·38	24·24	26·375	26·5
Actual mean pressure H. P. Cyl. ..	24·93	31·91	34·72	36·374	37·867	37·93	37·97
Actual mean pressure L.P. Cyl. ..	5·69	7·83	9·17	11·15	13·258	14·14	14·37
Loss at Admission in lbs., H. P. Cyl. .	5·5	5·5	4·5	4·5	4·5	4·5	4

+ Gain. — Loss in lbs.

It must be made plain at the outset that the factors for receiver pressure only, as given above, are for temporary use for comparing one engine with another.

A rule will be given subsequently for finding the receiver pressure without the aid of factors. Should the cut off requisite come between any of the tenths, such as $5\frac{1}{2}$ tenths or $6\frac{1}{2}$ tenths, the difference between any of the nearest factors must be taken and that divided accordingly.

The line "gain or loss at terminals" gives how much the actual terminal pressure in the diagrams examined, was above or below that due to the actual boiler pressure and the cut off as expressed in the headings.

The nominal cut offs given, are the means of what the actual cut off was, and if all engines have the same differences in actual and nominal cut off, the plan must be the right one. Engines that differ in this respect of course would require a different table made for them. Here it is seen that at some cut offs the steam actually in the cylinder at the end of the stroke varies from theory, according to the nominal cut off, from 3 pounds over measure to 12 pounds under measure.

The "factor for receiver pressure" is the fraction which the receiver pressure is of the actual terminal pressure in the high pressure cylinder. As the actual terminal pressure is what is multiplied by the fraction it might be thought that the receiver pressure would be modified according to the low pressure cut off, but the factors are given as for a constant low pressure cut off $= \cdot 583$. The factors are not altered for different initial pressures.

The table will apply very accurately always to the same engine and approximately to engines of different proportions of capacities.

The "gain or loss" is taken from the actual diagrams, it will be always so many pounds on or off as it is in Table I., and as given in the following illustrations.

Let R be the rate of expansion in the high pressure cylinder, P the initial absolute pressure, and T the absolute terminal pressure, and G the gain or amount that the pressure of release is greater at this rate of expansion than the theoretical absolute terminal pressure, and X the factors.

Then $\frac{P}{R} = T$

$(T + G) \times X =$ the actual absolute terminal pressure.

The rate of expansion at $\frac{3}{10}$ being 3·333, and, as the initial absolute pressure is 80 lbs., the gain is 2·25 lbs. and the factor ·61.

Then $\frac{80}{3.333} + 2{\cdot}25$ lbs. $= 26{\cdot}25$ lbs. the actual absolute terminal pressure, and $26{\cdot}25 \times {\cdot}61 = 16$lbs. the actual absolute receiver pressure. The above result, minus 15 lbs. the atmospheric pressure, shows that there is 1 lb. of steam in the receiver cutting off at $\frac{3}{10}$ of the stroke in the high pressure cylinder.

A glance at the line of factors from right to left shows that the figures are gradually becoming greater. This is partly due to the variations in receiver pressure between each tenth, and partly to the variations at terminals between each tenth. The drop in the receiver being the fall in pressure that takes place from the moment the exhaust or eduction port opens till a steady pressure is maintained in the receiver or cylinder or whatever the receptacle may be, and if it is measured as follows, it will be found to be greater at $\frac{7}{10}$ than at $\frac{3}{10}$ of the stroke.

The theoretical terminal pressure at $\frac{7}{10}$ of the stroke with an initial absolute pressure of 80lbs. is 56·02lbs, so that 56·02—10·147 lbs. = 45·873 lbs., the actual absolute terminal pressure.

Now, as the actual absolute receiver pressure is 26·375 lbs., we have 45·873 — 26·375 = 19·498 lbs.

$$\frac{19{\cdot}498 \times 100}{45{\cdot}873} = 42{\cdot}5 \text{ per cent. which represents the}$$

fall or drop.

Again, the theoretical terminal pressure, at $\frac{3}{10}$ of the stroke with an initial absolute pressure of 80lbs. is 24 lbs. and 24 + 2·25 lbs. = 26·25 lbs., the actual absolute terminal pressure at this grade.

And, as the absolute receiver pressure is 16 lbs.,

We have, 26·25—16 = 10·25 lbs.

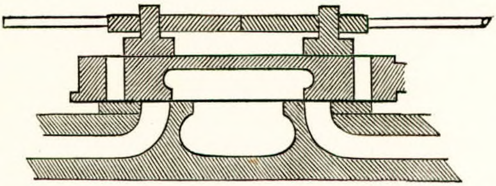
And $\frac{10{\cdot}25 \times 100}{26{\cdot}25}$ = 39·047 per cent., which represents the fall or drop at $\frac{3}{10}$ of the stroke.

It is greater then at $\frac{7}{10}$ than it is at $\frac{3}{10}$ of the stroke by, say, the difference in per cent., or 42·5—39·047 = 3·453 per cent.

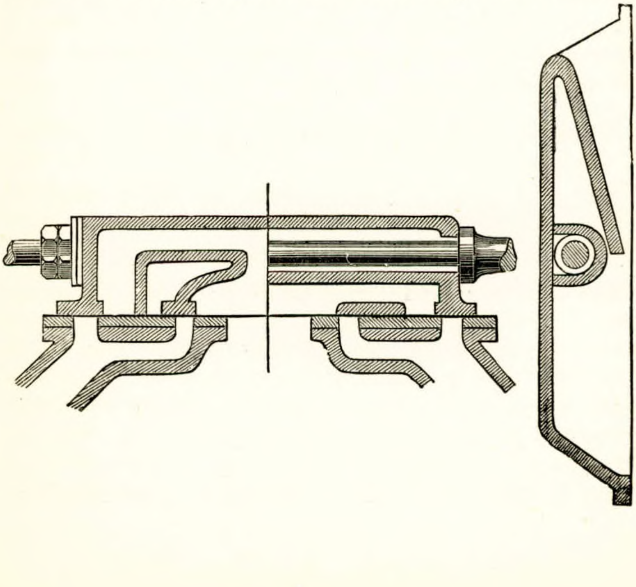
If the steam is cut off at $\frac{2}{10}$ of the stroke, the increase will be 50·15 per cent. Experience teaches that it would not be advisable to carry expansion beyond this with a high pressure without increasing the stroke. At lower rates, varying from $\frac{2 \cdot 7}{10}$ to $\frac{4}{10}$, depending upon the stroke and power, there is no jarring of any consequence, the motions are more regular, and the effect better. But this is less economical. The special trade a vessel is intended for is doubtless a consideration, but much rests with the designer.

In order to get the whole of the useful work out of the steam, it may be worked down to 2 lbs. per square inch, but 5 lbs. is about the lowest in good practice at the point of release, cutting off at $\frac{2}{10}$ of the stroke.

The back of the high pressure valve of No. 1 engines, instead of having, as is usually the case, one or two ports extending across the valve for the greater part of its breadth, both top and bottom, or wherever placed, by which steam is admitted and cut off, is constructed with a series of angular ports, and the expansion slide has also a series of angular ports through it, corresponding with the ports in the main slide valve. Consequently the expansion valve is moved across the back of the main slide by the gearing attached, and this movement either increases or reduces the area opening according to which way it is moved. The up and down motion or travel effects the requisite cut off, and that very effectively when the edges of the ports are fairly good. This arrangement is worked direct from the end of the crank shaft, dispensing with the usual eccentric gearing bracket and quadrant. Figure 1 shows a high pressure slide valve and ordinary expansion valve, which can be



varied at will by means of a right and left screwed spindle. Expansion valves, however, are generally fixed similar to an ordinary slide valve, and varied at will by the quadrant gearing above referred to. Figure 2 shows the usual low pressure slide valve, which is double ported, and admits steam past the outer edge and through



suitable apertures in the back of the valve at the same time. It has two ports open to exhaust at the same time; consequently, with the same stroke as the high pressure slide valve, it has double the total area open for steam and for exhaust alternately. But now to show how the rule is applied for finding the mean pressures of compound engines in accordance with what has been formerly stated. It may be stated here that it will be much the same in the case of triple and quadruple expansion engines, with the exception that appropriate tables would be required for the different systems. The working of No. 1 engines it will be remembered corresponds with the facts stated in Table 1. Then since 80 lbs. is the initial absolute pressure used, and the ratio of the low pressure to the high pressure cylinder being in the relation of 3·114 to 1, the cut off in the high pressure cylinder is ·726 of the stroke, the rate of expansion being 1·377. The cut off in the low pressure cylinder is ·5833, the rate being 1·714, the total back pressure in the low pressure cylinder is 3·5 lbs., and the loss between the boiler and admission to the high pressure cylinder is 4 lbs. The stroke is 42 inches.

$\frac{80}{1{\cdot}377} = 58{\cdot}09$ lbs., the theoretical absolute terminal pressure $(58{\cdot}09 - 12 \text{ lbs.}) \times {\cdot}575$ factor $= 26{\cdot}5$ lbs., the actual absolute receiver pressure.

$$\frac{1 + \text{Hyp. Log. } 1{\cdot}37}{1{\cdot}37} = {\cdot}959$$

$$(80 - 4) \times {\cdot}959 - 26{\cdot}5 = 46{\cdot}384 \text{ lbs.}$$

And $46{\cdot}384 \times {\cdot}819 = 37{\cdot}97$ lbs., the actual mean pressure in the high pressure cylinder.

$$\frac{1 + \text{Hyp. Log. } 1{\cdot}71}{1{\cdot}71} = {\cdot}898$$

$(26{\cdot}5 \text{ lbs.} \times {\cdot}898 - 3{\cdot}5 \text{ lbs}) \times {\cdot}708$ factor $= 14{\cdot}37$ lbs., the actual mean pressure in the low pressure cylinder. The above then are the practical results found by a single method. Though simple, there is, nevertheless, much thought and labour in connexion with it. Should it be desirable to know what would be the difference in the receiver pressure by having a later cut off in the low pressure cylinder, say at ·62 of the stroke, it will be found near enough by inverse proportion as follows:—

The stroke being 42 inches, and the mean cut off in the low pressure cylinder 24·5 inches on ·5833 of the stroke, and the receiver pressure 26·5 lbs., what will be the receiver pressure if the cut off is ·62 of the stroke? As ·5833 is to 26·5 lbs., so is ·62 to the receiver pressure required:—

$$\frac{26{\cdot}5 \times {\cdot}5833}{{\cdot}62} = 24{\cdot}931 \text{ lbs.}$$

which makes a difference of 1·569 lbs. in the receiver, and which is equivalent to about 1·01 lbs. per inch in this instance. Proceed similarly in all other instances, and the result found will be a close approximation of the practical even to ·8 of the stroke.

Later a rule may be given for finding the cut off to maintain a certain pressure in the receiver as formerly stated, as it is considered better to proceed now to show what are the effective initial loads, also the mean load on both pistons, &c. Let A be the area of the low pressure cylinder, and 3·114 the ratio of low pressure to high pressure cylinder.

Effective mean pressure in H. P. cylinder 37·97 lbs.

" " " L. P. " 14·37 lbs.

Effective initial load on H. P. piston = $(80 - 26{\cdot}5) \dfrac{A}{3{\cdot}114} = 17{\cdot}18$ lbs.

Effective initial load on L. P. piston = $(26{\cdot}5 - 3{\cdot}5) = 23{\cdot}00 \times A$.

Effective initial load on both pistons = $37{\cdot}97 \times \dfrac{A}{3{\cdot}114} \times 14{\cdot}37 \times A = 26{\cdot}563$ lbs. $\times$ A.

Efficiency of the system ·635 nearly.

In table II. are given the factors for another engine with a proportionately smaller receiver to show how they differ in the two cases.

EXPANSION TABLE II.

TENTHS.	$\frac{3}{10}$	$\frac{4}{10}$	$\frac{5}{10}$	$\frac{6}{10}$	$\frac{7}{10}$	$\frac{7\frac{1}{4}}{10}$
Rates of Expansion . .	3·333	2·5	2	1·666	1·428	1·379
Factors for Receiver press. .	·668	·653	·663	·648	638	·638
Gain or Loss at Terminals	+3·375	+2·8	—·641	—3·218	—6·856	—8·708
Factors for H.P. Cylind. mean pressure . .	1·07	·958	·905	·87	·834	·799
Factors for L.P. Cylind. mean pressure . .	·605	·605	·56	·56	·55	·55
Total back pressure in L.P. Cylinder . .	4·75	5	5	5	5	5
Loss at Admission in H.P. Cylinder . .	6·62	6·62	6·6	6·5	6·5	6·5

+ Gain. — Loss in lbs.

This would be all very well if the engines were to be worked always at full gear, but it would be neither so satisfactory nor so beneficial under different circumstances. There is a mean, and I might say that the most effective points have been struck by our leading practical engineers. The work can be better equalised in the triple expansion system, but this may be dealt with later on. The foregoing table shows the working of compound engines of 500 nominal horse power, by one of our best makers, from $\frac{3}{10}$ to $\frac{7}{10}$ of the stroke.

Data will be found from tables 1 and 2 near enough the mark for practical purposes. Seeing that the factor for the high pressure cylinder mean pressure, corresponding to $\frac{7}{10}$ of the stroke, is less in table 2 than that corresponding to the same rate in table 1, the receiver pressure of No. 2 engines is higher, but as the rate of expansion increases or with an earlier cut off the factors rise higher than they do in case 1.

Mr. A. SOMMERVILLE

(Member).

Mr. McMillan has given us all an example of energy and industry which we would do well to follow. Every one who has heard the summary read by Mr. Ruthven to-night must see what a vast amount of research and great length of time it must have taken to prepare the paper itself, as well as the patience and labour required to work those formulæ in such a complete manner.

My own practical experience tells me that diagrams are not much to be relied upon, even when read by men who consider themselves experts in slide valves, air pumps, and all other kinds of diagrams.

I have been with engines that worked sadly out of order, and yet the diagrams did not show anything materially wrong, but when we opened the engines for examination we found such vagaries as broken piston springs, cod and tongue pieces out of place, and slide valves passing steam, and yet the diagrams were silent as to these defects.

Some years since I had the pleasure of expressing my opinion about diagrams to a celebrated engineer and shipbuilder on the Clyde, the first, or one of the first, to introduce inverted direct acting engines in ocean steamers, and who held that diagrams are a true measure of the power of engines, also that all internal defects were shown, that, indeed, the indicator diagram was the engineer's stethoscope. This I strongly maintained was not altogether correct, on the following grounds :—

If we go with a fair wind down the Channel, and we find the engines are making 56 revolutions per minute, indicating 2,500 h.p., burning 50 tons of coal per day, and, if without altering anything about the engines, we merely turn the ship round head to wind, will we not find the engines making probably 54 revolutions, indicated horse power down to 2,200, and consumption of coal increased 12 per cent. ?

Again, in the Navy we have ships showing on their trial trips enormous indicated over their nominal horse power, does this not mean a fine pitched propeller with small area of blade, giving a high piston speed, and consequently a high horse power ?

I sailed in a steamer some time ago, and when deeply laden, with the engines working full open, she steamed $11\frac{1}{2}$ knots and indicated 1,200 h.p., but under precisely the same conditions, excepting that the ship was on a light draught, the indicated horse power increased to 1,600 and the speed to 14 knots.

If Mr. McMillan, whom I consider an expert at diagrams, will give me a practical explanation of those vagaries not showing on the diagrams, I will be more inclined to accept the proposition that indicator diagrams are of sufficient value to base accurate data upon.

THE HONORARY SECRETARY.

With reference to the proportions of cylinders made by different makers of compound engines, these certainly do vary considerably, as the following figures will show:—

COMPOUND ENGINES.

Reference.	Pressure.	Diameter of Cylinders.	Stroke.	N.H.P.	Ratio of Areas. H.P. to L.P.
A	90	56 in. & 105 in.	72 in.	1000	1 to 3·515
B	90	58 „ & 100 „	63 „	800	1 „ 2·972
C	90	36 „ & 70 „	45 „	250	1 „ 3·78
D	90	35½ „ & 69 „	48 „	267	1 „ 3·77
E	90	50 „ & 90 „	60 „	354	1 „ 3·24
A	80	32 „ & 58 „	36 „	175	1 „ 3·285
B	80	54 „ & 94 „	60 „	650	1 „ 3·03
C	80	48 „ & 87 „	54 „	500	1 „ 3·285
E	80	32 „ & 62 „	42 „	185	1 „ 3·753
F	80	44 „ & 78 „	46 „	265	1 „ 3·142
A	75	34 „ & 60 „	39 „	180	1 „ 3·114
B	75	50 „ & 86 „	54 „	500	1 „ 2·958
C	75	34 „ & 60 „	45 „	200	1 „ 3·114
E	75	$49\frac{7}{32}$ „ & $90\frac{3}{16}$ „	47¼ „	500	1 „ 3·361
A	71	46 „ & 72 „	48 „	360	1 „ 2·449
A	70	60 „ & 104 „	60 „	700	1 „ 3·004
B	70	34 „ & 60 „	42 „	180	1 „ 3·114
C	70	37 „ & 66 „	42 „	200	1 „ 3·181
E	70	36 „ & 63 „	45 „	230	1 „ 3·062
A	65	34 „ & 60 „	39 „	180	1 „ 3·114
B	65	36½ „ & 63 „	45 „	221	1 „ 2·979
B	60	47 „ & 82 „	42 „	346	1 „ 3·043
C	60	35½ „ & 62 „	39 „	180	1 „ 3·05
F	60	50 „ & 86 „	54 „	500	1 „ 2·958

TRIPLE EXPANSION ENGINES.

Reference.	Pressure.	Diameter of Cylinders.			Stroke.	N.H.P	Ratio of Areas.	
							H.P. to I P.	I.P. to L.P.
		in.	in.	in.				
B	160	31	51 &	84	54 in.	530	1 to 2·706	1 to 2·22
G	160	28	46 &	76	51 ,,	450	1 ,, 2·69	1 ,, 2·72
H	160	38	$61\frac{1}{2}$ &	100	66 ,,	1100	1 ,, 2·61	1 ,, 2·64
I	160	24	39 &	64	42 ,,	300	1 ,, 2·64	1 ,, 2·69
A	150	41	63 &	100	72 ,,	1000	1 ,, 2·36	1 ,, 2·51
G	150	34	56 &	90	60 ,,	550	1 ,, 2·71	1 ,, 2·57
J	150	$40\frac{1}{2}$	$63\frac{1}{2}$ &	102	72 ,,	1000	1 ,, 2·46	1 ,, 2·58
I	150	29	47 &	76	51 ,,	500	1 ,, 2·62	1 ,, 2·61
B	170	20	$33\frac{1}{2}$ &	54	42 ,,	151	1 ,, 2·806	1 ,, 2·59
C	170	33	52 &	$86\frac{1}{2}$	60 ,,	700	1 ,, 2·48	1 ,, 2·76
B	180	20	$33\frac{1}{2}$ &	54	42 ,,	151	1 ,, 2·806	1 ,, 2·59
J	180	22	36 &	60	48 ,,	300	1 ,, 2·67	1 ,, 2·77

It is in my opinion a fallacy on which the proportions advocated in the paper are based. An indicator diagram is not a safe guide to trust to where strict accuracy of details is absolutely necessary, and especially where the diagrams are taken by different engineers in different steamers and under different conditions. In order that formulæ may be of value, the bases must be beyond question, or arbitrary. Even in matters of speculation this must hold good, or it will be impossible for those who take part in the controversy to argue from the same premises.

Mr. McMILLAN has gathered together a great many points, and has been at considerable pains to collect the data and place the results of his observations before us. We are indebted to him for doing so, and also for calling attention to a subject which affords food for considerable study and discussion. There is no doubt that many engines have been designed and constructed with too much capacity of receiver, such being detrimental to the economical working of the engine. In some cases such as this, the receiver capacity has been reduced by being built up with bricks and cement. An indicator diagram is very useful, and indicates fairly well how an engine is working, but it does not seem to me to be sufficiently reliable for constructing formulæ upon, at the same time, the results given by Mr. McMillan are both interesting and useful.

Mr. McMILLAN.

It appears that on examination of certain engines Mr. Sommerville found certain vagaries or freaks in connection with the piston springs and tongue pieces, together with a leaky slide valve, and yet the indicator diagrams were silent to all this.

Indicator diagrams do not point to either broken or slack springs, or displaced tongue pieces, unless steam is passing the piston owing to that. The slide valve could not be passing much steam when under pressure, or it would have shown on the diagram. The practical ear is, perhaps, the best indicator for detecting such slight defects as have been mentioned. The celebrated engineer referred to valued his indicator, I believe, because the lines it traced spoke volumes to him. The indicator cards enabled him to see wherein lay his errors in design, or how much he improved on previous practice.

What Mr. Sommerville says we shall find, should we steam out a certain distance seawards with a fair wind and return against it, the engines working during the whole time at the same rates of expansion, seems to me contrary to both reason and practice.

If the speed is reduced to the extent of two revolutions per minute, steaming against the wind, the horse power must be less and the consumption per day will be less. If by steaming at the rate of two revolutions less per minute we save 100,000 cubic feet of steam per day of twenty-four hours, the consumption must be less, considering the rate of expansion, and the quality of the coal is the same.

If I understand him aright, Mr. SOMMERVILLE means that the engines were full open when the ship was heavily laden as well as when light. The cut offs were the same, but the conditions in respect to draft were vastly different. Under such circumstances the indicator diagrams would simply show the amount of steam used per stroke, and the difference would not be great per stroke.

Surely Mr. SOMMERVILLE expected that the revolutions would be much increased with a light ship, the same amount of steam being used per stroke as when heavily laden, and that in consequence the horse power would be increased, the differenee being in the resistance and the power required.

Indicator diagrams are the most correct measure of the power applied to move the pistons of steam engines known at present, and they are of great practical value to marine engineers. It would give me much pleasure if Mr. SOMMERVILLE, or any other engineer, would furnish us with a handy machine that would show what the former describes as vagaries, but until then I fear we must continue to appeal to both ear and judgment for such cases.

N.B.—Since the foregoing Paper was read and condensed to the form in which it is now printed, the author (Mr. McMILLAN) and Mr. SOMMERVILLE have both passed away from our midst, otherwise it is probable that several points which appear to require further elucidation from the author's view, would have been made clear.

J. A.