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"Economy Trials of a Non-Condensing Steam-Engine: Simple, Compound and Triple."

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THE trials about to be described have been made by the Author with one of his central-valve engines.

They were planned to determine certain points in connection with its economical performance at various speeds and with various steam-pressures; also to obtain figures for comparison with the results gathered from other types of steam-engine. In the course of the first few trials, many questions arose in connection with initial condensation in the cylinders; and in order, if possible, to learn something of the effect of surface, range of temperature, and speed of rotation in increasing or diminishing this condensation, the original group of trials has been added to considerably. Subsequent trials will include condensing as well as non-condensing tests, trials on the brake, and trials in connection with the dynamo-machines for driving which the central-valve engine has been used.

This Paper deals with the non-condensing trials, and includes trials of the same engine as a simple, a compound, and a triple-expansive engine. These have been made with steam-pressures varying from 40 lbs. to 170 lbs. per square inch, and at from 110 to 420 revolutions per minute.

The Author is not aware of any trustworthy experiments having been previously made with engines running at so high a speed. The trials at high steam-pressures have also, so far as he knows, been few and isolated. His aim was to show, with some approach to accuracy, the effect of varying steam-pressures and varying ratios of expansion.

The central-valve engine is well adapted for the purpose, for it can be readily altered from a simple to a compound or a triple-expansive engine. The low-pressure piston and valves remain the same throughout these changes; but a second or third smaller piston and set of valves is added for compound or triple working, all being coupled to the original crank and eccentric. The point of cut-off in any of the cylinders can readily be altered, and the engine can be driven at widely varying speeds.

The main group of trials has been made at 400 revolutions per minute, because, above that speed the indicator-diagrams are not so steady, and, for the purpose which the Author had in view, it was essential that the diagrams should be measured with some accuracy to determine the steam present at various points in the expansion. In practice, it is, however, much more usual to run these engines at 500 revolutions per minute, and, from the changes which occur between 120 and 400 revolutions, it is not difficult to foreshadow the effect of a further increase in speed on the general economy of the engine.

In the trials, the power which has been measured is the indicated power, and this has been compared with the feed-water pumped into the boiler. The indicated power is clearly the right point to start from, as it is only by reference to indicator diagrams that the various losses can be located. Most questions of importance can be settled with reference to the power thus ascertained, and but few trials are necessary to determine the comparative values of indicated and brake powers. The Author has, however, thought it desirable to give details of a trial recently made to ascertain the useful electrical power obtained from various indicated powers in the same engine.

The doubtful points which the Author hoped to settle, for his own engine at any rate, by the trials which form the subject of the present Paper, may be briefly stated thus:—

1st. To ascertain under various conditions, the percentage of feed-water pumped into the boiler, which can be accounted for by the indicator in the cylinder to which the steam is first admitted, and the percentage which cannot be so accounted for owing to leakage, radiation, priming, and initial condensation.

2nd. To ascertain the effect of speed of rotation, area of exposed surface, steam-pressure, and range of temperature on initial condensation and on economy generally.

3rd. To fix, as nearly as possible, the best practical ratio of expansion with any given steam-pressure.

4th. To ascertain what percentage of the work, due theoretically from the steam shown by the indicator, is actually obtained under various conditions.

5th. To determine the steam-pressure beyond which the expansion can be most economically carried on in two cylinders instead of one cylinder, and the pressure beyond which it may be desirable to employ three cylinders.

6th. To ascertain the ratio of the work done by each lb. of steam, to the work thermo-dynamically due from it under the limiting conditions of each trial.

The trials were so arranged that one observer only was necessary, and thus it became possible to have some of the more important trials checked by independent engineers, under whose eyes every observation for any one trial could be brought. In this way personal error was as far as possible eliminated.

The gentlemen who kindly consented to make these trials were Mr. MacFarlane Gray, Professor Kennedy, Mr. Druitt Halpin, Professor Unwin, F.R.S., and Mr. Wilson Hartnell, and the results of their trials have been, as far as possible, embodied with the Author's in the tables.

A description of these engines has been published,[1] and it is, therefore, unnecessary to describe them in detail now. It may be well, however, to mention that the valves for all three cylinders are actuated by one eccentric placed on the crank-pin. They are placed in the interior of the piston-rod, which is hollow. The cut-off is effected at full piston-speed, by ports in the hollow rod passing into metallic rings in the cylinder ends.

The point of cut-off can be altered by hand or by the governor; but, as in the trial engine the effect of each range of expansion was to be observed, the cut-off was fixed by raising or lowering the metallic rings at the end of the cylinder by washers of various thicknesses placed beneath them. The washer was stamped with the distinguishing number of the trial for which it was made, in order that any trial which appeared to conflict with others, or the results of which for other reasons it was desirable to check, could be reproduced with certainty. Most of the cushioning of the reciprocating parts, for reasons given in the above Proceedings, is done in a chamber apart from the working cylinder.

The Author has brought the experimental engine for exhibition, in order to give a clearer idea of its construction. The working parts are exactly in the condition in which they were when taken out at the conclusion of the trials.

[1] Minutes of Proceedings Inst. C.E. vol. lxxxiii. p. 182.

It will be convenient to divide the Paper into five heads:—

(A) A brief consideration of the work theoretically due from a given quantity of steam at various pressures, and exhausting into the atmosphere (no allowance being made for any excess of back-pressure).

(B) A description of the main groups of trials made at 400 revolutions per minute with the engine working as a simple, a compound, or a triple-expansive engine, and with ratios of expansion which appeared from theoretical considerations to be best suited to the various steam-pressures employed.

(C) The difference between the work actually obtained and that theoretically due from the steam.

(D) A description of certain trials made to determine the amount of moisture in the steam, and the losses likely to arise from leakage and radiation, and to ascertain approximately the effect of speed of rotation, area of exposed surface, and range of temperature on the amount of steam condensed initially and up to the point of cut-off.

(E) A description of a group of trials made to determine the probable value of automatic expansion-gear in a compound engine.

(A) Work Theoretically Due from a Given Quantity of Steam Exhausting into the Atmosphere.

Neglecting the variation of specific heat of water, the usual formula for total heat in steam, calculated for arithmetical convenience from zero Fahrenheit, is $1{,}115 + \cdot 3\,t$, where $t =$ the temperature Fahrenheit. Taking absolute temperature $= 461 +$ Fahrenheit, and adapting the above to absolute temperature, which will be denoted by θ, then—

$$\text{Total heat in steam} = 1{,}438 + \cdot 3\,\theta.$$

Working with steam at absolute temperature A, expanding to absolute temperature B, and exhausting against the pressure due to steam at the latter temperature, the heat-units U thermodynamically due in the shape of work from 1 lb. of steam of the quality for which Regnault determined the latent heat, are, as expressed by Mr. J. MacFarlane Gray:—

$$U = \left(\frac{1{,}438 - \cdot 7\,A}{A} + \frac{A - B}{A + B}\right)(A - B).$$

Taking the heat-unit as 770 foot-pounds, the result is 2,571 heat-units = 1 HP.-hour.

The lbs. weight of steam per HP.-hour are therefore—

$$W = \frac{2{,}571}{U}.$$

The results worked out for various pressures according to this formula are shown by the lowest curve on Plate 3, Fig. 1, the ordinates measured from the base-line to the curve giving the lbs. of steam per HP.-hour required thermo-dynamically with the various absolute steam-pressures indicated.

The comparison about to be made between the weight of steam actually used to produce 1 HP. for an hour, and the weight shown in Plate 3, Fig. 1, is with the most perfect engine possible, working under the ideal conditions assumed in the works of Clausius, Rankine, Tait, and other authorities.

The words used by Clausius in Chapter XI, sect. 3, under the head of "Assumptions for the purpose of simplification," will best explain what this perfect engine is.[1]

In sect. 5, p. 241, "Determination of the work done during a single stroke," he finally arrives at the following formula :—

$$W' = m_1\,\rho_1\,\frac{T_1 - T_0}{T_1} + M\,C\left(T_1 - T_0 + T_0 \log\frac{T_0}{T_1}\right),$$

[1] "For the purpose of this investigation, we will assume, as has usually been done, that the cylinder is a non-conducting vessel, and so neglect the exchange of heat which takes place during each stroke between the walls of the cylinder and the steam.

"The vapour within the cylinder can never be anything but steam of a maximum density, with a certain admixture of water. For it is evident, from the conclusions of Chapter VI, that during the expansion which takes place in the cylinder after it is shut off from the boiler, the steam cannot pass into the superheated condition, because no heat is imparted to it from without, but must rather partially condense. It is true that there are certain other processes, to be mentioned later, which tend to produce a slight superheating, but this is prevented from taking place by the fact that the steam always carries with it into the cylinder a certain amount of water in the form of spray, with which it remains in contact. *The exact amount of this water is of no importance*; and since, for the most part, it is diffused through the steam in fine drops, and therefore readily participates in the changes of temperature which the steam undergoes during expansion, no important error will be introduced if, at each moment under consideration, we assume that the temperature of the whole mass of vapour in the cylinder is the same.

"Further, to avoid too great complexity in the formulæ, we will first determine the whole work done by the steam-pressure, without examining how much of in which W' = the whole work done during the cyclical process by the steam-pressure, or in other words by the heat, in a case where the expansion continues until the steam has cooled by expanding from the temperature of the boiler to that of the pressure against which the engine exhausts. Rankine's formula for this[1] is—

$$U = \int_{\phi_A}^{\phi_B} (T_1 - T_2)\, d\, \phi.$$

Their statements agree with the simpler one by Mr. MacFarlane Gray, whose $\theta\,\phi$ diagram shows the same geometrically.

In this diagram, Fig. 1, the vertical ordinates are absolute temperature, and the area is energy in heat-units. The horizontal dimension is the quantity which in Rankine's formula is denoted by ϕ, and in Clausius's works by S, and named by him "Entropy." In Fig. 1 the feed-temperature B (and that of the back-pressure) is 673° absolute, or 212° Fahrenheit. The heat represented by the areas K + M is that which is necessary to raise the temperature of the feed-water to the temperature of the boiler-steam A (829° absolute or 368° Fahrenheit). The heat represented by the areas L + N is the latent heat added at the higher temperature to effect evaporation. All the formulas quoted

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this is actually useful work, and how much is expended on the engine itself in overcoming friction and in actuating the pumps required, besides the one shown in the figure, for the proper working of the machine. This latter part of the work may be subsequently determined and deducted from the whole, in the manner shown later on. It may further be remarked, with regard to the friction between the piston and cylinder, that the work expended upon this is not to be regarded as wholly lost. For since heat is generated by this friction, the inside of the cylinder is thereby kept hotter than it otherwise would be, and the power of the steam increased accordingly.

"Lastly, since it is desirable to understand the working of the most perfect machine possible, before inquiring into the influence of the various imperfections which occur in practice, we will, in this preliminary investigation, make two further assumptions, which may afterwards be withdrawn. The first is that the inlet pipe from the boiler to the cylinder, and the outlet pipe to the condenser or to the atmosphere, are so large, or else the speed of the engine is so slow, that the pressure within the end of the cylinder connected with the boiler is always equal to the pressure in the boiler itself; and similarly that the pressure within the other end is always equal to that in the condenser, or to the atmospheric pressure, as the case may be. The second is that there are no clearance or waste spaces to affect the result."—"The Mechanical Theory of Heat." By R. Clausius. Translated by Walter R. Browne, M.A., p. 238.

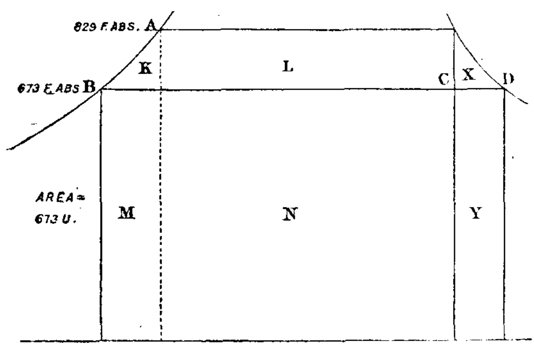
[1] "A Manual of the Steam-Engine and other Prime Movers." Article 266, p. 344.

are different ways of showing that the whole work done during the cyclical process by the most perfect steam-engine, under the ideal conditions stated by Clausius, is that represented by the K + L, and the efficiency of such an engine is $\frac{K + L}{K + L + M + N}$.

The work represented by the area K + L will accordingly be taken as the standard, and the practical results obtained with the trial engine will be compared with it. Mr. Gray's formula is a geometrical deduction from the diagram, on the assumption that the line A B is a straight line, an assumption which does not entail any appreciable error between the limits of temperature usual in steam-engines.

In planning the trials, the first point to settle was the best ratio of expansion at which to make tests at each steam-pressure. The Author decided not to be guided by existing practice on this point, but to fix the best theoretical ratio of expansion for each pressure, and to make a series of trials at about that ratio and another at a slightly lower one; for it is evident that the best

Fig. 1.



ratio in practice must be lower than any fixed theoretically without reference to the increasing cylinder-condensation with high grades of expansion. The trials being in this instance non-condensing ones, the back-pressure was decided by the atmospheric pressure on the day of each trial.

The boiler was constructed to carry a pressure of 160 lbs. per square inch above the atmosphere, which limited the working-pressure; and the trials have been made between 40 lbs. and 170 lbs. absolute.

With reference to the best number of expansions, the Author will refer to a diagram which he has used for some years when settling the best ratio of expansion in any particular case. Fig. 2 shows approximately the work theoretically obtainable from 1 cubic inch of steam at 50 lbs. absolute pressure, used in a cylinder of 1 inch cross sectional area and of various lengths. It will be seen immediately why this unit is used instead of a unit of weight.

The heights of the vertical ordinates A G, B H, C I, show the total work theoretically due from the steam, in a cylinder 1 inch, 2 inches, and 3 inches long, the horizontal line X Y dividing the useful work from that wasted in displacing the atmosphere. The ordinates are measured from a diagonal base-line, because the work wasted in displacing the atmosphere varies directly as the length of the cylinder. The loss from this cause is therefore constantly increasing, the rate being the same for each addition to the length of the cylinder.

The gain due to increased expansion for each such increase is however a steadily decreasing one, and before the one balances the other it is necessary to stop. The ordinates D G, E H, F I, represent the useful work theoretically obtainable from steam of 50 lbs. absolute pressure used non-expansively, expanded twice, and expanded three times, before being exhausted into the atmosphere. The highest point in the curve gives the theoretically best number of expansions. It will be seen that, although the gain due to doubling the original volume of the steam is considerable, there is little subsequent gain, and that after its volume has been trebled, there is a loss for each subsequent addition to the length of the cylinder.

Plate 3, Fig. 2, shows similarly the work theoretically obtainable from 1 cubic inch of steam at various pressures, and with various ratios of expansions. The curves are for pressures of 50, 70, 90, 110, 130, 150, 170, and 190 lbs. absolute. The line crossing the diagram cuts the pressure-curves at the points which indicate, in the Author's opinion, the extreme ratio of expansion likely to show any gain in practice. These points give a ratio of expansion equal in each case to the absolute pressure in lbs. per square inch divided by 25, and it was accordingly decided to make a series of compound trials with ratios of expansion $= \dfrac{\text{absolute steam-pressure}}{25}$, and another series with ratios of expansion $= \dfrac{\text{absolute pressure} - 10}{25}$.

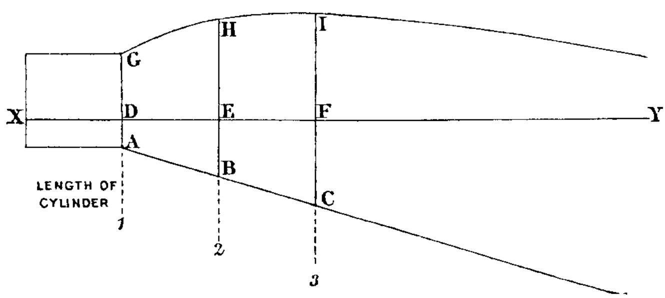
A more exact expression for the theoretically best ratio of expansion would be

$$\left(\frac{P}{B + F}\right)^{\frac{m}{m+1}},$$

where P = absolute pressure during admission, B = back-pressure, F = the pressure necessary to overcome the friction of the engine, and where the relations of p and v are expressed by the equation $p^m v^{m+1}$ = constant.

It should be stated that the work represented by the vertical ordinates in Fig. 2 is that due from steam expanded adiabatically, and that it is calculated on the assumption that $p^6 v^7$ = constant. This is, of course, not true for all pressures, or for all ratios of expansion; but it is quite near enough to fix approximately the best ratios for the trials.

FIG. 2.

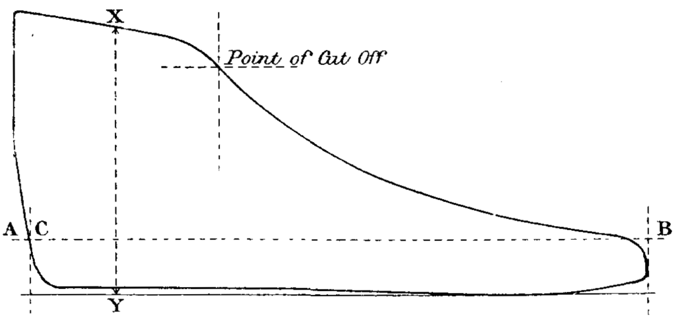


The results of these trials will be compared, later in this Paper, with the more correct curve shown by Plate 3, Fig. 1.

The Author's previous experience had led him to believe that, even with comparatively low pressures in non-condensing engines, there was marked economy from carrying on the expansion in two cylinders. It was, however, more doubtful whether any great gain could be expected from the use of three cylinders with pressures below 200 lbs. absolute; and it was very questionable if, with the boiler-pressure in this case available, any gain at all could be anticipated. The compound trials have, therefore, been made in the greatest numbers, and have been arranged to overlap the simple trials at one end of the scale of pressure, and the triple at the other.

The curves in Plate 3, Fig. 2, apply equally to simple, compound, and triple trials, for they ignore all questions of loss in passages leading from one cylinder to the other. The nominal ratio of expansions in each trial was fixed by dividing the volume of steam, at the terminal pressure, exhausted from the low-pressure cylinder at each stroke, by the capacity of the high-pressure cylinder at the point of cut-off, questions of clearance being neglected. The volume exhausted is not, of course, the whole volume of the low-pressure cylinder, but that proportion of it which is represented by C B in Fig. 3; C B being the length measured along the line A B, from the point B, at which that line cuts the expansion curve produced to the end of the stroke, to the point C, where it cuts the compression curve. This proportion was assumed to be approximately 0·95 of the stroke. As a matter of fact it was rather more than this, and the true ratio of expansion will in almost every case be found to be greater than the nominal ratio.

FIG. 3.



The questions which the Author has had frequently to decide in estimating the probable power and economy of an engine under given conditions are the following:—

1. What proportion of the theoretical diagram due to the steam present at cut-off is likely to be obtained in practice?

2. What addition must be made to the weight of steam shown by the diagram to be admitted, in order to cover that proportion of the steam which is also admitted, it is true, but only to be condensed immediately, owing to the abstraction of heat by water present, or by the comparatively cool cylinder-walls?

In a compound engine a larger percentage of the feed-water is accounted for by the diagram than in a simple engine; but owing to the greater loss due to passages and to other causes peculiar to the compound engine, the diagrams from such an engine must always, when considered with reference to the steam present at cut-off, show less work than those from a simple one.

The absence of sufficient data has rendered it very difficult to answer the above questions with any approach to accuracy, and therefore, in tabulating the results of the present series of trials, the Author has given these figures as clearly as possible. They must not, however, be taken as accurate, or even approximate, for other types of engine, as the surfaces of the cylinder and passages, and the amount of water retained in the cylinder vary so much in different types, as to make it impossible to lay down any general rules. It seems probable that in most engines the changes in the amount of initial condensation will follow the variations in speed and range of temperature, as in this case, but that the exact amount under certain conditions can only be experimentally determined for each type.

In one important particular the trial engine has a decided advantage over most others; the action of the steam on the underside of the high-pressure piston is equivalent really to a third stage of expansion; in other words, the compound engine is in a better position with respect to division of temperature than an ordinary compound engine; while in the triple engine the expansion may be said to be carried on in five stages, although these stages are not all equal.

(B) The Method of making the Trials.

The feed-water was measured in the following way:—

A tank of sufficient capacity to hold all the water for a trial was mounted on one of Avery's 3-ton weighing-machines, and filled by a flexible pipe, which was removed before the trial began. The weighing-machine has been carefully checked by standard weights in the course of the trials. The weight of the tank and water was noted at the commencement of each trial, the water being shut off from the feed-pump, with which the tank was connected by a flexible pipe. The desired load was then put on the engine, which drove a dynamo-machine, at other times employed to light Messrs. Willans and Robinson's works. The load being adjusted by regulating the resistance in the main circuit, the engine was allowed to run until the water in the gauge-glass reached a mark on a scale about half way down the glass. The time was then noted, the speed counter which registered the total revolutions during the run was put in gear, and the pipe connecting the tank and feed-pump opened.

A few minutes before the conclusion of the trial the water was worked up about $\frac{1}{2}$ inch above the mark at which it started, and the pump was shut off from the tank. The water was then allowed to fall in the glass until it reached the mark at which it started, when the time was noted and taken at the conclusion of the trial. The weight of the tank and of the water was then recorded, and the counter was thrown out of gear.

Thus the conditions at the commencement and at the conclusion of the trial were identical, the only source of error being that due to the observation of the height of the water in the boiler. Supposing a mistake of $\frac{1}{16}$ inch to be made at the commencement and the end of a trial, which is practically impossible, the total error would amount to 20 lbs. of water over the whole run, and a possible error of about 1 per cent. would be the result; it is extremely improbable, however, that the average error is more than 0·25 per cent.

The feed-pump was of the Worthington direct-acting type, and was supplied with steam from another boiler. With this pump it was possible to keep the feed-water almost exactly to a mark on the glass; in fact, it need never have been touched during a trial if it had not been that the pressure in the boiler driving it varied from time to time.

The power of the boiler was such that it was possible to keep the pressure nearly constant; it rarely varied 2 lbs. in the course of a trial. The indicator diagrams, of which three sets were taken each hour, are therefore practically identical. They were taken in all cases by the Crosby indicator, and it is not too much to say that without that indicator such trials would have been impossible. It would be difficult to speak too highly of the working of these beautiful instruments, which produce perfectly clear and measurable diagrams at the speed at which the majority of the tests have been made.

The springs have been frequently checked hot by a standard pressure-gauge, and out of a large number only one spring showed any appreciable error.

Other Crosby indicators were used by engineers, who from time to time made check trials, and the results in all cases agreed with those obtained with indicators employed by the Author. All diagrams have been measured at least twice by planimeters. The Author's planimeter has been checked by running round a square scored on a sheet of copper by Captain Sankey, R.E. This square was scored by the machine used for the sheet lines of the Ordnance maps at Southampton, and was proved by micrometer comparison with the standard ordnance foot to be practically correct.

As to personal error in measuring the diagrams, it is found possible, starting with the planimeter wheel at zero, to go over a dozen diagrams, and then, having taken the reading for their collective areas, to work round the same diagrams backwards, without appreciable error, the wheel returning to zero. The Author believes that in most trials the measurements are within 0·5 per cent. of the truth. Mr. MacFarlane Gray has checked a large portion of the diagrams, and his measurements agree within very narrow limits with those taken by the Author, who would here acknowledge the very great help which Mr. MacFarlane Gray has given him in the course of the present trials, both by advice and by personal assistance. The Author also desires to say that much of the accuracy with which the figures for these trials have been ascertained is due to the great care and pains bestowed on them by his assistant, Mr. P. A. Low, Assoc. M. Inst. C.E.

(B) Description of the Main Series of Trials, Simple, Compound, and Triple, all at 400 Revolutions per Minute.

The general dimensions of the trial engine are:—

	Square inches.
Net area of h. hp. piston	34·500
,, ,, ,, underside . .	31·416
,, of hp. piston	71·472
,, ,, ,, underside . .	65·973
,, of lp. piston	141·340
	Inches.
Length of stroke	6
	Cubic inches.
Capacity of trunk clearance h. hp.	11·1
,, ,, ,, hp.	15·0
,, ,, ,, lp.	26·0
Capacity of cylinder clearance h. hp. . .	14·8
,, ,, ,, hp. . . .	30·0
,, ,, ,, lp. . . .	33·6

The nomenclature adopted shows whether the engine was worked simple, compound, or triple, and also the conditions aimed at as to pressure and ratio of expansion, C $\frac{80}{3\cdot2}$ signifying that the engine was worked compound, with an intended mean absolute pressure in the cylinder during admission of 80 lbs., and an intended ratio of expansion of 3·2. In the same way S $\frac{110}{4 \cdot 4}$ indicates a simple trial with 110 lbs. admission pressure and 4·4 expansions. T $\frac{170}{6 \cdot 4}$ signifies a triple trial 170 lbs. pressure and 6·4 expansions. The 14-inch cylinder is always called the lp. cylinder, the 10-inch the hp. cylinder, and the 7-inch the h. hp. cylinder.

The figures for the simple trials are given in Table I, and the results as to indicated HP., water per indicated HP. per hour, water accounted for at cut-off per indicated HP. per hour, and percentage of water missing at cut-off, are shown graphically by Plate 3, Fig. 3.

In the same way the results of the series of compound trials are given in Table II, and are shown by Plate 3, Figs. 4 and 5.

The results of the triple trials are given in Table III, and are shown by Plate 3, Fig. 6.

The mean pressure during admission was taken as the starting-point, as the angle of the steam line during admission varied, owing to the same steam-pipe being used throughout, and the throttling being consequently greater with the larger cylinder.

The mean admission temperature has been taken as the higher temperature in calculating the efficiency of the engine. If the boiler temperature had been taken, the results would have been rather more unfavourable to the simple engine, which now has the benefit of any superheating or drying of the steam due to wire-drawing in the pipe. Judging, however, from the compound trial, C $\frac{60}{4}$ Throttled, Table VII, in which the steam in the boiler was 130 lbs. above the atmosphere, and that in the cylinder 60 lbs. absolute, any advantage to be derived from such drying or superheating is very slight. The figures may, therefore, be taken as substantially the same as they would have been with a larger steam-pipe, and as being fairly comparable with the compound figures. The efficiency can readily be calculated from the boiler temperature, as the pressure in each case is given.

The revolutions were in all cases as nearly as possible 400 per minute, being below the speed at which the engines ordinarily run, in order that the diagrams might be more accurately measured.

The first trials were the compound ones. The series, in which $\frac{P}{25}$ = ratio of expansion, were made with pressures at 10-lb. intervals, commencing with $C\,\frac{80}{3\cdot 2}$, and concluding with $C\,\frac{150}{6}$.

The series, in which $\frac{P - 10}{25}$ = ratio of expansion, commenced with $C\,\frac{90}{3\cdot 2}$, and went up to $C\,\frac{160}{6}$.

The simple trials were made with $\frac{P}{25}$ = ratio of expansion between 40 lbs. absolute pressure and 110 lbs., and the triple trials between pressures of 140 lbs. absolute and 170 lbs., beyond which pressure the boiler used was not suitable.

It is unnecessary to refer at length to the Tables, but a few points need explanation.

In Tables I, II, and III, *line* 4, the figures "Mean absolute pressure during admission," have been obtained by measuring the mean height of all diagrams taken during the trial, at a point half way between the commencement of the stroke and the point of cut-off, the measurements being taken from the steam line to the atmospheric line (line X Y, Fig. 3), and this measurement being added to the barometric pressure.

Line 5.—"Corrected ratio of expansion" has been found by dividing the volume of steam at terminal pressure discharged from the LP. cylinder by such a volume of steam at mean admission pressure, as would agree with the steam shown by the indicator to be present (as steam) at the point of cut-off. This method of calculating the expansions takes account of any steam required to fill clearances, but not of steam condensed during admission. The clearance steam is sometimes a + and sometimes a − quantity, according as the cushion pressure is higher or lower than the cut-off pressure.

Line 14.—"Total mean pressure referred to cylinder without clearance" has been found by dividing the mean pressure referred to LP. piston-area (*line* 13) by the percentage of the LP. cylinder, which was used to calculate the ratio of expansion, and which is shown by the length CB. in Fig. 3. This percentage was found to be in almost all cases 0·97 or 0·975 of the whole cylinder.

Line 15.—"Theoretical mean pressure" is calculated from the corrected ratio of expansions, on the assumption that steam expands adiabatically according to the formula $p^6 v^7$ = constant, and it is therefore only approximate; though the error is but a slight one in any particular case.

Line 16.—"Percentage of theoretical mean pressure actually obtained" has been found by dividing the figures in *line* 14 by those in *line* 15.

Line 21.—In calculating the "Feed-water per indicated horse-power-hour accounted for by indicator at cut-off," every diagram has been measured with the greatest care, and the calculation sheets appended will explain the method adopted.

In judging of the probable accuracy of the results as a whole, it must not be overlooked that the measurement of the steam, shown by the diagrams, constitutes a very useful check on the feed-water measurements, the diagram measurements having been taken by one person and calculated by another, neither of whom was present at the trial.

There are two clearance spaces in each cylinder to take into account, one an annular passage in the trunk between the cut-off ports and the valve ports, and the other the true cylinder clearance. The former must be treated as filled from the back-pressure to the pressure at which the diagram is measured each revolution, and the latter from the cushion-pressure to the pressure at which the diagram is measured; in the compound and triple engines this latter is frequently a minus quantity, even in diagrams measured at the point of cut-off.

Line 22.—The figures in this line, "Water theoretically required per indicated horse-power per hour," are calculated for the mean admission pressure from the same formula as the ordinates in Plate 3, Fig. 1.

Line 34.—The "heat-units missing per hour at the point of cut-off" are calculated on the assumption that all feed-water, not shown by the indicator to be present at cut-off, is present as water in the cylinder at that time and at the temperature due to the pressure indicated. This is not true as regards leakage, but it has been found practically impossible to separate leakage from the other causes which go to make up "the missing quantity."

Line 35.—"The heat-units missing per stroke" have been inserted in order to facilitate the comparison between the missing heat and the changes in temperature and surface of the cylinder walls which are usually supposed to account for it.

The same numbers have been given where possible to the corresponding lines in all the Tables, so that the results of trials under various conditions can be readily compared.

Throughout the whole series no hitch of any kind occurred, and it has been necessary in only very few cases to repeat any trial; where this has been done the mean of the two trials is given in the Tables.

In the case of the compound trials, in Table II, two were made twice $C\frac{100}{4}$ and $C\frac{130}{5 \cdot 2}$; all others are the results of single trials. $C\frac{150}{6}$ is a trial made by Professor Kennedy, whose measurements for the steam present were taken rather later in the stroke than the Author's, which accounts for the missing quantity being rather larger than shown by the figures in the trials on each side of it.

The success of the precautions adopted to ensure accuracy will be evident on an examination of the results as shown by the diagrams, the greatest divergence from a fair line in the curves for water per indicated HP. and water accounted for by the indicator per indicated HP. being about $1\frac{1}{2}$ per cent.

In the simple trials the results were not quite so uniform, much more water was present in the cylinder, and the larger quantity of water evidently had a bad effect in rendering many of the indicator diagrams unsteady, and increasing the initial condensation.

The triple trials were only made at three steam-pressures; but they are interesting in connection with the question of initial condensation. In the simple trials in Table I the missing quantity of steam at cut-off varied from 11·7 per cent. with 40 lbs. pressure to about 30 per cent. with 100 lbs. and 110 lbs. pressure. At 80 lbs. pressure and 3·2 expansions the missing quantity was 23·7 per cent. in the case of the simple engine; on the other hand it was reduced under the same conditions to 5·2 per cent. in the compound engine.

The missing quantity varied in the compound trials in Table II from 5·2 per cent. in the case of $C\frac{80}{3 \cdot 2}$ to 17 per cent. in the case of $C\frac{160}{6}$. On making the same trial triple, the condensation was reduced from 17 per cent. to under 5 per cent.

There can be no question that the compound engine is very superior in economy to the simple one for pressures as low even as 80 lbs.; and in practice, where questions of wear and leakage begin to tell, the difference must be still more marked.

(C) The Difference between the Work actually obtained and that theoretically due from the Steam.

The Author will now compare the results obtained in the main groups of trials with those previously shown to be theoretically possible.

In the first place, it may be well to see how closely the theoretically best ratio of expansion agrees with that which is found to give the best results in practice. In order to carry out this comparison, trials have been made with 130 lbs. mean admission-pressure and (nominal) ratios of expansion from 4 to 10·8. Plate 3, Fig. 7, shows the results of these trials graphically, the upper line as before giving the HP., the second the feed-water per HP.-hour, and the third the steam per HP.-hour accounted for by the high-pressure indicator, the lowest line giving the percentage of total feed-water missing at the point of cut-off, as in the diagrams for the other groups of trials. Little difference is shown in economy between the trials named $C\,\frac{130}{4\cdot 8}$, $C\,\frac{130}{5\cdot 2}$, $C\,\frac{130}{5\cdot 6}$, and $C\,\frac{130}{6}$, which is what would be anticipated from theoretical considerations.

Table IV gives the results of the trials in the form described for the main groups. It will be seen that with the same steam-pressure the percentage of steam initially condensed in the high-pressure cylinder varies from 9 to 25 per cent., owing to the increased range of temperature, and in spite of the diminished surface with early cut-off.

Plate 3, Fig. 8, shows one of the theoretical curves in Plate 3, Fig. 2, viz., that for 130 lbs. steam separate from the rest for the purpose of comparing it with the actual results. These results are indicated by the black dots, through which a fair line has been drawn. They are corrected for the true ratios of expansion as given in Table IV. They are in all cases far below the work theoretically possible, the distance between the upper and the lower lines indicating the difference.

This loss may be broadly divided into two parts :—

1st. The difference between the weight of feed-water used and the weight of steam present at cut-off, constituting what has been called "the missing quantity."

2nd. The difference between the work (represented by the area of a theoretical diagram) due from the steam present at the point of cut-off in the high-pressure cylinder, and the work obtained in practice.

This broad distinction is not maintained throughout the expansion, because the steam initially condensed is partially or wholly re-evaporated before the end of the stroke, and thus the area of the diagram is in practice increased. Plate 3, Fig. 8, shows this clearly. The heights of the upper and of the lower lines give for each ratio of expansion the comparative values of work possible and work done; while the difference between the heights of the two upper lines shows the proportion of steam which is inoperative at the moment when the cut-off takes place, its latent heat being at that moment stored in the cylinder-walls, or in the water present in the cylinder.

The approach of the two lower lines to one another at the higher expansion indicates the re-evaporation of a larger quantity of water. Thus the condensation at the commencement is not all loss; and this is especially the case in a compound engine, where the re-evaporation, even if not completed during the stroke, is completed in time to be of use in the cylinder below it.

The figures in *lines* 28 and 29 afford a comparison between the missing quantities in the high- and in the low-pressure cylinders with various ratios of expansion. When the ratio of expansion is low the missing quantity in the high-pressure cylinder is smaller than in the low-pressure one. As it is increased the quantities become more nearly equal (in spite of the fact that a larger and larger proportion of the work is done in the high-pressure cylinder, and consequently that the liquefaction due to work in the high-pressure cylinder is greater), until with six expansions they are equal. Beyond that point the steam present at cut-off in the low-pressure cylinder is a far larger proportion of the feed-water than that present in the high-pressure (Plate 4, Fig. 1).

On referring to Table IV, "Percentage of theoretical mean pressure actually obtained," it will be seen that by the comparatively greater re-evaporation of the original missing quantity, the area of the actual diagram has been increased from 90·8 per cent. at 4·46 expansions to 98·1 per cent. at 8·38 expansions.

It may be well to consider the principal causes for these two kinds of loss, which may be named initial losses, and loss in the form of the diagram. Initial losses may be due to leakage, radiation, or cylinder condensation, and they may also be apparently increased, owing to water carried over from the boiler with the steam.

Steam is probably never produced perfectly dry from a boiler, so that it is very important to know how far the steam used in the course of these trials differed from that of which the latent heat was determined by Regnault, and at Mr. Gray's suggestion calorimetric tests were made to determine this point. These tests will be described later, but it may be well to state here that they showed that the water in the steam from the boiler could certainly not be taken as more than 1 per cent.

The second class of losses, namely, those in the form of the actual as compared with the theoretical diagram, includes losses from back-pressure (in excess of that of the atmosphere which, being a necessary loss in non-condensing engines, has been already allowed for), losses from friction in the steam-passages between the high-pressure and the low-pressure cylinders, losses due to drop in the receiver, if such exist, and losses due to condensation of steam from radiation, or to greater initial condensation during admission to the later cylinders than occurred in the case of the first cylinder. On the other hand, as has been already explained, these losses are reduced by the re-evaporation of the missing quantities, or by the partial recovery in the later cylinders, of the steam leaking past the first piston. Some of these losses, such as those between the two cylinders, are peculiar to the compound engine, and they must be more than neutralized by gains of another kind before the compound engine can show an economical superiority over the simple one.

On the other hand, the compound engine has the following distinct advantages:—Losses due to clearance can be reduced, losses from leakage are minimised, and initial condensation is reduced, owing to the narrower range of temperature in each cylinder.

The generally accepted reason for the proved superiority of the compound engine is the reduced range of temperature; and the confirmation of that view which these trials afford is most satisfactory.

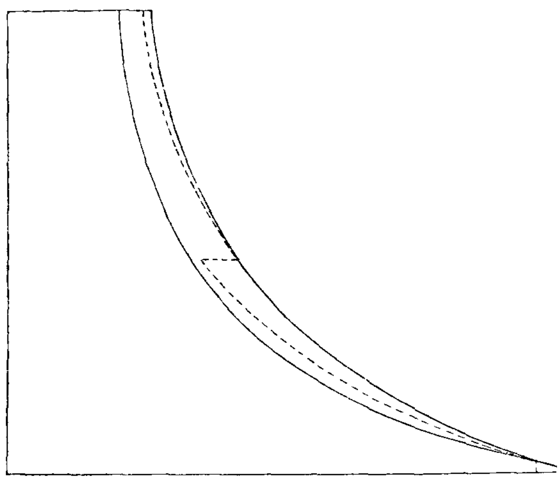
The largely increased surfaces of a compound engine, as compared with those of a simple engine, make it at first difficult to understand the great gain from reduced range of temperature. The matter is often regarded as if the increased surfaces must be set against the reduced range; but this is not the case, or only to a very limited extent. The steam initially condensed in the high-pressure cylinder is re-evaporated during expansion, or

L 2

during the exhaust stroke, and if there is (as, of course, there must be) condensation during admission to the low-pressure cylinder, it must be considered as a repetition of the earlier action at a later stage in the expansion, and not an addition to it. Fig. 4 shows this; the large difference between the theoretical diagram and the full line representing the loss from initial condensation in the simple engine, while the smaller losses shown by the dotted lines denote the losses from the same cause in the compound engine.

Taking the lowest pressure at which the engine was worked compound $C\frac{80}{3\cdot 2}$, it will be seen from Table II that the "missing

FIG. 4.



quantity" was $5\cdot 2$ per cent. In the corresponding simple trial $S\frac{80}{3\cdot 2}$, Table I, the quantity was $23\cdot 7$ per cent., showing that the proportion of feed-water present as steam in the cylinder at the point of cut-off was greater in the compound than in the simple by 18 per cent. There is not, however, a gain to this extent by using the compound engine, for the water per HP.-hour is in the one case $29\cdot 67$, and in the other $26\cdot 17$, or a clear gain for the compound of $11\cdot 8$ per cent., the loss from increased condensation being to some extent recovered in the simple engine by increased re-evaporation, and the gain in the compound being partially neutralized by the losses peculiar to the compound engine. Such figures as these, obtained from trials under conditions peculiarly favourable to the simple engine, lead the Author to think that the number of simple engines made in the future, where economy is an object, will be very small. The only question to consider now is that of the point in pressure and expansion at which it will be necessary to add a third cylinder.

The missing quantity in the compound engine was 17 per cent. with 160 lbs. pressure, and six expansions.

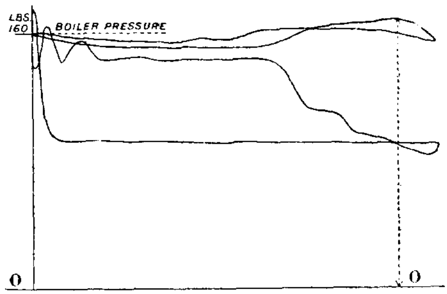
Under the same conditions it was in the triple engine, Table III, only about 5 per cent., and although up to that point the gain was almost neutralized by the losses from the new set of passages, it is obviously only necessary to go a little higher in pressure to get considerably greater economy than would be possible with the compound engine. An initial condensation of 5 per cent. is believed to be an unusually low value even for a triple expansive engine, but it is confirmed by almost all the trials; the only important exception being the one made by Professor Unwin, where the missing quantity was 10·3 per cent. A comparison of the figures for this trial T $\frac{170}{6\cdot 4}$, December 17th, 1887, with those for the other trials made under the same conditions on January 12th and 16th, 1888, shows, however, that the larger missing quantity in the earlier trial was due to leakage.

The Author had hardly realized, when these trials were made, the importance of large steam-ports in dealing with very high-pressure steam. This question is often treated as one of area of passage as compared with cylinder-area at certain speeds of piston; but with wide variations in pressure it is evidently most important to consider the density of the fluid being dealt with. The loss of pressure due to the passages is far greater with high pressures than with lower ones, and the diagrams from the 14-inch cylinder, with the various steam-pressures employed in it, show most clearly the effect of the greater density of the steam.

The loss of pressure from steam-pipe and steam-ports in the case of the simple-engine trials is almost exactly proportional to the density of the boiler-steam.

The h. high-pressure diagram (Fig. 5) gives some idea of the action of high-pressure steam; the diagram above it shows how the steam, piled up in the steam-chest after cut-off owing to its velocity in the steam-pipe, reaches a pressure exceeding the boiler-pressure by 10 lbs. It is evident that the area of a steam-pipe or passage ought to be regulated rather by the weight of steam

FIG. 5.



passing through it, than by the velocity with which any particular measure of it passes.

(D) A DESCRIPTION OF CERTAIN TRIALS MADE TO ACCOUNT FOR THE MISSING QUANTITY.

A number of trials have been made to determine approximately the effect on initial condensation of increasing or diminishing the surface exposed, the range of temperature, and the speed of rotation; but it may be well to consider first one or two causes which account in part for the missing quantity, and which cannot readily be distinguished from initial condensation when the diagrams are measured. Leakage is one of these, radiation and consequent loss in the steam-pipe and steam-chest is another, and a third is priming.

The Author attempted to measure the leakage past the piston and valves by blocking the engine in various positions, and noting the fall of water in the gauge-glass on the boiler over a given time. It was found, however, that the steam leaving the boiler under these circumstances was too small a quantity to be ascertained accurately. The largest quantity measured in this way (and in it was necessarily included the loss from radiation in the steam-pipe and steam-chest) was 15 lbs. per hour. It was certain that the leakage measured in this way would not agree with the leakage when the piston was in motion; and, taking into consideration the fact that the 15 lbs. per hour represented little more than one-tenth of the total missing quantity under approximately similar conditions, when the engine was at work, the Author thought it would not be safe to attempt to sub-divide the missing quantity at cut-off. A separator was originally placed on the steam-pipe for the purpose of abstracting any water condensed in the pipe, or carried over as priming from the boiler. The total quantity caught was, however, only 3 lbs. in nine hours, probably not more than could be accounted for by the additional surface of the separator, and the pipe was therefore led direct to the engine in all but the earliest trials.

The question of moisture in the steam, as compared with that present in Regnault's experiments, was the subject of a series of calorimetric tests. In making these a steam-pipe from the boiler was led into the feed-tank, which was mounted on a weighing-machine, and into the tank, filled with cold water, a certain weight of steam was blown from the boiler. For the purpose of ascertaining the weight of water and tank, the ordinary scale was used, but when weighing the comparatively small amount of steam blown into the tank, the weighing-machine was merely used as a balance, the steam taking the place of two $\frac{1}{2}$ cwts., which have since been compared with a standard.

The method adopted was as follows:—

The empty tank of iron was first weighed, and on each trial the weight of the tank full, or partly full of water, was noted, and the temperature taken. In consideration of the difference in the specific heats of iron and of water, the net weight of the tank was taken as equal to a certain weight of water, and this weight, together with that of the water in the tank, was considered to be raised from the lower to the higher temperature by the incoming steam.

The weight of water, and of tank taken as water, having been ascertained, two $\frac{1}{2}$ cwts. were placed alongside the tank on the weighing platform, and the whole was then balanced. The balance was extremely delicate in proportion to the weight of water and tank, about 3 tons; and it was found that by plunging the fingers of one hand up to the second joint into the water in the tank, the beam could be made to rise, and on removing them to fall again. When the balance was adjusted, the two $\frac{1}{2}$ cwts. were removed; the steam from the boiler was then caused to blow into the open air, and when it was evident that it was coming from the pipe perfectly dry the time was noted, and the steam-cock on the boiler was regulated until it allowed the steam to escape at a pre-arranged rate, the rate being ascertained to be roughly correct by noting the fall of water in the gauge-glass on the boiler.

The rate being adjusted, the cock was turned to allow the steam to enter the tank, to which it was admitted by a pipe leading nearly to the bottom, the lower end of the pipe being perforated. An electric bell was kept ringing as long as the scale-beam was at rest, but when it rose, thus indicating that the weight of steam blown in equalled the weight (1 cwt.) which was removed previous to the test, the bell stopped, and the cock having been turned off as quickly as possible, the mixture was well stirred and the temperature taken. The scale was then again carefully balanced by taking away or adding, if necessary, a small quantity of water in a can, which was afterwards weighed and deducted from or added to the 112 lbs. intended to be blown into the tank.

The weight of water in all cases except one was over 2 tons, and the weight of steam blown in being considerable, it was hoped that the results would be more consistent than those of other calorimetric tests previously made in the same way, but on a smaller scale, and this fortunately proved to be the case.

Table X shows the general results of the tests. The steam from the boiler usually contained from 99 per cent. to 100 per cent. of dry steam, the calculations being based on the assumption that the steam used by Regnault was free from moisture.

In the Table—

g = gross weight of tank and water.
p = absolute pressure of steam.
t_1 = the temperature (Centigrade) of the cold water.
t_2 = the temperature (Centigrade) of the water after the steam has been blown in.
t_3 = the temperature (Centigrade) of the steam.
w = effective cold-water weight, including weight of tank taken as water.
u = weight of steam blown in.
x' = percentage of steam of Regnault's quality in the mixture of steam and water from the boiler, not corrected for variation of specific heat of water.
x = same as x', but with Bosscha's correction for the variation of specific heat of water.

$$x' = \frac{(t_2 - t_1)\,w - (t_3 - t_2)\,u}{(606\cdot5 - 0\cdot695\,t_3)\,u},$$

$$x = \frac{(t_2 - t_1)\,w - (t_3 - t_2)\,u - 0\cdot00011\left[(t_2^2 - t_1^2)\,w - (t_3^2 - t_2^2)\,u\right]}{(606\cdot5 - 0\cdot695\,t_3)\,u}.$$

The thermometers used for these tests were specially made for Mr. Gray. In some of them 1° Centigrade read 1 inch on the scale, and therefore it was possible to obtain the readings very accurately. They were compared at the time of the test with standard thermometers, kindly brought from Southampton by Captain Sankey, R.E., and have also been compared with the thermometer used for testing the standards of length at the Ordnance Office. The readings were corrected for the cooling ascertained to take place from radiation, but this correction was only a small one, owing to the large quantity of water employed, and hardly affects the general result.

It will be seen that in this matter of priming, as in the cases of leakage and of radiation, very small quantities are dealt with, and the Author has therefore thought it better to treat the whole of the steam missing, according to the diagrams at cut-off, as one quantity, rather than to attempt to subdivide it throughout the trials. The particulars given will, however, be some indication of the values of priming, leakage, and radiation in the pipes and steam-chest.

Neglecting these small disturbing elements, the general effects of surface, range of temperature, and speed of rotation on the missing quantity, may now be considered as a whole.

The most complete trials have been made with the compound engine. It will be noticed from Table II that in the main groups of compound-engine trials there are several fluctuating elements, all of which have probably some effect on the missing quantity. Between the low-pressure trials $C\frac{80}{3 \cdot 2}$ and $C\frac{90}{3 \cdot 2}$, and the high-pressure ones $C\frac{150}{6}$ and $C\frac{160}{6}$, the range of temperature to which the initial or clearance surfaces are exposed varies from $-5°$ to 35°. These surfaces amount in this case to 1·771 square foot.

The surface of the cylinder-walls, exposed up to the point of cut-off, varies in the same group of trials from 1·027 to 0·532 square foot, the range of temperature to which these surfaces are exposed being from 31° to 77°. The speed of rotation in all the trials in Table II is approximately 400 revolutions per minute.

The density of the admission steam differs considerably in the experiments, the weight per cubic foot at the lowest pressure being little more than one-half of the weight at the highest pressure.

The missing quantity in lbs. of water per hour at the point of cut-off in the HP. cylinder varies from 33·9 lbs. to 129·3 lbs., the abstraction of heat, being in the case of C $\frac{80}{3 \cdot 2}$ represented by 1·271 thermal-unit per stroke, and in the case of C $\frac{160}{6}$ by 4·657 thermal-units per stroke.

In the group of trials, Table IV, one of the variable elements in Table II is absent, viz., the alteration in the initial density of the steam. In that Table, which gives the results of trials with 130 lbs. steam-pressure and different ratios of expansion, the initial surfaces are exposed to ranges of temperature of 13° in the case of C $\frac{130}{4}$, and 45° in C $\frac{130}{8}$. The surfaces exposed during admission are subjected to ranges of 52° in the case of C $\frac{130}{4}$, and of 81° in C $\frac{130}{8}$. The admission surface varies from 0·799 square foot to 0·4 square foot.

In this group of trials the missing quantity of water per hour varies from 74·6 lbs. in the case of C $\frac{130}{4}$ to 133·8 lbs. in that of C $\frac{130}{8}$. The thermal-units abstracted per stroke in these two cases are 2·679 and 4·848 respectively.

Table VIII and Plate 3, Fig. 10, give particulars of trials at 160 lbs. pressure and various ratios of expansion.

In Table VII, which gives the results of trials at four expansions and various steam-pressures, there is no variation in the surface exposed, as the cut-off takes place at the same point in the stroke in all cases. In these trials, however, the density of the steam varies, the weight being 0·1442 lb. per cubic foot in the case of C $\frac{60}{4}$, and 0·2909 lb. per cubic foot in C $\frac{130}{4}$.

The range of temperature for the initial surface varies from 4°·5 to 13°, and that of the admission surface from 40°·5 to 52°. The total missing quantity in lbs. of water per hour in this group of trials alters very slightly, and the fluctuations are only such as may be due to variable losses from leakage and radiation, for they do not appear to follow any clearly-defined course. The results of this series of trials are shown by Plate 3, Fig. 9.

Table IX and Plate 3, Fig. 11, give particulars of trials at 5·6 expansions and various pressures.

Now in the first group, Table II, the conditions are such that all the elements supposed to influence initial condensation vary except speed of rotation. In the group Table IV there is a second important exception, viz., change in initial density; and in the third group (Table VII) an equally important exception, viz., change in surface; while in this group the changes in initial density reach their maximum.

The Author is unwilling to suggest any theory to account for the various results shown in these Tables, his object being rather to record the facts as nearly as they could be ascertained, for comparison with the results of other trials. He would however direct attention to the very small effect which changes in amount of exposed surface, or in the density of the steam, appear to have on the missing quantity reduced to heat-units.

If surface is an important factor in this matter, it is surely not unreasonable to expect a marked effect from the greater density of the steam in the high-pressure trials in Table VII, than in those made with lower steam-pressure; for the steam being denser in one case than in the other, a greater number of steam molecules should be able to impart their heat to each square inch of surface. There does not, however, appear to be any such effect from the increasing density of the steam in the trials recorded in Table VII, and the figures for the three groups of trials will show that the missing quantity in heat-units per stroke follows nearly the range of temperature to which the admission surfaces are exposed.

The Author thinks, therefore, that water is likely to prove a far more important factor than surface at such speeds as 400 revolutions per minute. Surface as a resting place for water is no doubt an important element, especially if it be rough surface, and to a certain extent, no doubt, the iron has some effect; but these and other experiments led the Author to think that where, as in some of these trials, all the condensation takes place in one-thirtieth of a second, the presence of a, comparatively speaking, constant weight of water in the cylinder is the main factor, and that such a body of water is sufficient to account for the missing heat in the groups of compound trials.

It is not to be expected that it will be entirely or exactly accounted for by the water, because leakage can hardly fail to increase with the higher pressures, and radiation also will increase a little; but, speaking generally, it may be said that a body of water raised from the exhaust temperature to the mean admission temperature, during admission, each stroke need not exceed in weight 0·06 lb., or be less than 0·04 lb. to account for the results in the Tables. Here the weight is given in each case, and for those who think that the metal of the cylinder is the principal factor in initial condensation the figures, corrected for the difference in the specific heats of iron and of water, may be useful as a rough indication of the depth to which the action may penetrate the metal.

It might be more correct to take the range of temperature from cut-off to exhaust, and the pressure of the cushion-steam would probably be a factor also, although the period of compression is such a short one, in which case the weight of water need not vary so much as by the figures given above.

It seems to the Author that the presence of water, in larger or in smaller quantities in the cylinder, may be the explanation of the difference in the initial condensation of various engines.

Low-speed engines are evidently in a worse position than quick-running engines in this respect; as in the former the water has a better chance of accumulating on the surface and in corners of steam-passages than in the latter, in which it must be more diffused amongst the steam, and consequently more readily cleared out during exhaust.

Whether initial condensation is due to water mainly or to surface, the effect of increased speed is a clear gain. At the lower speeds in all probability the effect of surface is felt as well as that of water, so that the loss in heat-units per stroke is largely increased.

Table VI gives the results for the compound trials at various speeds, and Plate 3, Figs. 12, show the results graphically. Owing to the enlargement of the diagram (due largely to re-evaporation during expansion) in the low-speed trials, the absolute quantity of water per HP.-hour accounted for by the high-pressure diagram decreases as the speed is reduced, but the water actually used per indicated HP.-hour is considerably increased.

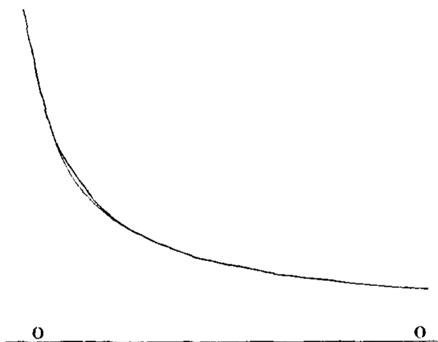
The feed-water, in the case of C $\frac{110}{4}$ and 120 revolutions per minute, is 24·73 lbs. per indicated HP.-hour, and in that of C $\frac{110}{4}$ and 400 revolutions 21·37, a clear gain at the higher speed of 14 per cent. These figures are borne out by the other compound trials made at various speeds C $\frac{90}{3 \cdot 2}$ and C $\frac{130}{4 \cdot 8}$. In these cases no change was made in the engine except in speed.

The simple-engine speed trials, Table V, show the same thing, the missing quantity being in the case of S $\frac{110}{4 \cdot 4}$ 120 *revolutions*, 44·5 per cent. of the whole feed-water.

It will be noticed that, in the compound-engine speed trials for any particular pressure and ratio of expansion, the total initial condensation is practically constant per hour at all speeds, and that if the speed of the engine be doubled the percentage of steam initially condensed is, roughly speaking, halved.

In the case of the simple-engine speed trials this does not appear

Fig. 6.



to be so, as the missing quantity per hour is always smaller at the lower speeds, although still a larger percentage of the feed-water used than at the higher one.

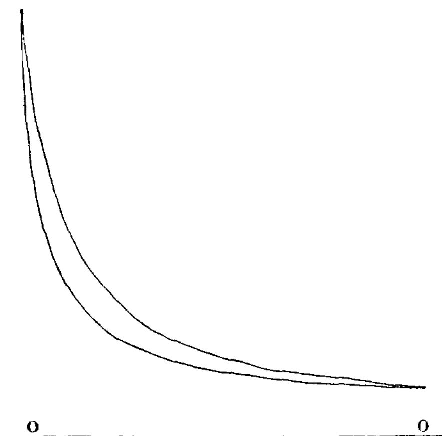
The greater absorption of heat during compression (whether by cylinder-walls or by water in the cylinder) was very noticeable in the slow-speed trials, and can be ascertained by comparing the temperature of the cushion-steam at various speeds.

The important part which may be played by water in absorbing heat in a cylinder is shown by Figs. 6, 7, and 8, the indicator diagrams in which are enlarged from those taken from the air cushion-chamber.

Fig. 6 is the normal diagram when air only is present; in this it is impossible to distinguish the lines traced by the indicator pencil on the up and down strokes. The expansion and compression are adiabatic; in other words, there is no loss of power and no appreciable transmission of heat from the compressed air to the walls of the cushion-cylinder.

In the case of Fig. 7 a considerable quantity of water was purposely injected, and as the temperatures of the air and water

FIG. 7.

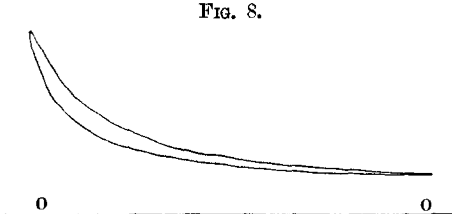


do not correspond throughout the stroke, the power given up by the air during the down stroke is less than that put into it during the up stroke.

Fig. 8 is a diagram from a cushion-cylinder in which steam was used for experiment instead of air.

From these diagrams it would appear, that with air the transmission of heat is inappreciable unless water be present; but that in the case of steam there is always a sufficient body of water, probably deposited mainly as a film on the cylinder-walls, to cause considerable loss.

FIG. 8.



It is impossible to do more in the limits of this Paper than just allude to these various points; but the Author hopes that the Tables and diagrams will enable others to compare his results with their own. When the condensing-engine trials have been completed, and the effects of larger ranges of temperature are considered, it may be possible to throw a little more light on a very complicated subject.

(E) The Economical Value of Automatic Expansion-Gear in Compound Non-Condensing Engines.

The gain likely to result from the use of automatic expansion-gear on the high-pressure cylinder of a compound engine depends largely on the amount of initial condensation. If there were no loss from this cause, the gain from the use of automatic gear would be considerable; but, owing to the increase in the percentage of steam initially condensed with high ratios of expansion, cases may easily occur in which there is practically no gain at all.

The Author thought that it might be interesting to compare the results of the series of trials made with 130-lb. steam and various ratios of expansion with those of the series made with a fixed ratio of expansion and various steam-pressures. Plate 3, Fig. 13, shows the results of these trials graphically. The base-line gives the power of the engine, and the ordinates measured from this base-line show the lbs. of water per indicated HP.-hour. The upper full curve gives the feed-water per indicated HP.-hour when the power is reduced by working at a lower pressure, and the lower full curve the feed-water in the same way when the power is reduced by varying the expansion. The upper and lower dotted curves show the weight of steam per indicated HP.-hour accounted for by the high-pressure diagrams. It will be seen that the highest power is that due to 130-lb. steam and four expansions; at this, which is assumed to be the maximum power of the engine, the economy of the two methods is equal.

As the power is reduced the method of varying the expansion has a slight advantage, until at three-quarters power the gain is about 7 per cent. From this point the gain diminishes, until below half-power it is very small. With an ordinary slide-valve, indeed, the increased friction and less regular motion would most likely convert such a slight gain as there is into a loss. On looking merely at the dotted curves for water apparently used, it would appear that there was a great gain from the use of increased expansion, and the gain would be a real one if it were not for the great increase in initial condensation with the higher ratios of expansion.

The initial condensation in this case is small compared with what is usually found in steam-engines, and the case is therefore peculiarly favourable to the automatic gear. If the initial condensation were doubled, as it would be in the engine when running at half speed, the gain would disappear, and the advantage be entirely on the side of the variable-pressure diagram, which represents the ordinary throttled engine. That diagram is, indeed, somewhat unfair to the throttled engine, as the boiler-pressure was varied; and therefore each lb. of feed-water required less heat to convert it into steam in this case for the lower powers than in the case of the variable expansive series.

A trial or two made with the full steam-pressure in the boiler, but throttled on its way to the engine, appeared to show some advantage over the trials with reduced boiler-pressure as plotted. The advantage, however, was not enough to encourage the Author to make an extended series of trials; undoubtedly, however, the results exhibited by Plate 3, Fig. 13, show the case for throttling at its worst.

These results will not surprise those who have given much attention to the subject; but it may be well to point out that there is an essential difference, in the matter of automatic expansion-gear, between simple and compound non-condensing engines. In the former the range of temperature is practically fixed by the initial pressure on the one hand, and the atmospheric pressure on the other, and an earlier cut-off does not increase this range. In the compound engine, however, every step in the direction of increased expansion is accompanied by increased range of temperature, and, as has been shown, by increased initial condensation. The equal division of temperature between the two cylinders, which is the essential feature of the compound engine, is very soon lost, the low-pressure diagram becoming smaller and smaller as the ratio of expansion becomes greater.

It should be stated that the engine with which these trials were made is not in any way a special one, but a standard engine taken out of store for the purpose.

The present trials have naturally suggested many improvements, which are likely to lead to increased economy in the future; but it was thought better to make no alterations in the engine as the trials proceeded, in order that no corrections should be necessary in comparing one trial with another.

In conclusion, it has been shown: 1. That the percentage of feed-water not present as steam at cut-off may vary with steam practically dry, from less than 5 per cent. to more than 40 per cent., according to the conditions of working.

2. That at 400 revolutions per minute, its amount in any given cylinder is determined mainly by the range of temperature in that cylinder, and that it does not appear to be affected by variations in the density of the steam, or by changes in the area of exposed surface. These facts seem to suggest that the action of the cylinder-walls is not seriously felt at high rotative speeds, the cause of the condensation being rather to be looked for in the alternate heating and cooling of a small body of water retained in the cylinder.

On the other hand, at comparatively low speeds, the missing quantity, considered as a percentage of the feed-water, increases very fast, which may possibly be due to the action of the walls becoming at such speeds a more important factor.

In the low-speed single trials this increase is not so marked as in the low-speed compound ones, which is consistent with the view just expressed; for the walls in the smaller cylinder bear a larger proportion to the contained steam than in the case of the low-pressure cylinder.

It has also been conclusively proved that, up to 400 revolutions per minute, at any rate, high speed is favourable to economy.

3. In the trial engine, working compound at 400 revolutions per minute, the ratio of expansion found to give the best results in practice did not differ much from the one which theoretical considerations suggested; but it is probable that at lower speeds the variation would be much more marked.

4. As to the percentage of work due from the steam present at cut-off, which was obtained in the trials, the Author has shown that this varied from about 85 per cent. in the triple, and 87 per cent. in the low-pressure compound trials, to considerably over 100 per cent. in the low-speed simple ones; the increase being mainly caused by the evaporation of the water initially condensed.

5. In a non-condensing engine, running at a speed of 400 revolutions per minute, it is worth while to employ two cylinders, even with pressures as low as 80 lbs. absolute; and at pressures of 160 lbs. absolute there is a slight gain, which is likely to increase

with the pressure, from the use of three cylinders. It is evident also that, at low speeds, the number of stages in which the expansion is carried on must be increased, if anything like the same efficiency is to be maintained.

6. As to the general question of thermo-dynamic efficiency, which has been defined as the ratio of the work done by each lb. of steam to the work thermo-dynamically due from it, when expanded from the mean admission to the exhaust temperature, in these trials the highest efficiency with a simple engine was obtained at the lowest pressure, being equal, at 40 lbs. pressure, to 81 per cent.; and at 110 lbs. pressure, to 68·8 per cent.

With the compound engine, the highest efficiency, 82·5 per cent., was obtained in the trials at 110 lbs. pressure, from which point the efficiency fell both with lower and higher pressures, being only 80·8 per cent. at 80 lbs., owing to increased loss in passage, and only 77 per cent. at 160 lbs., owing to increased initial condensation.

Finally, at 170 lbs. pressure, the triple engine gave an efficiency of about 78 per cent., and, as it is probably an increasing one, the best result is not likely to be obtained much below 200 lbs. pressure.

The many causes of loss in the steam-engine make it doubtful whether an efficiency of 80 per cent. can be greatly exceeded; but it is evident that, if it is to be maintained at greater total ranges of temperature than have been available for these trials, it must be by the judicious combination of high rotative speed, and a suitable number of stages of expansion.

Only one condensing-engine trial has so far been made; in this 170 lbs. absolute steam-pressure was used, and the consumption of steam was 15·1 lbs. per HP.-hour; but it is expected that better results than this will be obtained with cylinders of more suitable proportions.

Experiments with jackets will hereafter be tried in some of the condensing trials. The Author, however, does not expect any marked economy to result at the higher speeds.

Plate 4, Fig. 2, gives a comparison between the external electrical power and the power indicated in the cylinders, in the case of a Willans engine combined with a Siemens dynamo recently supplied to the Admiralty and tested at various powers. The speed was approximately the same in all the trials, but the current was varied from about 100 amperes to about 450 amperes, the electromotive force being in all cases about 80 volts.

The efficiency of the whole apparatus, that is $\frac{\text{E. HP.}}{\text{I. HP.}}$, varied from about 66 per cent. at the lowest power to 82·3 per cent. at the highest power; so that, taking the thermo-dynamic efficiency of the engine at 80 per cent., as above, the heat-equivalent of the external electrical HP. would be equal to nearly 66 per cent. of the heat-equivalent of the work thermo-dynamically due from the steam in the case of the full power trials.

The Paper is accompanied by numerous diagrams, from which Plates 3 and 4 and the Figs. in the text have been prepared.

TABLE I. SIMPLE SERIES, 400 REVOLUTIONS.

Trial letter, intended ratio of expansion, and intended absolute mean admission pressure		$S\frac{40}{1\cdot57}$	$S\frac{50}{2\cdot17}$	$S\frac{70}{2\cdot8}$	$S\frac{80}{3\cdot2}$	$S\frac{90}{3\cdot6}$	$S\frac{100}{4}$	$S\frac{110}{4\cdot4}$
Date of trial	1	Dec. 9	Dec. 6	Dec. 8	Nov. 30	Nov. 30	Dec. 7	Dec. 5
Barometric back-pressure (lbs.)	2	14·49	14·46	14·57	14·66	14·75	14·64	14·74
Boiler pressure above atmosphere	3	36·25	51·0	74·0	85·0	97·0	110·0	122·0
Cylinder pressure (mean absolute during admission)	4	**40·88**	**50·65**	**68·67**	**78·66**	**92·65**	**98·14**	**106·34**
Ratio of expansion (corrected)	5	1·79	2·43	3·09	3·45	3·85	4·22	4·57
Point of cut-off in hp. cylinder	6							
,, ,, lp. ,,	7	0·604	0·437	0·339	0·296	0·264	0·2375	0·216
Duration of trial (minutes)	8	180	270	242	169	180	176	298
Temperature of engine-room, Fahrenheit	9	68	65	65	64	58		
Mean pressure on hp. piston	10							
,, ,, ,, ,, (under side)	11							
,, ,, lp. piston	12							
Total mean pressure referred to lp. piston	13	**19·6**	**22·62**	**29·14**	**31·06**	**36·83**	**36·87**	**38·61**
,, ,, ,, ,, cylinder without clearance	14	20·51	23·57	30·28	32·47	38·28	37·95	40·04
Theoretical mean pressure for corrected ratio of expansion on assumption that $p^6 v^7$ = constant	15	20·87	23·58	30·41	33·59	38·28	38·25	39·76
Percentage of theoretical mean pressure actually obtained	16	**98·2**	**99·96**	**99·5**	**96·6**	**100·0**	**99·2**	**100·72**
Revolutions per minute (mean during trial)	17	393·5	408·4	409·1	403·17	400·9	397·7	406·16
Indicated horse-power	18	**16·51**	**19·77**	**25·51**	**26·8**	**31·61**	**31·49**	**33·55**
Feed-water used per hour (lbs.)	19	706·0	711·0	830·8	795·2	850·0	877·7	874·4
,, ,, indicated horse-power-hour (lbs.)	20	**42·76**	**35·96**	**32·57**	**29·67**	**26·89**	**27·8**	**26·0**
,, per indicated horse-power-hour accounted for by lp. indicator at cut-off	21	37·74	29·03	23·92	22·62	20·21	19·7	18·36

Water theoretically required per indicated horse-power-hour for steam expanded from mean admission to exhaust temperature	22	34·67	28·62	23·02	21·15	19·24	18·66	17·9
Percentage efficiency	23	**81·08**	**79·58**	**70·6**	**71·6**	**71·5**	**67·1**	**68·8**
Lbs. of water collected per hour lp. steam chest	24	4·6	6·5	2·7	..	8·0	7·2	2·8
,, ,, accounted for by lp. indicator at cut-off	25	623·1	574·0	610·3	606·4	639·0	603·8	616·0
,, ,, ,, ,, lp. ,, at 0·604 of stroke	26	623·1	565·0	598·0	603·5	651·2	634·4	640·6
,, ,, ,, ,, lp. ,, at end of stroke	27	632·6	586·0	670·9	642·6	690·0	671·9	686·7
Lbs. of water not accounted for by lp. indicator at cut-off	28	82·9	137·0	220·5	188·8	211·0	273·9	258·4
,, ,, ,, ,, lp. ,, at 0·604 of stroke	29	82·9	146·0	232·8	191·7	198·8	243·3	233·8
,, ,, ,, ,, lp. ,, at end of stroke	30	73·4	125·0	159·9	152·6	160·0	205·8	187·7
Percentage of total feed-water missing at cut-off lp. cylinder	31	**11·7**	**19·3**	**26·5**	**23·7**	**24·8**	**31·25**	**29·56**
,, ,, ,, ,, at 0·604 of stroke lp. cyl.	32	11·7	20·53	28·02	24·1	23·3	27·7	26·7
,, ,, ,, ,, at end of stroke lp. cyl.	33	10·4	17·58	19·26	19·2	18·83	23·44	21·53
Heat-units missing per hour at cut-off lp. cylinder	34	77,213	126,588	200,544	170,494	189,140	244,866	229,976
,, ,, ,, stroke ,, lp. cylinder	35	**3·269**	**5·166**	**8·168**	**7·048**	**7·863**	**10·261**	**9·436**
Initial surface exposed in square feet lp. cylinder	36	2·77	2·77	2·77	2·77	2·77	2·77	2·77
Temperature Fahrenheit during admission (mean)	37	268	281	301	310	322	326	332
,, ,, reached by cushion steam	38	267	256	279	248	272	279	285
Range of temperature to which initial surface is exposed	39	1	25	22	62	50	47	47
Weight of steam at mean admission pressure, lbs. per cubic foot	40	0·0992	0·1213	0·1613	0·1832	0·2136	0·2243	0·2430
Surface exposed during admission, square feet	41	1·422	1·336	0·798	0·696	0·622	0·559	0·508
Temperature Fahrenheit during exhaust	42	212	212	214	214	215	214	213
Range of temperature to which admission surface is exposed	43	56	69	87	96	107	112	119
Weight of water in cylinder which at above range of temperature would account for abstraction of heat	44	0·058	0·074	0·093	0·073	0·073	0·091	0·079
Pressure (absolute) at point of cut-off	..	35·7	42·8	58·2	65·4	76·3	80·2	87·1
Pressure (absolute) at end of stroke	..	22·1	20·2	22·9	22·12	23·8	23·6	23·5
Mean back-pressure (absolute)	..	14·8	14·46	15·4	15·4	15·6	15·4	15·0

TABLE II.—COMPOUND SERIES. $\frac{P}{25}$ = RATIO OF EXPANSION

Trial letter, intended ratio of expansion, and intended absolute mean admission pressure		C $\frac{80}{3\cdot2}$	C $\frac{90}{3\cdot2}$	C $\frac{90}{3\cdot6}$	C $\frac{100}{3\cdot6}$	C $\frac{100}{4}$
Date of trial	1	Oct. 19	Oct. 19	Oct. 12	Oct. 14	Oct. 20 and Dec. 21
Barometric back-pressure (lbs.)	2	14·97	14·94	14·55	14·63	14·7
Boiler pressure above atmosphere	3	80·2	90·5	90·4	100·5	104·6
Cylinder pressure (mean absolute during admission)	4	**78·59**	**87·54**	**90·0**	**98·83**	**97·75**
Ratio of expansion (corrected)	5	3·35	3·5	4·02	4·0	4·42
Point of cut-off in hp. cylinder	6	0·6	0·6	0·523	0·523	0·470
„ „ „ lp. cylinder	7	0·604	0·604	0·604	0·604	0·604
Duration of trial (minutes)	8	180·0	123·0	176·5	118·0	124 and 240·5
Temperature of engine-room, Fahrenheit	9	..	..	..	..	..
Mean pressure on hp. piston	10	20·24	22·592	25·29	28·05	28·7
„ „ „ „ (under side)	11	5·96	7·198	6·29	6·86	6·39
„ „ „ lp. „	12	16·06	19·156	14·87	17·53	15·5
Total mean pressure referred to lp. piston	13	**29·08**	**23·934**	**30·71**	**34·93**	**33·0**
„ „ „ „ „ cylinder without clearance	14	29·82	34·8	31·49	35·82	33·85
Theoretical mean pressure for corrected ratio of expansion on assumption that $p^6 v^7$ = constant	15	34·08	38·1	35·4	40·56	36·43
Percentage of theoretical mean pressure actually obtained	16	**87·5**	**91·3**	**88·9**	**88·3**	**92·9**
Revolutions per minute (mean during trial)	17	400·0	401·08	397·6	401·5	405·3
Indicated horse-power	18	**24·91**	**29·14**	**26·14**	**30·0**	**28·65**
Feed-water used per hour (lbs.)	19	652·0	703·9	639·8	690·0	653·4
„ „ „ „ indicated horse-power-hour (lbs.)	20	**26·17**	**24·16**	**24·47**	**23·0**	**22·81**
Feed-water per indicated horse-power-hour accounted for by hp. indicator at cut-off	21	24·8	22·81	22·61	21·56	20·45
Water theoretically required per indicated horse-power-hour for steam expanded from mean admission to exhaust temperature	22	21·15	19·86	19·58	18·6	18·7
Percentage efficiency	23	**80·8**	**82·2**	**80·0**	**80·8**	**82·0**
Lbs. of water collected per hour lp. steam chest	24	10·0	9·5	11·5	11·0	11·0
Lbs. water accounted for by hp. indicator at cut-off	25	618·1	666·1	591·2	647·0	586·1
Lbs. water accounted for by lp. indicator at cut-off	26	553·1	596·9	536·1	543·3	546·5
Lbs. water accounted for by lp. indicator at end of stroke	27	559·8	603·5	546·8	571·8	557·6
Lbs. water not accounted for by hp. indicator at cut-off	28	33·9	37·8	48·6	43·0	67·3
Lbs. water not accounted for by lp. indicator at cut-off	29	98·9	107·0	109·7	146·7	107·4

$\frac{P - 10}{25}$ = **RATIO OF EXPANSION.** 400 REVOLUTIONS.

$C\frac{110}{4}$	$C\frac{110}{4\cdot4}$	$C\frac{120}{4\cdot4}$	$C\frac{120}{4\cdot8}$	$C\frac{130}{4\cdot8}$	$C\frac{130}{5\cdot2}$	$C\frac{140}{5\cdot2}$	$C\frac{140}{5\cdot6}$	$C\frac{150}{5\cdot6}$	$C\frac{150}{6}$	$C\frac{160}{6}$
Oct. 20	Oct. 21	Oct. 22	Oct. 26	Oct. 27	Oct. 27 and Nov. 28	Oct. 28	Nov. 2	Nov. 3	Nov. 12	Nov. 5
14·9	14·96	15·02	14·94	14·78	14·72	14·6	14·35	14·14	..	14·5
113·85	113·0	124·5	124·8	135·4	133·6	143·7	145·2	155·0	..	165·0
109·3	**108·96**	**121·27**	**119·92**	**130·58**	**129·9**	**139·7**	**141·85**	**149·84**	..	**158·5**
4·51	4·9	4·84	5·26	5·33	5·87	5·8	6·2	6·15	..	6·46
0·470	0·427	0·427	0·392	0·392	0·362	0·362	0·336	0·336	0·313	0·313
0·604	0·604	0·604	0·604	0·604	0·604	0·604	0·604	0·604	0·604	0·604
123·0	121	181	124	189	181·5 and 124	184	120	179	307	177
..	..	..	67	68	..	73	64	71	..	67
32·1	32·531	36·78	35·992	39·0	37·61	41·525	41·0	43·86	44·87	46·02
7·2	7·513	7·856	6·654	7·4	6·2	7·983	8·68	7·02	7·28	7·77
18·7	16·0	18·0	16·07	18·8	16·8	18·22	17·3	19·2	16·84	19·29
38·26	**35·95**	**40·26**	**37·37**	**41·85**	**38·65**	**42·94**	**42·02**	**44·62**	**42·88**	**46·04**
39·24	36·87	41·29	38·32	42·92	39·64	44·04	43·09	45·76	43·8	47·22
41·39	38·22	44·6	41·17	45·52	41·53	46·2	44·5	48·36	..	49·2
94·8	**96·4**	**92·5**	**93·0**	**94·2**	**95·4**	**95·3**	**96·8**	**94·6**	..	**95·9**
402·9	402·66	402·7	404·15	405·5	401·9	398·66	405·1	404·05	402·1	401·2
33·0	**30·98**	**34·73**	**32·33**	**36·31**	**33·25**	**36·64**	**36·43**	**38·59**	**36·95**	**39·55**
705·3	663·4	721·3	688·5	736·5	673·27	734·4	714·5	750·8	722·4	759·0
21·37	**21·41**	**20·76**	**21·29**	**20·35**	**20·26**	**20·0**	**19·64**	**19·45**	**19·56**	**19·19**
19·34	18·9	18·6	18·6	17·9	17·38	17·17	16·7	16·76	15·87	15·92
17·65	17·68	16·8	16·87	16·25	16·26	15·72	15·61	15·23	..	14·87
82·5	**82·5**	**80·9**	**79·2**	**80·0**	**80·25**	**78·6**	**79·4**	**78·3**	..	**77·4**
12·0	8·5	13·0	14·5	16·5	14·8	17·5	17·0	15·0	16·8	16·5
638·3	588·4	645·7	602·5	650·1	577·9	631·9	609·2	636·8	586·6	629·7
590·5	575·5	580·1	558·1	595·7	553·2	588·1	573·3	596·2	..	597·3
588·3	546·4	595·1	577·9	607·3	558·0	605·6	589·6	595·3	..	610·5
67·0	75·0	75·6	86·0	86·4	95·3	102·5	105·3	114·0	135·8	129·3
114·5	87·9	141·2	129·9	140·5	114·9	146·3	141·2	154·4	..	161·7

TABLE II.—

Trial letter, intended ratio of expansion, and intended absolute mean admission pressure		C $\frac{80}{3\cdot2}$	C $\frac{90}{3\cdot2}$	C $\frac{90}{3\cdot6}$	C $\frac{100}{3\cdot6}$	C $\frac{100}{4}$
Lbs. water not accounted for by lp. indicator at end of stroke	30	92·2	100·4	93·0	118·2	95·8
Percentage of total feed-water missing at cut-off hp. cylinder	31	**5·2**	**5·0**	**7·6**	**6·23**	**10 2**
Per cent. of total feed-water missing at cut-off lp. cylinder	32	15·17	15·2	17·42	21·25	16·36
Per cent. of total feed-water missing at end of stroke lp. cylinder	33	14·14	14·25	14·53	17·13	14·65
Heat-units missing per hour at cut-off hp. cyl.	34	30,510	33,804	43,424	38,227	59,948
,, ,, stroke at cut-off hp. cylinder	35	**1·271**	**1·404**	**1·819**	**1·586**	**2·459**
Initial surface exposed in hp. cylinder square feet	36	1·771	1·771	1·771	1·771	1·771
Temperature F. during admission (mean)	37	310	318	320	326	325·5
,, ,, reached by cushion-steam	38	315	323	320	328	315·5
Range of temperature to which initial surface is exposed	39	−5	−5	0	−2	10
Weight of steam at mean admission pressure, lbs. per cubic foot	40	0·1829	0·2029	0·2079	0·2258	0·2241
Surface exposed during admission, square feet	41	1·027	1·027	0·889	0·889	0·799
Temperature Fahrenheit during exhaust	42	279	285	279	284	278
Range of temperature to which admission surface is exposed	43	31	33	41	42	47·5
Weight of water in cylinder which at above range of temperature would account for abstraction of heat	44	0·041	0·042	0·044	0·038	0·051
Pressure (absolute) at point of cut-off hp. cyl.	..	71·21	79·94	81·21	88·82	88·25
,, ,, ,, ,, lp. cyl.	..	31·35	33·7	30·0	31·3	30·27
Pressure (absolute) at end of hp. stroke	..	45·22	..	..	..	..
,, ,, ,, lp. stroke	..	19·3	20·8	18·9	19·6	19·2
Mean back-pressure (absolute) hp. cyl.	..	48·3	53·7	48·25	52·79	48·65
,, ,, ,, lp. cyl.	..	15·8	15·4	15·6	15·1	14·97

continued.

$C\frac{110}{4}$	$C\frac{110}{4\cdot4}$	$C\frac{120}{4\cdot4}$	$C\frac{120}{4\cdot8}$	$C\frac{130}{4\cdot8}$	$C\frac{130}{5\cdot2}$	$C\frac{140}{5\cdot2}$	$C\frac{140}{5\cdot6}$	$C\frac{150}{5\cdot6}$	$C\frac{150}{6}$	$C\frac{160}{6}$
117·0	117·0	126·2	110·6	129·2	115·2	128·8	124·9	155·5	..	148·5
9·5	11·3	10·5	12·5	11·7	14·2	13·9	14·7	15·1	18·8	17·0
16·25	13·3	19·6	18·9	19·1	17·0	19·9	19·76	20·6	..	21·3
16·59	17·64	17·5	16·07	17·55	17·12	17·54	17·49	20·69	..	19·55
59,560	66,412	66,261	75,723	75,686	83,637	89,513	91,926	99,180	..	112,103
2·464	2·748	2·742	3·122	3·110	3·468	3·742	3·781	4·091	..	4·657
1·771	1·771	1·771	1·771	1·771	1·771	1·771	1·771	1·771	1·771	1·771
334	334	342	341	347	346·5	353	353	358	..	362
323	318	324	320	326	320·5	326	324	326	..	327
11	16	18	21	21	26	27	29	32	..	35
0·2495	0·2486	0·2752	0·2724	0·2951	0·2935	0·3142	0·3187	0·3355	..	0·3539
0·799	0·726	0·726	0·666	0·666	0·615	0·615	0·571	0·571	0·532	0·532
285	279	285	280	285	281	286	282	285	..	285
49	55	57	61	62	65·5	67	71	73	..	77
0·050	0·050	0·048	0·051	0·05	0·053	0·055	0·053	0·056	..	0·060
95·5	95·0	105·0	104·6	113·3	110·13	119·6	120·0	126·1	..	133·7
33·2	32·2	32·5	31·15	33·1	31·35	32·7	31·8	33·2	..	33·4
..	..	..	..	..	..	..	..	48·8	..	..
20·2	18·7	20·4	19·7	20·6	19·2	20·7	20·0	20·2	..	20·9
53·6	48·7	53·4	49·6	54·0	50·15	54·7	51·2	53·74	..	53·7
15·7	15·7	15·4	15·77	15·4	15·2	15·4	15·2	15·0	..	14·9

TABLE III.

A, Professor Kennedy. B, Professor Unwin. C, Mr. Hartnell. D, Mr. MacFarlane Gray.

Trial letter, intended ratio of expansion, and intended absolute mean admission pressure		$T\frac{150}{5\cdot 6}$	$T\frac{150}{6}$	$T\frac{160}{6}$	$T\frac{160}{6\cdot 4}$	$T\frac{170}{6\cdot 4}$	$T\frac{170}{6\cdot 4}$	$T\frac{170}{6\cdot 4}$
				A		B	C	D
Date of trial	1	Dec. 14 and Jan. 17	Dec. 19	Jan. 14	Dec. 16	Dec. 17	Jan. 12	Jan. 16
Barometric back-pressure (lbs.)	2	14·74	14·5	14·99	14·47	14·7	15·04	14·88
Boiler pressure above atmosphere	3	150·15	151·9	160·0	160·8	172·0	172·5	
Cylinder pressure (mean absolute during admission)	4	151·9	149·65	159·49	158·07	..	172·54	
Ratio of expansion (corrected)	5	5·97	6·259	6·158	6·507	..	6·52	
Point of cut-off in h. hp. cylinder	5A	0·694	0·647	0·647	0·606	0·606	0·606	0·606
" " hp. "	6	0·604	0·604	0·604	0·604	0·604	0·604	0·604
" " lp. "	7	0·604	0·604	0·604	0·604	0·604	0·604	0·604
Duration of trial (minutes)	8	178 and 289	241	300	300	240	184	223
Temperature of engine-room, Fahrenheit	9	61·5	57	..	72	65	56	60
Mean pressure on h. hp. piston	9A	32·94	33·85	39·91	36·71	39·11	44·533	43·908
" " h. hp. " (under side)	9B	14·58	14·1	15·06	12·5	11·359	16·11	14·680
" " hp. "	10	22·74	21·66	22·04	22·5	26·1	23·55	22·728
" " hp. " (under side)	11	6·69	6·25	6·8	6·64	7·158	6·39	6·886
" " lp. "	12	15·27	14·266	15·045	14·5	16·454	15·55	*16·63
Total mean pressure referred to lp. piston	13	41·11	39·498	42·424	40·675	45·032	44·855	*45·311
" " " cylinder without clearance	14	42·16	40·51	43·51	41·71	46·18	46·00	
Theoretical mean pressure for corrected ratio of expansion on assumption that $p^6 v^7$ = constant	15	49·94	47·09	51·36	48·9	..	53·97	
Percentage of theoretical mean pressure actually obtained	16	84·54	86·02	84·7	85·3	..	85·2	
Revolutions per minute (mean during trial)	17	405 6	409·0	401·2	408·4	414·85	400·4	416·46
Indicated horse-power	18	35·69	34·58	36·44	35·56	40·041	38·45	40·42
Feed-water used per hour (lbs.)	19	702·6	669·2	696·4	682·6	749·0	709·5	745·0
" " indicated horse-power-hour (lbs.)	20	19·68	19·35	19·11	19·19	18·7	18·45	18·42
" per indicated horse-power-hour accounted for by h. hp. indicator at cut-off	21	18·64	18·29	18·26	17·88	16·77	17·52	
Water theoretically required per indicated horse-power-hour for steam expanded from mean admission to exhaust temperature	22	15·16	15·25	14·57	14·9	..	14·36	
Percentage efficiency	23	77·05	78·8	76·2	77·6	..	77·8	77·7

Lbs. of water collected per hour hp. steam chest	23A	5·04	7·3	3·18	4·5	3·8	4·5	2·1	
,, ,, ,, ,, lp. ,, ...	24	14·53	15·75	15·7	13·5	14·6	13·5	15·4	
,, ,, accounted for by h. hp. indicator at cut-off . .	25	665·3	632·6	665·5	636·0	671·7	673·9		
,, ,, ,, ,, hp. ,, ,, . .	25A	599·1	589·1	582·6	600·4	665·2	600·6		
,, ,, ,, ,, lp. ,, ,, . .	26	547·1	532·1	533·3	531·8	594·4	537·7		
,, ,, ,, ,, lp. ,, at end of stroke .	27	575·9	550·8	566·9	554·3	637·25	572·8		
Lbs. water not accounted for by h. hp. indicator at cut-off . .	28	37·3	36·6	30·9	46·6	77·3	35·6		
,, ,, ,, ,, hp. ,, ,, . .	28A	103·5	80·1	113·8	82·2	83·8	108·9		
,, ,, ,, ,, lp. ,, ,, . .	29	155·45	137·1	163·1	150·8	154·6	171·8		
,, ,, ,, ,, lp. ,, at end of stroke	30	126·65	118·4	129·5	128·3	111·75	136·7		
Per cent. of total feed-water missing at cut-off h. hp. cylinder .	31	5·33	5·46	4·43	6·84	10·32	5·01		
,, ,, ,, ,, hp. ,, .	31A	14·84	11·95	16·32	12·06	11·20	15·33		
,, ,, ,, ,, lp. ,, .	32	22·12	20·48	23·41	22·11	20·64	24·21		
,, ,, ,, ,, end of stroke lp. cylinder	33	18·01	17·68	18·57	18·81	14·92	19·25		
Heat-units missing per hour at cut-off h. hp. cylinder . . .	34	32,122	31,549	26,521	40,000	..	30,345		
,, ,, ,, stroke ,, h. hp. ,, . . .	35	1·316	1·285	1·101	1·632	..	1·263		
Initial surface exposed in h. hp. cylinder, square feet . . .	36	1·065	1·065	1·065	1·065	..	1·065		
Temperature Fahrenheit during admission (mean)	37	359	358	363	362	..	369		
,, reached by cushion steam	38	363·0	361·5	363·0	361·0	..	366·0		
Range of temperature to which initial surface is exposed . .	39	—4	—3·5	0·0	1·0	..	3·0		
Weight of steam at mean admission pressure, lbs. per cubic foot	40	0·3399	0·3351	0·3559	0·3531	..	0·3833		
Surface exposed during admission, square feet	41	0·840	0·783	0·783	0·733	..	0·733		
Temperature Fahrenheit during exhaust	42	334·5	330·0	333·0	331·0	..	335·0		
Range of temperature to which admission surface is exposed .	43	24·5	28·0	30·0	31·0	..	34·0		
Weight of water in cylinder which at above range of temperature would account for abstraction of heat	44	0·053	0·045	0·036	0·052	..	0·037		
Pressure (absolute) at point of cut-off h. hp. cylinder . . .	..	148·9	146·1	156·0	155·7	..	168·6	165·9	
,, ,, ,, ,, hp. ,, . . .	..	70·3	68·5	69·0	69·7	..	71·4	71·2	
,, ,, ,, ,, lp. ,, . . .	..	30·35	29·2	29·9	29·2	..	30·2	30·5	
Pressure (absolute) at end of stroke h. hp. cylinder	..	105·1	96·4	102·8	96·0	..	104·5		
,, ,, ,, ,, hp. ,, ...	..	46·15	43·8	45·4	44·4	..	47·1		
,, ,, ,, ,, lp. ,, ...	..	19·52	18·5	19·4	18·6	..	19·6	19·6	
Mean back-pressure (absolute) h. hp. cylinder	..	110·3	103·4	107·5	105·8	..	110·4	110·4	
,, ,, ,, hp. ,, ...	..	49·0	47·5	48·4	47·2	..	49·2	49·4	
,, ,, ,, lp. ,, ...	..	15·35	14·8	15·3	14·8	..	15·4	15·3	

* Corrected for excess of barometric pressure over 14·7 lbs.

TABLE IV.—COMPOUND SERIES. STEAM 130 LBS. PRESSURE WITH VARIOUS EXPANSIONS.

Trial letter, intended ratio of expansion, and intended absolute mean admission pressure	—	$C\frac{130}{4}$	$C\frac{130}{4\cdot4}$	$C\frac{130}{4\cdot8}$	$C\frac{130}{5\cdot2}$	$C\frac{130}{5\cdot6}$	$C\frac{130}{6}$	$C\frac{130}{8}$
Date of trial	1	Oct. 21	Oct. 26	Oct. 27	Oct. 27 and Nov. 28	Nov. 2	Nov. 4	Nov. 28
Barometric back-pressure (lbs.)	2	14·95	14·99	14·78	14·72	14·35	14·24	14·745
Boiler pressure above atmosphere	3	134·8	135·0	135·4	133·65	135·0	135·0	135·5
Cylinder pressure (mean absolute during admission)	4	**128·85**	**129·19**	**130·58**	**129·97**	**131·1**	**130·09**	**128·15**
Ratio of expansion (corrected)	5	4·46	4·85	5·33	5·87	6·10	6·52	8·38
Point of cut-off in hp. cylinder	6	0·470	0·427	0·392	0·362	0·336	0·313	0·233
,, ,, lp. ,,	7	0·604	0·604	0·604	0·604	0·604	0·604	0·604
Duration of trial (minutes)	8	186	182	189	124 and 181·5	182	180	200
Temperature of engine-room Fahrenheit	9	..	..	68	68	68	72	69
Mean pressure on hp. piston	10	39·57	40·14	39·0	37·61	37·8	37·8	32·2
,, ,, ,, ,, (under side)	11	8·95	8·27	7·4	6·2	5·86	6·2	4·5
,, ,, lp. ,,	12	21·8	19·8	18·8	16·8	16·18	14·4	8·25
Total mean pressure referred to lp. piston	13	**45·97**	**43·95**	**41·85**	**38·62**	**38·01**	**36·47**	**26·6**
,, ,, ,, ,, cylinder without clearance	14	47·14	45·07	42·92	39·64	39·16	37·40	27·28
Theoretical mean pressure for corrected ratio of expansion on assumption that $p^6 v^7$ = constant	15	51·9	48·57	45·52	41·53	40·71	37·79	27·8
Percentage of theoretical mean pressure actually obtained	16	**90·8**	**92·7**	**94·2**	**95·46**	**96·1**	**98·9**	**98·1**
Revolutions per minute (mean during trial)	17	406·8	405·0	405·5	401·9	402·6	400·0	404·45
Indicated horse-power	18	**40·03**	**38·0**	**36·31**	**33·25**	**32·76**	**31·23**	**23·03**
Feed-water used per hour (lbs.)	19	830·9	778·3	736·5	673·25	654·7	634·6	534·0
,, ,, indicated horse-power-hour (lbs.)	20	**20·75**	**20·48**	**20·35**	**20·26**	**20·0**	**20·32**	**23·14**
,, per indicated horse-power-hour accounted for by hp. indicator at cut-off	21	18·9	18·4	17·9	17·38	17·13	16·56	17·37

Water theoretically required per indicated horse-power-hour for steam expanded from mean admission to exhaust temperature	22	16·32	16·3	16·25	16·26	16·2	16·26	16·37
Percentage efficiency	23	**78·6**	**79·5**	**80·0**	**80·2**	**81·0**	**80·0**	**70·7**
Lbs. of water collected per hour lp. steam chest	24	13·0	12·5	16·5	14·87	17·25	14·5	9·25
,, ,, accounted for by hp. indicator at cut-off	25	756·3	699·0	650·1	577·9	561·2	517·3	400·2
,, ,, ,, ,, lp. ,, ,,	26	663·9	624·2	595·7	558·2	549·7	517·6	435·6
,, ,, ,, ,, lp. ,, at end of stroke	27	670·7	632·2	607·3	558·0	571·1	524·5	456·7
Lbs. water not accounted for by hp. indicator at cut-off	28	74·6	79·3	86·4	95·3	93·52	117·3	133·8
,, ,, ,, ,, lp. ,, ,,	29	167·0	154·1	140·5	114·9	105·0	117·0	98·4
,, ,, ,, ,, lp. ,, at end of stroke	30	160·2	146·1	129·2	115·2	83·6	110·1	77·3
Percentage of total feed-water missing at cut-off hp. cylinder	31	**8·9**	**10·18**	**11·7**	**14·25**	**14·28**	**18·4**	**25·0**
,, ,, ,, ,, ,, lp. ,,	32	20·1	19·8	19·1	17·0	16·0	18·5	18·43
,, ,, ,, ,, ,, end of stroke lp. cyl.	33	19·27	18·77	17·55	17·12	12·76	17·35	14·47
Heat-units missing per hour at cut-off hp. cylinder	34	65,394	69,498	75,686	83,637	82,017	102,989	117,677
,, ,, ,, stroke ,, hp. ,,	35	**2·679**	**2·86**	**3·110**	**3·468**	**3·395**	**4·291**	**4·848**
Initial surface exposed in hp. cylinder square feet	36	1·771	1·771	1·771	1·771	1·771	1·771	1·771
Temperature F. during admission (mean)	37	346	346	347	346	347	347	346
,, reached by cushion steam	38	333	328	326	320	319	314	301
Range of temperature to which initial surface is exposed	39	13	18	21	26	28	33	45
Weight of steam at mean admission pressure, lbs. per cubic foot	40	0·2909	0·2918	0·2951	0·2935	0·2958	0·2937	0·2895
Surface exposed during admission, square feet	41	0·799	0·726	0·666	0·615	0·571	0·532	0·4
Temperature Fahrenheit during exhaust	42	294	288	285	281	279	275	265
Range of temperature to which admission surface is exposed	43	52	58	62	65	68	72	81
Weight of water in cylinder which at above range of temperature would account for abstraction of heat	44	0·051	0·049	0·05	0·052	0·049	0·06	0·059
Pressure (absolute) at point of cut-off hp. cylinder	..	112·5	113·0	113·3	112·2	111·1	109·2	106·4
,, ,, ,, ,, lp. ,,	..	36·7	34·67	33·1	32·2	30·9	29·1	24·4
Pressure (absolute) at end of stroke hp. cylinder								
,, ,, ,, ,, lp. ,,	..	22·7	21·4	20·6	19·9	19·6	18·0	15·6
Mean back-pressure (absolute) hp. cylinder	..	61·5	56·3	54·0	51·0	48·5	45·3	38·5
,, ,, ,, lp. ,,	..	15·5	14·99	15·4	15·1	15·4	14·8	15·4

TABLE V.—SIMPLE

Trial letter, intended ratio of expansion, and intended absolute mean admission pressure		$S\frac{50}{2 \cdot 174}$		
Date of trial	1	Dec. 6	Dec. 7	Dec. 6
Barometric back-pressure (lbs.)	2	14·46	14·57	14·44
Boiler pressure above atmosphere	3	51·0	44·0	40·25
Cylinder pressure (mean absolute during admission)	4	**50·65**	**49·55**	**49·04**
Ratio of expansion (corrected)	5	2·43	2·3	2·25
	6			
Point of cut-off in lp. cylinder	7	0·437	0·437	0·437
Duration of trial (minutes)	8	270	153	122
Temperature of engine-room, Fahrenheit	9	65	60	50
	10			
	11			
	12			
Total mean pressure referred to lp. piston	13	**22·62**	**21·72**	**23·13**
" " " to cylinder without clearance	14	23·57	22·27	23·72
Theoretical mean pressure for corrected ratio of expansion on assumption that $p^6 v^7$ = constant	15	23·58	23·83	24·11
Percentage of theoretical mean pressure actually obtained	16	**99·96**	**93·4**	**98·4**
Revolutions per minute (mean during trial)	17	**408·4**	**200·6**	**110·5**
Indicated horse-power	18	**19·77**	**9·32**	**5·47**
Feed-water used per hour (lbs.)	19	711·0	389·4	251·8
" " indicated horse-power hour (lbs.)	20	**35·96**	**41·78**	**46·03**
" per indicated horse-power accounted for by indicator at cut-off	21	29·03	31·79	30·16
Water theoretically required per indicated horse-power-hour for steam expanded from mean admission to exhaust temperature	22	28·62	29·15	29·37
Percentage efficiency	23	**79·58**	**69·7**	**63·8**
Lbs. of water collected per hour lp. steam-chest	24	6·5	8·0	6·0
" " accounted for by lp. indicator at cut-off	25	574·0	296·3	165·0
Lbs. of water accounted for by lp. indicator at 0·604 of stroke	26	565·0	294·0	164·9
Lbs. of water accounted for by lp. indicator at end of stroke	27	586·0	321·0	179·2
Lbs. of water not accounted for by lp. indicator at cut-off	28	137·0	93·1	86·8
Lbs. of water not accounted for by lp. indicator at 0·604 of stroke	29	146·0	95·4	86·9

SPEED TRIALS.

S $\frac{70}{2\cdot8}$			S $\frac{90}{3\cdot6}$			S $\frac{110}{4\cdot4}$		
Dec. 8 14·57 74·0 **68·67** 3·09 0·339 242 65	Dec. 9 14·47 66·5 **71·07** 3·087 0·339 152 66	Dec. 8 14·40 62·0 **69·1** 2·99 0·339 127 68	Nov. 30 14·75 97·0 **92·65** 3·85 0·264 180 58	Nov. 30 14·7 84·7 **88·46** 3·74 0·264 118 ..	Dec. 1 14·93 80·0 **89·43** 3·72 0·264 178 66	Dec. 5 14·74 122·0 **106·34** 4·57 0·216 298 ..	Dec. 2 14·98 112·0 **108·98** 4·46 0·216 119 ..	Dec. 5 14·74 105·4 **108·72** 4·32 0·216 123 ..
29·14 30·28 30·41 **99·5**	**32·0** 32·97 32·08 **102·3**	**32·9** 33·74 31·69 **93·9**	**36·83** 38·28 38·28 **100·0**	**35·1** 36·2 36·94 **98·0**	**38·0** 38·97 37·42 **104·1**	**38·61** 40·04 39·76 **100·72**	**42·8** 44·05 41·68 **105·68**	**44·31** 45·44 42·99 **105·7**
409·1 **25·51**	**205·2** **14·05**	**112·7** **7·937**	**400·9** **31·61**	**223·0** **16·75**	**122·8** **9·98**	**406·16** **33·55**	**223·7** **20·49**	**138·0** **13·09**
830·8 **32·57** 23·92 23·02 **70·6** 2·7 610·3 598·0 670·9	483·35 **34·4** 22·55 22·5 **65·4** 5·0 316·9 332·6 371·7	323·1 **40·7** 22·08 22·9 **56·2** 5·0 175·1 188·6 217·4	850·0 **26·89** 20·21 19·24 **71·5** 8·0 639·0 651·2 690·0	465·25 **27·77** 20·8 19·71 **70·9** .. 350·1 355·9 374·5	339·8 **34·05** 19·61 19·64 **57·6** .. 195·8 211·8 242·2	874·4 **26·0** 18·36 17·9 **68·8** 2·8 616·0 640·6 686·7	618·6 **30·19** 17·41 17·68 **57·66** 4·0 356·8 398·5 456·2	408·8 **31·22** 17·41 17·7 **56·6** 3·0 226·9 252·5 283·5
220·5 232·8	166·45 150·57	148·0 134·5	211·0 198·8	115·15 109·35	144·0 128·0	258·4 233·8	261·8 220·1	181·9 156·3

TABLE V.—SIMPLE-ENGINE

Trial letter, intended ratio of expansion, and intended absolute mean admission pressure		S $\frac{50}{2\cdot174}$		
Lbs. of water not accounted for by lp. indicator at end of stroke	30	125 0	68·4	72·6
Per cent. of total feed-water missing at cut-off lp. cylinder	31	**19·3**	**23·9**	**34·5**
Per cent. of total feed-water missing at 0·604 of stroke lp. cylinder	32	20·53	24·5	34·5
Per cent. of total feed-water missing at end of stroke lp. cylinder	33	17·58	17·57	28·8
Heat-units missing per hour at cut-off	34	126,588	85,931	79,433
,, ,, ,, stroke at cut-off	35	**5·166**	**7·139**	**11·97**
Initial surface exposed in square feet	36	2·77	2·77	2·77
Temperature, Fahrenheit, during admission (mean)	37	281	280	279
,, ,, reached by cushion steam	38	256	247	243
Range of temperature to which initial surface is exposed	39	25	33	36
Weight of steam at mean admission pressure, lbs. per cubic foot	40	0·1213	0·1198	0·1196
Surface exposed during admission (square feet)	41	1·336	1·336	1·336
Temperature (Fahrenheit) during exhaust	42	212	213	212
Range of temperature to which admission surface is exposed	43	69	67	67
Weight of water in cylinder which at above range of temperature would account for abstraction of heat	44	0·074	0·106	0·178
Revolutions per minute	..	408·4	200·6	110·5
Pressure (absolute) at point of cut-off	..	42·8	43·6	43·8
Pressure (absolute) at end of stroke	..	20·2	21·7	22·1
Mean back-pressure (absolute)	..	14·46	15·2	14·7

SPEED TRIALS—*continued.*

$S\frac{70}{2\cdot8}$			$S\frac{90}{3\cdot6}$			$S\frac{110}{4\cdot4}$		
159·9	111·65	105·7	160·0	90·75	97·6	187·7	162 4	125·3
26·5	**34·44**	**45·81**	**24·8**	**24·75**	**42·5**	**29·56**	**42·33**	**44·5**
28·02	31·16	41·63	23·3	23·4	37·7	26·7	35·58	38·2
19·26	13·07	32·71	18·83	19·52	28·70	21·53	26·26	30·65
200,544 **8·168**	150,973 **12·262**	134,384 **19·864**	189,140 **7·863**	101,216 **7·564**	129,168 **17·523**	229,976 **9·436**	232,556 **14·338**	161,254 **19·475**
2·77	2·77	2·77	2·77	2·77	2·77	2·77	2·77	2·77
301	303	302	322	319	319	332	333	333
279	278	280	272	264	263	285	272	275
22	25	22	50	55	56	47	61	58
0·1613	0·1666	0·1623	0·2136	0·2050	0·2067	0·2430	0·2486	0·2482
0·798	0·798	0·798	0·622	0·622	0·622	0·508	0·508	0·508
214	213	212	215	214	214	213	214	214
87	90	90	107	105	105	119	119	119
0·093	0·136	0·220	0·073	0·072	0·166	0·079	0·120	0·163
409·1	205·2	112·7	400·9	223·0	122·8	406·16	223·7	138·0
58·2	60·1	60·6	76·3	74·6	75·6	87·1	90·0	93·1
22·9	25·2	26·8	23·8	23·1	27·0	23·5	28·1	28·4
15·4	15·1	14·65	15·6	15·5	15·4	15·0	15·5	15·5

TABLE VI.—COMPOUND-ENGINE

Trial letter, intended ratio of expansion, and intended absolute mean admission pressure		$C\frac{90}{3 \cdot 2}$			
Date of trial	1		Oct. 19	Nov. 23	Nov. 24
Barometric back-pressure (lbs.)	2		14·94	14·76	14·64
Boiler pressure above atmosphere	3		90·5	85·16	83·5
Cylinder pressure (mean absolute during admission)	4		**87·54**	**90·41**	**90·84**
Ratio of expansion (corrected)	5		3·5	3·42	3·41
Point of cut-off in hp. cylinder	6		0·604	0·604	0·604
" " lp. "	7		0·604	0·604	0·604
Duration of trial (minutes)	8		123	123	150
Temperature of engine-room, Fahrenheit	9		..	59	..
Mean pressure on hp. piston	10		22·59	27·1	27·94
" " " " (under side)	11		7·19	9·71	10·18
" " lp. "	12		19·15	18·83	19·74
Total mean pressure referred to lp. piston	13		**33·93**	**37·05**	**38·39**
" " " " cylinder without clearance	14		34·8	38·0	39·37
Theoretical mean pressure for corrected ratio of expansion on assumption that $p^6 v^7$ = constant	15		38·1	41·03	41·35
Percentage of theoretical mean pressure actually obtained	16		**91·3**	**92·6**	**95·2**
Revolutions per minute (mean during trial)	17		**401·08**	**210·8**	**122·0**
Indicated horse-power	18		**29·14**	**16·73**	**10·02**
Feed-water used per hour (lbs.)	19		703·9	422·92	270·8
" " indicated horse-power-hour (lbs.)	20		**24·16**	**25·27**	**27·02**
Feed-water per indicated horse-power-hour accounted for by hp. indicator at cut-off	21		22·85	22·07	21·55
Water theoretically required per indicated horse-power-hour for steam expanded from mean admission to exhaust temperature	22		19·86	19·52	19·48
Percentage efficiency	23		**82·2**	**77·2**	**72·0**
Lbs. of water collected per hour lp. steam chest	24		9·5	15·6	9·2
Lbs. of water accounted for by hp. indicator at cut-off	25		666·1	369·3	216·0
Lbs. of water accounted for by lp. indicator at cut-off	26		596·9	318·0	186·3
Lbs. of water accounted for by lp. indicator at end of stroke	27		603·5	341·1	197·5
Lbs. of water not accounted for by hp. indicator at cut-off	28		37·8	53·62	54·8
Lbs. of water not accounted for by lp. indicator at cut-off	29		107·0	105·0	84·5

SERIES. SPEED TRIALS.

	$C\frac{110}{4}$				$C\frac{130}{4\cdot8}$		
	Oct. 20	Nov. 24	Nov. 25		Oct. 27	Nov. 26	Nov. 25
	14·9	14·64	14·6		14·78	14·64	14·64
	113·85	105·0	102·6		135·4	124·0	120·0
	109·3	**109·14**	**110·08**		**130·58**	**128·84**	**128·79**
	4·51	4·5	4·36		5·33	5·22	5·22
	0·470	0·470	0·470		0·392	0·392	0·392
	0·604	0·604	0·604		0·604	0·604	0·604
	123	125	116		189	124	150
	..	64	60		68	70	63
	32·1	35·33	35·15		39·0	40·2	40·86
	7·2	9·0	9·62		7·4	10·83	9·83
	18·7	18·20	19·5		18·8	18·54	20·03
	38·26	**40·234**	**41·73**		**41·85**	**43·89**	**45·24**
	39·24	41·26	42·8		42·92	45·01	46·4
	41·39	41·77	44·49		45·52	45·66	45·66
	94·8	**98·7**	**96·2**		**94·2**	**98·5**	**101·6**
	402·9	**211·98**	**123·8**		**405·5**	**216·4**	**130·86**
	33·0	**18·26**	**11·06**		**36·31**	**20·33**	**12·67**
	705·3	422·4	273·6		736·5	432·5	300·0
	21·37	**23·13**	**24·73**		**20·35**	**21·27**	**23·67**
	19·34	18·42	18·49		17·9	17·19	16·65
	17·65	17·67	17·52		16·25	16·3	16·31
	82·5	**76·3**	**70·8**		**80·0**	**76·6**	**68·8**
	12·0	15·0	9·3		16·5	16·9	5·8
	638·3	336·4	204·5		650·1	349·5	211·0
	590·5	309·4	187·7		595·7	318·8	202·2
	588·3	326·2	196·5		607·3	334·8	217·8
	67·3	86·0	69·1		86·4	83·0	89·0
	114·5	113·0	85·9		140·5	113·6	97·8

TABLE VI.—

Trial letter, intended ratio of expansion, and intended absolute mean admission pressure			C $\frac{90}{3\cdot 2}$		
Lbs. of water not accounted for by lp. indicator at end of stroke	30		100·4	81·9	73·3
Percentage of total feed-water missing at cut-off hp. cylinder	31		**5·0**	**12·6**	**20·2**
Per cent. of total feed-water missing at cut-off lp. cylinder	32		15·2	24·8	31·2
Per cent. of total feed-water missing at end of stroke lp. cylinder	33		14·25	19·35	27·07
Heat-units missing per hour at cut-off hp. cylinder	34		33,804	47,898	48,919
Heat-units missing per stroke at cut-off hp. cylinder	35		**1·404**	**3·786**	**6·682**
Initial surface exposed in hp. cylinder square feet	36		1·771	1·771	1·771
Temperature F. during admission (mean)	37		318	320	320
„ reached by cushion steam	38		323	307	303
Range of temperature to which initial surface is exposed	39		−5	13	17
Weight of steam at mean admission pressure, lbs. per cubic foot	40		0·2029	0·2087	0·2096
Surface exposed during admission, square feet	41		1·027	1·027	1·027
Temperature Fahrenheit during exhaust	42		285	283	283
Range of temperature to which admission surface is exposed	43		33	37	37
Weight of water in cylinder which at above range of temperature would account for abstraction of heat	44		0·042	0·102	0·180
Revolutions per minute	..		401·08	210·8	122·0
Pressure (absolute) at point of cut-off hp. cyl.	..		79·94	81·5	82·5
„ „ „ „ lp. cyl.	..		33·7	33·5	33·9
Pressure (absolute) at end of stroke hp. cyl.					
„ „ „ „ lp. cyl.	..		20·8	21·9	21·9
Mean back-pressure (absolute) hp. cylinder	..		53·7	51·8	51·8
„ „ „ lp. cylinder	..		15·4	15·5	15·4

continued.

	$C\frac{110}{4}$				$C\frac{130}{4 \cdot 8}$		
	117·0	96·2	77·1		129·2	97·7	82·2
	9·5	**20·3**	**25·2**		**11·7**	**19·1**	**29·66**
	16·25	26·7	31·4		19·1	26·4	32·6
	16·59	22·77	28·17		17·55	22·59	27·41
	59,560	76,282	61,153		75,686	72,824	78,053
	2·464	**6·000**	**8·230**		**3·110**	**5·611**	**9·94**
	1·771	1·771	1·771		1·771	1·771	1·771
	334	334	334		347	346	346
	323	306	305		326	309	314
	11	28	29		21	37	32
	0·2495	0·2490	0·2511		0·2951	0·2909	0·2914
	0·799	0·799	0·799		0·666	0·666	0·666
	285	281	282		285	282	284
	49	53	52		62	64	62
	0·050	0·113	0·158		0·050	0·087	0·160
	402·9	211·98	123·8		405·5	216·4	130·86
	95·5	92·7	96·2		113·3	110·9	111·6
	33·2	32·4	33·7		33·1	32·9	34·3
	20·2	20·9	21·4		20·6	21·1	22·6
	53·6	50·2	51·2		54·0	51·4	52·9
	15·7	15·2	15·0		15·4	15·6	15·0

TABLE VII.—COMPOUND-ENGINE SERIES. FOUR EXPANSIONS, VARIOUS PRESSURES, BOILER PRESSURE VARIED.

Trial letter, intended ratio of expansion, and intended absolute mean admission pressure	..	C $\frac{60}{4}$	C $\frac{70}{4}$	C $\frac{80}{4}$	C $\frac{90}{4}$	C $\frac{100}{4}$	C $\frac{110}{4}$	C $\frac{120}{4}$	C $\frac{130}{4}$	C $\frac{60}{4}$ Throttled.
Date of trial	1	Nov. 8 and Dec. 21	Dec. 29	Nov. 8	Dec 30.	Oct. 20 and Dec. 21	Oct. 20	Jan. 2	Oct. 21	Dec. 22
Barometric back-pressure (lbs.)	2	14·609	14·84	14·64	14·8	14·7	14·902	14·39	14·95	14·75
Boiler pressure above atmosphere	3	60·75	75·0	82·3	96·0	104·6	113·85	128·2	134·8	130·0
Cylinder pressure (mean absolute during admission)	4	**60·98**	**72·64**	**81·09**	**89·8**	**97·75**	**109·3**	**120·62**	**128·85**	**60·95**
Ratio of expansion (corrected)	5	4·51	4·63	4·42	4·57	4·42	4·51	4·56	4·46	4·76
Point of cut-off in hp. cylinder	6	0·470	0·470	0·470	0·470	0·470	0·470	0·470	0·470	0·470
,, ,, lp. ,,	7	0·604	0·604	0·604	0·604	0·604	0·604	0·604	0·604	0·604
Duration of trial (minutes)	8	156 and 188	237	150	242	124 and 240·5	123	177	186	180
Temperature of engine-room, Fahrenheit	9	61	48	68	53	63	..	57	..	56
Mean pressure on hp. piston	10	15·6	18·233	23·05	25·58	28·7	32·1	36·0	39·6	15·05
,, ,, ,, ,, (under side)	11	3·268	4·233	5·14	5·516	6·39	7·2	8·33	8·94	3·058
,, ,, lp. ,,	12	6·056	9·041	10·91	12·975	15·5	18·7	20·22	21·8	5·80
Total mean pressure referred to lp. piston	13	**15·45**	**20·22**	**24·94**	**28·462**	**33·00**	**38·26**	**42·28**	**45·97**	**14·825**
,, ,, ,, ,, cylinder without clearance	14	15·84	20·73	25·57	29·19	33·85	39·24	43·36	47·14	15·205
Theoretical mean pressure for corrected ratio of expansion on assumption that $p^6 v^7 =$ constant	15	16·81	21·99	27·79	31·18	36·43	41·39	47·43	51·9	15·64
Percentage of theoretical mean pressure actually obtained	16	**94·0**	**94·2**	**92·0**	**93·6**	**92·9**	**94·8**	**91·4**	**90·8**	**97·2**
Revolutions per minute (mean during trial)	17	399·9	413·1	399·8	405·7	405·3	402 9	409·6	406·8	400·3
Indicated horse-power	18	**13·22**	**17·88**	**21·35**	**24·71**	**28·65**	**33·0**	**37·07**	**40·03**	**12·705**
Feed-water used per hour (lbs.)	19	438·8	510·3	537·6	608·4	653·45	705·3	798·6	830·9	410·6
,, ,, indicated horse-power-hour (lbs.)	20	**33·17**	**28·54**	**25·18**	**24·62**	**22·81**	**21·37**	**21·54**	**20·75**	**32·32**
,, per indicated horse-power-hour accounted for by hp. indicator at cut-off	21	27·7	24·19	22·61	21·36	20·45	19 34	18·9	18·9	27·35

Water theoretically required per indicated horse-power-hour for steam expanded from mean admission to exhaust temperature	22	24·93	22·2	20·78	19·58	18·7	17·65	16·81	16·32	24·95
Percentage efficiency	23	**75·15**	**77·7**	**82·5**	**79·5**	**82·0**	**82·5**	**78·0**	**78·6**	**77·1**
Lbs. of water collected per hour lp. steam chest	24	9·37	14·75	8·25	15·5	11·0	12·0	12·0	13·0	9·0
" " accounted for by hp. indicator at cut-off	25	361·8	432·5	482·8	528·0	586·1	638·3	700·8	756·3	347·5
" " " " lp. " "	26	375·3	445·5	447·4	511·4	546·5	590·5	626·0	663·9	376·4
" " " " lp. " at end of stroke	27	383·2	461·4	457·6	525·7	557·6	588·3	617·4	670·7	384·9
Lbs. water not accounted for by hp. indicator at cut-off	28	77·0	77·8	54·8	80·4	67·3	67·0	97·8	74·6	63·1
" " " " lp. " "	29	63·5	64·8	90·3	97·0	107·4	114·5	172·5	167·0	34·2
" " " " lp. " at end of stroke	30	55·6	48·9	80·1	82·7	95·8	117·0	151·1	160·2	25·7
Percentage of total feed-water missing at cut-off hp. cyl.	31	**17·49**	**15·25**	**10·1**	**13·22**	**10·2**	**9·5**	**12·25**	**8·9**	**15·36**
" " " " lp. cyl.	32	14·3	12·69	16·8	15·94	16·36	16·25	21·6	20·1	8·33
" " " " end of stroke lp. cyl.	33	12·62	9·57	14·88	13·6	14·65	16·59	18·92	19 27	6·26
Heat-units missing per hour at cut-off hp. cylinder	34	70,337	70,525	49,320	72,038	59,948	59,560	86,161	65,394	57,768
" " " stroke " hp. cylinder	35	**2·931**	**2·845**	**2·055**	**2 959**	**2·459**	**2·464**	**3·505**	**2·679**	**2·405**
Initial surface exposed in hp. cylinder square feet	36	1·771	1·771	1·771	1·771	1·771	1·771	1·771	1·771	1·771
Temperature F. during admission (mean)	37	293·5	305·0	312·0	320·0	325·5	334·0	341·0	346·0	293·5
" reached by cushion steam	38	289·0	298·0	304·0	310·0	315·5	323·0	330·0	333·0	288·0
Range of temperature to which initial surface is exposed	39	4·5	7·0	8·0	10·0	10·0	11·0	11·0	13·0	5·5
Weight of steam at mean admission pressure, lbs. per cubic foot	40	0·1442	0·1699	0·1888	0·2075	0·2241	0·2495	0·2736	0·2909	0·1442
Surface exposed during admission, square feet	41	0·799	0·799	0·799	0·799	0·799	0·799	0·799	0·799	0·799
Temperature (Fahrenheit) during exhaust	42	253·0	263·0	267·0	272·0	278·0	285·0	289·0	294·0	254·0
Range of temperature to which admission surface is exposed	43	40·5	42·0	45·0	48·0	47·5	49·0	52·0	52·0	39·5
Weight of water in cylinder which at above range of temperature would account for abstraction of heat	44	0·072	0·067	0·045	0·061	0·051	0·050	0·067	0·051	0·061
Pressure (absolute) at point of cut-off hp. cylinder	..	53·2	62·1	71·5	77·5	84·55	95·5	103·9	112·5	51·2
" " " " lp. cylinder	..	21·5	24·7	25·89	28·6	30·27	33·2	34·4	36·7	21·6
Pressure (absolute) at end of stroke hp. cylinder										
" " " " lp. cylinder	..	13·41	15·56	16·2	18·0	19·2	20·2	21·7	22·7	13·45
Mean back-pressure (absolute) hp. cylinder	..	31·9	37·1	40·0	43·3	47·16	53·6	57·1	61·5	32·0
" " " lp. cylinder	..	16·0	15·9	15·7	15·4	14·97	15·7	14·7	15·5	16·0

TABLE VIII. / TABLE IX.

Trial letter, intended ratio of expansion, and intended absolute mean admission pressure		Table VIII. $C\frac{160}{5\cdot2}$	Table VIII. $C\frac{160}{5\cdot6}$	Table VIII. $C\frac{160}{6}$	Table IX. $C\frac{130}{5\cdot6}$	Table IX. $C\frac{140}{5\cdot6}$	Table IX. $C\frac{150}{5\cdot6}$	Table IX. $C\frac{160}{5\cdot6}$
Date of trial	1	Nov. 1	Nov. 3	Nov. 5	Nov. 2	Nov. 2	Nov. 3	Nov. 3
Barometric back-pressure (lbs.)	2	14·34	14·175	14·502	14·35	14·35	14·143	14·175
Boiler pressure above atmosphere	3	162·6	165·0	165·0	135·0	145·2	155·0	165·0
Cylinder pressure (mean absolute during admission)	**4**	**155·44**	**157·29**	**158·5**	**131·1**	**141·85**	**149·84**	**157·29**
Ratio of expansion (corrected)	5	5·57	6·07	6·46	6·1	6·2	6·15	6·07
Point of cut-off in hp. cylinder	6	0·362	0·336	0·313	0·336	0·336	0·336	0·336
" " lp. "	7	0·604	0·604	0·604	0·604	0·604	0·604	0·604
Duration of trial (minutes)	8	120	179	180	182	120	180	179
Temperature of engine-room, Fahrenheit	9	72	74	67	68	64	71	74
Mean pressure on hp. piston	10	47·8	47·28	46·02	37·8	41	43·86	47·28
" " " " (under side)	11	11	7·77	7·77	5·86	8·68	7·02	7·77
" " lp. "	12	21·0	20·13	19·29	16·18	17·3	19·2	20·13
Total mean pressure referred to lp. piston	**13**	**50·226**	**47·53**	**46·04**	**38·01**	**42·02**	**44·62**	**47·53**
" " " " cylinder without clearance	14	51·51	48·74	47·22	39·16	43·09	45·76	48·74
Theoretical mean pressure for corrected ratio of expansion on assumption that $p^6 v^7 =$ constant	15	55·37	51·95	49·2	40·71	44·5	48·36	51·95
Percentage of theoretical mean pressure actually obtained	**16**	**93·03**	**93·8**	**96·0**	**96·1**	**96·8**	**94·6**	**93·8**
Revolutions per minute (mean during trial)	17	421·66	411·3	401·2	402·6	405·1	404·05	411·3
Indicated horse-power	**18**	**45·3**	**41·85**	**39·55**	**32·76**	**36·43**	**38·59**	**41·85**
Feed-water used per hour (lbs.)	19	869	801	759	654·7	714·5	750·8	801
" " indicated horse-power-hour (lbs.)	**20**	**19·18**	**19·14**	**19·19**	**20·0**	**19·64**	**19·45**	**19·14**
" per indicated horse-power-hour accounted for by hp. indicator at cut-off	21	16·67	16·35	15·92	17·13	16·7	16·76	16·35
Water theoretically required per indicated horse-power hour for steam expanded from mean admission to exhaust temperature	22	15·0	14·9	14·87	16·2	15·61	15·23	14·9
Percentage efficiency	**23**	**78·2**	**77·8**	**77 4**	**81**	**79·4**	**78·3**	**77·8**

Lbs. of water collected per hour lp. steam chest . .	24	14·5	14·5	16·5		17·25	17	15	14·5
" " accounted for by hp. indicator at cut-off	25	755	684·3	629·7		561	608·8	636·8	684·3
" " " " lp. " "	26	674·2	635·2	597·3		549·7	573·3	596·2	635·2
" " " " lp. " at end of stroke	27	707·97	650·1	610·5		571·1	589·6	595·3	650·1
Lbs. water not accounted for by hp. indicator at cut-off	28	113·7	116·7	129·3		93·5	105·7	114·0	116·7
" " " " lp. " "	29	194·88	165·8	161·7		105·0	141·2	154·4	165·8
" " " " lp. " at end of stroke	30	161·03	150·9	148·5		83·6	124·9	155·5	150·9
Per cent. of total feed-water missing at cut-off hp. cyl.	**31**	**13·25**	**14·6**	**17·2**		**14·3**	**14·7**	**15·3**	**14·6**
" " " " " lp. cyl. .	32	22·4	20·7	21·3		16·04	19·76	20·6	20·7
" " " " end of stroke lp. cyl.	33	18·54	18·84	19·55		12·76	17·49	20·69	18·84
Heat-units missing per hour at cut-off hp. cylinder. .	34	98,520	101,213	112,103		82,017	91,926	99,180	101,213
" " " stroke " hp. cylinder. .	**35**	**3·894**	**4·095**	**4·656**		**3·396**	**3·781**	**4·091**	**4·095**
Initial surface exposed in hp. cylinder, square feet . .	36	1·771	1·771	1·771		1·771	1·771	1·771	1·771
Temperature Fahrenheit during admission (mean) . .	37	361	362	362		347	353	358	362
" reached by cushion steam	38	333	329	327		319	324	326	329
Range of temperature to which initial surface is exposed	39	28	33	35		28	29	32	33
Weight of steam at mean admission pressure, lbs. per cubic foot	40	0·3476	0·3156	0·3539		0·2958	0·3187	0·3355	0·3516
Surface exposed during admission, square feet . . .	41	0·615	0·571	0·532		0·571	0·571	0·571	0·571
Temperature Fahrenheit during exhaust.	42	291	287	285		279	282	265	287
Range of temperature to which admission surface is exposed	43	70	75	77		68	71	73	75
Weight of water in cylinder which at above range of temperature would account for abstraction of heat .	44	0·055	0·054	0·06		0·049	0·053	0·056	0·054
Pressure (absolute) at point of cut-off hp. cylinder. .	..	135·5	133·1	133·7		111·1	120·0	126·1	133·1
" " " " lp. cylinder . .	..	36·0	34·6	33·4		30·9	31·8	33·2	34·6
Pressure (absolute) at end of stroke hp. cylinder . .	..	..	..	..		..	..	48·8	..
" " " " lp. cylinder . .	..	23·09	21·27	20·9		19·6	20·0	20·2	21·27
Mean back-pressure (absolute) hp. cylinder	..	58·4	55·8	53·7		48·5	51·2	53·74	55·8
" " " lp. cylinder	..	15·8	14·73	14·9		15·4	15·2	15·0	14·73

TABLE X.

Date.	No.	g.	p.	t_1.	t_2.	t_3.	w.	u.	x'.	x.	Duration of Blow in Minutes.	Remarks.
Nov. 19 . .	1	46–1–15	105	8·505	24·30	166·0	4,416	110·375	99·82	99·96	7·0	
„ 19 . .	2	44–1–4	173	23·35	39·3	188·0	4,181	109·4	96·87	96·38	7·0	Steam blown into warm water.
„ 26 . .	3	50–3–14	160	11·6	25·81	184·0	4,919	110·38	99·27	99·488	8·0	
Dec. 3 . .	4	50–3–16	139	12·00	25·82	178·0	4,921	110·2	96·29	96·46	4·0	Steam blown in very fast. Water in boiler high.
„ 3 . .	5	51–3–15	154	15·6	29·4	183·0	5,032	109·97	99·66	99·76	15·0	
„ 12 . .	6	51–2–14	162	9·10	23·32	185·0	4,947	111·1	98·63	98·93	4·5	Water in boiler low.
„ 22 . .	7	53–3–22	178	9·827	23·501	188·8	5,263	112·0	100·4	100·72	12·0	
Feb. 15 . .	8	50–2–21	65	10·77	25·19	147·8	4,898	112·0	100·8	100·48	9·0	
„ 25 . .	9	50–0–15	127	8·18	22·73	174·0	4,836	110·8	99·63	99·873	9·0	

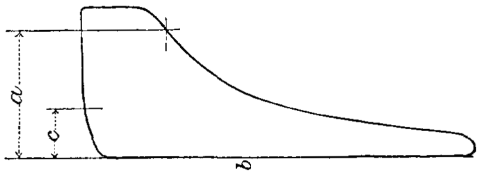
CALCULATIONS.

Point of cut-off . . . 0·216. Cylinder lp.

Trial. . . . . S $\frac{110}{4\cdot4}$ (100 revolutions).

Date . . . . Dec. 5, 1887.

FIG. 9.



X = barometric pressure . . . . = 14·74 lbs.

a = pressure above atmosphere at point of cut-off . . mean of 7 diagrams . . . . = 78·35 „

b = pressure above atmosphere at middle of exhaust stroke, mean of 7 diagrams . . . . = 0·77 „

c = pressure above atmosphere at point of admission, mean of 7 diagrams . . . . = 30·58 „

$A = X + a$ . . . . = 93·1 „

$B = X + b$ . . . . = 15·5 „

$C = X + c$ . . . . = 45·3 „

w^1 = weight of steam per cubic foot at A pressure . . = 0·2146

$w^1 - w^2$ = 0·1743

w^2 = „ „ „ B „ . . = 0·0403

$w^1 - w^3$ = 0·1054

w^3 = „ „ „ C „ . . = 0·1092

W^1 = „ „ accounted for by volume swept by piston

$$= \frac{\text{point of cut-off} \times 6 \times \text{R per hour} \times \text{area} \times w^1}{1{,}728} = \frac{0\cdot216 \times 6 \times 8{,}280 \times 141\cdot34 \times 0\cdot2146}{1{,}728} = 188\cdot3 \text{ lbs.}$$

W^2 = weight of steam accounted for by trunk clearance

$$= \frac{\text{capacity of trunk clearance} \times \text{R per hour} \times (w^1 - w^2)}{1{,}728} = \frac{26 \times 8{,}280 \times 0\cdot1743}{1{,}728} \;.\;.\;.\;.\;.\; = 21\cdot7 \text{ „}$$

W^3 = weight of steam accounted for by cylinder clearance

$$= \frac{\text{capacity of cylinder clearance} \times \text{R per hour} \times (w^1 - w^3)}{1{,}728} = \frac{33\cdot6 \times 8{,}280 \times 0\cdot1054}{1{,}728} \;.\;.\;.\;.\;.\; = 16\cdot9 \text{ „}$$

S = total weight of steam accounted for by indicator $= W^1 + W^2 + W^3$ . . . . = 226·9 „

y = feed-water used per hour . . . . = 408·8 „

$\frac{S}{y}$ = percentage of water accounted for by indicator . = 55·5

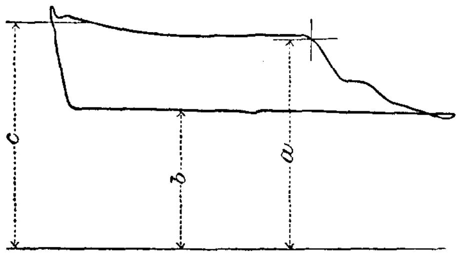
Water missing . . { 181·9 lbs.
44·5 per cent.

CALCULATIONS.

Point of cut-off . . . 0·647. Cylinder h.hp.

Trial . . . . $T\frac{150}{6}$.
Date . . . . Dec. 19, 1887.

FIG. 10.



X = barometric pressure . . . . = 14·5 lbs.

a = pressure above atmosphere at point of cut-off, mean of 12 diagrams . . . . = 131·56 „

b = pressure above atmosphere at middle of exhaust stroke, mean of 12 diagrams . . . . = 88·9 „

c = pressure above atmosphere at point of admission, mean of 12 diagrams . . . . = 142·03 „

$A = X + a$ . . . . = 146·1 „

$B = X + b$ . . . . = 103·4 „

$C = X + c$ . . . . = 156·5 „

w^1 = weight of steam per cubic foot at A pressure . . = 0·3276 „

$w^1 - w^2$ = 0·0908

w^2 = „ „ „ B „ . . = 0·2368

$w^1 - w^3$ = −0·0223

w^3 = „ „ „ C „ . . = 0·3499

W^1 = „ „ accounted for by volume swept by piston $= \dfrac{\text{point of cut-off} \times 6 \times \text{R per hour} \times \text{area} \times w^1}{1{,}728}$
$= \dfrac{0·647 \times 6 \times 24{,}541 \times 34·5 \times 0·3276}{1{,}728}$ = 623·0 lbs.

W^2 = weight of steam accounted for by trunk clearance $= \dfrac{\text{capacity of trunk clearance} \times \text{R per hour} \times (w^1 - w^2)}{1{,}728}$
$= \dfrac{11·1 \times 24{,}541 \times 0·0908}{1{,}728}$ = 14·3 „

W^3 = weight of steam accounted for by cylinder clearance $= \dfrac{\text{capacity of cylinder clearance} \times \text{R per hour} \times (w^1 - w^3)}{1{,}728}$
$= \dfrac{14·8 \times 24{,}541 \times -\ 0·0223}{1{,}728}$ = −4·7 „

S = total weight of steam accounted for by indicator $= W^1 + W^2 + W^3$ . . . . = 632·6 „

y = feed-water used per hour . . . . = 669·2 „

$\dfrac{S}{y}$ = percentage of water accounted for by indicator . = 94·54

Water missing . . { 36·6 lbs.
5·46 per cent.

[DISCUSSION.

TRIALS OF A NON-CONDENSING STEAM-ENGINE.

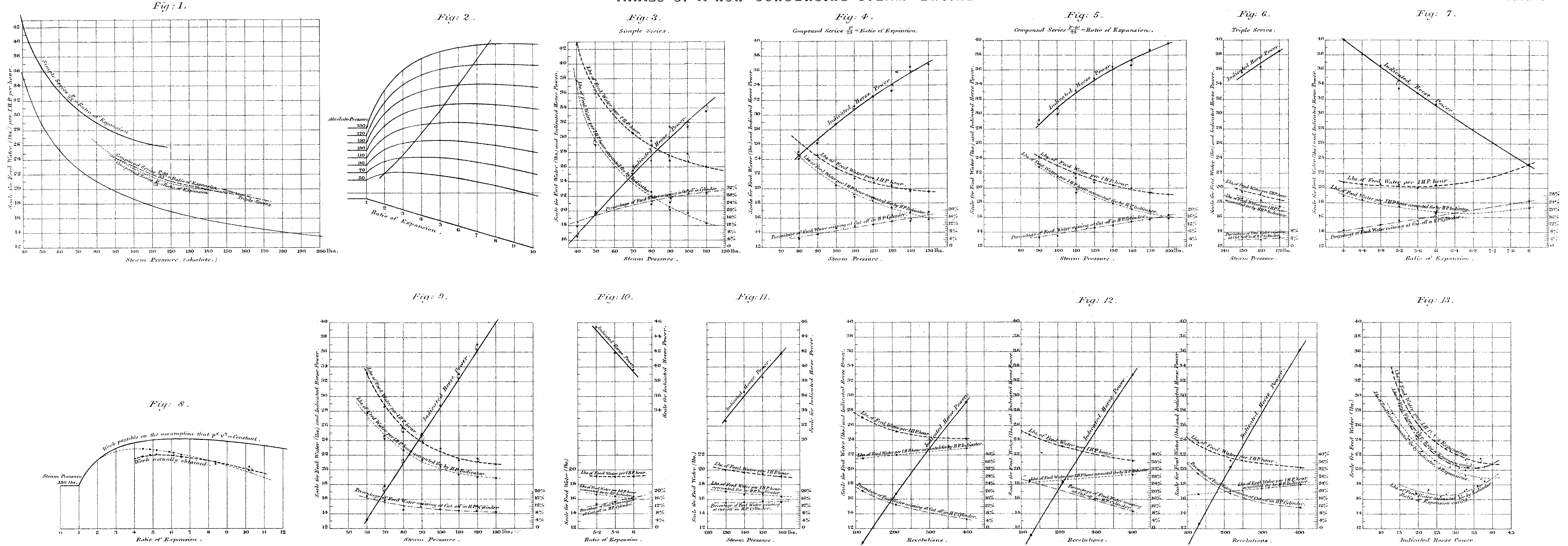


Fig: 1.

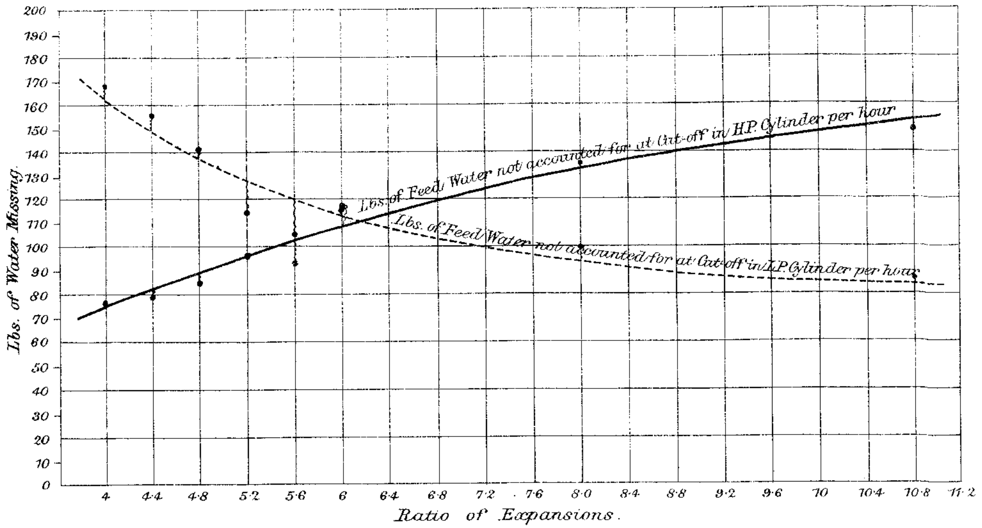


Fig: 2.

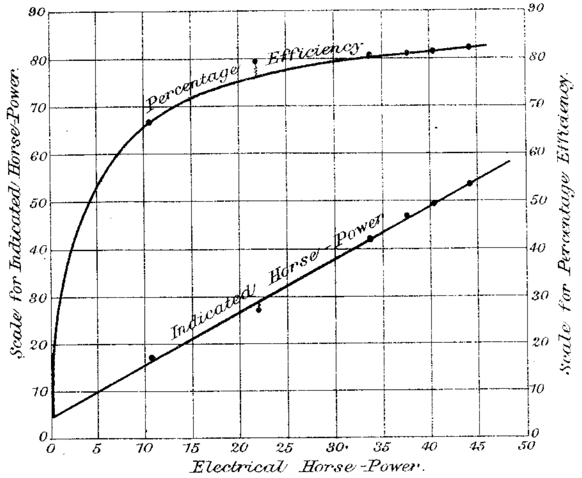
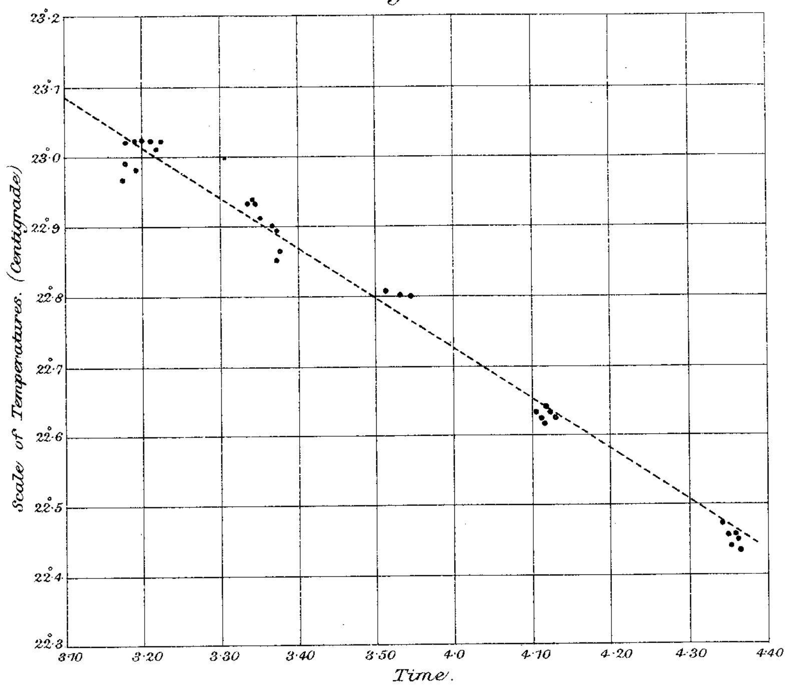


Fig: 3.



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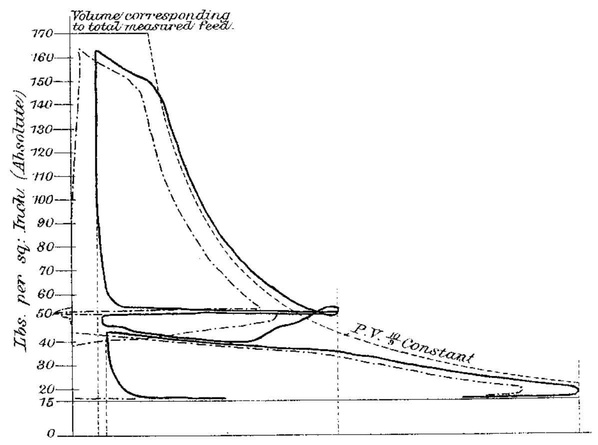


Fig: 5.

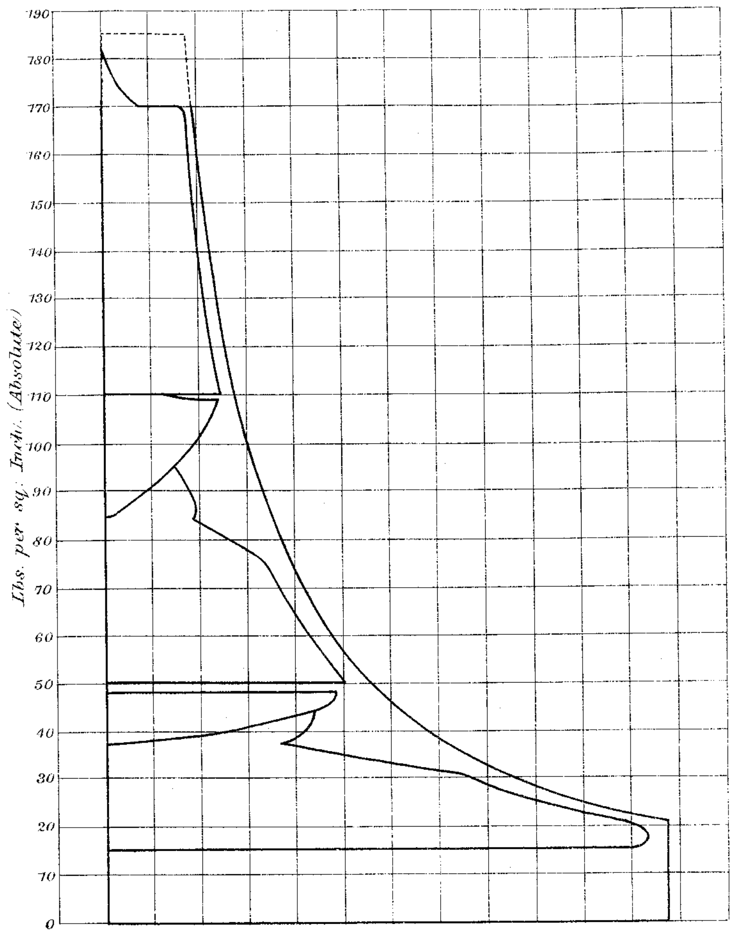
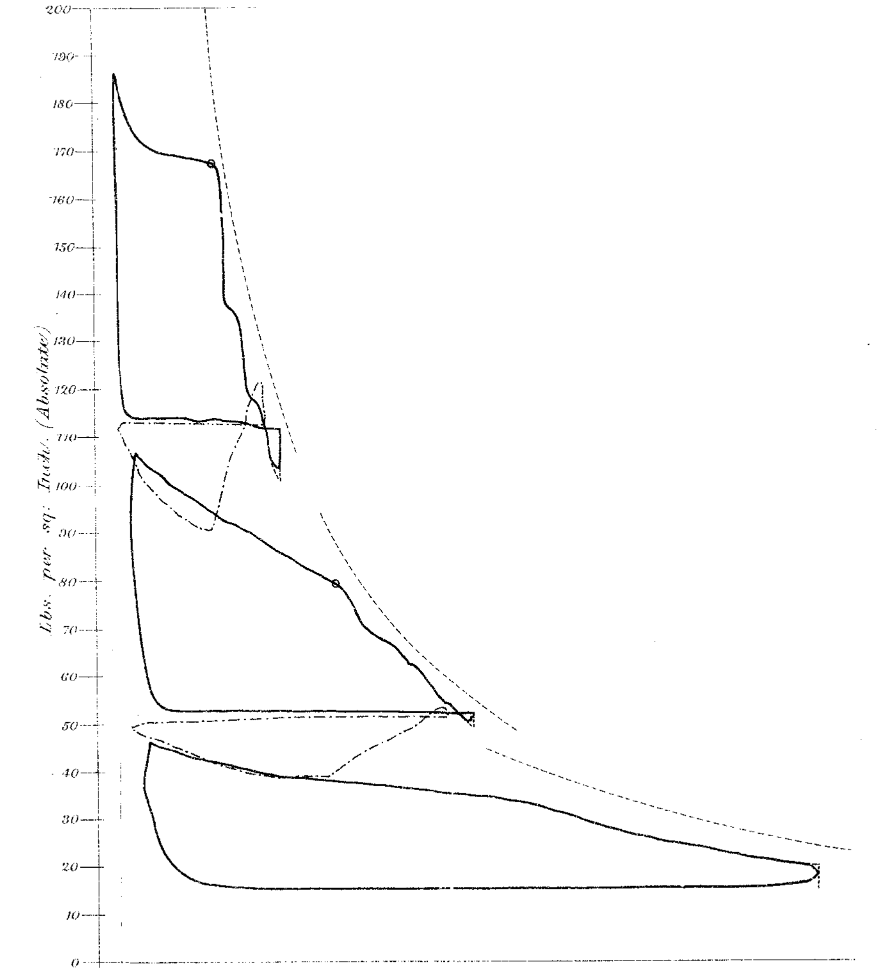


Fig: 6.



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