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JOHN CLARKE HAWKSHAW, M.A., Member of Council, in the Chair.

(*Paper No. 3024.*)

"On the Law of Condensation of Steam deduced from Measurements of Temperature-Cycles of the Walls and Steam in the Cylinder of a Steam-Engine."

By HUGH LONGBOURNE CALLENDAR, M.A., F.R.S., and JOHN THOMAS NICOLSON, B.Sc.

IN the discussion of steam-engine trials, it has been customary to compare the measured cylinder-feed with the quantity of steam indicated by the cards, with a view to deduce the so-called "missing quantity" of steam, and to arrive at the consequent loss of available energy resulting from condensation and re-evaporation. The question of cylinder condensation has also been elaborately discussed from a theoretical standpoint, on the assumption that the temperature-cycle of the metal surface is the same as that of the steam.

The object of the experiments recorded in this Paper, which were carried out in the summer of 1895, in the thermodynamic laboratory of the McDonald Engineering Building, at the McGill University, was the measurement of the cyclical interchange of heat between the walls and the steam. From a knowledge of the wall temperature-cycle, the amount of condensation taking place at a particular point under definite conditions can be readily ascertained, and the nature of the action can be studied separately, without confusion with other effects which are taking place in other parts of the cylinder at the same time. The mean temperature of the walls in different parts of the cylinder was also measured. A comparison of the results leads to a simple relation between the mean wall temperature, speed and condensation, which appears to hold through a considerable range of conditions.

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The engine used in the experiments, Figs. 1, Plate 6, was made by the Robb Engineering Company, of Amherst, N.S. It has a large slide-valve with double ports, and a relief-back of the Porter type; and is designed to run at a speed of 250 revolutions per minute. In ordinary working, the speed is regulated by the governor on the fly-wheel, which automatically varies the travel and angular advance of the valve, in proportion to the load. For these experiments the governor was disconnected, and the valve-eccentric set to give the cut-off desired in each experiment. The principal dimensions are: Stroke, 12 inches; diameter of cylinder, 10·5 inches; piston displacement, 0·601 cubic foot; clearance volume, 0·060 cubic foot; slide-valve, 10·74 inches by 13·5 inches; steam-ports, 9·5 inches by 1·5 inch. In the present trials the engine was made single-acting by the addition of a brass lap 1·19 inch wide to the crank end.

CYLINDER-WALL TEMPERATURE-CYCLES.

The measurement of cylinder temperatures by means of mercury thermometers has been carried by Mr. Bryan Donkin[1] to a high degree of refinement. From a careful study of those experiments it was concluded that it would not be possible to obtain any more definitely quantitative results by the use of mercurial thermometers. The more laborious, but at the same time more powerful and pliable, methods of electrical thermometry were therefore adopted.

The Thermo-Electric Method.—This method has been applied by Professor E. Hall, of Harvard College;[2] but the Authors' results, obtained by a modification of that method, differ so considerably from his, that it is necessary to explain somewhat fully the points of difference between the two methods. In a thermo-electric circuit, the metal between the hot and cold junctions should be of continuously uniform quality, since different specimens even of the same metal are known to differ materially in thermo-electric power, and it was found necessary to take the greatest precautions to secure the most uniform quality in the materials used for the circuit. In the Hall method of inserting the thermo-junction, it would be a matter of some difficulty to secure absolute freedom from leakage. It is stated that leakage was noticed on several occasions, and it is evident that the intermittent rushes of steam

[1] Minutes of Proceedings Inst. C.E., vol. cvi. p. 264.

[2] Transactions of the American Institute of Electrical Engineers, 1891, vol. viii. p. 236.

through and about the junction might entirely change the temperature conditions. In order to avoid this source of trouble as far as possible, the junction swere made with the cast iron of the cylinder itself, holes being bored in the solid metal to within a carefully measured distance of the inner surface. This method has the further advantage of not introducing any foreign material into the thermo-electric circuit.

In measuring the cyclical variations of temperature, the conditions of running should be the same when the observations are being taken at different points of the stroke, and at different depths in the metal. It is necessary to keep the engine running as steadily as possible, and to take the observations of the cycle in rapid succession. It is also important to take the indicator diagrams and the other observations of temperature simultaneously with the wall-cycle. A special cover was therefore cast for the cylinder, and fitted with eight thermo-couples at different depths. Any one or any two of these junctions could be connected to the galvanometer and the external cold junction, by a suitable arrangement of mercury-cups, and a contact-maker could be set to close the galvanometer circuit at any point of the stroke, and for any desired fraction of a revolution.

Since the observations had to be taken in rapid succession, the well-known compensation method of Poggendorf, in which the electromotive force of the thermo-couple is balanced against that of a steady current flowing in a uniform potentiometer wire, was adopted. The paraffin bath containing the external junction was maintained constantly at or very near 212° F., by a jacket of steam at atmospheric pressure. With a contact duration of only $\frac{1}{30}$ revolution, it was possible to read the galvanometer to nearly one-tenth of a degree.

Cycle Contact-Maker.—The cycle contact-maker, Figs. 2, Plate 6, closed the galvanometer circuit for a small fraction of a revolution at any desired point of the cycle. A pair of insulated revolving brushes were connected by a copper wire. One brush made contact with a central copper tube, the other with a number of copper sectors of different lengths let into the circumference of a wooden disk. A galvanometer circuit could thus be closed by any of these sectors, so as to alter the duration of the contact. The point of the cycle at which the brush made contact was determined by the setting of a divided circle. The contact-maker could be inserted in the circuit of either of two galvanometers, according as it was desired to observe a steam-cycle with the platinum thermometer, or a wall-cycle with one of the thermo-couples.

Experimental Cylinder Covers.—The cylinder-cover was provided with a steam-tight jacket, Figs. 3, Plate 6. Eight holes for the junction wires, arranged in a circle of 1·5 inch radius round the centre of the cover, were bored to depths of 0·01 inch, 0·02 inch, 0·04 inch, 0·08 inch, 0·16 inch, 0·32 inch, 0·64 inch, respectively, measured from the inner surface of the cover. The test wires were insulated with mica and india-rubber tube and washers, except at the bottom of the holes, where they made contact with the cast iron of the cover. In passing through the jacket, they were protected by steel tubes screwed and soldered into the metal of the cover. The central tube, seen projecting from the cylinder inside the jacket, was for clearance of the piston steam-thermometer to be subsequently described.

The Thermo-Couples.—Commercially pure nickel wire was first tried for the thermo-junctions, but the electromotive force of this impure nickel with cast iron was found to be three times less than with wrought-iron wire. This implied that pure iron wire would make with cast iron a much better thermo-junction than German silver, or even than pure nickel. It was, therefore, decided to use wrought-iron wire for the thermo-couples. Particular care was exercised in the choice of the materials. The iron wire used was from the same hank, and was carefully annealed after being bent into position. The cast-iron connections for the external junction were made by planing rods out of the middle of a 4-inch bar, cast at the same time and from the same ladle as the cylinder cover itself.

From experiments on the cylinder, the formula for E, the electromotive force of the cast-iron and wrought-iron junction, one of the junctions being at $t°$ C., and the other at 100° C., was found to be:—

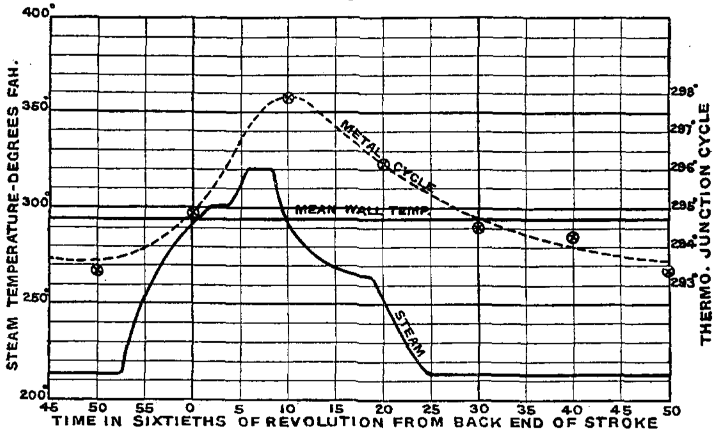
$$E = 1692 - 17 \cdot 86\, t + 0 \cdot 0094\, t^2, \text{ in microvolts.}$$

As a general rule, the thermo-couples were applied chiefly to the measurement of small differences of temperature at a known mean temperature $t°$ C. It was sufficient to use the simpler formula for the change of electromotive force per degree, namely:—

$$\frac{d\,E}{d\,t} = 17 \cdot 90 - 0 \cdot 0190\, t, \text{ microvolts per } 1° \text{ C.}$$

Observations of Wall-Cycles in the Cover.—The first observation of a wall-cycle is shown in *Fig. 5.* The full curve, corresponding to the scale on the left, gives the steam temperatures as deduced from the indicator-diagram taken simultaneously. The dotted curve, corresponding to the scale on the right, gives the variations of temperature observed at a depth of 0·01 inch in the cover, at a speed of 100 revolutions per minute. The observations are shown at (×). The cycle shows a range of only 4·3° F., corresponding to an absorp-

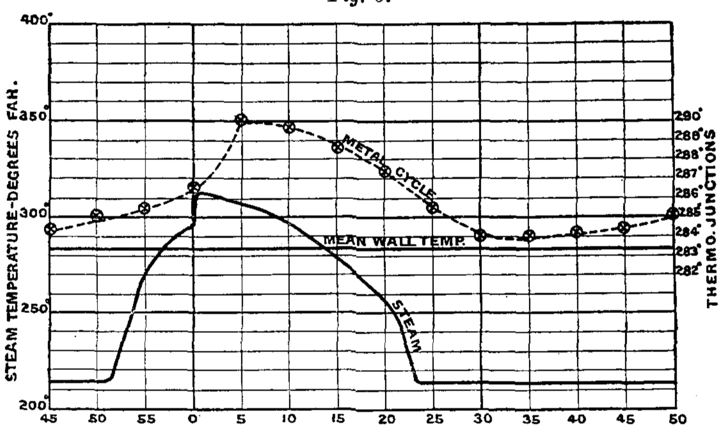
Fig. 5.



Speed, 100 revolutions per minute.

METAL CYCLE AT $\frac{1}{100}$ INCH COVER.

Fig. 6.



Speed, 46 revolutions per minute; throttled.

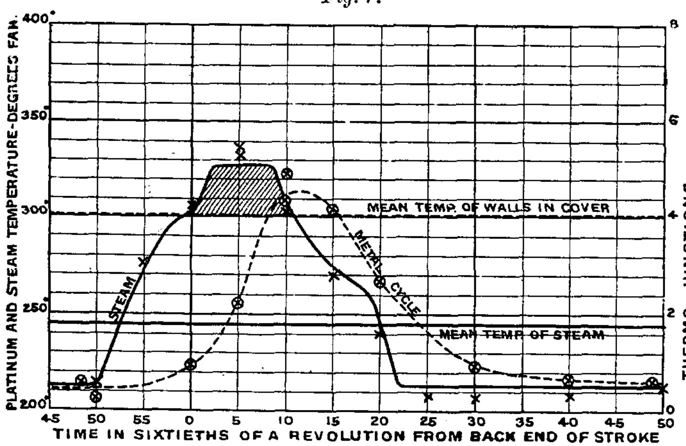
METAL CYCLE AT $\frac{1}{25}$ INCH COVER.

tion of 1 thermal unit F. per square foot per cycle, or a condensation of about $\frac{1}{1000}$ lb. of steam. The same observation was repeated, when a range of about 4° F. was again found.

In taking the observations for the wall-cycle, shown in *Fig. 6*, the differential method was adopted for the first time, which accounts for the greater smoothness of the curve.

In this method the thermo-electric circuit consisted simply of a small portion of the cast iron of the cylinder and of two iron wires making contact at different depths in the metal, at points which are at nearly the same mean temperature. Only a small difference of temperature, for which the thermo-electric method is admirably suited, is therefore observed. By this method, nearly all the troubles that otherwise arise from the slow changes

Fig. 7.



Speed, 73·4 revolutions per minute.

METAL CYCLE AT $\frac{1}{25}$ INCH COVER.

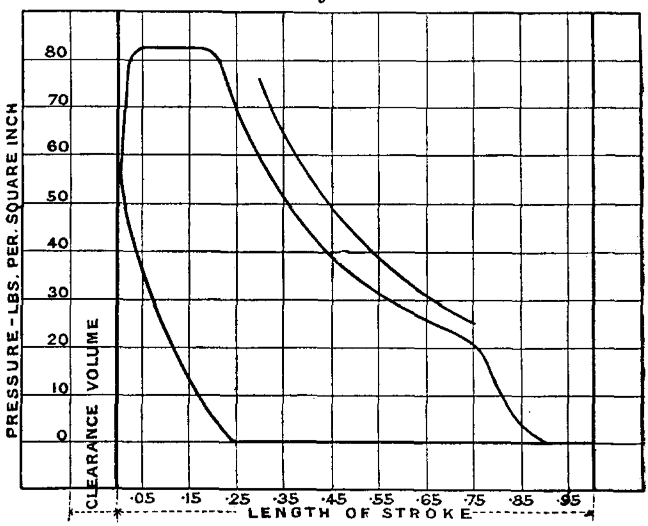
of conditions always in progress are avoided, since a gradual change of temperature will affect the metal at a depth of ½ inch to the same extent as the surface of the metal; whereas the rapid cyclical changes are practically evanescent at the greater depth. For the remainder of the experiments the differential method was always used for the wall-cycles. The cycle shown in *Fig. 7* is an illustration of an observation taken about a month later, at a depth of 0·039 inch, and a speed of 73·4 revolutions per minute.

The shaded area of the steam curve, above the line representing the mean temperature of the cover, may be called the "Condensation Area," as it appears to determine the amount of condensation taking place. For the cycles shown in *Figs. 5* and *6*, the test-wires were pressed against the metal of the cover. For

Fig. 7 and subsequent observations, the apparatus had been entirely rearranged. The holes had been bored and measured afresh, new wires of different lengths had been fitted, and all the joints and thermo-junctions had been carefully soldered with pure tin. This insured a good and permanent contact, but otherwise no difference in the observed temperature cycles could be detected. *Fig. 8* shows an average card belonging to this trial, from which the cycle curve of *Fig. 7* was deduced. Several trials were run with this valve-setting.

Wall-Cycles on Barrel-Surface at Side.—A set of eighteen thermo-

Fig. 8.



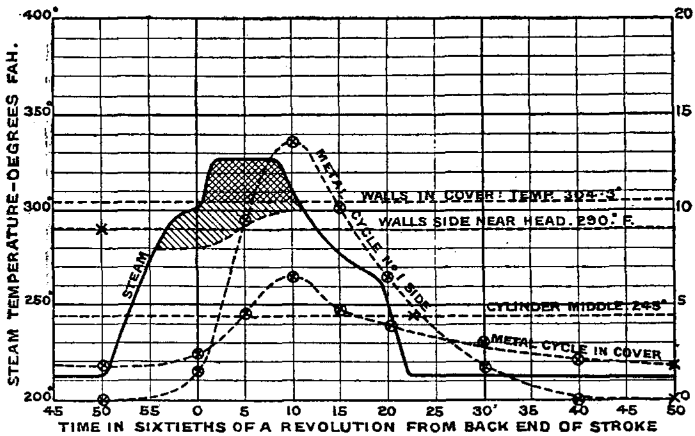
Speed, 73·4 revolutions per minute; cut-off 0·200.

INDICATOR DIAGRAM.

junctions were fitted along the side of the cylinder to observe the wall-cycles and the distribution of temperature along the barrel-surface. The arrangement of the side junctions is shown in Figs. 4, Plate 6. To facilitate making and changing connections, the majority of the test-wires were connected to mercury-cups in a slip of wood fixed to the cylinder inside the cast-iron lagging. At the back end of the stroke, and at 4 inches, 6 inches and 12 inches along the side from it, pairs of junctions were inserted, one bored to leave 0·04 inch of metal, and the other ½ inch at each point. The remainder of the junctions, at 2 inches, 8 inches, 10 inches, 14 inches and 16 inches respectively, were holes bored to the depth of ½ inch, into which iron wires could be inserted for observing the length-distribution of temperature. There were four similar holes at points above and below on a line round the middle of the cylinder. Three vertical holes, 2 inches deep, were also bored in the metal of the side, at distances of 1 inch, 7·5 inches and 15 inches along the stroke, for the insertion of mercury or platinum thermometers, for calibrating the side junctions, and for observing the mean wall temperatures.

Comparison of Cycles on Cover and Side.—The curves in *Fig. 9* illustrate cycles observed in the same trial. The mean temperature of the side walls opposite the clearance space was 14·3° F. lower than that of the cover. Junction No. 1, side, was situated at this

Fig. 9.



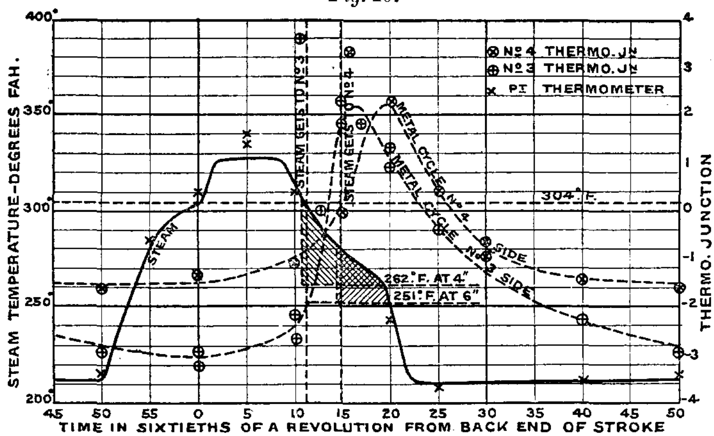
Speed, 43·9 revolutions per minute.

METAL CYCLES $\frac{1}{25}$ INCH IN COVER AND No. 1 SIDE.

point at the back end of the stroke, and was exposed to the same steam as the cover. The ranges observed, at the cover and side, were 4·9° F. and 13·5° F. respectively; the speed being 44 revolutions per minute. The depths of the junctions were, cover, 0·039 inch, and side, 0·037 inch respectively. This particular cycle at the side was the largest observed throughout the trials. It corresponds to a range of about 20° F. at the surface of the metal. The "condensation area" for each junction is shaded as in the previous example. It will be observed that the temperature of the wall begins to rise near the end of compression, shortly after condensation begins, and reaches a maximum shortly after cut-off near the end of the condensation period. The fall due to re-evaporation is nearly as rapid as the rise. In drawing the lower boundary of the condensation area for this cycle, the probable surface temperature is taken instead of the mean; but the difference of area is small. The magnitude of the condensation area would be little affected by taking the mean temperature instead of the surface temperature, but the cycle curve would not then correspond so well, allowing for lag, with the probable rate and epoch of condensation.

Side-Wall Cycles beyond Cut off.—Fig. 10 gives an example, on a temperature scale two and a half times as large, of cycles observed on junctions Nos. 3 and 4, at 4 inches and 6 inches along the cylinder from the back end. The lower observations at the points

Fig. 10.



Speed, 45·6 revolutions per minute.

Metal Cycles in Nos. 3 and 4 Side.

0, 10, 15 and 20, for No. 3, were taken after the observation of cycle No. 4, and illustrate the order of accuracy attainable in the measurement of these cycles. It will be noticed that the temperature begins to rise at each point before steam reaches it, if the piston is assumed to fit accurately. This may be explained partly by the friction of the hot piston, partly by a probable small leakage of steam, and partly by the fact that the piston-rings are about $\frac{1}{2}$ inch from the face of the piston. In estimating the condensation areas for these junctions, $\frac{1}{2}$ inch is allowed for the imperfect fit of the piston; that is to say, it is assumed that the steam reaches the junction when the face of the piston is $\frac{1}{2}$ inch behind the corresponding point of the stroke. The thickness of metal at these junctions was 0·037 inch and 0·039 inch respectively, and is probably accurate to $\frac{1}{500}$. The speed was 45·6 revolutions per minute.

Effect of Later Cut-off.—A few trials only were run at a later cut-off—three at one-third, and two at one-half. The data of these trials were not in general sufficiently complete to afford a satisfactory basis of comparison. With a later cut-off the range of the wall-cycles observed on the cover did not differ materially from those previously recorded. The longer steam contact was compensated by a higher wall temperature. On the side wall, near the back end of the cylinder, the rise of temperature was nearly 20° F. at one-half cut-off as compared with one-fifth. The range of the cycle was reduced from 11·0° F. to 9·2° F., at a depth of 0·037 inch in the metal, and a speed of 49 revolutions per minute. At 4 inches and 6 inches along the cylinder the ranges of the wall-cycles, at the same depth and cut-off, were increased to 7·2° F. and 5·0° F. respectively, as the full-pressure steam reached these points of the wall. The temperature of the middle of the cylinder was raised nearly 30° F. as compared with one-fifth cut-off, but the comparison could not be made quite satisfactorily owing to slightly different conditions of steam pressure. The total condensation, including the later portions of the stroke, was probably increased somewhat, but at the same time the cylinder feed was more than doubled.

Temperature Distribution and Steady Heat Flow.

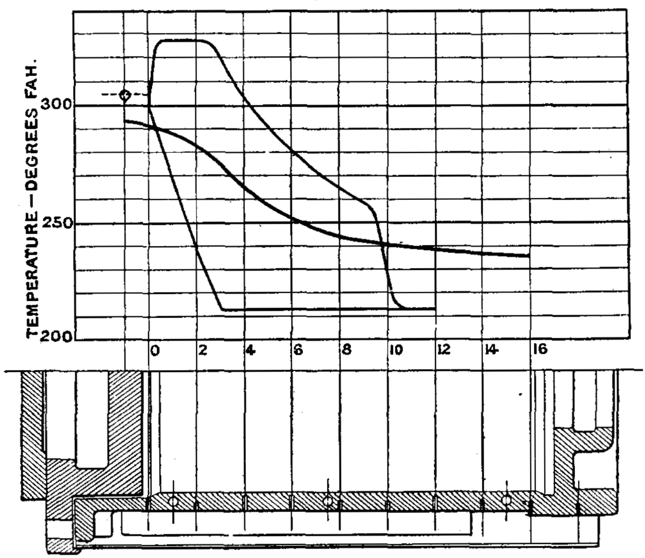
Outward Temperature Gradients.—Careful measurements of the temperature gradients were made in various parts of the cylinder with a view to deduce the steady flow of heat. From the mean of several observations in the thickness of the cover, which was protected by an air-jacket, a probable gradient of 0·55° F. per inch was deduced, a value which corresponds fairly well with the probable external loss. At the points 4 inches and 6 inches along the side, the temperature of the inner surface was 2·4° and 1·1° F. respectively lower than that of the outer surface. This curious and at first sight paradoxical result, means that the cylinder wall at these points was losing heat to the steam.

Temperature Gradients on Barrel Surface.—The longitudinal distribution of temperature for one-fifth cut-off, single-acting, is shown in *Fig. 11*, together with a corresponding diagram of steam temperatures deduced from the average indicator-diagram, with a mean speed of 45·6 revolutions per minute.

On several occasions special experiments showed that a change of speed produced little change in the distribution of temperature. The mean temperature at the centre of the cover in the same trials is shown by the mark (–), at 305° F., opposite the section of the cover.

Effect of Longitudinal Conduction, and Piston Convection.—The greater part of the condensation on the barrel-admission surface in a small single-acting engine is probably due to the lowering of temperature caused by conduction and convection along the cylinder. In this set of observations the maximum gradient of 9·3° F. per inch occurs at a point a little after cut-off, and corresponds to a loss of heat by the admission surface of 11·2

Fig. 11.



SECTION OF CYLINDER AND COVER, SHOWING POSITION OF THERMO-JUNCTIONS AND THERMOMETERS.

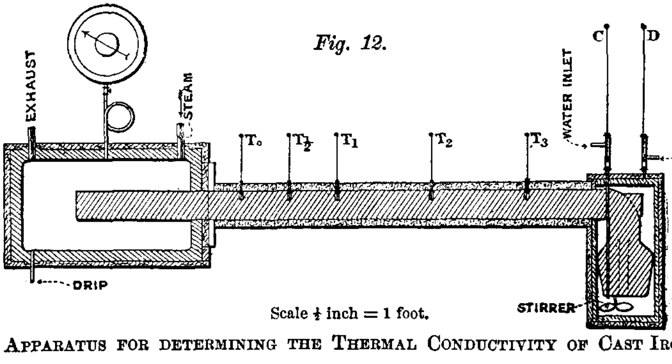
thermal units per minute. The cyclical convection of heat by the piston is a factor which may be of some importance in a single-acting engine. It will depend on the surface of the piston, on the closeness of the contact, and on the difference of temperature between the ends of the cylinder. In the present case the mean difference of temperature was 45° F. at one-fifth cut-off, and nearly 65° F. at one-half cut-off. The curved surface of the piston was approximately 1 square foot. The effect of piston convection may be traced in the form of the cycles observed at one of the side junctions. In a double-acting engine, where both ends of the cylinder are at nearly the same temperature, the effect of piston convection would be of very little importance. In a high-speed single-acting engine like the Willans engine, it is probable that piston convection might be the most important factor in cooling the barrel-admission surface, and causing initial condensation.

Abstraction of Heat by the Condensation of Wet Steam.—For a distance of nearly 6 inches along the side, while the engine was running at one-fifth cut-off, the external surface was nearly 1° F. on the average hotter than the internal. Heat, supplied by conduction, was being abstracted from the inner surface at a rate of at least 5 thermal units per square foot per minute. On stopping the engine the temperature along this belt immediately began to rise, and continued rising for some minutes. The heat abstracted by evaporation is greater than that supplied by condensation over this part of the surface. The probable explanation is to be found in the wetness of the steam due to adiabatic expansion. (See p. 185.)

Conductivity and Specific Heat of Cast Iron.

The electrical resistance of cast iron was found to be nearly ten times greater than that of wrought iron. Considerable difference

Fig. 12.



Apparatus for determining the Thermal Conductivity of Cast Iron.

in the thermal conductivity was therefore to be expected. Observations on the cylinder showed that the conductivity of cast iron was probably some 30 per cent. less than the value generally assumed. It was therefore desirable to attempt a special determination of this important constant by the most accurate methods. With this object the apparatus shown in *Fig. 12* was designed. The metal used was a 4-inch bar of iron, cast from the same ladle as the cylinder cover.

Two independent methods were employed—(1) The calorimetric method, in which the quantity of heat transmitted is directly measured; and (2) that of Ångström, which depends on observation of the propagation of temperature waves.

The Calorimetric Method.—In terms of the units employed, the result of this method may be stated as follows:—The quantity of heat conducted across a plate of cast iron 1 foot square and 1 inch thick, for a difference of temperature of 1° F. between its faces, amounts to 5·65 thermal units (pound-degree-F.) per minute, the plate being at a temperature of 104° F. (The value generally assumed for wrought iron is 7·5 in the same units.) Expressed in terms of C.G.S. units centigrade, the values become, cast iron, 0·117, wrought iron, 0·155.

The Ångström Method.—The diffusivity or thermometric conductivity is the ratio of the calorimetric conductivity k to the thermal capacity c of unit volume. The unit volume was taken to be a plate 1 foot square and 1 inch thick, in order to harmonize with the units employed for the calorimetric conductivity. Observations were taken at mean temperatures of 215° F. and 130° F., from which was derived for the probable variation of the diffusivity with temperature the formula:

$$\frac{k}{c} = 1{\cdot}42 - 0{\cdot}0010\,t,$$

where t is the temperature in degrees Fahrenheit.

Specific Heat of Cast-Iron.—From a specially devised series of experiments there was deduced for the specific heat s of the specimen of cast iron at a temperature $t°$ F., the formula:

$$s = 0{\cdot}1090 + 0{\cdot}000060\,t,$$

which is probably correct to 1 per cent. between 200° and 350° F. Combining this with the result previously given for the diffusivity, the following values for the thermal capacity c of a plate 1 foot square and 1 inch thick, and for the calorimetric conductivity k are obtained:—

TABLE I.—CONDUCTIVITY AND SPECIFIC HEAT OF CAST IRON.

Temperature.	Specific Heat s.	Thermal Capacity c.	Diffusivity $\frac{k}{c}$.	Conductivity k.
° F.				
100	0·115	4·21	1·32	5·55
150	0·118	4·32	1·27	5·48
200	0·121	4·43	1·22	5·40
250	0·124	4·54	1·17	5·31
300	0·127	4·65	1·12	5·21
350	0·130	4·76	1·07	5·10

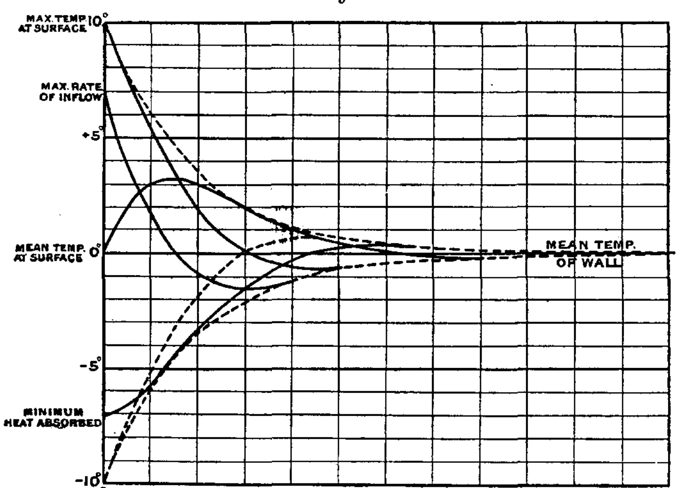
Density, Thermal Capacity and Composition of the Bar.—The density of the bar gives a weight of 36·6 lbs. for a plate 1 foot square and 1 inch thick. The values for c given in Table I, are found by multiplying the specific heat s by 36·6. It will be observed that the values of the thermal constants of cast iron vary to a considerable extent with the temperature. For calculating the Tables it is necessary to assume certain average values of these constants and the numbers, $k = 5·4$, $c = 4·5$, $\frac{k}{c} = 1·20$ are taken, which correspond to a temperature of 220° F. The composition of the bar is shown by the appended chemical analysis.

	Per cent.
Total carbon	3·08
Graphite . .	2·86
Silicon . .	2·89
Phosphorus .	1·05
Manganese .	0·85
Sulphur . .	0·022
Total .	7·89

CONDENSATION.

Cyclical Heat Absorption.—The curves given in *Fig. 13* illustrate

Fig. 13.



Speed, 42 revolutions per minute; range, 20° at surface.

TEMPERATURE DEPTH CURVES; CAST-IRON CYLINDER.

the cyclical absorption of heat by the metal of a cast-iron cylinder at a speed of 42 revolutions per minute, for a simple harmonic variation of surface temperature with a range of 20°. The full curves show the simultaneous values of the temperature at different depths for four typical points of the cycle. The heat absorbed or rejected by the metal between any two points of the cycle is proportional to the area included between the corresponding temperature-depth curves. The dotted boundary curves have the equation $t = \pm\ e^{-mx}$, and show the rate of diminution of the range of temperature t with increase of depth x. The index coefficient m is given by the formula $m = \sqrt{\frac{\pi n c}{k}}$, where n is the number of revolutions per minute. In the case figured, $m = 10 \cdot 5$. The wave-length of the temperature-oscillation is $\frac{6}{10}$ inch. At this depth the retardation amounts to one complete period, and the range is reduced to less than one five-hundredth part of its value at the surface. The wave-length in each case may thus conveniently be regarded as the practical limit of penetration of the heat-waves.

Numerical Values for Cast Iron.—The following Table has been calculated for a cast-iron cylinder, with a simple-harmonic cycle of 10° F. range at the surface, at various speeds, assuming the values $k = 5 \cdot 4$, $c = 4 \cdot 5$, $\frac{k}{c} = 1 \cdot 20$, in the units given. The values for any other range are directly proportional to the range. For calculating the various columns the numerical formulas used are:—

Index coefficient $m = 1 \cdot 618\ \sqrt{n}$. Wave-length $= \frac{6 \cdot 28}{m}$.

Temperature range at 0·040 inch depth $= 10° \times e^{-0 \cdot 040\, m}$.

Heat absorbed in thermal units Fahr. per square foot per cycle $= 10° \times \frac{3 \cdot 18}{m}$.

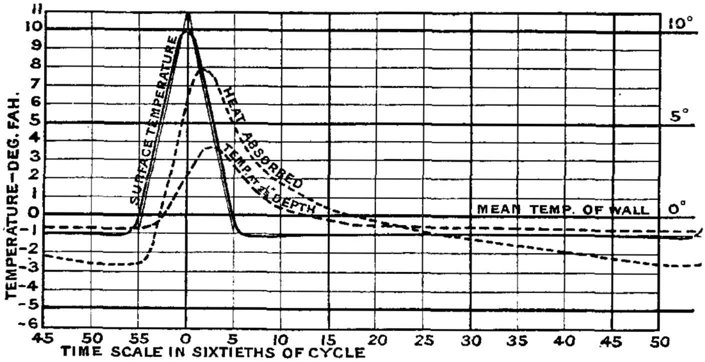
TABLE II.—CYCLICAL HEAT ABSORPTION AT DIFFERENT SPEEDS IN CAST IRON.

Revolutions per Minute.	Index Coefficient.	Wave-Length (Penetration).	Temperature Range at 0·040 inch.	Heat Absorbed in Thermal Units per Square Foot.	
n.	m.	Inch.	Per 10° Surface.	Per Revolution.	Per Minute.
25	8·09	0·777	7·25	3·93	98
40	10·24	0·613	6·63	3·11	124
50	11·44	0·549	6·32	2·78	139
60	12·54	0·501	6·05	2·53	152
70	13·54	0·464	5·82	2·35	164
80	14·47	0·434	5·60	2·20	176
90	15·35	0·409	5·41	2·07	187
100	16·18	0·388	5·24	1·97	197
150	19·82	0·317	4·52	1·61	242
200	22·88	0·275	4·00	1·40	280
300	28·02	0·224	3·26	1·14	342
400	32·36	0·194	2·75	0·98	392
500	36·18	0·174	2·35	0·88	440

The above Table is used in calculating the results of the wall-cycles.

Effect of the Form of the Cycle.—The formulas given, refer to a simple-harmonic wave, which is propagated without change of

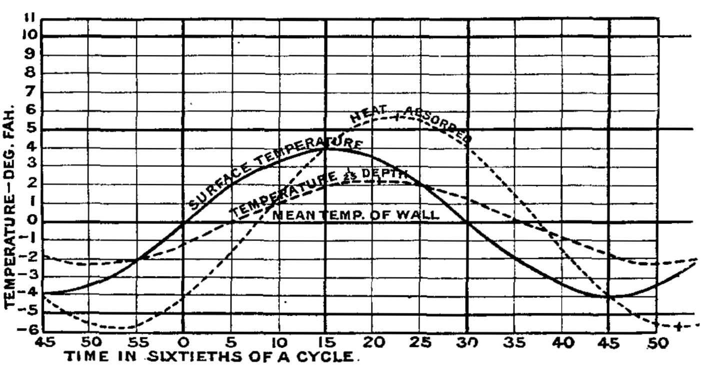
Fig. 14.



TRIANGULAR CYCLE FOR A RANGE OF 4·5° AT $\frac{1}{25}$ INCH IN METAL.

form. Cycles of any other form must be analyzed by the Fourier method into their simple-harmonic components. In order to compare the results of assuming an entirely arbitrary and peculiar

Fig. 15.



SIMPLE-HARMONIC CYCLE FOR A RANGE OF 4·5° AT $\frac{1}{25}$ INCH.

cycle, instead of the simple-harmonic, the coefficients of the first twelve terms of the series, representing the triangular cycle shown in *Fig. 14*, were calculated by the Fourier method. This series, when plotted, gives a curve, *Fig. 14*, rising to a maximum of 10° at the point O, and giving a very nearly constant temperature of $-1°$ for $\frac{5}{6}$ths of the cycle. The corresponding curve of heat-absorption, and the curve of temperature at a depth of 0·040 inch, may be compared with the simple-harmonic cycle, *Fig. 15.* The temperature range of *Fig. 14* at a depth of 0·040 inch is found to be 4·5°, corresponding to a surface range of 11°, at a speed of seventy-seven revolutions in cast-iron. For the same range at a depth of 0·040 inch, the simple-harmonic surface-range is 8° only. The surface-ranges differ considerably, but the quantities of heat absorbed are nearly equal.

TABLE III.—RESULTS OF OBSERVED WALL-CYCLES.

Trial Mark and Cut-off.	Revolutions per Minute.	Thermo-Junction Depth.	Temperature Ranges. Observed at Junction.	Temperature Ranges. Calculated for Surface.	Heat (Q) Absorbed per Square Foot per Cycle.	Maximum Card Temperature.	Mean Wall Temperature.	Condensation Area (A), Degree-Seconds per Cycle.	Ratio, $\frac{Q}{A}$.
		Ins. Ins.	°	°		°	°		
I$\frac{1}{5}$	100·0	0·010 C	4·3	5·1	1·00	318	295	1·50	0·67
II$\frac{1}{5}$	102·0	0·010 C	3·8	4·5	0·88	319	296	1·45	0·61
VIII$\frac{1}{3}$	46·0	0·040 C	6·1	9·4	2·75	313	287	4·28	0·64
X$\frac{1}{3}$	77·0	0·037 S 0	11·0	18·9	4·25	317	277	5·60	0·76
XVI$\frac{1}{5}$	73·4	0·039 C	4·0	6·8	1·56	324	301	2·58	0·61
XVII*a*	70·4	0·039 C	4·0	6·7	1·57	335	310	2·64	0·60
XVII$\frac{1}{5}$	70·4	0·013 C	5·6	6·6	1·55	335	310	2·61	0·59
XVII*b*	97·0	0·039 C	3·3	6·2	1·26	329	305	1·90	0·66
XVIII	45·6	0·039 S 6	3·5	5·4	1·56	329	251	2·43	0·64
XVIII	45·6	0·037 S 4	4·7	7·2	2·07	327	262	3·45	0·60
XIX	43·8	0·039 C	4·9	7·5	2·20	329	305	4·20	0·53
XIX	43·8	0·037 S 0	13·5	20·2	6·00	328	291	9·13	0·65
XX*a*	47·7	0·039 C	4·6	7·2	2·05	331	307	3·80	0·54
XX*a*	47·7	0·037 S 0	11·0	17·2	4·90	331	293	7·65	0·64
XX*a*	47·7	0·037 S 4	4·3	6·7	1·91	331	265	3·31	0·58
XX*b*	81·7	0·039 C	3·4	6·0	1·31	331	306	2·32	0·56
XX*b*	81·7	0·037 S 0	8·3	14·2	3·10	330	292	4·60	0·67
XX*b*	81·7	0·037 S 4	3·5	6·0	1·31	331	264	2·03	0·65

The above Table includes all the observations of wall-cycles, for which complete data were available. In the third column C stands for cover, and S for side. For the latter junctions, the distance in inches along the side is also given. In trial X, the cycle at S, 0 inch, was taken against one of the junctions in the cover. The maximum card temperature is deduced from an indicator-diagram taken in the middle of the cycle. The mean wall temperature is that at the middle of the wall at the position of the junction, from observations taken before and after the cycle. These temperatures are probably right to 0·5° F.

Relation between Temperature, Speed and Condensation.—A comparison of the temperature-ranges at the surface of the metal, with the steam-cycles and the mean wall temperatures, appears to demonstrate that, even at the lowest speed of these trials, and making allowance for the form of the surface cycle, the time does not suffice for the steam to raise the temperature of the surface of the wall more than a small part of the way up to its own temperature. The largest surface-range of 20° F., observed in Trial XIX at forty-four revolutions, with a condensation lasting for nearly $\frac{1}{3}$ second, would raise the surface to 301° F. only, the temperature of the steam being 328° F. during the latter half of the interval. It is hardly possible to suppose this resistance to the passage of the heat from the steam to the metal, to be due to the existence of a surface film of oil or water. Assuming such a film to have a conductivity only one-hundredth of that of iron, it would require a thickness of at least 0·020 inch to produce the observed effects, if the surface of the film itself were instantly raised to the steam temperature. The cylinder was frequently examined immediately after a run, but the interior was invariably found clean and nearly free from grease. The grease film was certainly less than one-thousandth of an inch in thickness. With regard to water, the case is perhaps stronger. The maximum observed absorption of 6 T.U. per square foot, would correspond to the condensation of a film about one-thousandth of an inch thick. If the resistance is due to a water film, the evaporation must be incomplete, and a film must remain from stroke to stroke. Further, this must also take place so uniformly and consistently as to give perfectly regular resistance in all parts of the cylinder, and throughout the trials. The steadiness and consistency of the readings are perhaps the best proof, but there is other strong evidence, that such a film was not present.

The obvious inference from the observations is that the rate of condensation of steam is physically limited, and it is necessary to assume a provisional law of condensation in order to compare the results. As a simple and workable hypothesis, and for other reasons, the rate of condensation of steam on a metal surface was assumed to be proportional to the difference of temperature, and to be independent of the pressure. This assumption would make the amount of condensation taking place on any part of the walls, proportional per cycle to the average excess of temperature of the steam, multiplied by the time during which the temperature is above that of the walls. This product is found by measuring the condensation area on the cycle diagram included between the curves representing the steam and the wall-surface cycles. It is conveniently measured by the product of the time in seconds into the average difference of temperature in degrees. If this area on the cycle-diagram is measured in degrees of temperature, and in sixtieths of a revolution, the result gives the condensation area in degree-seconds per minute. This result divided by the revolutions per minute gives, further, the condensation area in degree-seconds per cycle. These condensation areas have been measured for each of the observed wall-cycles, and are given in the last column but one of Table III. The last column gives the ratio of the heat Q absorbed per cycle to the condensation area A in each case. The approximate constancy of this ratio would appear to indicate that the hypothesis is at least a first approximation to the truth.

Method of Estimating the Total Condensation at any Epoch.—It is evident from Table III that the condensation area is fairly proportional to the amount of condensation taking place at any point over the range covered by the experiments. It is further assumed, if a vertical line is drawn on the diagram of the cycle corresponding to the position of the piston at any epoch, the condensation areas measured to that line may be taken as proportional to the condensation which has taken place at each point of the surface up to the epoch considered. It is possible in this manner to arrive at a fair estimate of the amount of condensation at or shortly after cut-off, before re-evaporation has commenced. There is, further, evidence that the re-evaporation follows the same law, and may be in many cases treated in a similar manner.

Numerical Application of Method of Condensation Areas.—To make an estimate of the condensation in Trials XVI-XX, at 0·250 of the stroke, shortly after cut-off, and at 0·700 of the stroke, shortly before release, the mean wall temperatures are assumed to be independent of the speed, provided that there is no wire-drawing or other change of conditions. Hence the condensation areas, measured on the cycle-diagram in degrees of temperature and in sixtieths of a revolution, are the same or nearly so at different speeds. It is sufficient, therefore, for this estimate to take the area so measured and multiply it by 0·61, the mean of the values of the ratio $\frac{Q}{A}$ in Table III, in order to deduce the cyclical heat-absorption for the parts of the surface considered.

The following Table gives the total heat-absorption on the clearance surfaces. The portions of the barrel surface subsequently exposed by [the motion of the piston, require to be treated somewhat differently. The temperature of the piston was specially determined by means of a platinum thermometer inserted through a hole in the piston-rod.

TABLE IV.—CYCLICAL HEAT-ABSORPTION FOR CLEARANCE SURFACES.

Portions of the Surface Considered.	Area of Surface.	Mean Temperature.	Condensation Area per Square Foot per Minute.	Heat Absorbed in Thermal Unit Fahr. per Minute.
	Square Feet.	° F.	° Seconds.	
Cover, face 10·5 inches diameter .	0·60	305	185	68
" side 3·0 "	0·70	305	185	79
Piston, face 10·5 inches diameter .	0·60	295	300	110
" side 0·5 "	0·11	295	300	20
Barrel, side 3·0 "	0·71	297	285	123
Counterbore 0·5 "	0·12	291	380	28
Ports and valve . "	0·90	305	185	102
Sums and Means . .	3·74	301	231	530

At 0·250 of the stroke, 3 inches of the barrel surface have been exposed by the motion of the piston. Estimating the total contribution of this portion of the surface at 55 T.U.F. per minute up to 0·250 of the stroke, there is a total heat-absorption of 585 T.U.F. per minute at 0·250 of the stroke, which corresponds approximately under the conditions of the trials, to the condensation of 0·65 of a pound of steam per minute. The condensation per cycle at this point for any of the trials considered, may be obtained by dividing this result by the revolutions per minute. It will be observed that the clearance surfaces contribute about 90 per cent. of the total condensation.

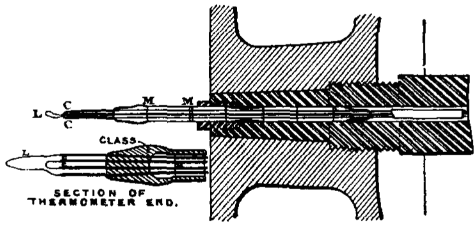
Estimate of Re-evaporation during Expansion.—It is not possible to make an estimate of re-evaporation with the same degree of probability as condensation, but some idea may be gained, by a similar method, of the amount of re-evaporation that has taken place before release. A probable excess of re-evaporation over condensation equivalent to nearly 300 T.U.F. per minute is found, and the quantity of heat rejected by the metal at release and during the exhaust period, generally called the exhaust waste, may be estimated at 250 T.U.F. per minute, under the conditions of the present trials.

STEAM CYCLES.

The Steam Thermometer.—The thermometers, which were sufficiently sensitive to follow the changes of temperature of the steam throughout the stroke, were made of very fine platinum wire, $\frac{1}{1,000}$ inch in diameter. The method of attachment to the piston is shown in *Fig. 16.*

Temperature Cycle of Steam in Cover.—The first set of observa-

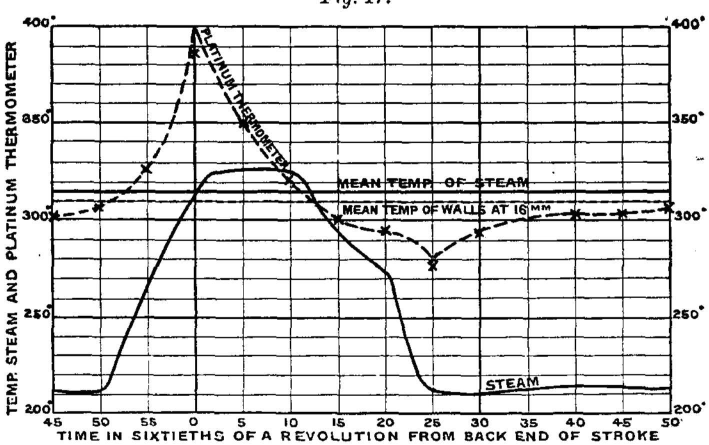
Fig. 16.



SECTION OF PISTON SHOWING PLATINUM THERMOMETER.

tions with one of these thermometers was made with the instrument fixed in a hole in the cylinder-cover. The results are exhibited in *Fig. 17*. The indicated steam temperatures are shown by the full curve, the temperatures of the platinum thermometer

Fig. 17.

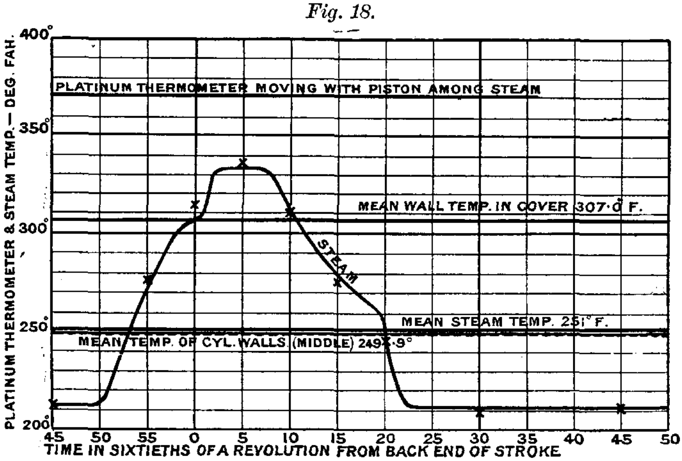


PLATINUM THERMOMETER IN STEAM IN $\frac{3}{8}$-INCH HOLE IN COVER.

by the dotted curve. The temperature scale is the same for both. The most striking feature of the platinum curve is the great superheating shown at the end of compression. During admission the temperature rapidly falls. At, and shortly after, cut-off the thermometer invariably showed a temperature 2° or 3° below the indicator. It soon, however, rises above the indicated curve, and, with the exception of a sudden drop at release, due to adiabatic expansion, continually approaches the temperature of the walls during the exhaust period. The actual superheating of the steam during compression must have been greater than that shown by the thermometer for two reasons. The method of observation gives the mean temperature during a certain interval of contact, generally $\frac{1}{30}$ revolution, and cannot therefore reproduce a very short and sharp maximum. Secondly, although the thermometer was certainly very sensitive, the lag must have been appreciable on a rise of 100° F. taking place in 0·1 second. It is remarkable that the effect of radiation from the cool surrounding walls is not more noticeable. To test the effect of pure radiation, as compared with that of convection, on these thermometers, a special experiment was made, when it was found that the rate of loss of heat by pure radiation for this very fine wire, at temperatures between 200° F. and 350° F., was between fifty and one hundred times less than that due to convection. The possible error due to direct radiation from the surrounding walls does not, therefore, amount to more than 1° or 2°, and the thermometer is really indicating the temperature of the steam around it. The peculiar characteristics of the platinum curve in *Fig. 17* were verified on several occasions, and with different settings of the valve, the results observed in every case being similar.

Piston Steam Thermometer.—To observe the temperature of the main body of the steam at a distance from the walls, a similar thermometer was attached to the piston in the manner shown in *Fig. 16.* The thermometer projected from the piston for a distance of about 3 inches, and was received at the back end of the stroke in a tube 1 inch in diameter in the centre of the cover. The indications of this thermometer at different speeds and at different settings of the valve, were in remarkably close agreement with the card. Systematic differences, however, were always observed, which, from their consistency and from the great number of observations, cannot be attributed either to errors of the indicator or of the thermometer. The curve in *Fig. 18* is drawn from the card taken simultaneously with the observations of the platinum thermometer. The differences observed are almost too small to be shown. They are much greater, however, than the uncertainty of the observations themselves. The temperatures at admission and release were much steadier than in the cover. At most other points of the stroke it was possible to take readings to 0·1 degree, and the extreme variations often did not amount to more than 0·5 degree for several minutes. The superheating shown during compression was small; the greatest amount was shown in the middle of the admission period. The centre of the steam appears to have been at this epoch generally 4° F. or 5° F. above the indicator. *Fig. 18* shows the smallest value recorded, namely, 2° F. Other examples will be found in *Fig. 7* and *Fig. 10.* Throughout the expansion curve, the readings of the platinum thermometer were between 2° F. and 3° F. lower than the indicator. At the end of the stroke, and for part of the exhaust period, the temperature fell to between 207° F. and 208° F., but recovered to 212° F., or close to the barometric temperature, before the end of exhaust. This may have been due to a real lowering of pressure,

Fig. 18.



Curve showing agreement between Indicator and Platinum Thermometer.

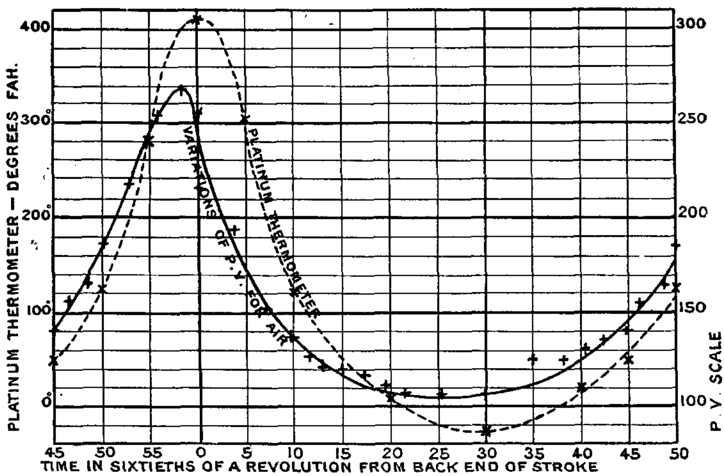
owing to the rapidity of condensation in the surface condenser at a pressure slightly below that of the atmosphere. The corresponding difference of pressure is only 1 lb. The lowering of temperature of the steam during expansion appears, on consideration, to be too large and regular to be explained by any error or lag of the indicator; lag of the thermometer would have the opposite effect.

Testing the Indicators.—The indicators were tested as nearly as possible under the conditions of the trials, and were carefully adjusted; they were daily oiled and cleaned, and tested for friction and back-lash. At the comparatively low speeds of the trials, it is hardly possible that they should have so considerable a lag as would be required to explain the difference between the indicator and the platinum thermometer.

Test of Sensitiveness.—A test of the sensitiveness of the platinum thermometer, in which the engine was run with air instead of steam in the cylinder, with a view to determine the probable amount of lag, is illustrated in *Fig. 19.* The lag could not have been greater than 2° with the temperature of the air rising at the rate of 100° per second, but it is not possible to obtain any form of indicator sufficiently sensitive and accurate to perform this test satisfactorily.

Superheating due to Wire-drawing.—*Fig. 20* gives an illustration of a different kind of steam-cycle, taken during the measurement

Fig. 19.



Temperature Cycle with Air in Cylinder.

of a valve leak, and showing a very unexpected amount of superheating. The temperature of the steam leaking into the cylinder under these conditions, as measured by the platinum thermometer on the piston, is shown by the dotted curve. The full curve below shows the temperatures deduced from the indicator on the assumption that the steam was saturated.

Conclusions.—From the steam-cycle observed in the hole in the cover, it appears that, even within $\frac{1}{10}$ inch from the walls, the temperature of the steam is greatly affected by adiabatic compression and expansion, but that during comparatively quiescent periods of the cycle, such as the exhaust, the steam close to the walls is heated nearly to the wall temperature. In a single-acting non-condensing engine, with a moderate degree of compression, water is not likely to collect in the corners and recesses of the clearance; and the clearance contents, consisting chiefly of superheated steam, cannot be regarded as a primary cause of condensation of the admission steam. The fact that the platinum thermometer, after falling below the indicator at cut-off, crosses it again at a temperature slightly below that of the walls, and then rises considerably above it, shows that re-evaporation from the cover is probably complete some time before release, and that evaporation from a highly heated wall is probably a process of a rapid and explosive character. From the piston steam thermometer, the temperatures deduced for the indicated pressures

Fig. 20.

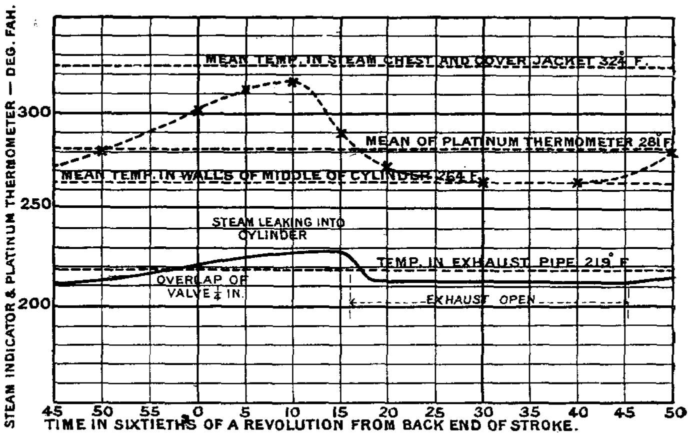


Diagram illustrating Superheating due to Wire-drawing.

seem to represent very fairly the average state of the main body of the steam, but the steam is probably slightly superheated during compression and admission, and slightly supersaturated during expansion and exhaust. The effects observed are probably too large to be explained by lag of the indicator. The superheating of the steam during admission may be partly explained by the further compression of the already superheated cushion steam, which in the present case formed one-fifth of the cylinder contents. It is also partly due to the kinetic energy of the inrush. In any case it is evidence that the steam supply from the boiler was fairly dry. Under the conditions of the trials it is impossible that the steam supply could have been superheated. In fact, thermometers in the steam-pipe and steam-chest indicated the normal temperature. The results of tests with a number of different thermometers showed the temperature of the steam during expansion, and during the early part of the exhaust, to be lower than that deduced from the indicator.

Dynamical Equilibrium of Expanding Steam.—Apart from the statical condition of equilibrium between steam and suspended water-drops, which depends on the size of the drops and on the value of the surface tension, there is also a dynamical condition in the case of rapidly expanding steam, which has not, so far as the Authors know, been previously noticed. When steam is expanding adiabatically, it requires, as is well known, a condensation equivalent to nearly one thermal unit per pound per degree Fahrenheit of fall, to keep it up to the saturation temperature. This condensation must take place chiefly on the surface of drops already formed. The temperature of the drops can be maintained only by continual evaporation. Unless the steam condensed is at a lower temperature than the drops, there can be no balance of condensation, and the temperature of the drop cannot fall. The lowering of steam-temperature required will evidently be proportional directly to the rate of condensation, and inversely to the surface exposed by the drops. Since the drops are at once foci of condensation and foci of heat, there must be powerful obstructive influences at work, and the lowering of temperature of the steam may therefore be considerable. In the absence of more certain indications these obstructive influences may be assumed similar in magnitude to those which limit the rate of condensation of steam on a metal surface. It is possible to make a numerical estimate of the order of the lowering of temperature. At n revolutions per minute, the initial rate of fall during expansion, in the experiments at one-fifth cut-off, may be taken as about $10\,n$ °F. per second, and the required balance of condensation as $8\ n$ thermal units per pound per second. If the wetness of the steam is 1 per cent., and the average diameter of the drops 0·000024 inch, the surface exposed per pound of steam would be nearly 480 square feet, and the lowering of the steam temperature at 100 revolutions would be about 2·7° F. The lowering required increases in direct proportion both to the speed and the diameter of the drops. With nearly dry steam at high speeds the initial lowering may be considerable, especially if the drops are large and few. If it be admitted that the temperature of rapidly expanding steam may fall considerably below its saturation temperature, it affords a possible explanation of a certain loss of efficiency in high-speed engines. From the behaviour of other vapours in a supersaturated condition, the pressure, if the steam remained dry, would be much more reduced for a given expansion of volume than it would be if the steam had time to maintain itself by condensation at its saturation temperature.

The statical condition mentioned affords an explanation of the unwillingness of steam to condense otherwise than on dust nuclei, or on drops of water already formed. It will be observed, however, that in order to account for a lowering of 2 or 3 degrees, the drops would have to be of so minute a size as to be invisible in the most powerful microscope. The linear dimensions of the drops actually occurring in steam-engine practice are probably between one hundred to one thousand times greater.

For the dynamical condition, on the contrary, the larger the drops the less the surface they expose, and the greater the fall of temperature required. A diameter of $\frac{1}{10,000}$ inch would mean a fall of about 50° F. below the saturation temperature at 400 revolutions per minute and wetness 1 per cent. In the absence of accurate knowledge of the properties of steam under these conditions, it is not possible to say exactly how much missing steam such a fall of temperature would account for. It would probably be between 5 per cent. and 10 per cent., according to the extent of the drop surface. The subsequent recovery of temperature, as more drops were formed and condensation proceeded, would simulate the effect of re-evaporation. The initial fall of temperature at a high speed, however considerable, would probably be of very short duration.

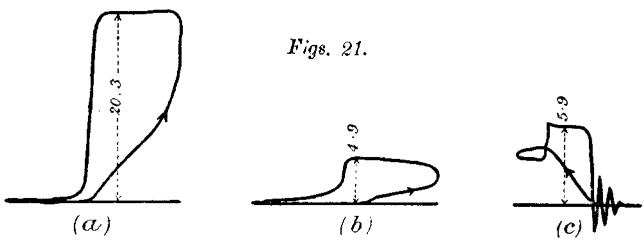
It is not improbable that steam in this supersaturated condition may tend to condense more readily on any surfaces exposed to it, which happen to be below the saturation temperature, than would be the case with ordinary wet saturated steam. The temperature of the steam itself may have some influence, as well as that of the surface on which it is deposited. For the same degree of adiabatic expansion, the heat abstracted from the walls would be the same, whether the steam condensed is dry and supersaturated, or wet and saturated, provided that in the latter case the suspended moisture is deposited along with the steam. The amount of condensation, however, might be greater in the case of the supersaturated steam, if the temperature of the steam itself has any influence.

Supposing the steam is many degrees below its saturation temperature during rapid expansion, no thermometer, however sensitive, could indicate the whole extent of the phenomenon. The rapid motion of the steam and the piston might tend to cool it, but the condensation on its surface would tend to keep it near the saturation temperature corresponding to the pressure. The lag of the thermometer would also make the reading too high. If, on the other hand, it is inconceivable that the thermometer should indicate anything but the saturation temperature, the differences observed must be due to a real difference of pressure, owing to the rapid vortical motion of the steam, between the centre and the circumference of the cylinder contents. Such differences must exist, and have often been regarded as important. It is probable that further experiments of this nature might throw some light on the question.

Valve and Piston Leakage.

The Slide Valve.—In estimating the amount of condensation in the cylinder by comparing the measured feed per revolution with the steam indicated by the diagrams, the valve and piston leak is generally assumed to be negligible. The effect of leakage,

Figs. 21.



however, is in many ways so similar to that of condensation, that one may readily be mistaken for the other; and no estimate of condensation deduced from diagram and feed measurements can have any claim to consideration, unless the state of the valves and piston in respect of leakage is simultaneously investigated. In addition to trying the stationary test for leakage, which is very easily applied, the leakage was measured as accurately as possible under the actual conditions of running. The stationary test was found to be of little or no value.

Preliminary Leakage Tests.—To measure the leakage into and out of the cylinder, with the valve in motion under the conditions of running, but just not admitting steam directly, the piston was blocked and cards were taken, the barrel motion being obtained from the valve spindle, and the engine being driven by a motor. *Figs. 21*, *a* and *b*, are sample indicator-diagrams taken at the back end of the cylinder, in the above manner. The first, *a*, showing a maximum of 20·3 lbs., was taken on July 29th, diagram *b* on

August 29th, after the valve had been very carefully scraped and refitted; diagram *c* is one taken at the crank end on the same date.

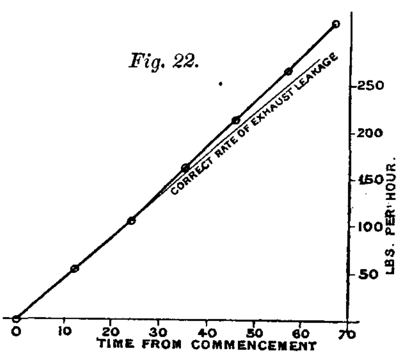
The following were the conditions of the test:—

Date.	Indicator-diagram (Type).	Average Pressure.		Maximum Temperature.	Volume of Cylinder.	Revolutions per Minute.	Leak.
		Gauge.	Cards, Maximum.				
				°	Feet.		Lbs. per Hour.
July 29 . .	*a*	97·0	19·2	301	0·360	41·8	38·6
August 29 .	*b*	81·0	4·87	316	0·648	75·7	30·6
August 29 .	*c*	81·0	5·3	..	0·063	75·7	3·6

A smaller indicated pressure in the second case was nearly compensated by the higher speed and greater volume, so that the resulting leak deduced is not very far from being proportional to the difference of pressure under which the leak took place. It would appear that the leakage is not merely a question of such minute differences of fit as those corrected in the scraping.

Direct Exhaust Leakage.—Preliminary trials showed the direct leakage of steam from the steam-chest into the exhaust to be by far the largest and most important. In order to measure this leakage as nearly as possible under the conditions of running, both the steam ports were blocked with lead, and the valve was driven by an electric motor, the piston being disconnected. The following are the results of two experiments made with the same valve setting as for the later series of trials at one-fifth cut-off. In the first trial, 112 lbs. were condensed in 25·17 minutes, at a gauge pressure of 91 lbs. per square inch, and the rate of leak appeared to increase slightly as the oil-film was gradually dissipated.

Fig. 22.



The effect of the dissipation of the oil-film is well illustrated in the second trial under the same conditions, for which the results of the separate weighings of the feed have been plotted in the curve shown in *Fig. 22.* In this trial 317·5 lbs. were condensed in 66·42 minutes, at a gauge pressure of 80·5 lbs. The initial rate of leak, however, which was taken as the correct value, agrees very closely with the previous trial.

To compare the results of different leakage trials, the leakage is provisionally assumed proportional to the difference of pressure. In the equation, leak in lbs. per hour $= k \times$ (difference of pressure); the constant k, which may be called the "rate of leak" of the valve, gives the leak per hour per pound difference of pressure. The rate of the direct exhaust leak for this particular setting of the valve is:—$k = 2 \cdot 98$. The round number $k = 3 \cdot 00$ for estimating the leakage corrections is evidently within the limits of error of the measurements. No trace of this leakage appears on the indicator-diagrams, which were of the most perfectly regular type throughout, as shown in *Fig. 8.*

Leakage of Unbalanced Slide-Valves.—With a view to investigate this question further, exactly similar tests were made on the smallest and largest valves of a quadruple-expansion engine. The H.H.P. valve gave a leak of 38 lbs. per hour, with a pressure difference of 100 lbs. between the steam-chest and the exhaust pipe. The L.P. valve gave 41 lbs., and 29 lbs. per hour, with pressure differences of 34 lbs. and 21 lbs. respectively. These valves are ordinary unbalanced slide-valves, with large bearing and guiding surfaces, and the fitting throughout is undoubtedly of a high order. The low-pressure valve was proved to be absolutely steam-tight when stationary.

Provisional Law of Leakage.—By applying the law of transpiration of liquids, assuming that the leakage takes place chiefly in the form of water, results were at once obtained which, considering the nature of the measurements, are remarkably consistent both for the balanced and for the unbalanced valves. The leakage of a liquid through a fissure of nearly uniform thickness, depending on the nature of the water- and oil-film, should be proportional directly to the difference of pressure and the perimeter of the port, and inversely to the width of the bearing surfaces. The latter factor is somewhat difficult to estimate for a moving valve, but for the present purpose the values given in the following Table are probably sufficiently approximate. If the fissure through which the leak takes place is of nearly the same thickness in each case, the rate of leakage k per lb. pressure per hour should be proportional to the perimeter p of the port, divided by the mean overlap l. Thus $k = \frac{C p}{l}$, where C is a coefficient depending on the nature of the oil-film, which should be the same for the same type of valve. It might possibly be nearly the same for valves of different types, if it depends on some physical property of a mixture of oil and water.

TABLE V.—COMPARISON OF RATES OF LEAKAGE.

Valve considered.	Perimeter of Port, p.	Overlap (mean), l.	Ratio, $\frac{p}{l}$.	Observed Rate of Leak, k.	Deduced Value of $C = \frac{kl}{p}$.
	Inches.	Inches.			
Balanced, Robb	72	0·5	144	3·00	0·021
Unbalanced H.H.P. . . .	30	1·5	20	0·38	0·019
„ L.P.	65	1·0	65	1·20	0·019
„ „	65	1·0	65	1·38	0·021

The leakage coefficients observed with the different valves, are at least of the same order of magnitude. If so, we are justified in concluding that the leakage probably takes place in the form of water, and is proportional to the difference of pressure. It would also appear probable that such leakage is the normal state of things with a moving valve, and that the excessive leakage observed with the balanced valve, is not a defect peculiar to this type, but is simply a consequence of its comparatively large size. At the high speed for which this engine was designed, the leakage would not be a very serious matter. If the leakage were simply due to accidental tilting motions, or to bad fitting, it is difficult to see why the value of the coefficient C should be of the same order for valves of such different types. If, on the other hand, the leakage is not purely accidental, but follows a regular law, it is a matter of such practical importance as to be well worthy of further investigation.

Leakage into Cylinder after Cut-off.—From the experiments described on p. 31, it is possible to obtain an independent verification of the value of the coefficient C. With $l = \frac{1}{2}$ inch, and $p = 20$ inches, the values are $C = 0 \cdot 022$, and $C = 0 \cdot 020$ respectively. By means of the time integral of the expression $\frac{Cp\,(P_0 - P)}{l}$, the leakage taking place into the cylinder during expansion between the points 0·25 and 0·70 of the stroke, for trials XVI.–XX., amounts approximately to 6·0 lbs. per hour. The smallness of this result is due to the fact that the difference of pressure is small just after cut-off when the overlap is least.

Piston-Leak under Conditions of Running.—Under running conditions, the steam leaking past the piston was found to amount to 15·3 lbs. per hour under a mean pressure of 33 lbs. The piston-leak taking place during expansion between 0·25 and 0·70 of the

TABLE VI.— COMPARISON OF CARDS AND FEED WITH WALL-CYCLES.

Single-Acting Non-Condensing Trials. Cut-off at 0·200 of stroke.
Clearance, 10 per cent.; Release, 0·750; Expansions (cut-off to release) 2·83.

——	1	2	3	Means.	4	5	6	Means.	7
1. No. of trial . . .	XIX	XVIII	XXa	1, 2, 3	XVIIa	XVI	XXc	4, 5, 6	XVIIb
2. Duration in minutes	37·0	68·0	55·0	..	79·0	76·0	35·0	..	25·0
3. Mean revs. per minute	43·8	45·7	47·7	45·7	70·4	73·4	81·7	73·7	97·0
4. Mean gauge pressure	87·9	89·2	94·4	90·8	98·1	92·0	94·2	95·1	96·0
5. Gross feed per rev. .	·1422	·1437	·1483	..	·1094	·1036	·1000	..	·0856
6. Leakage correction .	·1004	·0976	·0990	..	·0697	·0627	·0576	..	·0494
7. Corrected feed per rev.	·0418	·0461	·0493	·0462	·0397	·0409	·0424	·0407	·0362
8. Calculated cushion steam	·0107	·0104	·0103	..	·0099	·0098	·0100	..	·0105
9. Total weight of fluid expanding in cylinder	·0525	·0565	·0596	·0567	·0496	·0507	·0524	·0505	·0467
10. Indicated weight at 0·250	·0407	·0414	·0437	..	·0418	·0394	·0408	..	·0393
11. Indicated weight at 0·700	·0466	·0456	·0488	..	·0460	·0436	·0454	..	·0426
12. Increase of indicated weight	·0059	·0042	·0051	..	·0042	·0042	·0046	..	·0033
13. Adiabatic condensation	·0019	·0020	·0021	..	·0020	·0019	·0020	..	·0019
14. Indicated evaporation	·0078	·0062	·0072	·0069	·0062	·0061	·0066	·0062	·0052
15. Calculated evaporation	·0076	·0073	·0070	·0073	·0048	·0046	·0041	·0046	·0035
16. Indicated condensation	·0118	·0151	·0159	·0146	·0078	·0113	·0116	·0099	·0074
17. Calculated condensation	·0148	·0142	·0136	·0142	·0092	·0089	·0080	·0088	·0067
18. Water present at 0·250 {per cent. of feed (7) .	28·3	32·7	32·3	31·7	20·0	27·7	27·3	24·3	20·4
19. Water present at 0·250 {per cent. of fluid (9) .	22·5	26·8	26·7	25·8	15·7	22·3	22·1	19·6	15·9
20. Indicated HP. . .	4·10	4·34	4·78	4·43	7·02	6·67	7·71	7·00	8·81
21. Lbs. per I.HP. hour	26·8	29·1	29·5	28·6	23·8	27·1	26·9	25·7	23·8
22. Condensation per I.HP. hour . .	7·6	9·5	9·5	9·1	4·8	7·5	7·3	6·3	4·8

stroke, under the conditions of trials XVI–XX, amounts to only 2·5 lbs. per hour, the greater part of the leak occurring during admission.

Comparison of Indicated and Calculated Condensation.—The results given in Table VI, with the exception of the first four and the last five lines, are expressed in terms of the weight of steam and water per cycle in lbs.

The differences of the weights in lines 9 and 10 are given in line 16. They depend entirely on the rate of leakage assumed for applying the very large leakage correction given in line 6. Considering the magnitude of this correction, it is most remarkable that the weights in line 16 should have so great a degree of consistency. A variation of only 3 per cent. in the rate of leak would be sufficient to explain the largest discrepancy from the mean. The order of consistency shown under such differences of speed and pressure, and probably also of lubrication, from trial to trial, is perhaps the strongest proof that could be given that the phenomenon of valve leakage is subject to regular laws, and deserves much more attention than it has hitherto received. Considering the range of speed covered by the trials, there is strong evidence that the rate of leakage is nearly independent of the speed of reciprocation of the valve. The evidence that it is simply proportional to the pressure is much less conclusive. Line 17 gives the total condensation at 0·250 of the stroke, calculated from the results of the observed wall-cycles for three low-speed trials. This assumes that the cyclical condensation per minute is independent of the speed provided that the temperature conditions remain unchanged. It is evident, on comparing the numbers in lines 16 and 17, that the indicated condensation agrees as closely as can be expected with this view. There is perhaps a slight indication of a greater rate of condensation at the higher speeds, but this is partly accounted for by the higher mean pressure and temperature corresponding to these trials. The results in line 17 were calculated to correspond to a gauge-pressure of 90 lbs.

The Nature and Effects of Valve Leakage.—The foregoing experiments would make it appear probable that a moving valve, however well fitted, is subject to a regular leakage of a peculiar type, which has not been previously suspected. The leakage appears to take place in the following manner. So long as the valve is stationary, the oil-film may suffice to make a perfectly tight joint, but as soon as it begins to move, the oil-film becomes broken up and partly dissipated. Water is being continually condensed on the colder parts of the surface exposed by the motion of the valve. This water works its way through, and breaks up the oil-film under the combined influence of the pressure and the motion. The continual re-evaporation taking place in the exhaust tends to keep the valve and the bearing-surfaces of the seat cool, and to maintain the leaking fluid in the state of water. The exhaust steam from the cylinder has the same tendency. The coefficients of viscosity of steam and water at the temperatures which occur in a steam-engine are not accurately known. But whereas that of steam increases with rise of temperature, that of water diminishes very rapidly. It is not improbable that the quantity of water which can leak through a given crack under a given difference of pressure, may be from twenty to fifty times greater than the quantity of steam which can leak under similar conditions. This agrees with well-known facts in regard to leakage, and explains how it is that the leakage in the form of water is so great. A few simple experiments were made with regard to the transpiration of water and steam under the conditions in question, and the leakage in the form of water was more than twenty times as great, the water being at a temperature below boiling point. The motion both of the water and the steam, owing to the high velocity, was certainly turbulent or eddying, which would have the effect of greatly increasing the resistance as compared with that due to viscosity, if the motion were steady. For the case of steady motion, comparative tests were made of the relative values of the viscosity of water, cold and hot. The measurements were not sufficiently accurate to give the law of the variation of the viscosity with temperature above 212°; but it appeared that the viscosity at 212° F. was only one quarter of that at 62° F., and that it continued to diminish very rapidly. Under the actual conditions of the valve-leak experiments, the water leak is more likely to have been between forty or fifty times the steam leak. An explanation is thus furnished of a possible form of leakage, indirectly due to condensation and re-evaporation—so many times greater than the steam leakage, which, alone, engineers have been in the habit of contemplating, that it might well claim attention on its own merits, apart from the very limited number of valves on which it has hitherto been possible to make direct experiments.

The analysis of a large number of observations, in addition to the few made by the Authors, leads to the conclusion that all valves leak more or less when in motion, and that in many cases the greater part of the missing quantity is to be attributed to leakage of this description. Whatever the precise manner in which the leak takes place, it appears to be nearly proportional to the difference of pressure, and to be in most cases independent of the speed. In any case, it appears probable that the leakage is connected in some way with the condensation taking place on the valve surfaces. If so, it may evidently be greatly reduced, if not entirely cured, by jacketing, or otherwise heating the valve-seat, to minimise the condensation.

These views have an important bearing on the design of valves. For low-speed engines, separate steam- and exhaust-valves should possess advantages over the ordinary slide-valve. The superiority of the compound engine would also appear to be partly due to the great reduction of possible leakage.

General Conclusions.

On the Rate of Condensation of Steam.—The most important general conclusion to be derived from the experiments is that the rate of condensation of steam on a metallic surface is limited, and is proportional to the difference of temperature. The small increase observed in the ratio $\frac{Q}{A}$, p. 165, with increase of speed, may be most naturally explained as due to the diminished thickness of the water-film deposited, and the smaller range of the metal cycle. The slightly smaller values observed on the cover as compared with the sides, though possibly due to slight differences in soldering or in the form of the cycle, may also be attributed to the state of the surface. Taking these factors into consideration, the probable rate of condensation of steam on a clean and dry metal surface is found to be 0·74 thermal units per second per square foot per 1° F. difference of temperature at 300° F. Expressed in more familiar quantities, the rate above given would correspond in a surface condenser to the condensation of 27 lbs. of dry steam per square foot per hour for a difference of temperature of 10° F. between the steam and the surface.

Condensation in Terms of Temperature Distribution.—Assuming that the amount of condensation is limited by the rate of condensation of steam given by the above law, the problem of estimating the amount of condensation taking place in any given engine with any given steam cycle, is reduced to the (comparatively) simple problem of determining the temperature distribution on the cylinder walls while the engine is running. Given the temperature distribution, the condensation is inferred by measuring the condensation areas on the cycle diagram.

Limit of Cyclical Condensation for any given Cycle.—That it should be necessary to observe the temperature distribution in any case, in order to be able to deduce the cyclical condensation, may appear at first sight a somewhat disappointing result. The form, however, of the law of condensation and re-evaporation as deduced from the experiments, leads directly to a limiting value of the cyclical condensation, and gives a result of striking simplicity, which is undoubtedly applicable to a large number of important cases. If the conditions of external and internal heat loss are supposed to be such that the mean temperature of the clearance surface, on which the greater part of the initial condensation takes place, is reduced to the mean of the steam cycle, it is plain that the condensation and evaporation areas on the cycle diagram will be equal. If the temperature of the clearance surface falls below this point, evaporation will be incomplete. Water will then accumulate in the cylinder until a balance is attained by the mechanical removal of the excess. It is obvious that all steam condensed on the surface, and then mechanically rejected in the form of water, represents the communication of a quantity of heat to the walls, equivalent to the total heat of the steam condensed, and thus rejected, reckoned from the temperature at which the water is thrown off from the walls. The quantity of heat thus communicated per lb. of water rejected, may be between twenty and fifty times greater than that communicated by the condensation and subsequent re-evaporation of an equal quantity of steam.

If the engine starts cold, and the surfaces are gradually heated by the action of the steam, it is clear, from the same considerations, that the rise of temperature up to this point, so long as water is being mechanically rejected, will be extremely rapid. The mean of the steam cycle is, therefore, on the provisional law, a natural minimum of temperature for the wall surface, corresponding to a maximum limit of condensation for any given cycle.

In order to deduce the limiting value of the condensation per square foot per hour for any given cycle, it is simply necessary to draw the cycle diagram corresponding to the indicator-diagram, and to rule across it the line representing the time average of the steam temperature, as shown, for instance, in *Fig. 18.* The area above this line is the maximum condensation area corresponding to this particular cycle. The maximum value of the condensation, measured in thermal units per hour, is forty-five times this area measured in degrees F. and sixtieths of a cycle. As a general rule, the condensation must be less than this limit, because the temperature range of the surface of the metal, which is far from negligible at low speeds, has the effect of diminishing the condensation area, and also the available evaporation area to a nearly equal extent, although it does not materially affect the condition that the mean temperature of the wall surface should be the same as that of the steam-cycle.

If t' is the mean temperature of condensation, and t'' that of re-evaporation, and L', L'', the corresponding heats of vaporization, $L'' - L' = 0 \cdot 70\ (t' - t'')$, approximately. The quantity of heat Q given up to the walls by condensation and re-evaporation per pound of steam, is given by the formula, $Q = 0 \cdot 30\ (t' - t'')$. If t° is the mean temperature of the walls, the quantity of heat given up to the walls by each pound of steam condensed at t' and rejected without re-evaporation, is $L' + t' - t^\circ$. If the range $t' - t''$ is 50° F., this quantity is more than fifty times as great as Q. The mean temperatures of condensation and evaporation, t' and t'', are found by a process analogous to that of finding the centres of gravity of the corresponding areas. Each temperature is weighted in proportion to its difference from that of the wall surface, which determines the rate of condensation at each point of the cycle. Following the usual notation for the centre of gravity formula,

$$t' = \frac{\Sigma\, t\, (t - t^\circ)}{\Sigma\, (t - t^\circ)}$$

where t, t°, are the temperatures of the steam and walls at any point of the cycle. The quantity of heat given to the walls by the condensation and re-evaporation of dry steam, is given by the formula, $Q = 0 \cdot 30\ (t' - t'')$, per pound. A more elaborate or apparently exact formula, is useless, because the value of the constant $0 \cdot 30$ in this expression, being the value of the change in the total heat of steam per 1°, is one of the most uncertain elements in the whole theory of the steam-engine. According to the experiments of Regnault and Griffiths, the mean value of this constant between 30° and 100° C. should be $0 \cdot 40$, but it appears not improbable that its value diminishes with rise of temperature.

It has often been pointed out that, as a result of the comparatively small rate of increase of the total heat of steam with rise of temperature, a relatively small loss of heat, or a slight change of conditions, is competent to account for a considerable change in the initial condensation. The balance is extremely delicate, and is very easily turned. The further possibility, that there should be a limit of condensation, results from the form of the law of condensation, and could not have been foreseen so long as the rate of condensation was regarded as infinite. When this limit is reached, the conditions as regards increase of condensation are extremely stable. If the limiting range $t' - t''$ is 50° F., which is not uncommon in compound engines, a loss of 15 T.U. is sufficient to account for each pound of initial condensation and re-evaporation at this range. But if at this point the rate of loss of heat is suddenly doubled, the initial condensation will be increased by less than 2 per cent. If, on the other hand, the rate of loss of heat were reduced to one-quarter, the initial condensation and the range of temperature between condensation and re-evaporation would each be reduced to one-half of their limiting values. It is also interesting to observe that the form of the law of condensation would make the limiting value of the condensation in any cylinder depend chiefly on the temperature range in that cylinder. Of all the results which have been empirically established with regard to cylinder condensation, this result has always been regarded as the most certain. To express the result more accurately, according to this law, the limiting value of the condensation, when measured in pounds per hour, should vary as the area included between the steam-temperature cycle curve and the line representing the mean temperature of the steam-cycle. The limiting value should also increase slightly with increase of speed, because the temperature range of the metal surface is reduced, and the effective condensation area is thereby increased.

Correction for the Metal Surface Cycle.—It is clear that no simple formula can be constructed to take account of all the possible varieties of cycle. The correction is not large, and might be neglected if it were not that it varies with the speed and with the point of the cycle considered. In the majority of cases which occur in practice, the maximum point of the wall-surface temperature cycle is found to coincide very nearly with the point of cut-off, and the point at which the steam-cycle curve crosses the wall-surface curve is generally very near the point of release. For these two points of the cycle, in the case of limiting condensation, a sufficiently approximate correction can be applied for the effects of variation of speed by the following simple method. The condensation area is measured from the line of mean wall-temperature, and up to the line of cut-off or release as required. The area so measured is then reduced in the proportion $\frac{(1 + \sqrt{n})}{(3 + \sqrt{n})}$ for the point of cut-off, and in the proportion $\frac{\sqrt{n}}{(3 + \sqrt{n})}$ for release. The correction at the point of cut-off varies from 25 per cent. at $n =$ twenty-five revolutions per minute to 9 per cent. at four hundred revolutions per minute, if the steam-cycle remains the same. This formula assumes a cast-iron cylinder. The main effect of making the correction is to increase the value of the condensation constant from 0·61 T.U. to 0·74 T.U. per degree-second per square foot, as deduced from Table III. It also has the effect of making the observations agree rather better with the calculations at higher speeds, both in Tables, pp. 19 and 34.

Effect of Initial Wetness of the Steam.—In view of the observations on the effect of the condensation of wet steam, it is interesting to make an estimate of the possible increase of condensation thereby produced. It is, however, necessary to make a few assumptions, which are probably not in all cases justifiable.

Let x be the dryness fraction of the steam condensed at a temperature t', and let it be assumed that the proportion of suspended moisture $(1 - x)$ is all deposited on the walls together with the condensed steam. Let it be further assumed that the whole of this water is re-evaporated non-explosively at an average temperature t''. The mean temperature of the wall-surface, t°, will be somewhere between these two extremes, and will be modified by conduction from the neighbouring parts of the cylinder. To simplify the conditions the point considered may be situated on the clearance surface near the middle of the cover, or the cylinder to be so large and thin that the effect of conduction may be neglected. The other losses may also be supposed negligible in comparison with that due to re-evaporation.

If L′ L″ are the latent heats of vaporization at the temperatures t' t'', $L'' - L' = 0{\cdot}70\ (t' - t'')$. The heat supplied by condensation per pound of steam condensed is L′. The weight of water deposited by the condensation of 1 pound of steam is by hypothesis $\frac{1}{x}$, and supplies heat to the amount $\frac{(t' - t'')}{x}$ in cooling from t' to t''. The heat abstracted by re-evaporation of this water is $\frac{L''}{x}$. The balance of heat abstracted is therefore $\frac{L'' - t' + t''}{x} - L'$, which may be written in the form $\frac{L'(1 - x)}{x} - \frac{0{\cdot}30\ (t' - t'')}{x}$. Unless the temperature of the wall is maintained by external agency, it will therefore continue to fall until $(t' - t'') = \frac{L'(1 - x)}{0{\cdot}30}$. For instance, the effect of 5 per cent. wetness is to lower the wall temperature until the range $t' - t''$ is 150° F.

Effect of Water in the Cylinder.—After the accumulation of water in the cylinder has commenced, it is not quite so clear whether the effect of the water would be to increase or diminish the total condensation. It appears probable there would not be a great change. In a working cylinder, the water could not accumulate to any considerable thickness, except in special pockets. If the water were present in sufficient quantity to be thrown into spray, and thoroughly mixed up with the steam, so as to expose a large surface to its action, the water so broken up would almost certainly be carried out of the cylinder with the steam, in proportion to the minuteness of its subdivision. The film left on the surface would, therefore, probably be very thin, and would not seriously affect the result, either in the direction of increase or diminution. To test this supposition, the law of condensation was applied to calculate the mean wall temperatures and the amounts of condensation observed by Donkin in his "Revealer" experiments, in those cases in which water was probably present. The results thus calculated agreed with the observations within the limits of error of the measurements.

Case of Limiting Condensation.—It would appear from the above considerations that the case of limiting condensation in which the re-evaporation is incomplete requires to be treated separately, not only on account of its superior simplicity, but also because the possible variation of the condensation under these conditions differs so greatly from the case in which re-evaporation is complete. In order to test whether any given cylinder is actually in this condition, it is only necessary to insert a thermometer in some convenient hole in the metal of the clearance surface, and to compare the temperature indicated with the mean of the steam cycle. If the clearance surface is found to be at or near the critical temperature, the limiting value of the condensation has probably been reached, and can be approximately calculated by the above method. Next to the average indicator diagram of the trial, the most important datum required for the application of the method is the extent of the clearance surface. In cases where this is not known, and the diagrams are not given, but only the initial and exhaust pressures, a rough estimate of the probable limiting condensation in lbs. per hour, double-acting, may be formed by multiplying the temperature range by four times the area of the piston-face in square feet.

The high-pressure cylinder in Table VII was jacketed with its own exhaust steam. This fact, however, would hardly be enough to account for the missing quantity being nearly twice as

TABLE VII.—ILLUSTRATION OF LIMITING CONDENSATION.

Trial of "Ville de Douvres."[1]

Compound, 2,977 I.HP.; Feed per hour, 61,800 lbs.; Revolutions per minute, 36·8.

	High	Low
1. Cylinder considered	High Pressure.	Low Pressure.
2. Point of stroke	Cut-off	Release.
3. Percentage of feed missing	19·4	27·5
4. Missing quantity . . . . lbs. per hour	12,000	17,000
5. Mean of steam temperature cycle . . . . ° F.	287	205
6. Range ,, ,, ,, ... ,,	86	81
7. Range of pressure . . . . lbs. per square inch	80	24
8. Clearance surface . . . . square feet	85	211
9. Condensation area . . . . ° seconds	794	804
10. ,, corrected for metal cycle	625	540
11. Thermal units absorbed per square foot per minute .	460	400
12. Clearance condensation . . . . . . lbs. per hour	5,200	10,500
13. Barrel surface exposed . . . . . . square feet	47	140
14. Mean condensation area, corrected	230	200
15. Barrel condensation . . . . lbs. per hour	1,100	2,500
16. Total ,, ... ,, ,, ,,	6,300	13,000
17. Remainder to be accounted for	5,700	4,000

great as the condensation limit. Leakage of the high-pressure valves or piston appears probable. The remainder, line 17, in the case of the low-pressure cylinder, corresponds fairly well with the probable wetness of the steam due to adiabatic expansion combined with partial drying due to friction, re-evaporation and other causes. The examination of a number of such cases leads to the following general conclusions. In the high-pressure cylinder, it is not generally likely, owing to the probable dryness of the initial steam, and also to the greater probability of partially explosive evaporation, that the condensation should reach its limiting value. Nevertheless, in the great majority of cases, the missing quantity at cut-off in the high-pressure cylinder is much greater, often many times greater, than can be accounted for on the supposition of limiting condensation, according to this method of analysis. To account for this, the condensation must either be proportional to the density of the steam, or the greater part of the missing quantity represents leakage. The former supposition conflicts with the Authors' experiments and those of Donkin, but is in agreement with those of some other observers. The latter supposition would generally require rates of valve and piston leakage similar to those observed, and would appear the most natural explanation in the light of these experiments.

[1] *Proceedings of the Institution of Mechanical Engineers, 1892, p. 158.*

In the low-pressure cylinder of an unjacketed engine, the condensation may frequently have its limiting value, owing to the initial wetness of the steam. It is also much more important than the leakage for two reasons. The condensation is greater, because the initial surface exposed is much larger. The leakage is less, because the difference of pressure on the valves and piston is much smaller. It is also evident that the perimeter of the ports and piston, upon which the leakage mainly depends, varies directly as the linear dimensions, whereas the surface exposed for condensation varies as the square of the diameter.

Case of Partial Condensation or Complete Re-evaporation.—It is evident, from the smallness of the results obtained in the small single-acting engine at low speeds, and from many similar results obtained by other observers, that, in the case of the simple engine, when working at a moderate ratio of expansion, the initial condensation is often very far below its limiting value. Re-evaporation is probably complete on the clearance surfaces, either at release, or at a very early period in the exhaust, and the walls are probably dry for most of the return stroke. The case of complete re-evaporation, or of partial condensation, as it may be called to distinguish it from the special case of limiting condensation, does not admit of the same simplicity of treatment as the limiting case. The temperature conditions are evidently far less stable, and the amount of cyclical condensation, which depends on the balance of heat loss and supply, is liable to be affected to a much greater extent by the variations of the conditions of running, and by differences of type and arrangement in different engines. It would, therefore, be unsafe to attempt to apply the results deduced from any one engine, under special conditions, to any other engine. At the same time it is possible that some light may be thrown on a very complicated problem by the careful consideration of the results observed in a particular case. The amount of cyclical condensation would appear not to be greatly affected by a moderate variation of speed. Some increase is to be expected, both on account of the diminished range of the metal cycle and also on account of the greater convective action of the exhaust steam; but the former cause partly tends to compensate itself by producing a higher wall-temperature, and the latter depends so much on the initial pressure and on other conditions that it cannot be satisfactorily represented by a formula.

Effect of Varying the Conditions of Running.—Assuming as a first approximation that the cyclical condensation is independent of the speed, it may most conveniently be expressed either in thermal units per minute or in pounds of water per hour.

(*a*) *Variation of Cut-off.*—From observations made at one-half, one-third, and one-fifth cut-off, it is inferred that, when the engine is working single-acting, non-condensing, the cyclical condensation measured per minute at cut-off is to a first approximation independent of the ratio of expansion. If the condensation is measured as a percentage of the indicated steam at cut-off (excluding cushion steam), this result is equivalent to the statement that the percentage condensed increases nearly in direct proportion to the ratio of expansion, defined as being the ratio of the volume occupied by the feed steam at cut-off to the volume of the cylinder. It is not convenient to measure the condensation as a percentage, either of the whole cylinder contents or of the whole cylinder feed, because these involve cushion steam and leakage. In a double-acting engine, the condensation on the barrel surface is necessarily less, and the temperature of the clearance surfaces higher and less variable. The change of each term would therefore be less, and it would seem probable that a similar compensation would occur, leaving the condensation at cut-off unaltered by change in the ratio of expansion. The formula is very attractive on account of its simplicity, which is the first desideratum in a formula of this kind intended to cover roughly a variety of conditions. Its applicability to the case of the double-acting engine is not to be suggested were it not that it appears to represent very fairly many of the most reliable results. In a large class of engine trials, the effect of varying the ratio of expansion r, on the observed percentage z of the cylinder feed condensed at cut-off, is closely represented by the semi-empirical formula of Thurston, $z = a\sqrt{r}$, where a is a constant depending on the other conditions. The numerical value of a for the engine would be 15. According to the Authors' formula $z = \dfrac{100\,c\,r}{100 + c\,r}$, where c has the numerical value 10 in the present case. It is remarkable that these two formulæ, which are at first sight so totally dissimilar, should give results not differing by more than 2·5 per cent. throughout the whole range, from three to twenty expansions. From three to one expansions, the Authors' formula would appear to be preferable, as that of Thurston generally gives results which are too high as compared with experiment.

(*b*) *Double- versus Single-Acting.*—It is possible to draw general conclusions from a consideration of the effect on the distribution of temperature. In a double-acting trial there would be practically no effect of convection of heat by the piston. The gradient of longitudinal conduction would be halved at an early cut-off, and would practically disappear at a late cut-off. The effect would be to raise the temperature of the barrel portion of the admission surface very materially. The condensation on the piston would also be reduced, probably to less than one-half. Against these reductions the effect of the piston-rod at the crank end, and of conduction of heat to the framework of the engine, are to be set, which would tend to lower the temperature at that end as compared with the back end of the cylinder. It is evident that the condensation reckoned as a percentage of the steam would be considerably reduced. For the engine under review, the reduction is estimated at 30 per cent.; that is to say, the total condensation per minute, instead of being doubled, would be increased by about 40 per cent. as an outside estimate. In the case of large engines, the effect of conduction being negligible, the percentage saving by double action would be less.

(*c*) *Variation of Initial Pressure.*—The initial pressure was not sufficiently varied to give any direct information on this point, but experiments show indirectly that the effect is much more complicated than might be supposed. The external loss of heat from the cylinder would be increased nearly in proportion to the increase in the difference of temperature from the surroundings. The internal loss by re-evaporation and by the exhaust steam might be modified in a very different way. The experiments would appear to indicate, as has been suggested by Kirsch, that re-evaporation from the more highly-heated portions of the walls is of an explosive character; that is to say, that a portion of the water-film is blown off the walls without abstracting a full equivalent of its latent heat of vaporization. Condensing at atmospheric pressure with an initial steam temperature of 330° F. to 325° F., the temperature of the cover was over 300° F., and the platinum thermometer in the cover appeared to show that re-evaporation was complete almost as soon as the indicated temperature fell below this point. This observation at once suggested the partially explosive character of the evaporation as a possible explanation of the high temperature attained by the cover. The same explanation probably applies to a less extent to the hotter parts of the barrel surface, which appeared to have been gaining much more heat by condensation than they were losing by re-evaporation. On the cooler parts the balance of heat supplied would be simply that due to the small difference in total heat between the steam condensed at a higher and evaporated at a lower temperature. At one-fifth cut-off the evaporation apparently ceased to be explosive at a temperature between 270° F. and 260° F. The steepness of the temperature gradient along the sides of the cylinder cannot otherwise be satisfactorily accounted for. If the possibility of explosive evaporation at higher temperatures is admitted, depending partly on the diminished surface tension of the water and partly on the greater density of the steam, it is clear that the condensation may not necessarily increase continuously with increase of initial pressure. This result was arrived at independently of Kirsch, from the evidence of the observations. Kirsch makes the suggestion, not from direct experiment, but as a possible explanation of the smallness of the condensation observed in practice as compared with that which would theoretically be required, supposing that the surface of the walls were raised to the temperature of the steam, on the usual assumption that the rate of condensation of steam is practically infinite. Without further evidence, it would not be fair to conclude that re-evaporation at higher temperatures is always of this character. The conditions of the hole in the cover are not quite the same as those of the plane surface; but the observations suggest a possibility which obviously requires consideration. On one occasion a curious effect was accidentally obtained as an illustration of the possible consequences of explosive re-evaporation. When the electric-lighting engine was unexpectedly shut off, the boiler-pressure rapidly rose nearly 10 lbs. above the usual limit. This produced a rise of temperature of 6° F. at the back end of the cylinder, but the temperature at the middle of the cylinder rose more than 15° F., from 264° F. to nearly 280° F. The other conditions of running were unchanged with the exception of a slight increase of speed. As a general rule the changes of temperature at this point of the side were less than those at the back end of the cylinder. It is difficult to account for this abnormal rise of temperature, except by supposing that the re-evaporation at this point ceased to be normal, and became partially explosive. It will be observed that the change of temperature took place at that part of the scale which for other reasons appears to be the critical point in a non-condensing engine.

(*d*) *Variation of Wetness.*—It has long been recognized that the presence of water in the cylinder or of priming in the steam, must have the effect of increasing condensation. In a small single-acting engine, at low speeds, and without special precautions as to lagging the cylinder or drying the steam, practically conclusive evidence was obtained that the action of the metal alone was competent to produce the observed effects. At the same time an illustration was furnished of the abstraction of heat by the condensation and re-evaporation of wet steam, which shows that initial wetness of the steam is probably one of the most powerful factors in increasing cylinder condensation. Provided that the initial wetness is small, and that the lowering of the wall-temperature produced by it is not sufficient to greatly change the other conditions upon which the balance of heat depends, it is possible to represent the effect in a simple manner by combining the formula already given with the law of condensation. If $t' - t^\circ$ be the mean difference of temperature between the steam and the walls during condensation when the steam is dry, the balance of heat supplied by condensation and evaporation is approximately $0 \cdot 60\ (t' - t^\circ)\ W'$ per hour, where W′ is the weight condensed. For a small lowering of wall-temperature, the balance of heat required would not be greatly reduced, and the condensation would be increased nearly in the same proportion as the difference of temperature $t' - t^\circ$. Upon these assumptions, it is found that, for initial steam of a percentage dryness $100\,x$, the initial condensation, as compared with that due to dry steam, is increased by a percentage given by the formula $\dfrac{100\ L\ (1 - x)}{1 \cdot 2\ (t' - t^\circ)}$. If the temperature difference for dry steam is 30 F°. and L = 900, the effect of 1 per cent. of wetness would be to increase the initial condensation by 25 per cent. In the majority of partial condensation cycles, the errors involved in the above assumptions are of such a nature as to make this formula hold through a somewhat wider range than would otherwise be the case, but it cannot be trusted beyond 50 per cent. increase, and should be regarded, in any case, as showing rather the general nature of the effect than its absolute magnitude.

(*e*) *Variation of Back Pressure.*—For a given initial pressure, the wetness of the exhaust steam, due to adiabatic expansion, will depend on the back pressure. The cooling of the internal surfaces during exhaust, apart from re-evaporation, will depend on the wetness of the exhaust steam quite as much as on its temperature. From the reports of trials in which the jacketed surfaces of cylinders, valve-chests, and receivers are given, it is possible to estimate that the rate of abstraction of heat by wet steam in motion under such conditions does not probably ever exceed 1 T.U. per square foot per minute per 1° F. difference of temperature, and may be very considerably less. If the wetness per cubic foot, and not per pound, of the steam, is considered, it would appear probable that the cooling effect of the exhaust steam in a condensing engine may often be actually less than in a non-condensing, but that on the average there is no decided difference.

(*f*) *Effect of Compression.*—If there were no interchange of heat between the walls and the steam, compression to the initial pressure would restore the cushion steam to its initial state. The volume at compression in the present case was one-third of the volume of the cylinder and clearance, and probably included at least two-thirds of the heat abstracted by the exhaust steam. This would explain the very considerable superheating of the steam observed during compression close to the walls. The effect of an early compression may thus be regarded as equivalent to a considerable reduction of the loss of heat due to the exhaust steam, which is probably, next to initial wetness, the most potent factor in abstracting heat from the walls. The effect of an early release is probably similar to that of an early compression in reducing condensation, though it acts in a different way. In the case of partial condensation an early release allows less time for the condensation of wet steam on the colder parts of the walls towards the end of the stroke, and more time for the walls to dry before the return stroke of the piston, a condition probably unfavourable to piston leakage. It is possible that an early release may diminish the condensation and the exhaust waste sufficiently to more than compensate for the loss of area on the indicator diagram. In the cases of limiting condensation, on the contrary, the effect of an early release may be to increase the condensation, and to lower the temperature of the walls by increasing the available evaporation and condensation areas.

Effect of Superheating.—It would appear improbable that superheated steam can supply much heat to walls which are below the saturation temperature, except in so far as it condenses on the walls. Since the superheat is a very small proportion of the total heat, it may be naturally supposed that the rate of condensation of superheated steam is not very different from that of saturated steam. If the superheat be s°, and the saturation temperature of the steam t', the quantity of heat supplied to the walls by the condensation and re-evaporation of a weight, W′, of superheated steam will be $Q = 0 \cdot 60\ (t' - t^\circ) + 0 \cdot 5\ s$ per pound, where t° is the mean wall-temperature, and the specific heat of steam is taken as $0 \cdot 5\ s$. The law of condensation, making the same assumptions as in the case of initial wetness, leads to a similar formula for the reduction of the initial condensation by a small degree of superheating. If $(t' - t^\circ)$ represents, as before, the difference of temperature between the steam and the wall-surface when the engine is using saturated steam with the same cycle, the percentage reduction of the initial condensation, effected by the use of slightly superheated steam, is given by the expression $\frac{100\,s}{2 \cdot 4\,(t' - t^\circ)}$. This formula may be taken as holding approximately for the clearance surface, provided that s is not greater than $\frac{1}{2}\,(t' - t^\circ)$. The reduction of condensation in the case in which $s = t' - t^\circ$ amounts to about 33 per cent. instead of 41 per cent. as given by the above formula. The difference of temperature between the steam and the clearance surface is diminished approximately in the same proportion.

Besides diminishing the initial condensation by raising the temperature of the wall-surface, the use of superheated steam diminishes the wetness during expansion, and therefore considerably reduces the exhaust waste and the abstraction of heat by the condensation of wet steam towards the end of the stroke. It is also probable that it may tend to diminish leakage.

Effect of Jacketing.—The effect of jacketing a cylinder with steam at boiler-pressure, is to raise the temperature of the jacketed walls very nearly to that of the boiler if the jackets are working properly. According to the law here proposed, the condensation on the jacketed surfaces would be practically negligible, and the clearance surfaces are by far the most important. In large engines it would consequently be of little use to jacket the sides, but in small engines the clearance surfaces would also be heated by conduction so as to be practically jacketed. It is not improbable that a part of the economy due to jacketing, especially in small engines, is owing to the reduction of leakage. The valves and valve-seats become so heated by conduction that the possible water-leakage is minimised. From the same point of view, the drying of the steam in jacketed receivers must have a beneficial tendency, as there is then less water available to cool the valve-surfaces by re-evaporation in the exhaust.

Variation of Size and Surface.—The effect of variation of surface exposed, and particularly of the extent of the clearance surface, is probably different according as the condensation is of the partial or limiting type. If the main factor in the abstraction of heat from the clearance surface is the condensation of wet initial steam, as is probably the case when the condensation has its limiting value, the amount of heat abstracted and the initial condensation will vary simply as the surface exposed. In this case it is of primary importance to know the full extent of the clearance surface on which the greater part of the condensation takes place. In order to reduce the loss as far as possible, the extent of the clearance surface should be reduced to a minimum. If, on the other hand, the initial steam is dry, and the condensation is of the partial type, the extent of the clearance surface is a matter of much less consequence, because increase of surface has the effect of raising the temperature. It is probably best, in the case of partial condensation, to neglect differences of clearance surface in comparing different engines, and to take the clearance surface at each end of the cylinder as being πd^2, where d is the diameter. The actual clearance surface cannot be more than 50 per cent. less than this, and is seldom more than 50 per cent. greater. For all practical purposes the equivalent clearance surface forms a sufficient basis of comparison. But the barrel surface exposed up to cut-off, allowing for the time of exposure, may be very simply represented, if desired, by the addition of the term $l\,d\,c$, where c is the cut-off fraction, and l the stroke. The state of the surfaces is not important if the rate of condensation of steam is regarded as the main factor in limiting the amount of heat absorbed. The presence of a thin film of grease or rust may make the cycles observed at a given depth in the metal of smaller range, but will not really make much difference in the amount of condensation, unless the film is so thick and obstructive as to greatly increase the surface range of temperature, which is probably seldom the case.

Effect of Conduction.—In two similar cylinders of different linear dimensions, but with the same distribution of temperature, the loss of heat from the admission surface by conduction, will be proportional to the thickness of the metal. The initial loss due to conduction, reckoned as a percentage of the steam, for cylinders of the same thickness, will vary inversely as the cube of the linear dimensions. It is necessary to exercise caution in applying the results deduced from small engines to large. In order to estimate the probable effect of conduction, the temperatures have been compared with those obtained by Donkin with a cylinder 6 inches in diameter, and 8 inches stroke. The Authors conclude that the effects of conduction in their engine are not to be neglected as compared with larger machines, but probably do not amount to more than five or ten per cent., and are not such as to seriously vitiate the general nature of their conclusions. In cylinders of different shapes, under similar conditions of running, the loss due to conduction at cut-off reckoned as a percentage of the indicated steam, would vary as $\frac{t}{d\,l^2}$, where t is the thickness of the metal, d the diameter, and l the length of the cylinder. For instance, the effect of conduction in the cylinder above mentioned, would be nearly five times as great as in that of the Robb engine.

Formulæ of Condensation and Leakage.—From the foregoing considerations it will be evident that, when the required data are available, or when it is possible to observe the temperature distribution, no formula can be regarded as being at all satisfactory. It is, nevertheless, convenient to have a simple approximate formula for the purpose of making rough estimates and comparisons, and also as exhibiting the results of the investigation in a brief and compact form. In the case of limiting condensation, when neither the cards nor the extent of the clearance surface are given, the limiting value of the condensation W′ at cut-off, expressed in pounds per hour, may be estimated from the formula,

$$W' = \pi\, d^2\, (t' - t''),$$

where d is measured in feet and t in degrees Fahr. This formula assumes that the clearance surface is $\pi\, d^2$, but makes an allowance of 20 per cent. for the barrel surface. It also assumes that the cut-off is at or near mid-stroke, and that the drop of pressure during admission is small. The factor $(t' - t'')$ is supposed to represent the total range of the steam. The latent heat of the steam is taken as 900. The result, thus estimated, may be corrected by substituting the proper value of the latent heat in each case, and may then be reduced in the proportion $\frac{(1 + \sqrt{n})}{(3 + \sqrt{n})}$, to allow for the effect of the probable range of the metal cycle.

The case of partial condensation may be roughly represented by the formula,

$$W' = C \times S = S\,(t' - t^\circ),$$

where C is the condensation in pounds per hour at cut-off per square foot of total equivalent clearance surface S of the cylinder, supposed unjacketed. The condensation factor C is a function of the initial and exhaust temperatures and of the external conditions, but may be taken as being approximately independent of the speed and the ratio of expansion. Apart from superheating or jacketing, the value of C is probably most affected by the degree of compression. The condensation factor C may also be interpreted as the mean difference of temperature $(t' - t^\circ)$, between the walls and the admission steam, reduced to one-half cut-off.

The effect of size and surface, and of double or single action, may be supposed included in the expression for the surface S. This factor should be taken as $2\,\pi\,d^2$ double-acting, and as $\pi\,d^2$ single-acting.

The effect of jacketing in large engines may be represented by simply omitting the jacketed area from the factor S; but the effect of conduction in small engines cannot be satisfactorily included in the formula. If W is the weight of feed in pounds per hour, accounted for by the indicator at cut-off, and W° the total missing quantity per hour, W + W° represents the total cylinder feed. If the condensation W′ can be estimated, either by a formula or by observing the temperature distribution, the remainder W° − W′, which may conveniently be represented by the symbol W″, may be most probably attributed to leakage. According to the Authors' experiments, the main part of the leakage at any point of the stroke may be represented by the formula,

$$W'' = L\,(p' - p''),$$

where $(p' - p'')$ represents the difference of pressure between the valve-chest and the exhaust. If the factor L cannot be measured, it may be estimated by the method of p. 177, from the mean overlap and the perimeter of the ports. Since the area of the steam-ports is generally designed to vary as d^2 N, where N is the piston speed, L may be taken as being proportional to $d\,\sqrt{N}$ in similar engines.

Illustrations of Partial Condensation.—As a test of the validity of the method of reducing to the equivalent clearance surface in the case of partial condensation, and as showing the relative unimportance of the barrel surface, the following cases are cited, having been selected by Cotterill (*loc. cit.*, p. 334) for a similar

TABLE VIII.—CONDENSING TRIALS.

Authorities and Trial Mark.	Stroke, *l*.	Diameter, *d*.	Revolutions per Minute.	Cut-off, *c*.	Absolute Pressure at Cut-off.	Missing Quantity. Lbs. per Hour, W°.	Missing Quantity. Per Cent. of Feed.	Equivalent Clearance Surface, S.[1]	Condensation Factor, C.
	Feet.	Feet.						Sq. Ft.	°
Mair . . (L)	5·5	2·67	20·3	0·26	46	770	29	33	23
,, . . (M)	5·5	2·67	20·3	0·26	46	1,200	37	52	23
Dallas . (D)	2·5	3·0	56·9	0·20	47	1,460	29	60	24
Hirn and (2)	6·5	2·0	30·5	0·26	54	850	30	32	27
Hallauer (5)	6·5	2·0	30·0	0·16	55	450	25	Superheat, 81° F.	
(HH.). . (6)	6·5	2·0	30·4	0·16	55	760	36	29	26

[1] Calculated by formula, $S = 2\,(\pi\,d^2 + l\,d\,c)$.

purpose. There is considerable range of speed and of relation of stroke to diameter, but the conditions of pressure are fairly comparable, and the leakage similar, and probably small as compared with the condensation. The engines are also sufficiently large to make the effect of conduction practically negligible.

In the trial Mair (L),[1] the sides and base of the cylinder were jacketed, which would have the effect of reducing the unjacketed clearance surface by about one-third. The condensation constants given by Cotterill for these three cases are, (M) 5·3, (D) 7·0, (HH) 3·4. This wide range of values is probably to be explained by his taking the condensation as proportional to the barrel surface. The agreement of the values of C in Table IX is as close as can be expected, and may be taken as showing that, in the case of partial condensation, the equivalent clearance surface is the better basis of comparison.

TABLE IX.—NON-CONDENSING TRIALS.

Authorities and Trial Mark.	Stroke, l.	Diameter, d.	Revolutions per Minute.	Cut-off, c.	Absolute Pressure at Cut-off.	Compression.	Missing Quantity. Lbs. per Hour W°.	Percentage of Feed.	Condensation Factors. Equivalent Clearance Surface S.	Actual Clearance Surface.	Factor, C.
	Feet.	Feet.							Sq. Ft.	SqFt.	
C and N (4–6) .	1·0	0·88	74	0·20	95	0·25	43	24	2·6	3·7	17°
Willans . . .	0·5	1·17	110	0·44	44	36°	87	35	4·6	2·8	19°
Simple $\frac{50}{2\cdot 2}$ . .	0·5	1·17	201	0·44	44	33°	93	24	4·6	2·8	20°
C.E. 1893, p. 174 .	0·5	1·17	408	0·44	43	25°	137	19	4·6	2·8	30°
Willans . . .	0·5	0·83	122	0·60	83	17°	55	20	2·4	1·8	23°
Compound $\frac{90}{3\cdot 2}$.	0·5	0·83	211	0·60	82	13°	54	13	2·4	1·8	23°
Loc. cit. . . .	0·5	0·83	401	0·60	80	—5°	38	5	2·4	1·8	16°
Gately and . (16)	3·5	1·5	68	0·41	65		390	11	18·5		21°
Kietzsch . (17)	3·5	1·5	69	0·42	50	?	800	24	18·5	?	43°
(Thurston) . (18)	3·5	1·5	69	0·40	40		372	16	18.5		20°
Col. English . . Mech. Eng., 1887 . Pl. 90, Fig. 5 . .	1·5	1·33	40	0·30	84	0·00	470	45	12.2	?	39°
J. W. Hill, (Peabody, p. 265). R.C.	4·0	1·5	75	0·16	99	0·08	1,040	29	16·1		65°
H. C.	4·0	1·5	76	0·14	100	0·12	1,100	34	16·0	?	69°
Whk.	4·0	1·5	76	0·17	91	0·08	1,124	32	16·2		70°

[1] Minutes of Proceedings Inst. C.E., 1884, vol. lxxix. p. 340.

Table IX illustrates the fact that the results obtained by the thermoelectric method, even when reduced to the equivalent clearance surface, are smaller than any similar results, except perhaps a few of Willans'. The effect of an early compression is also well marked. In the last three cases the steam is recorded as having been occasionally 5 per cent. wet, and the condensation was probably limiting.

Estimation of Leakage.—In the case of partial condensation, it is evident that the result may be so profoundly affected by differences of wetness or of compression, that it would not be justifiable to take the small result found in the case of the Authors' engine, and to assume that the larger results shown by other engines were to be entirely attributed to leakage. With the Robb engine, running double-acting under the same conditions, making allowance for piston convection, etc., the condensation factor ought not to exceed 13 lbs. or 14 lbs. per square foot of equivalent clearance surface per hour. This result is much smaller than that shown by any of the double-acting cases cited. Although this value of the condensation factor cannot be directly applied to the estimation of leakage in other engines, the formula may be of use in comparing different trials of the same engine, or similar trials of different engines, under strictly comparable conditions. In the case of limiting condensation, if the clearance surface and the steam cycle are known, a definite limit to the condensation is at once afforded. Any excess may be reasonably attributed to leakage. Unless the steam is known to be wet, the temperature of the walls should be observed, because it cannot otherwise be certain that the condensation is really limiting. If it happens to be partial the leakage will be under-estimated. If $C = 20$ is taken as being a probable average value of the condensation factor in the case of simple engines, the total loss would be given by a formula of the type

$$W^\circ = 40\,\pi\, d^2 + \frac{d\sqrt{N}\,(p' - p'')}{10}.$$

which would make the leakage relatively more important in small engines and at high piston speeds and pressures.

Prof. C. A. Smith, using a formula of the type $W^\circ = C' d\,(t' - t'')$ to represent the total losses in simple unjacketed engines, measured in lbs. per hour per degree of steam range and per foot of piston diameter, finds values of C' which vary between 1·85 and 4·72. The same series of trials may be represented on the Authors' formula by values of C from 15 to 25, and values of the leakage constant from 5 to 10. This does not appear to be an unlikely range of variation, but unless the condensation is often limiting in simple engines, the leakage must frequently be the greater.

The data required for the exact application of the proposed method of estimating leakage are not generally available in any extant trials; but to exemplify the general conclusions to be derived from the analysis, the triple-expansion trials of the experimental engines at Owens College, described by Osborne Reynolds,[1] may be selected, not only on account of the unimpeachable accuracy of the observations, but also because these engines would appear to have achieved some of the best performances on record for engines of so small a size. If it can be shown that, even in these engines, in spite of their record performance, a large part of the missing quantity is probably to be credited to leakage, it follows, *a fortiori*, that a similar cause may be suspected in more ordinary cases. These trials possess the further advantage of an unusually complete and unreserved record of all the leakage tests which were applied, and of the various minor leakages which were discovered and rectified from time to time.

The trials 41, 35, and 40, in which the receivers were jacketed but not the cylinders, will be taken as an illustration. The effect was probably to dry the steam completely at the low speed for the low-pressure cylinders, but less completely at the higher speeds. The following are the data for feed and receiver-jacket condensation in the three trials:—

CYLINDER JACKETS EMPTY. RECEIVER JACKETS AT BOILER PRESSURE.

Trial number	41	35	40
Boiler pressure, absolute	204	205	201
Feed per hour (hot well)	464	688	1,055
Jacket condensation, lbs. per hour	99	117	158

The following are the data for the three separate engines:—

Engine Number	I.			II.			III.		
Trial number	41	35	40	41	35	40	41	35	40
Expansions, r	2·7	2·3	2·0	2·4	2·4	2·2	2·7	3·0	2·6
Revs. per minute, n	146	229	322	127	215	320	109	184	276
Pressure range, $p' - p''$	137	132	123	45	50	52	20	21	23
Temperature range, $t' - t''$ degrees	83	78	72	68	70	69	123	115	107
Missing quantity, z° per cent.	40	29	22	41	38	30	51	48	32
Missing quantity, w° lbs. per hour	185	200	232	190	262	317	237	332	337
Condensation limit	43	41	39	83	88	88	310	300	285

[1] Minutes of Proceedings Inst. C.E., vol. xcix. p. 152.

The condensation limit given in the last line, in default of the cards, and of the actual area of the clearance surface, is estimated by the method explained. No allowance is made for the variation in the ratio of expansion, but since the three engines were similar, the values estimated may probably be taken as showing the relative order of magnitude of the condensation-limit to be expected in the three cylinders. Taking engine No. III, it would appear probable that at the lowest speed the steam was practically dry, and the condensation consequently partial. At the higher speeds, the drying in the receivers was probably insufficient, and the condensation had nearly, if not quite, reached its limit. In the case of engine No. I, the condensation-limit is obviously insufficient to explain the missing quantity. It is likely that both the condensation and the leakage increased slightly with increase of speed, but the leak appears to have been much more important than the condensation. The rate of leak required would have been between 1·0 lb. and 1·5 lb. per pound pressure per hour. A leak, roughly estimated at 30 lbs. per hour, was observed on one occasion with this valve standing. The rate of leak required for the valve when running would appear not unlikely, considering the type of the valve, and that it was specially designed for a high speed. Part of the leak may have been due to the piston and to the cut-off valve, but the pistons appear from other evidence to have been fairly tight. The valve was probably mainly responsible. In the case of engine No. II, the same observations are applicable. The rate of leak required would have been between 3 lbs. and 4 lbs. This would appear somewhat excessive, but it is recorded that leakage of the cut-off valve was inferred from the cards in this engine, and was rectified at a subsequent date. It is evident that a leak of the kind described as having been discovered in this valve might be expected to increase considerably with the speed, as the amount of leak would depend on the inertia. The rate of leak in engine No. III may have been upwards of 2 lbs., but is evidently much less important than the condensation, owing to the smaller pressure difference on the valve. If the leakage therefore is rejected, the rate of condensation of steam must increase rapidly with the density. Setting aside the experiments described in the present Paper, this would appear a perfectly tenable hypothesis. Unless, however, the surface of the cylinder has been underestimated, the metal in engine No. I seems hardly capable of accounting for the whole missing quantity at the lower speed, even if the temperature range of the surface of the metal were the same as that of the steam; at least, on any reasonable theory of condensation. The inferences drawn from the above case may be taken as typical of a great number of other cases which might be given.

Direct experiments by Colonel English[1] on initial condensation with a portable engine appeared to show a rate of condensation varying roughly as the density of the steam. The method employed was similar to that used by the Authors for measuring the exhaust leak, except that steam was admitted at each revolution to the clearance space at the back end of the cylinder, and that the sides of the cylinder were jacketed. The speed and pressure were considerably varied, but it was assumed in reducing the observations that the condensation per hour varied as the square root of the speed. The effect observed when measured per hour appears to be much more nearly independent of the speed. It also appears to be nearly a linear function of the pressure difference. At a mean density of 0·15 the results observed by Colonel English would correspond to the condensation of about 40 lbs. per square foot per hour, which is nearly four times the value found by the Authors at a higher density in an unjacketed cylinder. These results may be more naturally explained by supposing a comparatively moderate condensation varying as the temperature difference, combined with a much larger leakage varying as the pressure difference. The rate of leak thus required would be about 1·5 lb. per pound pressure per hour in the non-condensing experiments, and a slightly smaller rate in the condensing experiments. It may be observed that this rate of leak is the same as that found in one of the Authors' valves, which was proved to be absolutely steam-tight when stationary. It does not imply that the valve was in bad condition, or that the engine was at fault in any way.

Expression in Terms of the Steam and Feed.—It is frequently convenient to express the condensation and leakage in terms of the indicated feed and also of the total feed. The percentages of indicated feed being designated by y, and the percentages of total feed by z, the following notation is afforded:—

$$y^\circ = \frac{100\,W^\circ}{W},\; y' = \frac{100\,W'}{W},\; y'' = \frac{100\,W''}{W},\; y^\circ = y' + y''.$$

$$z^\circ = \frac{100\,W^\circ}{(W^\circ + W)},\; z' = \frac{100\,W'}{(W^\circ + W)},\; z'' = \frac{100\,W''}{(W^\circ + W)},\; z^\circ = z' + z''.$$

$$z^\circ = \frac{y^\circ}{(100 + y^\circ)},\; z' = \frac{y'}{(100 + y^\circ)},\; z'' = \frac{y''}{(100 + y^\circ)},\; y^\circ = \frac{z^\circ}{(100 - z^\circ)}.$$

[1] Proceedings of the Institution of Mechanical Engineers, 1887, p. 503.

If V is the piston displacement per revolution, n the revolutions per minute, w the density of the steam at cut-off in lbs. per cubic foot, and r the ratio of the volume of the feed steam at cut-off to the piston displacement, the formula for the indicated feed in lbs. per hour is $W = \frac{60\, n\, V\, w}{r}$. Also $W' = C\,S = \frac{y W}{100}$, $W'' = L\,(p' - p'')$. So that, $y' = \frac{100\, C\, S\, r}{60\, n\, V\, w}$, $y'' = \frac{100\, L\,(p' - p'')}{60\, n\, V\, w}$. The piston displacement V is to be taken as $\frac{\pi\, d^2\, l}{4}$ single-acting, and as $\frac{\pi\, d^2\, l}{2}$ double-acting. The value of the ratio $\frac{S}{V}$ is therefore $\frac{4}{l}$ either single-acting or double-acting.

If the rate of leakage L is taken as proportional to the product of the diameter of the cylinder and the square root of the normal piston speed N, $L = L'\, d \sqrt{N}$, where L' may be called the leakage constant, and has values which probably vary between 0·20 and 0·05 in engines of different types.

The formula for the leak percentage y'' may then be written in the simpler form, $y'' = \frac{L''\, r}{d \sqrt{N}}$, where L'' is a new constant, which is practically proportional to L' in all cases which occur in practice of engines running at their normal piston speed. This shows that the effect of leakage on the performance of different engines may be expected to diminish in proportion as the diameter and the square root of the piston speed are increased. If, however, an engine running at its normal speed is being compared with the same engine running below its normal speed, $y'' = \frac{L''\, r \sqrt{N}}{2\, d\, n\, l}$, which shows that the effect of leakage on the performance will increase in direct proportion as the speed is diminished.

The formula for the condensation percentage y' may similarly be written, $y' = \frac{C'\, r}{N\, w}$, where $C' = \frac{40\, C}{3}$, and N is the piston speed, normal or otherwise. Comparing this formula with the leakage formula, the effect of condensation on the performance is not diminished by increase of diameter, but is much more affected by increase of piston speed; it is also considerably diminished by increase of initial pressure, which is not the case with leakage.

The formula for the total percentage loss due to both causes is—

$$y^\circ = y' + y'' = \frac{C'\, r}{N\, w} + \frac{L''\, r}{d \sqrt{N}}.$$

It is interesting to compare this formula with that of Cotterill, $y^\circ = \frac{C'' \log_e r}{d \sqrt{n}}$, which is intended to represent the results of experiment on the assumption of negligible leakage.

As an illustration of the nature of the consequences involved in the assumption that the rate of condensation of steam varies as the density, the triple-expansion trials already cited may be taken. From Cotterill's formula the values of the condensation constant C″ for the three engines at the lowest speeds, are:—No. I, 3·4; No. II, 5·0; No. III, 11·0. As the steam was probably fairly dried by the receivers at this speed, it is difficult to see why the value of the condensation constant should be so very different for three engines of a similar type and speed, unless the rate of condensation of steam does not vary as the initial density.

Summary of Conclusions.—The observed wall-temperature cycles show that the range of surface temperature, and the interchange of heat between the walls and the steam, is determined chiefly by the temperature of the walls in each case, and by the finite rate of condensation of steam. For this finite rate of condensation, the value 0·74 B.T.U. per square foot per second per 1° F. difference of temperature, at 300° F., is obtained, a result approximately equivalent to 2·7 lbs. of steam condensed per square foot per degree per hour.

The form of the wall-cycles, and other evidence show that the law of re-evaporation is the same as that of condensation, and that both are probably independent of the pressure. The amount of condensation in any cylinder can, therefore, be deduced by the observation of the distribution of wall-temperature while the engine is running.

From the form of the law of condensation there appears to be for any cycle a limit of condensation when the temperature of the walls is the time average of the steam temperature-cycle. Under this condition of "limiting" condensation, the re-evaporation is incomplete, and the temperature of the cylinder is maintained by the mechanical rejection of condensed water, so that it cannot fall much below this point.

The condensation observed was far below this limiting value, and the initial steam was probably uniformly dry. But a marked effect due to the condensation of wet steam was observed, which leads to the inference that, owing to the abstraction of heat by the subsequent re-evaporation of the deposited wetness, the condensation must always be limiting in cases where the initial wetness of the steam is considerable.

The case of "partial," as opposed to limiting condensation, is probably more common in simple engines. The amount of initial condensation per hour in this case appears to be nearly independent of the speed and of the ratio of expansion, and to vary little with the initial and exhaust temperatures. Simple approximate expressions are given, deduced from the law of condensation, for the effect of initial wetness or of superheating of the steam, which are probably, together with the degree of compression, the most important factors in determining the result.

Illustrations have been given of the method by which, if the requisite data are available, the condensation at any point of the cycle may be correctly computed by means of the condensation areas on the temperature-cycle diagram. This method requires a knowledge of the extent of the clearance surface, and also, in the case of partial condensation, of the mean temperature of the surface, in addition to the steam cycle. The application of this method may be expected to throw light on other causes of loss, and particularly on the amount of leakage under the actual conditions of running, which, from these experiments, appear to be a much more important source of loss than is generally admitted.

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The Paper is accompanied by twenty-four drawings, from which Plate 6 and the *Figs.* in the text have been prepared.

[APPENDIX.

APPENDIX.

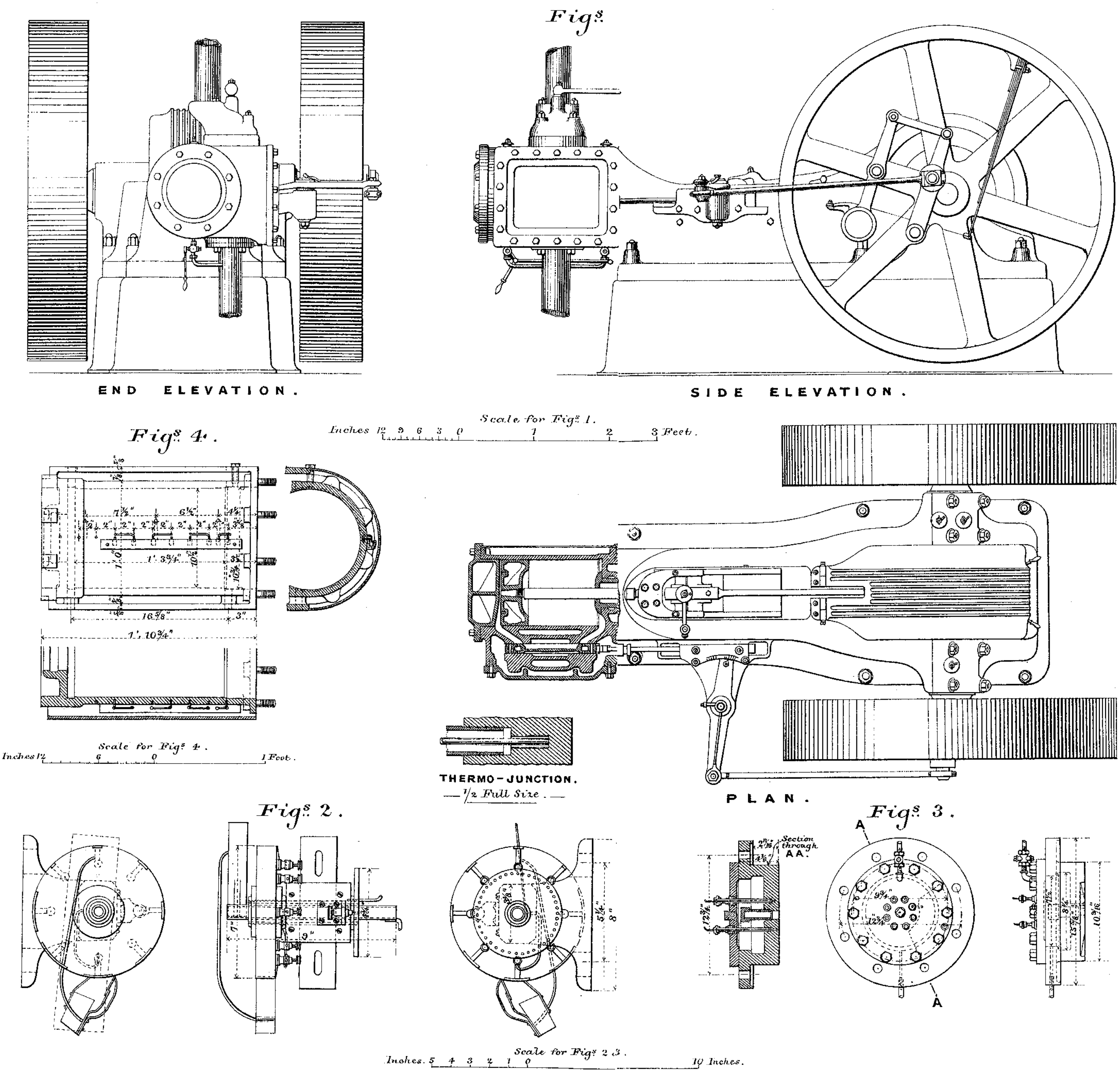
Verification of the Temperature Criterion for Limiting Condensation.

As a further confirmation of the law of condensation and re-evaporation proposed in the present Paper, it was evidently desirable to observe the temperature of the cylinder-walls of an engine, in some case in which water was undoubtedly present in the cylinder. The temperature of the walls in this case should be the same as the time-mean of the steam temperature. A case of this kind presented itself recently in the McGill workshop engine, which had to be run throttled, when the cut-off valve had been removed for repairs. The whole of the indicator-diagrams, on reduction, showed the temperature of the mean of the steam cycle to have been the same as that of the clearance surface, within the limits of error of the observations, although on some occasions diagrams were purposely taken when the load was suddenly changed. On November 13th the cut-off valve was replaced, and the condensation after this date apparently ceased to be limiting. The following Table gives a number of the results observed. It will be seen that the relation in question appears to hold over a considerable range of temperature so long as the condensation is limiting.

Wall Temperatures in Limiting Condensation.

Date, 1896.	Cut-off.	Remarks.	Temperatures of Steam Cycle.				—	Temperatures of Metal.
			Max.	Min.	Range.	Mean.		
			° F.	° F.	° F.	° F.		° F.
Oct. 17	¾	Throttled .	272	228	44	253	Non-condensing	254
,, 17	¾	Half open .	325	268	57	298	,,	299
Nov. 24	¾	Throttled .	283	241	42	260	,,	260
,, 5	¾	,, .	292	243	49	266	,,	267
,, 5	¾	Rising . .	277	232	45	253	,,	250
Nov. 6	¾	Falling . .	253	204	49	230	Condensing .	233
,, 6	¾	Throttled .	236	200	36	221	,, .	221
,, 9	¾	,, .	255	212	43	235	,, .	236
,, 10	¾	Rising . .	277	230	47	251	,, .	248
Nov. 13	⅓	Half open .	326	253	73	284	Condensing .	297
,, 13	⅓ .	Quarter open	312	232	80	265	,, .	283

[Discussion.


Figs.
END ELEVATION.
SIDE ELEVATION.
Scale for Figs. 1.
Inches 12 9 6 3 0 1 2 3 Feet.
Figs. 4.
Scale for Figs. 4.
Inches 12 6 0 1 Foot.
THERMO-JUNCTION.
1/2 Full Size.
PLAN.
Figs. 2.
Figs. 3.
Section through AA.
A
A
Scale for Figs. 2 3.
Inches. 5 4 3 2 1 0 10 Inches.