

EFFICIENCY OF INTERNAL-COMBUSTION ENGINES.

Final Report of a Committee appointed on the 6th of November, 1903, to consider and report to the Council on the Standards of Efficiency of Internal-Combustion Engines.¹

Members of the Committee :

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Report on Gas-Engine Trials at the National Gas-Engine Company's Works at Ashton.

(Adopted by the Council, 19 December, 1905.)

NOMENCLATURE.

- d difference between wet and dry bulb.
- f Glaisher's factor.
- t_1 temperature of air at anemometer.
- t_2 temperature of water entering calorimeter.
- t_3 temperature of gas at meter.
- t_4 temperature of water entering jackets.
- t_5 temperature of water leaving calorimeter.
- t_6 temperature of water leaving jackets.
- t_0 temperature of gases leaving calorimeter.
- w weight of dry air entering per hour.
- w_1 weight of steam associated with 1 lb. of dry air at dew-point.

¹ For the Preliminary Report of the Committee see Minutes of Proceedings Inst. C.E., vol. clxii. p. 307.

- w_2 weight of steam brought in by air per hour.
- w_3 weight of steam in each pound of exhaust gas saturated at t_0 .
- w_4 weight of water produced per hour by combustion of gas.
- w_5 weight of gas supplied per hour, assumed dry at entry.
- w_6 weight of dry exhaust gases per hour.
- w_7 weight of steam carried away per hour by exhaust gases.
- w_8 weight of cooling-water leaving calorimeter per hour, observed.
- w_9 weight of cooling-water entering calorimeter per hour.
- w_{10} weight of jacket-water entering and leaving per hour.
- p pressure of steam at dew-point.
- p_1 atmospheric pressure.
- p_0 pressure of dry air at anemometer.
- v volume of 1 lb. of dry air in cubic feet at t_1 and p_0 .
- V volume of air entering per hour measured by anemometer.
- h total heat of the steam associated with each pound of dry exhaust gases reckoned from 32°F .

1.—PRELIMINARY SUMMARY OF THERMAL EFFICIENCIES.

It was mentioned in the preliminary Report¹ that the National Gas-Engine Company, at the instance of Mr. Dugald Clerk, had placed three gas-engines at the disposal of the Committee for testing-purposes, and had been good enough to fit up the apparatus required for the tests.

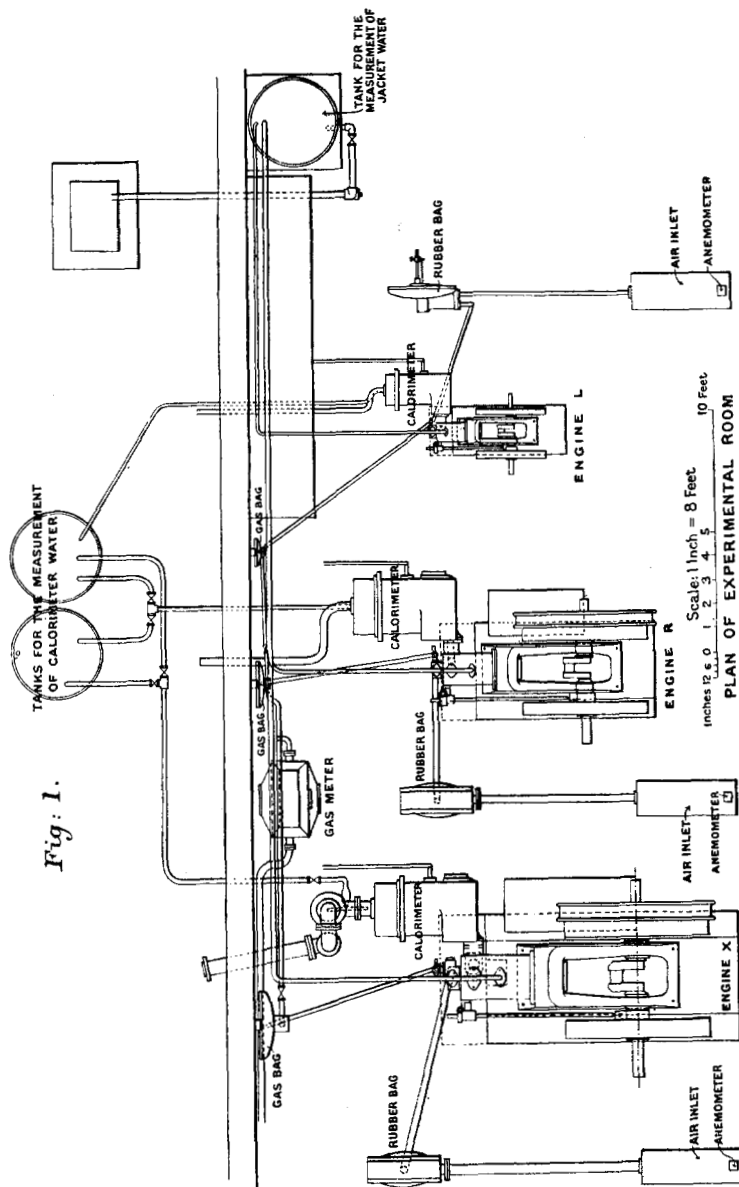
The engines, which are denoted by the letters L, R, X in the Tables, were arranged side by side in a testing-room (*Fig. 1*) completely supplied with apparatus for making the necessary measurements, the design of some of the plant being suggested by the Committee, and the plant being constructed by the Company.

The interesting feature of the trials was that the three engines were designed to work with equal compression-ratios, and, as will be seen from Table II., this was realized within 3 per cent. Their respective horse-powers differed widely. In round numbers the compression-ratio was $5\frac{1}{2}$ to 1, whilst the small engine was rated at 5 I.H.P., the middle-sized engine 25 I.H.P. and the largest engine 56 I.H.P.

One of the main objects of the Committee in carrying out the tests was to ascertain whether the ideal engine proposed in the Preliminary Report would be a satisfactory standard of comparison for engines varying so much in power and yet theoretically all of the same efficiency, since the compression-ratio was constant.

¹ Minutes of Proceedings Inst. C.E., vol. clxii. p. 307.

Anticipating the detailed description of the tests and the mass



of figures connected with them, the following Table gives the

figures concerned in this comparison. Two trials are given to each engine, one at full load, or, more strictly, the most economical load, and the other at half-load.

TABLE I.—THERMAL AND MECHANICAL EFFICIENCIES.

Designation of Engine .	L		R		X	
Trial Number	3	2	14	9	17	15
I.H.P.	3·6	5·72	14·5	25·9	34·1	56·3
B.H.P.	2·87	5·2	10·82	20·9	27·9	52·7
Mechanical efficiency .	0·80	0·90	0·75	0·80	0·82	0·94
British thermal units per hour equivalent to the I.H.P. $\times 2,545$.	9,166	14,557	36,902	65,915	86,784	143,283
British thermal units supplied per hour reckoned on the lower calorific value	32,260	49,630	117,200	187,700	267,500	450,600
Absolute thermal efficiency reckoned on I.H.P. .	28·0%	29·0%	31·5%	35·0%	32·5%	31·8%
Absolute thermal efficiency reckoned on B.H.P.	22·4%	26·1%	23·6%	28·0%	26·7%	29·9%
Clearance volume Maximum volume $= \frac{1}{r}$.	0·18	0·18	0·18	0·18	0·186	0·186
Thermal efficiency of air standard engine calculated from $\eta = 1 - \left(\frac{1}{r}\right)^{0·4}$	0·496	0·496	0·496	0·496	0·49	0·49
Relative efficiency from I.H.P.	56·4%	58·4%	63·5%	70·6%	66·3%	65·0%
Relative efficiency from B.H.P.	45·2%	52·6%	47·6%	56·4%	54·5%	61·0%
Gas per I.H.P.-hour . .	15·78	15·33	13·77	12·78	13·67	13·94
Gas per B.H.P.-hour . .	19·8	16·87	18·45	15·84	16·7	14·80
Ratio by volume of air to gas in explosive charge .	8·49	9·15	8·42	9·17	7·97	8·27

The gas per I.H.P. and B.H.P.-hour, and the ratio of air to gas in the explosive mixture, are added to the Table for the sake of comparison.

EFFICIENCIES CALCULATED FROM THE INDICATOR HORSE-POWER.

Rearranging the results in the Table, the efficiencies at full load and half-load are as follows:—

ENGINES WORKING AT ABOUT FULL LOAD.

	L	R	X
Indicator Horse-Power	5.72	25.9	56.3
Absolute thermal efficiency	0.290	0.350	0.318
Relative efficiency	0.584	0.706	0.650

ENGINES WORKING AT ABOUT HALF-LOAD.

	L	R	X
Indicator Horse-Power	3.6	14.5	34.1
Absolute thermal efficiency	0.280	0.315	0.325
Relative efficiency	0.564	0.635	0.663

The absolute thermal efficiency varies between 29 and 35 per cent. at full load and between 28 and 32½ per cent. at half-load. The efficiency relatively to the standard air-engine varies between 58½ and 70½ per cent. at full load and between 56.4 and 66.3 per cent. at half-load.

The values given for the indicator horse-power, and consequently those for the mechanical efficiency, are probably not very accurate, because the indicator-diagrams vary, and the mean of the limited number taken in a trial is not the true mean. It is at least probable that the mechanical efficiency was more nearly constant for the three engines than the figures in the Table indicate.

The mean mechanical efficiency of the full-load trials is 88 per cent. and that of the half-load trials is 79 per cent., values nearly in the ratio which they should bear if the frictional work is independent of the variation of load. If the indicator horse-power is calculated from the brake horse-power, assuming 88 per cent. for the mechanical efficiency at full load and 79 per cent. at half-load, the values given in the following Table are obtained:—

ENGINES WORKING AT ABOUT FULL LOAD.

	L	R	X
Indicator Horse-Power	5.91	23.8	59.9
Absolute thermal efficiency	0.303	0.323	0.338
Relative efficiency	0.611	0.650	0.690

ENGINES WORKING AT ABOUT HALF-LOAD.

	L	R	X
Indicator Horse-Power	3.64	13.7	35.4
Absolute thermal efficiency	0.287	0.298	0.337
Relative efficiency	0.579	0.600	0.687

It will be seen that the efficiencies thus calculated are more accordant than those previously given, but further consideration will be given to the question of the indicator horse-power in a subsequent part of the Report. The mean absolute efficiency above is 32.1 per cent. at full load and 30.7 per cent. at half-load. The mean relative efficiency is 65 per cent. at full load and 62.2 per cent. at half-load.

EFFICIENCIES CALCULATED FROM THE BRAKE HORSE-POWER.

The values given for the brake horse-power are not open to the same doubt as those for the indicator horse-power, and were no doubt accurately determined.

ENGINES WORKING AT ABOUT FULL POWER.

	L	R	X
Brake Horse-Power	5.2	20.9	52.7
Absolute thermal efficiency	0.261	0.280	0.299
Relative efficiency	0.526	0.564	0.610

ENGINES WORKING AT ABOUT HALF-POWER.

	L	R	X
Brake Horse-Power	2.87	10.82	27.0
Absolute thermal efficiency	0.224	0.206	0.267
Relative efficiency	0.552	0.476	0.545

The absolute thermal efficiency varies between 26.1 per cent. and 29.9 per cent. at full load and between 22.4 and 26.7 per cent. at half-load, being greater as the size of the engine increases, as would be expected. The mean absolute efficiency is 28.0 per cent. at full load and 24.2 per cent. at half-load.

The efficiency relatively to the air-engine standard at full power varies between 52·6 and 61·0 per cent., increasing regularly to a small extent as the engine increases in size. This might be anticipated, because it is likely that both the mechanical and thermal efficiency increase a little with the size of the engine. At half-load the efficiency relatively to the air-engine standard varies between 45·2 and 54·5 per cent., also increasing with the size of the engine. The mean relative efficiency at full load is 56·7 per cent. and at half-load 49·1 per cent., a difference which is also what would be expected.

The air-engine standard adopted by the Committee has been fully explained in the Preliminary Report. It may, however, be useful to point out that the absolute thermal efficiency is the fraction of the total heat of combustion which is converted into work, the remainder being rejected as heat. This efficiency is required in comparing thermal engines of different types. The efficiency relative to the standard air-engine merely shows how far the engine tried approaches the efficiency of an ideal engine working on approximately the same cycle as the engine tried, with the same compression, but without radiation and jacket-losses.

It would be desirable, but for one circumstance, to calculate the relative efficiency only from the indicator horse-power. But it appears that in the case of gas-engines, and especially gas-engines governed by hit-or-miss governors, the indicator-diagrams do not give as accurate results as is generally supposed. The diagrams vary much more than those of a steam-engine with a steady load, and the mean indicator horse-power, from the diagrams taken in a trial, may, it appears, differ a good deal from the real mean power. Hence, as the brake horse-power can be ascertained more exactly than the indicator horse-power, it is practically convenient to calculate the relative efficiency from the brake horse-power. In that case it must be remembered that, while the engine considered has mechanical frictional losses, it is compared with an ideal engine in which there are not only no thermal losses, such as jacket and radiation losses, but also no mechanical inefficiency.

This preliminary summary of the thermal efficiencies leads appropriately to a general description of the engines and apparatus employed in the tests, the tabular presentation of the observations, the method of reducing the data obtained in order to trace the heat-flow through the engines, and finally the presentation of the heat balance-sheets for nine tests.

2.—GENERAL DESCRIPTION OF ENGINES TESTED.¹

The three engines tested were of the standard four-stroke cycle type, built by the National Gas-Engine Co., Ltd., for cylinders of 5·5, 9 and 14 inches diameter respectively. The arrangements of the engines were identical in respect of the proportions of the combustion-space, the mechanism for the admission of the charge, and the exhaust-valves. In each case the charge was admitted by means of an inlet-valve opening into a port at the middle of the end of the combustion-space. Behind the inlet-valve was a gas-valve, and within the inlet-port the electric igniter was arranged, operating on the low-tension principle, with a Simms-Bosch magneto-instrument. The three engines were each provided with two electrical igniting arrangements, the L engine having the second igniter placed within the cylinder in a plug above the exhaust-valve. The R and X type engines had the second igniter placed at the side of the combustion-space, and not at the top. These second igniters were introduced, as it was intended at one time to make experiments with ignition inside the cylinder instead of in the port. These experiments, however, were not made by the Committee. The inlet-valve in each case was operated from the cam on the usual two-to-one shaft, which runs along the side of the cylinder, and the gas-valve was operated by a similar cam placed close to the main supply-valve cam. The engines were governed on the hit-or-miss method by the action of a centrifugal governor on a small block, which engaged with a pecker. The exhaust-valves were placed in the bottom of the combustion-space, and were operated also from the two-to-one shaft by levers. The exhaust-gases, after passing the exhaust-valve, traversed a water-cooled passage, the water-jacket of which was included in the jacket circulation of the engine. This is clearly seen in the section of the exhaust-calorimeter, Figs. 6, Plate 5. It will be noticed that with this arrangement heat is abstracted from the exhaust-gases after opening the exhaust-valve, before these gases arrive at the exhaust-calorimeter.

In the large engine, that is, the X type, the starting was accomplished by means of a hand pressure-pump, which communicated with a plug above the exhaust-valve. This plug contained a valve which allowed the access of mixture to the cylinder from

¹ Sunprints of the original drawings of the engines tested may be seen in the library of the Institution.

the hand-pump. The engine was started by placing the crank conveniently over the centre, pumping in a mixture of gas and air behind the piston, to a pressure slightly above that of the atmosphere, shutting the inlet-valve in the plug, and tripping the magneto-shield of the Simms-Bosch instrument. The electric spark then passed between the separated surfaces within the cylinder, and the engine started.

The Simms-Bosch magneto was of a well-known type, having a fixed armature, fixed permanent magnets, and movable shield. The shield was withdrawn and tripped by the operation of an adjustable pin rotating on a collar on the two-to-one shaft. This pin can be adjusted to vary the time of firing the charge as may be required.

It is needless to describe the operation of the engines. They act on the ordinary four-stroke cycle, and there was nothing in the mechanical arrangements which calls for special description. They are the well-known arrangements of the National Gas-Engine Co., as applied to engines of the three sizes selected by the Committee.

It may be stated, however, that in order to secure results comparable scientifically, the compression-spaces were carefully adjusted to have as nearly as possible similar proportions in the three engines. Care, too, was taken that the internal surface of the combustion-spaces should be smooth and clean, in order that no unknown element should enter into the observations due to irregularities in the castings. In the small engine, care was taken that the working parts were all very free, in order that no undue friction should affect the results.

TABLE II.—DIMENSIONS OF ENGINES.
(Captain Sankey and Professor Dalby.)

Designation of Engine	L	R	X
Clearance volume in cubic centimetres	850	3,920	12,680
Clearance volume in cubic inches (1 cubic inch = 16·387 cubic centimetres)	52	239	774
Diameter of cylinder inches	5·502	9·00	14·008
Stroke "	10·00	17·03	22·00
Area of cylinder square inches	23·78	63·62	154·1
Volume displaced by piston-stroke cubic inches	237·8	1,083	3,390
Total volume of cylinder "	289·8	1,322	4,164
Clearance. Percentage of total volume	17·94	18·08	18·59
Circumference of brake-drum feet	9·19	18·208	19·84
" " rope "	0·01	0·230	0·295
Effective circumference of brake "	9·200	18·438	20·141
Diameter of air-orifice in the measuring trunk, inches	4	8	12

The three engines were adjusted as to gas-supply, so that each engine should run as nearly as possible at its most economical load. With more gas the three engines could each give considerably more power, but such power would partake somewhat of the nature of an overload, and the consumption per brake-HP. would not be quite so low. In all the tests made, the igniter used was that in the admission-port.

3.—GENERAL ARRANGEMENT OF TESTING APPARATUS.

Fig. 2, Plate 5, shows a diagrammatic arrangement of the apparatus used in the testing of each engine. A novel feature in connection with this is the method adopted for measuring the air-supply to the engine by means of an anemometer placed in a box which forms an enlarged continuation of the air-suction pipe. The method of reducing the observations is described particularly in article 5 below. A novel form of exhaust-gas calorimeter, designed and fitted by the Company, was used to extract the heat from the exhaust gases; this was arranged on the principle first used by Prof. Bertram Hopkinson, and is described particularly in article 7 below, where drawings are given of the form used by the Committee, and of a slightly modified form which the Committee recommend for any further experiments. The water-measurements were made in tanks, which were calibrated by the Committee. All the weights, spring-balances, and thermometers used were calibrated. The thermometers are indicated in Fig. 2, Plate 5, by a capital T with subscript figures. All through the subsequent calculations and tables, the same subscript figures are retained for the temperatures registered by the corresponding thermometers. Thus t_3 is a temperature measured by the thermometer T_3 .

Table III. identifies the thermometers used, and Table IV. shows the corrections which had to be made on the observed temperatures. All the temperatures given in the Tables are temperatures which have been corrected by means of this Table from the original observations.

During the trials of the two smaller engines, the Committee did not sufficiently appreciate the effect of the heat brought in and carried away by the moisture in the air on the debit account of the heat balance-sheet. During the trials of the large engine, however, the hygrometric state of the air was found by means of a wet- and dry-bulb thermometer. In reducing the data of the trials on the smaller engines an assumption is made, based upon the measurements

TABLE III.—THERMOMETERS USED.

	Marks or Numbers on the Thermometers.			Range required at Ashton.
	L	R	X	
Exhaust Calorimeter—				
Water inlet. . .	1,167,113	1,167,115	1,167,115	43° to 65°
„ exit. . .	317,574	317,574	317,574	48° „ 95°
Gas exit. . .	1,167,115	J. Casartelli, A	<div style="display: inline-block; vertical-align: middle;"> { No mark Labelled, Correct at 50° </div>	48° „ 95°
Jackets—				
Water inlet. . .	1,167,113	1,167,113	1,167,113	43° „ 65°
Water exit. . .	No number	Marked J. Hicks No number Reads to 220° F.	No number	60° „ 180°
Meter.	Labelled D	Marked J. Casartelli	..	47° „ 54°
Air.	Spiral bulb	J. Hicks		

TABLE IV.—THERMOMETER CORRECTIONS. (Professor Callendar.)

—	32° F.	40° F.	50° F.	60° F.	70° F.	80° F.	90° F.	100° F.
J. Hicks 1,167,113 .	-0.1°	0	0	0	+0.2°	0	+0.3°	+0.3°
„ 1,167,114 .		Broken at Manchester.						
„ 1,167,115 .	-0.1°	-0.1°	0	0	+0.3°	+0.1°	+0.3°	+0.3°
„ 317,574 .	-0.2°	0	0	0	+0.2°	0	+0.1°	+0.1°
“Correct at 50°” .	-0.2°	0	-0.3°	+0.1°	+0.1°	0	-0.7°	-0.9°
J. Casartelli “D” .	-0.2°	0	0	+0.2°	+0.4°	+0.4°	+0.3°	+0.4
J. Casartelli “A” .		Not received from Manchester.						
J. Hicks, spiral bulb	-0.4°	-0.3°
J. Hicks, reading to } 220° }	Broken.	

made during the trials of the large engine, that the difference between the reading of the wet- and dry-bulb thermometers was 3 degrees.

The gas is assumed to be dry at entry. Even if it were saturated the amount of heat concerned would be negligibly small, because the weight of gas entering the cylinder is only about one-twentieth of the weight of air at full load, and at half-load only about one-fortieth.

The exhaust-gases are intimately mixed with the water in the calorimeter and escape from the calorimeter at temperatures varying between 70° F. and 100° F. It is assumed that they leave the

calorimeter completely saturated with moisture at the exhaust-temperature, and it is sufficiently accurate and more convenient if the calculation of the heat carried away is made on the supposition that the exhaust-gas is all air.

All these points are more fully explained in the articles below.

It may be stated at once that all heat quantities supplied to or rejected by the engine are reckoned from 32° F.

4.—HEAT SUPPLIED BY THE GAS. LOWER AND HIGHER CALORIFIC VALUE.

To estimate this the calorific value of the gas must be found, and the volume supplied and the temperature at which it is supplied to the engine must be observed.

The calorific value was determined at frequent intervals during the trials by means of the Junker calorimeter, a description of which and an example of the method of using it are given in Appendix I.

The volume was observed on the gas-meter, Fig. 2, Plate 5, and the temperature on the thermometer T_3 .

Before the heat-supply brought in by the stream of gas at atmospheric temperature in excess of that at 32° F., from which all the calculations are made, can be calculated, the density of the gas must be found, in order to reduce the observed volume to weight.

The density was found by calculation from analyses of the gas.

The analysis of the gas is shown in Table IA, Appendix II., with an example of the method of calculating the density. The density was found to be 0.038 lb. per cubic foot at 32° F. and 14.7 lbs. per square inch, and this is the value which has been used in the reduction of the results.

When the gas is burned, some steam is produced. If the products of combustion are cooled, as in the Junker calorimeter, this steam condenses and its latent heat appears as part of the heat produced by combustion. The heat measured when the moisture in the products of combustion is condensed is termed the higher calorific value. But in the working of gas-engines the products of combustion pass away at a high temperature and the latent heat of the steam does not appear as heat in the process, and is not available for the production of mechanical work. Hence, as stated in the Preliminary Report, the Committee decided to take as the calorific value the heat produced in combustion, exclusive of the latent heat of the steam forming part of the exhaust-gases. This is usually termed the lower calorific value.

In the heat balance-sheets (Appendix III.) the lower calorific value and the heat absorbed in the evaporation of water produced by combustion are given as separate items.

If G cubic feet of gas are supplied to the engine per hour at a temperature t_3 and pressure p_3 the weight w_5 of gas supplied to the engine is

$$w_5 = \frac{G \times 493 \times p_3}{(t_3 + 461) \times 14.7} \times 0.038$$

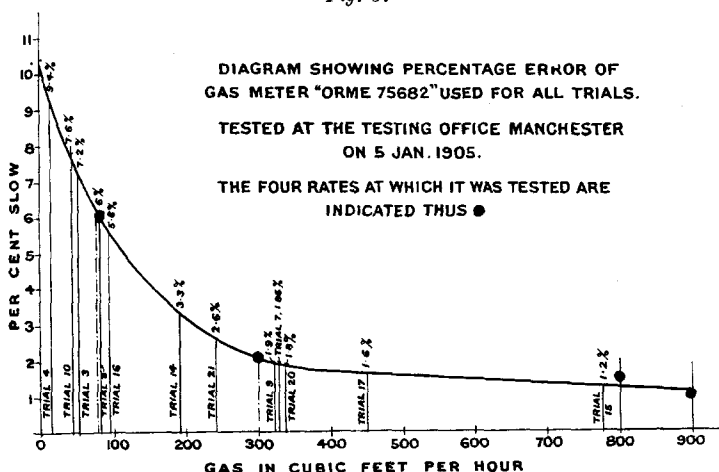
and the heat brought in by the gas-stream before explosion

$$= w_5 \times S (t_3 - 32)$$

S being the specific heat at constant pressure of the gas, taken equal to 0.508.

After the trials, the gas-meter used was taken to the official testing establishment of the Manchester Corporation for calibration. It was tested at four rates, namely 90, 300, 800 and 900 cubic feet per hour, and in each case was found to be slow. The percentage errors

Fig. 3.



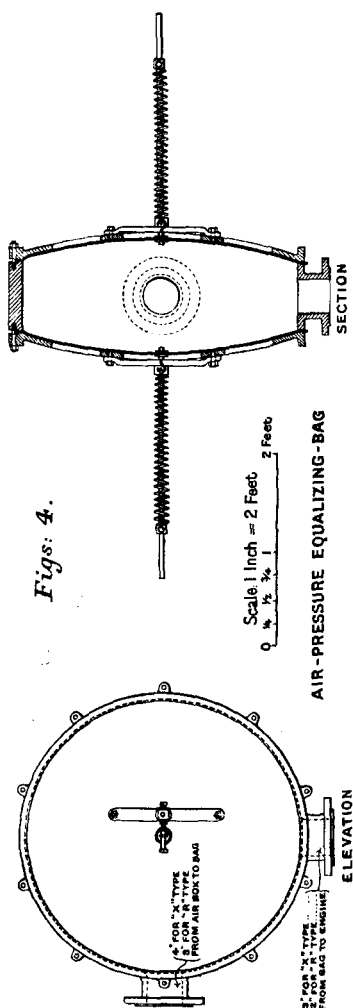
are plotted against the rates in Fig. 3, and the best curve is drawn through them. The observations made during the trials were corrected by this curve. Each trial is in fact plotted on the diagram, and the figure used to correct the results is written against it.

5.—AIR-SUPPLY.

The heat-supply carried into the engine with the air is the excess of the heat in the air above what there would be at 32° F.

In order to estimate the heat received by the engine, and also the heat rejected, it is necessary to observe the quantity of air supplied per hour, the barometric pressure and the temperature at which it is supplied, and also its hygrometric condition.

The measurement of quantity was made by means of a small anemometer, placed inside a wood trunk B, Fig. 2, Plate 5, which was attached to an extension of the air-suction pipe A of the engine. A large gas-bag, the details of which are shown in *Figs. 4*, was placed on the connecting-pipe in order to keep the flow of air at the anemometer as uniform in speed as possible. The fan of the anemometer was placed close to the air-inlet to the box, and its indications were read from without through a glass window let into the top of the box. The air-inlet was circular, and was cut out of a piece of sheet-iron so that the size could be easily adjusted to determine a rate of flow past the anemometer which would give a suitable rate of rotation to the fan. The pressure and temperature of the entering air were measured by



a barometer and a thermometer T_1 , whilst the hygrometric condition was taken by means of a wet- and dry-bulb thermometer.

At the engine trials, readings of the anemometer were observed

which gave the number of lineal feet of air passing the anemometer. These had to be reduced by some means to cubic feet of air. The method devised for doing this was simple and effective. When the engine was quite cold, it was driven by an electric motor through a belt placed on the fly-wheel. The gas-supply was cut off and the engine was allowed to draw in air through the orifice used in the trial, the cylinder in this way serving as a calibrating chamber. In some subsequent tests made by Mr. Dugald Clerk on the engine R, the cams were changed so that the cylinder became a single-acting air-pump without compression, and two of the orifices were separately calibrated on this engine at the mean speed of the actual trial. The dimensions of the box are given in Fig. 2, Plate 5.

In these anemometer calibrating trials the number of revolutions of the engine, the anemometer readings, and the duration of the run were observed. Indicator-diagrams also were taken. It was assumed that the proportion of the cylinder filled with air at atmospheric pressure at each stroke was given by the length on the indicator-diagram between the points at which the pencil-line crossed the atmospheric line. This distance may be termed the effective

TABLE V.—ANEMOMETER CALIBRATING TRIALS.

No. of Test.	Diameter of Orifice.	Anemometer Reading. Velocity of Air.	Revolutions of Engine per Minute.	Volume Displaced by Piston per Stroke.	Ratio of Effective to Total Stroke.	Volume of Air through Orifice per Minute.	Volume of Air corresponding to Anemometer Unit.
5	Inches. 4	Ft. per Min. 293·3	285·0	Cubic Feet. 0·1376	0·886	Cub. Feet. 17·26	Cubic Feet. 0·0588
6	"	268·0	263·0	"	0·920	16·65	0·0621
						Mean	0·0605
11	8	227·3	214·1	0·6273	0·892	59·9	0·263 ²
13	"	118·2	142·5	"	0·756	33·8	0·285
1 ¹	"	185·5	84·8	"	1·000	53·2	0·287
2 ¹	"	268·5	125·5	"	1·000	78·7	0·293
3 ¹	"	357·5	163·0	"	0·977	99·9	0·279
						Mean	0·281
18	12	193·8	142·2	1·962	0·927	129·3	0·667
19	"	214·3	155·6	"	1·000	152·6	0·712
4 ¹	"	138·0	165·0	"	0·988	102·3	0·741
5 ¹	"	166·0	193·0	"	0·977	118·3	0·713
						Mean	0·708

¹ These tests were made with special cams for the valve-gear so that the engine acted as a suction-pump without compression.

² This result is rather anomalous.

stroke. From these results the cubic feet of air per unit reading of the anemometer for each orifice used were obtained.

Although no theory of the rate of flow through the orifice was used in reducing the observations, it is desirable to examine whether a simple expression can be found for the volume of flow in terms of the anemometer reading and the area of the orifice used. The data given in Table V. were obtained in the calibrating trials.

If ω is the area of the orifice in square feet and v the velocity of the air at the anemometer in feet per second, then the volume flowing per second is

$$Q = c\omega v$$

where c is a coefficient of discharge which allows for the contraction at the orifice and the difference if any between the velocity given by the anemometer and the mean velocity in the contracted section of the current through the orifice, and also for any error in the graduation of the thermometer used.

Calculating c from the foregoing data, the following values are obtained:—

		Orifice.		
		4-Inch.	8-Inch.	12-Inch.
Values of c	{	0.676	0.756	0.849
		0.712	0.820	0.908
		..	0.821	0.942
		..	0.842	0.905
		..	0.801	..
Means		0.694	0.808	0.901

It is clear that c varies with the size of the orifice. This may be due partly to the anemometer not being in the plane of the most contracted section of the current. It is possible that in future tests it might be better to place the anemometer-fan at a distance from the orifice equal to half the diameter of the orifice. To a greater extent the variation of c is probably due to alteration of the coefficient of contraction as the ratio of the area of the orifice to that of the air-trunk varies. As the same air-trunk was used for all the orifices, c in these tests must vary with some function of d , the diameter of the orifice. Taking d in feet, it is found that

$$c = 0.903 d^{\frac{1}{2}} \text{ nearly,}$$

and the values of c are then

4-Inch.	8-Inch.	12-Inch.
$c = 0.686$	0.816	0.903

Using these values the discharge can be calculated for each of the calibrating trials, for comparison with the observed discharge. The following Table gives the results:—

COMPARISON OF OBSERVED AND CALCULATED AIR-DISCHARGE.

Test Number.	Diameter of Orifice in Inches.	Area of Orifice. Square Feet. <i>w</i>	Velocity of Air by Anemometer. Feet per Second. <i>v</i>	Observed Discharge of Air. Cubic Feet per Second. <i>Q</i>	Calculated Discharge of Air. Cubic Feet per Second. <i>c w v</i>	Error.
5	4	0·0872	4·888	0·288	0·292	+ 0·005
6	"	"	4·466	0·278	0·267	- 0·011
11	8	0·3485	3·788	0·998	1·076	+ 0·078
13	"	"	1·970	0·563	0·559	- 0·004
1	"	"	3·091	0·887	0·877	- 0·010
2	"	"	4·475	1·312	1·270	- 0·042
3	"	"	5·959	1·665	1·692	+ 0·027
14	12	0·7854	3·230	2·155	2·290	+ 0·135
19	"	"	3·571	2·543	2·531	- 0·012
4	"	"	2·300	1·705	1·631	- 0·074
5	"	"	2·766	1·972	1·963	- 0·009

It will be seen that the differences of the observed and calculated values are small, and sometimes positive and sometimes negative.

The fluid measured by the anemometer is not air. It is a mixture of air and water-vapour. The quantity of dry air in the mixture must be calculated from its observed hygrometric condition; the calculation at the same time gives the quantity of water in the mixture, and, these quantities being known, the heat carried into the engine by the air and by the water mixed with the air can be computed separately.

The method of making this computation is to take from a Table "Glaisher's factor" corresponding to the difference of reading of the wet- and dry-bulb thermometers. The product of this difference and the factor gives the number of degrees which the dew-point is below the observed temperature of the air. Thus in Trial 16 the difference between the readings of the wet- and dry-bulb thermometer was 3° F. Glaisher's factor for this difference is 1·9. The temperature of the air was 55° F. Hence the dew-point was—

$$55 - (3 \times 1·9) = 49·2 \text{ degrees.}$$

From the Steam Tables the pressure of water-vapour at 49·2 degrees is 0·172 lb. per square inch.

The barometer-reading was 30 inches = 14·74 lbs. per square inch,

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Consequently the pressure of dry air in the mixture at the anemometer was $14.74 - 0.172$ lbs. per square inch; that is, 14.55 lbs. per square inch.

Since the pressure at the dew-point is the pressure at the partly saturated temperature observed, namely 55 degrees, or 516° absolute, the volume of 1 pound of dry air at the anemometer may be computed from the usual expression

$$p v = 53.18 T,$$

where T is the absolute temperature, and p is the pressure in lbs. per square foot. Hence

$$\text{Volume of dry air per lb.} = \frac{53.18 \times 516}{14.55 \times 144} = 13.09 \text{ cubic feet.}$$

This 1 pound of dry air is mixed with an equal volume of water-vapour, which, as has been already found, is at a pressure of 0.172 lb. per square inch. From the Steam Tables the volume of 1 pound of steam at this pressure is $1,808$ cubic feet. Hence the weight of 13.09 cubic feet is 0.00722 lb.

Hence every pound of air which flowed past the anemometer into the engine carried, invisibly associated with it, 0.00722 lb. of steam. This weight is the weight of steam for complete saturation of the air at 49.2 degrees.

The heat-supply in the air may now be calculated thus:—

Let V be the number of cubic feet of air supplied per hour as computed from the anemometer readings.

Then

$$w = \frac{V}{13.09} \text{ pounds of dry air pass the anemometer per hour.}$$

$$w_2 = \frac{V}{13.09} \times 0.00722 \text{ pounds of steam per hour are carried by the air in addition.}$$

Taking the specific heat of air at constant pressure as 0.24 , and taking, from the Tables, the total heat H of a pound of steam at 49.2 degrees—

$$\text{Heat supplied by dry air per hour} = 0.24 \times w (t_1 - 32).$$

$$\text{„ „ „ steam in air} = w_2 \times H.$$

The actual figures from the trial in this case were—

$$\text{Heat supplied by dry air per hour} = 3,833 \text{ B.Th.U.}$$

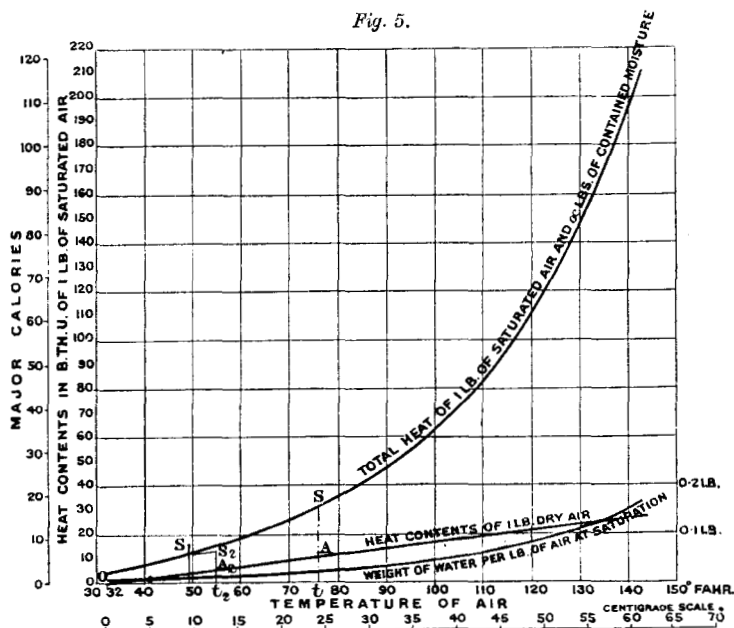
$$\text{„ „ „ steam in air} = 5,510 \text{ „}$$

$$\begin{array}{lcl} \text{Total heat brought into the engine} & & \\ \text{by the mixture} & = 9,343 & \text{„ per hour.} \end{array}$$

In practice it is unnecessary to make this detailed calculation, because Tables are available which give the heat-contents of 1 pound of air, and the heat of the steam associated with it, for various temperatures.

A curve, *Fig. 5*, has been drawn, from which the heat-contents of 1 pound of air and the associated quantity of steam may be read off with sufficient accuracy for all practical purposes of engine-testing. Temperature is set out horizontally. At any temperature t an ordinate is erected and a distance tA is set out to represent the heat-contents of 1 pound of dry air measured from 32° F., assuming the

Fig. 5.



specific heat of air to be 0.24 at constant pressure. Then a distance AS is set up to represent on the same scale the heat-contents of the steam carried by the air when completely saturated at t degrees. When t is varied the locus of A is a straight line passing through 32° , and the locus of S is a curve which does not pass through 32° , and whose shape depends upon the properties of steam as given in the Steam Tables. Points on it therefore have to be calculated for a series of temperatures.

Thus tS represents the heat-contents of 1 pound of dry air and the associated quantity of steam, which occupies the same volume as the 1 pound of air at temperature t degrees.

If in addition to the use of this curve some form of hygrometer is used, such as the Daniell hygrometer, by means of which the dew-point can be observed directly, the computation of the heat supplied to the engine by the stuff passing the anemometer is reduced to a very simple matter. It is interesting to note what a large amount of heat is carried by the invisible moisture in the air at high temperatures. Thus the diagram shows that air saturated at 140° has a total heat of about 195 B.Th.U. per pound.

To investigate the example just given, using the curve, suppose the dew-point by observation to be 49.2 degrees. Find 49.2 on the horizontal axis, *Fig. 5*, and draw a line S_1S_2 parallel to the air-line OA, cutting the ordinate through the observed temperature of the air in S_2 . t_2A_2 is the heat brought in by 1 pound of air; A_2S_2 is the heat brought in by the water mixed with the air.

These quantities read respectively—

Heat per lb. of dry air, at 55 degrees = 5.5 B.Th.U.

Heat brought in by steam associated

with each pound of air = 7 B.Th.U.

The dry air supplied in this test was 694 lbs. per hour.

Hence,

Heat supplied by dry air per hour = $694 \times 5.5 = 3,820$ B.Th.U.

Heat supplied by steam in air

per hour = $694 \times 7.9 = 5,480$ B.Th.U.

6.—HEAT BROUGHT IN AND TAKEN AWAY BY THE JACKET-WATER.

Water was supplied to the jackets from a small tank Q (*Fig. 2*, *Plate 5*), fitted with a ball-cock, and placed about 15 feet above the floor. The temperature of the entering water was taken by the thermometer T_4 inserted in the supply-pipe close to the jacket, and the temperature at exit was taken by the thermometer T_6 inserted in the pipe at the exit from the jackets. The jacket-water was led into a tank fitted with a gauge-glass open at the top, at the back of which a scale graduated in feet and decimals of a foot was placed. The divisions of this scale were calibrated by pouring in weighed quantities of water. It was found that the tank had a uniform capacity of 680.67 pounds of water per foot.

During the preliminary tests the jackets were connected directly to the water-main. The variation of pressure in the main was found to interfere considerably with the flow through the jackets. Hence,

for the subsequent tests the separate tank was installed in order to maintain a constant head on the jackets during any particular experiment.

If w_{10} is the weight of water flowing into the jacket per hour at the observed temperature t_4 the heat supplied to the engine by the incoming jacket-water is

$$w_{10}(t_4 - 32) \text{ B.Th.U.}$$

and if t_6 is the temperature at exit the heat carried away by the jacket-water per hour is

$$w_{10}(t_6 - 32) \text{ B.Th.U.}$$

When working with a hot jacket some of the jacket-water is carried away as vapour. To avoid losing this the pipe carrying the water away should be taken down into the measuring-tank so that the delivery-orifice is below the surface of the water, and a film of oil should be put on the surface of the water. In this way the vapour gets condensed in the tank, and evaporation from the surface is prevented.

7.—HEAT CARRIED AWAY BY THE EXHAUST GASES AND THE HEAT BROUGHT IN AND CARRIED AWAY BY THE CALORIMETER COOLING-WATER.

The hot gases were cooled by passing them through an exhaust-gas calorimeter. The construction of the calorimeters used in the trials is shown by the working drawings, Figs. 9 and 10, Plate 5. It will be observed that the calorimeter is water-jacketed right up to the connection with the engine exhaust-flange. The water is led in at a , and after passing through the jackets is led to the rose, R , Figs. 10, Plate 5, through the small orifices of which it spurts out to meet the stream of exhaust-gas. The water and gas then find their way out, passing various obstructions to ensure mixing and abstraction of heat from the gas until finally the gas, cooled down to about 90 degrees, escapes at G , and the water escapes at e , having about the same temperature as the escaping gas, although sometimes it was higher and sometimes lower. The general arrangement of the calorimeter is shown in Fig. 2, Plate 5, from which it will be seen that the water was brought into the calorimeter directly from the water-main, and was led to measuring-tanks placed outside the testing-room. These tanks were fitted with gauge-glasses and scales graduated in feet and decimals of a foot, and were calibrated

by pouring in weighed quantities of water. The calibrations obtained were:—

	Lbs. of Water per Foot.
No. 1 tank for exhaust calorimeter	680·82
No. 2 „ „ „ „	679·42

In the design of these calorimeters care should be taken that the thermometer measuring the temperature of the escaping cooling-water is placed so that the bulb is completely immersed in the water. As the pipe does not always run full, a pocket should be formed in the pipe to ensure this condition. Care should also be taken that the gas itself does not carry away water mechanically suspended in it. The drawings, Figs. 6, 7 and 8, Plate 5, show a form of calorimeter in which special attention is given to these two points, and which was used in some trials made by Mr. D. Clerk subsequent to those made by the Committee. The results showed that, although the calorimeters used by the Committee were not provided with these safeguards, the quantities concerned were measured without appreciable error.

In the reduction of the observations on the calorimeter, several minor points have to be observed. In the first place the water flowing out of the calorimeter, and which is measured in the measuring-tanks, is not exactly the quantity which enters it. The gases cooled to about 90° F. are at this temperature like a sponge as regards the absorption of water-vapour, and as they have been in intimate contact with the water during the whole time of the passage through the calorimeter it may be assumed that they will pass out completely saturated with moisture at the exhaust temperature. Now the amount of water required to saturate the exhaust-gases may be computed as though the gas was entirely air without introducing serious error, in which case the curve, *Fig. 5*, becomes available to find the amount required. A scale of weight of moisture per pound of dry air is given to the right of the diagram, from which it will be seen that at a temperature of 127 degrees 1 pound of gas completely saturated carries away 0·1 pound of water.

This amount is not however all abstracted from the water entering the calorimeter, for it will be observed that the air entering the engine cylinder brings in with it a definite weight of moisture, and the combustion of the gas produces a definite weight also. Hence the quantity actually abstracted from the cooling-water is the difference between the quantity required for the saturation of the exhaust-gases and the sum of the water produced by

combustion and that brought in by the air. To make the matter clear, consider the data of trial 16 (Table VII., Appendix III.).

The total weight, w_2 , of water brought in per hour by the air is	Lbs. 5.01
The total weight of water produced by combustion of the gases	} 5.73
per hour, w_4 , is	
Total	<hr/> 10.74 <hr/>

The total weight of water, w_7 , required for the complete saturation of exhaust-gas, at a temperature of 83 degrees, is, from the curve, *Fig. 5*, 0.025 lb. per pound of dry exhaust-gas, and the weight of dry exhaust-gas passing from the engine per hour is 692.5 lbs.; hence the total weight of water required per hour to saturate the exhaust-gases is $692.5 \times 0.025 = 17.31$ lbs. Hence the water abstracted from the cooling-water during its passage through the calorimeter is

$$17.31 - 10.74 = 6.57 \text{ lbs.}$$

The water entering the calorimeter is therefore the weight observed in the measuring-tanks; that is, 164.7 lbs. per hour plus this amount. The water entering the calorimeter is therefore 171.27 lbs. per hour.

Let w_8 be the weight of water flowing from the calorimeter and measured in the tanks.

Let m be the weight of water abstracted from the cooling-water during its passage through the calorimeter to assist in the complete saturation of the exhaust-gas $= w_7 - w_2 - w_4$. Let t_2 be the temperature of the water entering the calorimeter, and t_5 the temperature of the water leaving the calorimeter; then

$$\text{Heat brought into engine by cooling-water} = (w_8 + m) (t_2 - 32).$$

$$\text{Heat carried away from engine by cooling-water} = w_8 (t_5 - 32).$$

The heat carried away by each pound of the exhaust-gases at temperature t_0 may be read off the curve, *Fig. 5*, the assumption being that the gas is completely saturated at the exhaust temperature. This of course includes the heat, if any, abstracted in evaporating the water taken up by the exhaust-gases.

8.—HEAT CONVERTED INTO WORK.

Indicator-diagrams were taken at intervals during the trials, more with the object of studying the action of the engine than of computing the indicator horse-power. In the construction of the heat balance-sheets the basis of brake horse-power was taken. In each case the measurement of brake horse-power was made with a rope-brake of the usual type, the heavy weight hanging free and the other end of the rope being held by a spring-balance. In the case of the smallest engine a rope was too stiff to obtain good running, especially at low powers, and a flat band of webbing was used instead. The brake-ropes were placed on specially turned drums bolted to the flywheels in the case of the two larger engines, in order to be able to use water-cooling.

In the smallest engine the band was placed directly on the fly-wheel. All the weights and spring-balances used in the tests were calibrated.

The brake horse-power, reduced to British Thermal Units, is the quantity credited to the engine as work done.

9.—RADIATION.

Included under this heading is the radiation from the hot surfaces of the cylinder, trunk piston, etc., and the radiation from the bearings of the engine due to friction. In other words, the term includes the heat lost by direct radiation from the hot surfaces, together with the heat corresponding to the difference between the indicator horse-power and the brake horse-power. The difficulty of finding the indicator horse-power exactly led the Committee to define radiation in this way.

The method of measuring this quantity was to run the engine light and then to adjust the number of explosions per minute so that the temperatures maintained on the thermometers measuring the temperatures of the water to and from the jackets were respectively equal to those at a full-load trial. In this way the surface temperatures, allowing a very small quantity of water to flow through the jackets in order to obtain similar conditions to those during the normal working of the engine, were kept approximately the same in the two trials, although the inner temperatures would be lower because of the fewer explosions.

The method is of course only an approximate one, but it gives

some idea of the losses from this cause. It is to be understood that a radiation or no-load trial is worked out exactly in the same manner as a full-load trial, a balance-sheet being constructed. The heat unaccounted for in this balance-sheet is held to be loss due to radiation, and the figure so obtained is used in the full-load and half-load balance-sheets as the radiation loss.¹

10.—THE OBSERVATIONS.

A complete record of all the observations taken during the trials is given in Table VI., Appendix III., with the exception of the water-measurements, which are shown graphically in Figs. 11 and 12, Plate 5.

Figs. 12, Plate 5, show the observations on the jacket-water temperatures, and in each case the weight of water running through the jacket per hour is stated on the diagram.

Figs. 11, Plate 5, show the temperatures of the water entering and leaving the calorimeter, the temperature of the gas leaving the calorimeter and the weight of water leaving the calorimeter per hour.

Figs. 13, Plate 5, are records of temperatures taken by means of Professor Callendar's recording platinum thermometer. The variation of the temperature with the load and the variation with different mixtures of gas are shown instantly by the Callendar thermometer. The movement of the pencil is most fascinating. It is as though the recording thermometer were the pulse of the gas-engine. Every variation of working is detected and recorded. Unfortunately, owing to the absence of Professor Callendar, the apparatus was not used all through the trials. Professor Callendar came to Manchester as soon as he was able and fixed up the instrument, with the interesting results shown in the diagrams. It will be noticed that the lower curve shows the cooling of engine X during the night and gives some idea of the rate at which radiation takes place.

¹ To test the variation of the surface temperatures of the cylinder with the rate of flow through the jacket, Mr. Hayward made a subsequent series of experiments on a 10-B.H.P. engine at the Central Technical College. He found that, if the temperature of the upper outside surface of the cylinder was kept constant and the power of the engine varied from nothing to full load, the flow through the jacket being varied to secure this result, the mean temperature of the whole surface was lowered as the load increased, so that probably the radiation-losses were reduced. If this result is applicable to the Ashton trials, the radiation-losses at full load are probably a little over-estimated.

11.—SUMMARY OF OBSERVATIONS.

A complete summary of all the trials is given in Plate 6.

The Table is divided into seven vertical columns giving the summaries of the observations on :—

The indicator horse-power.

The brake horse-power.

The gas-supply.

The air-supply.

The jacket-water.

The exhaust calorimeter.

The composition of the exhaust-gases.

The I.H.P. is calculated in the usual way from the data given in the I.H.P. compartment, and a column is added to the right giving the gas per I.H.P.-hour.

The B.H.P. is calculated from the data in the B.H.P. compartment, and a column is added to the right giving the gas per B.H.P.-hour.

12.—MECHANICAL EFFICIENCY.

The mechanical efficiency values obtained from comparison of I.H.P. and B.H.P. of three Ashton engines are obviously incorrect; the three full-load tests show :

Engine	L	R	X
Mechanical efficiency . . .	0.90	0.80	0.94

There appears to be no reason why the R engine should show so low an efficiency as 0.80, and none why L and X should be so high as 0.90 and 0.94. Fortunately the observations made supply means of calculating these efficiencies by two other independent methods :

(1) By adding I.H.P. without load to B.H.P. at full load, assuming the friction as determined by the indicator to be the same when the engine is running without load as it is when the engine is fully loaded.

With engine L, Test 4, *No Load* shows that 0.96 I.H.P. maintains the speed of the engine at 291 revolutions per minute; reducing this to 258.9 revolutions per minute, the full-load speed, the I.H.P. necessary to drive the engine without load at 258.9

revolutions is found to be $\frac{258.9 \times 0.96}{291} = 0.85$ I.H.P.

The full load at 258·9 revolutions per minute is 5·2 B.H.P.

The I.H.P. is $5·2 + 0·85 = 6·05$

and the mechanical efficiency is $\frac{5·2}{6·05} = 0·86$

Calculated in this way the respective values of the mechanical efficiency are :—

Engine	L	R	X
Mechanical efficiency . . .	0·86	0·866	0·888

(2) By calculating from the full-load and half-load values of the B.H.P. and the gas-consumptions, assuming friction to be constant from half to full load.

The values required are :—

B.H.P. at full load.
 B.H.P. at half-load.
 Gas per hour at full load.
 Gas per hour at half-load.

The B.H.P. and the gas at half-load must be taken at the same number of revolutions per minute as in the full-load trial.

$$\frac{\text{Gas per hour at full load} - \text{Gas per hour at half-load}}{\text{B.H.P. at full load} - \text{B.H.P. at half-load}} = \text{Gas per I.H.P.-hour}$$

$$\text{Mechanical efficiency} = \frac{\text{Gas per I.H.P.-hour.}}{\text{Gas per B.H.P.-hour.}}$$

Calculated in this way the mechanical efficiencies are :—

Engine	L	R	X
Mechanical efficiency . . .	0·81	0·83	0·84

The mean values by (1) or (2) are :—

Engine	L	R	X
Mechanical efficiency . . .	0·835	0·848	0·864

Method (1) depends on the accuracy of the indicator ; but an error of, say, 5 per cent. only introduces an error of that amount in the friction value itself. In calculating mechanical efficiency from the total indicated power an error of 5 per cent. on the total may readily amount to 20 per cent. on the friction ; while by method (1) it is limited to the 5 per cent. on the friction value calculated. Method (2) gives the mechanical efficiency without reference to the

indicator, and it only assumes that the diagrams remain constant at the lighter load and that friction is constant between full and half-loads.

It seems clear, as has been already stated, that however carefully indicator-diagrams are taken in gas-engine trials, they do not furnish as accurate a value of the mean indicator horse-power as has been generally supposed.

13.—REDUCTION OF DATA TO FORM HEAT BALANCE-SHEETS.

The reduction of the data of nine experiments is exhibited in detail in Table VII., Appendix III. The order in which the successive thirty-three calculations are most conveniently made is exhibited in the Table. The Table is self-explanatory, but it may possibly be useful to go through the reduction of one set of observations in detail. For this purpose test 15 is selected.

The object of the first fifteen lines is to find the heat brought in by the air, from observation of the difference between the readings of the wet- and dry-bulb thermometers. In test 15, if the air had been dry at entry, the calculation of the volume per pound, line 8, would have been made with the pressure corresponding to the observed height of the barometer; that is, 14.74 lbs. per square inch. The corresponding volume would have been 12.26 cubic feet per lb. and the weight of air entering per hour would have been 591 lbs. instead of 583 lbs., corresponding to 3,520 B.Th.U. In the actual case, the air brought in 3,474 B.Th.U., and the water associated with it 4,970 B.Th.U. As already stated in article 5, these calculations can be materially reduced by using an instrument to observe the dew-point directly, and then using a curve like that shown in *Fig. 5*, but drawn to a larger scale.

The next eleven lines, namely lines 16 to 26, are arranged to lead up to the calculation of the weight of water entering the calorimeter, in order to find the heat brought to the engine by the entering calorimeter water. The question is complicated by the fact that the exhaust-gases mix with the cooling-water, and therefore abstract a small quantity of water to help to make up the water of saturation in the way already explained and illustrated in article 7. The weight of moisture in the air and the water produced by combustion also contribute towards the water of saturation, and these quantities are found in lines 12 and 19 respectively. The remaining calculations are simple and straightforward and call for no comment.

14.—HEAT ACCOUNTS.

The most important items for the nine experiments in Table VII. are brought together in a heat account in Table VIII. (Appendix III.). All but three items in this account are obtained from Table VII. These are items 1, 12 and 13.

Item 1, the heat due to the combustion of the gases, is derived from the experiments with the Junker calorimeter, and the observed quantity of the gas-supply per hour.

Both the quantities and the heat produced by combustion per hour are given in the "gas-supply" column, Plate 6.

Item 12, the heat-equivalent of the B.H.P., is found by multiplying the B.H.P. by 2,545.

Item 13, the radiation-loss, is the most open to question of any in the Table. It will be observed that the engine is run with no load, and, as already explained in article 9, the explosions and the water running through the jackets are severally adjusted until the same difference of temperature is obtained between the temperatures of the entering and leaving jacket-water.

The whole of the heat unaccounted for in the balance-sheet is then assumed to be lost in radiation, and this quantity is found by taking the difference between the heat supplied and the heat accounted for. Thus trial 16 requires 32,907 heat-units to be added to the credit side of the account to balance the account. This amount is then transferred to the accounts of trials 17 and 15, and is called the loss due to radiation.

Line 14 shows the amounts which are required to balance accounts when the several engines are running under load. The full-load trial of the large engine was only 6,787 units out of balance. These, it will be observed, had to be subtracted from the credit side to balance the account. It will be noticed that in the case of the small engine the account required the addition of heat to balance it, whilst in the case of the two larger engines the account overbalanced by the amounts shown, and indicated by having the negative sign prefixed to them.

15.—HEAT BALANCE-SHEETS.

In Table VIII., Appendix III., the heat quantities are reckoned as if all the fluids entered the engine at 32° F. Hence there are heat quantities, such as the respective differences in the total heat of the jacket and calorimeter cooling-water at 32° and at

the temperature at which it was actually used, which play no part in the working of the engine. The real waste due to the jacket is the quantity (11) in Table VIII. less the quantity (7). Similarly the real exhaust waste is the sum of the quantities (8) (9) and (10) less the quantities (2) (3) (4) (5) and (6). Table IX., Appendix III., calculated in this way gives the heat quantities concerned in the working of the engine, drawn up as a balance-sheet. It will be seen that the quantities unaccounted for are not large, and it is possible that if the very uncertain loss due to radiation, which is a comparatively large quantity, could have been more exactly determined, the quantities unaccounted for would have been smaller still.

In Table X., Appendix III., the heat quantities have been reduced to percentages of the heat supplied to the engine by the combustion of the gas. The no-load trials were made specially to determine by difference the radiation-loss, and in other respects are not of particular interest. The following Table presents a summary of the results of the other trials:—

	Percentages of Heat of Combustion.				
	Exhaust Waste.	Jacket Waste.	Radiation.	Total Waste.	Heat converted into Work.
<i>Full-Load Trials—</i>					
Engine L	35·3	23·5	7·6	66·5	26·7
„ R	40·0	29·3	10·0	79·4	28·3
„ X	39·5	25·0	7·3	71·7	29·8
Means	38·3	25·9	8·3	72·5	28·3
<i>Half-Load Trials—</i>					
Engine L	35·7	27·1	11·8	74·5	22·6
„ R	35·6	27·4	16·0	78·9	23·5
„ X	43·1	24·2	12·3	79·6	26·6
Means	38·1	26·2	13·3	77·7	24·2

The error of the tests, or heat unaccounted for, is not shown in this Table.

APPENDIXES.

APPENDIX I.

DETERMINATION OF THE CALORIFIC VALUE OF GAS WITH THE "JUNKER" CALORIMETER.

Description of Instrument.—In the "Junker" form of gas-calorimeter the heat generated by a Bunsen flame, in which the gas to be tested is consumed, is transferred to a stream of water passing through the instrument at a constant rate. The products of combustion are cooled by the water to the temperature of the surrounding air and radiation is reduced to a minimum by enclosing the apparatus in a plated and polished casing.

To determine the calorific value of a supply of gas, the following sets of observations are taken when the temperatures at all points in the instrument have become steady.

(1) The volume of gas (G) burnt in any interval and the pressure and temperature of the gas.

(2) The weight of water (W) passing through the calorimeter in the same interval.

(3) The inlet and outlet temperatures of the water, t_i and t_o .

(4) The weight of water formed by 1 cubic foot of gas.

(1) In the experiments made by the Committee the volume of the gas was measured with a small "wet" meter reading to $\frac{1}{100}$ cubic foot. The temperature was taken with a thermometer placed in the gas-space of the meter and the pressure above atmosphere with a manometer placed on the supply-side of the meter. The pressure and temperature of the gas were in all cases found to be nearly identical with those at the meter supplying the engines, and therefore no corrections on this score were necessary in calculating the heat supplied to the engine per cubic foot of gas.

(2) The weight of water (W) was measured by collecting the outflow from the calorimeter, while a convenient volume of gas (G) passed the meter.

(3) The inlet-temperature of the water, t_i , and the outlet temperature, t_o , were read on suitably-placed thermometers immediately before, during, and immediately after the collection of the water. If any large variation appeared in these the test was repeated.

(4) With the exception of the negligible quantity required for the complete saturation of the air passing the burner, all the water formed by combustion is condensed in the calorimeter and may be collected as it escapes from a small drain-pipe. Each gramme of this water gives up 0.536 major calorie of latent heat on condensation; and allowing for the cooling of the water from 100° C. to the temperature of the atmosphere this quantity becomes practically 0.6 major calorie.

All measurements were made in grammes, degrees centigrade, and cubic feet; hence for convenience the calorific value of the gas was first calculated in major calories per cubic foot and afterwards reduced to thermal units per cubic foot by multiplying by the constant 3.968. The calorific value of the gas per cubic foot reduced to 32° F. and 14.7 lbs. per square inch remained nearly constant during the trials, but the calorific value per cubic foot, as measured, varied considerably, owing to changes in the atmospheric temperature and pressure.

Example of a Test made by the Committee.

Date, 31 December, 1904.

Time, 10.45 A.M.

Temperature of gas, 44° F.

Manometer reading, 1.4 inch of water.

Barometer reading, 29.97 inches of mercury.

Gas burnt. Cubic Feet=G.	Water Collected Kilograms = W.	Inlet Temperature. C° = t_i	Outlet Temperature. C° = t_o
0.16	1.805	9.40	23.30
		9.40	23.32
		9.45	23.33
		9.50	23.31
	Mean	9.438	23.315

Higher calorific value of gas per cubic foot.

$$\begin{aligned}
 &= \frac{W(t_o - t_i)}{G} \\
 &= \frac{1.805 \times 13.877}{0.16} \\
 &= 156.5 \text{ major calories.} \\
 &= 622 \text{ B.Th.U.}
 \end{aligned}$$

25.3 grammes of water were formed from each cubic foot of gas burnt

∴ Lower calorific value of gas per cubic foot

$$\begin{aligned}
 &= 156.5 - 0.6 \times 25.3 \\
 &= 141.3 \text{ major calories} \\
 &= 561 \text{ B.Th.U.}
 \end{aligned}$$

Test of the Calorimeter.—After the trials the accuracy of the meter and calorimeter was tested at the Central Technical College by burning a stream of hydrogen produced directly from zinc and sulphuric acid.

The following results were obtained :—

Temperature of hydrogen gas, 54° F.

Manometer reading, 0.1 inch of water.

Barometer reading, 29.99 inches of mercury.

(1) 14 grammes of water were condensed while burning 0.65 cubic foot of hydrogen. This is equivalent to a condensation of 22.5 grammes of water per cubic foot at standard temperature and pressure. Pure hydrogen yields 22.7 grammes of water per cubic foot.

(2) Cubic feet of hydrogen burnt	0.120
Ditto reduced to 32° F. and 14.7 lbs. per sq. in.	0.115
Water collected in kilograms	0.786
Mean inlet temperature °C.	10.104
„ outlet „ °C.	22.690
Rise in temperature of water °C.	12.586

Hence higher calorific value of hydrogen as indicated by the calorimeter = 341 B.Th.U. per cubic foot.

Correct calorific value of hydrogen = 343 calories per cubic foot.

Error of calorimeter = 0.6 per cent.

This small error has been allowed for in calculating the heat quantities in line 1, Table VII. Appendix III.

APPENDIX II.

CALCULATION OF AIR-SUPPLY FROM ANALYSIS OF INCOMING AND EXHAUST GASES.

Samples of the incoming gas were taken on the 3rd and 5th of January. As the trials commenced on 28th December, 1904, and those for which the gas analysis was required were completed on 3rd January, 1905, the sample taken on the latter date is the only one used in these calculations. Table IA. gives the result of the analysis which was carried out by Mr. John L. Garle, of West-

TABLE IA.

Constituents.	Per Cent. by Volume.
Marsh-Gas (CH_4)	33.73
Unsaturated hydrocarbons	4.74
Hydrogen (H_2)	41.29
Carbon monoxide (CO)	7.13
Nitrogen (N_2)	10.22
Carbon dioxide (CO_2)	2.62
Oxygen (O_2)	0.27

minster. From this analysis Tables IIA., IIIA., and IVA. are calculated. Table IIA. gives the volumes of steam and carbonic acid produced by the complete combustion of 100 cubic feet of the gas, with just the required amount of oxygen. Table IIIA. gives the figures from which the specific heat of the gas is calculated. Table IVA. gives the percentage volumes of the dry products resulting from the combustion of 100 cubic feet of the gas with varying quantities of air. The results in this Table are plotted in *Fig. 14*, the abscissae being volumes of air per cubic foot of gas, and the ordinates being the percentage volumes of carbonic acid, oxygen and nitrogen formed by combustion. It will be seen from the diagram that the ordinate from a point on the base representing any proportion of air and gas in the mixture before combustion, cuts the curves in points which severally define the percentage of CO_2 , O_2 and N_2 ; therefore from an analysis of the exhaust-gases which gives the percentage volume of CO_2 , O_2 and N_2 , three values of the percentage volume of the air present at combustion can be inferred.

In the Ashton trials all three of these volumes were determined, the N_2 , however, only by difference. The air present at combustion has therefore been calculated only from the percentage volume of CO_2 and O_2 . The last four columns on Table VII. give these results. They cannot be expected to agree very closely with each other or with the independent measurement made by the anemometer, for, as has been already stated, only one sample of the incoming gases was available for analysis, and that was 7 days after the first trial. It is also questionable whether a fair sample of the exhaust gases was always obtained. It may have happened that a sample was taken just before or just after a miss had occurred; in the former case the proportion of air would be too high and in the latter case too low. Errors due to this cause would have much more effect on a half-power trial than on a full-power one, as there are so many more misses in the former case.

Fig. 14.

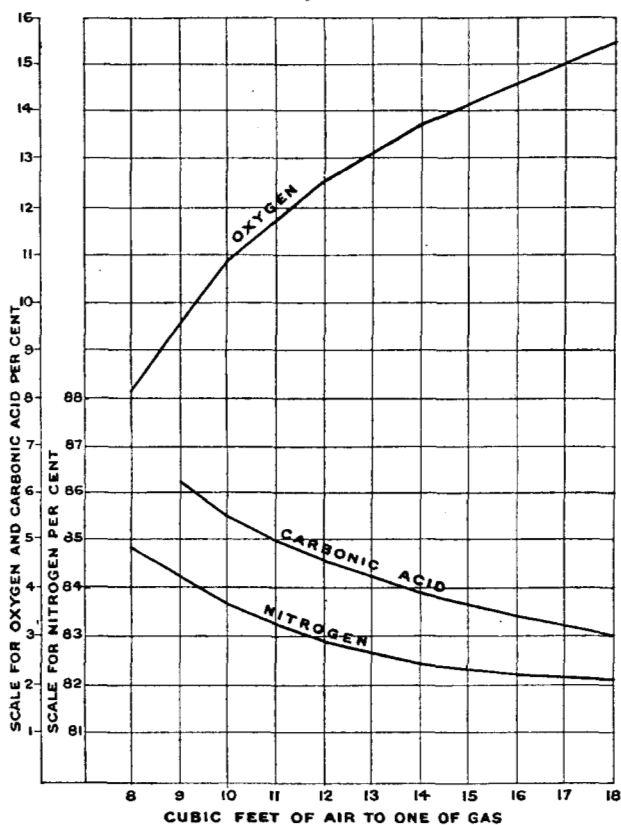


TABLE IIA.

Constituents of Gas.	Volume per Cent.	Oxygen for Complete Combustion.	Volume of Steam produced.	Volume of CO ₂ produced.
CH ₄	33.73	67.46	67.46	33.73
C ₂ H ₄	4.74	14.22	9.48	9.48
H ₂	41.29	20.64	41.29	..
CO	7.13	3.57	..	7.13
N ₂	10.22
CO ₂	2.62	2.62
O ₂	0.27
Totals . .	100.00	105.69	118.23	52.96

Therefore the volume of oxygen required for the complete combustion of 100 cubic feet of the gas is 105.69, less 0.27 free oxygen in the gas, which is equal to 105.42 cubic feet.

TABLE IIIA.

Constituents.	Volume per Cent.	Weight per Cubic Foot at 32° F. and 14·7 Lbs. pressure per Sq. In.	Weight per 100 Cubic Feet of Gas.	Specific Heat at Constant Pressure.	Heat for a Rise of 1 Degree per 100 Cubic Feet of Gas.
CH ₄	33·73	0·0455	1·535	0·593	0·910
C ₂ H ₄ *	4·74	0·0795	0·377	0·404	0·152
H ₂	41·29	0·00559	0·231	3·409	0·787
CO	7·13	0·0773	0·551	0·245	0·135
N ₂	10·22	0·0788	0·803	0·2438	0·195
CO ₂	2·62	0·1238	0·324	0·217	0·070
O ₂	0·27	0·0895	0·024	0·2175	0·052
		Totals	3·845	..	2·301

$$\text{Specific heat per pound } \frac{2\cdot301}{3\cdot845} = 0\cdot508$$

TABLE IV A.—SHOWING PERCENTAGE COMPOSITION OF EXHAUST-GASES DUE TO THE COMBUSTION OF 100 CUBIC FEET OF GAS WITH QUANTITIES OF AIR VARYING FROM 505 TO 1,800 CUBIC FEET.

Composition of Dry Gas before Combustion.			Dry Products of Combustion.					
Gas.	Air.		CO ₂		N ₂		O ₂	
	Oxygen.	Nitrogen.	Cubic Feet.	Per Cent.	Cubic Feet.	Per Cent.	Cubic Feet.	Per Cent.
	505·92							
100	105·42	400·5	52·96	11·4	410·72	88·60	0	0
	800							
„	166·7	633·3	„	7·0	643·52	84·9	61·28	8·1
	900							
„	187·4	712·6	„	6·2	722·8	84·3	82·0	9·5
	1,000							
„	208·3	791·7	„	5·5	801·9	83·7	102·9	10·8
	1,100							
„	229·1	870·9	„	5·0	881·1	83·3	123·8	11·7
	1,200							
„	250·0	950·0	„	4·6	960·2	82·9	144·6	12·5
	1,400							
„	291·6	1,108·4	„	3·9	1,118·6	82·4	186·2	13·7
	1,600							
„	333·3	1,266·7	„	3·4	1,276·9	82·0	227·4	14·6
	1,800							
„	375·0	1,425·0	„	3·0	1,435·2	81·6	269·6	15·4

* The unsaturated hydrocarbons are taken in the Table as C₂H₄. They should, from the analysis, have been taken as C_{3.38}H_{6.72}. The correction for this, however, is small, and the weight 3·845 in the Table has been used only in calculating one small item in the balance-sheets.

APPEN

TABLES VI.—TRIALS OF

Observer, Professor DALBY.								
Time.	Gas Meter.	Temperature.	Pressure.	Counter.	Right.		Left.	
					W.	w.	W.	w.
4.55		°F.	Inches of Water.		TRIAL 1.—			
5.5					42	10	42	8½
5.15					"	10½	"	9
5.25					"	"	"	9
5.30					"	"	"	9
					"	"	"	9
					TRIAL 2.—			
11.30				0·0749262	46	11½	49	11½
11.32								
11.36	000861·5	54	1·6		46	12	49	12
11.45	000874·2	"	"					
11.50				0·0751840				
11.55	000887·6	"	"		46	11½	49	12
12.5	000901·8	"	"					
12.10				0·0754457	46	10½	49	12
12.15	000915·7	"	"					
12.25	000929·5	"	"					
12.30				0·0757024	{ 46	9½	49	11
12.36	000945·0	"	1·75		44			
12.45	000957·6	"	"	0·0758965	44	8½	49	10½
					TRIAL 3.—			
2.30				0·0765445	21	3	21	1½
2.35	001062·6	54	1·5					
2.45	001071·6	"	"					
2.50				0·0768195	21	3	21	1½
2.55	001080·3	"	1·65					
3.5				0·0770253	21	3	21	1½
3.10	001093·3	"	"					
3.20	001102·3	"	"					
3.25				0·0773008	21	3	21	1½
3.30	001111·2	"	"	0·0773702	21	3	21	1½
					TRIAL 4.—			
3.50	001121·36	53·8	2·4	0·0774698		No	load	
4.0	001123·40	"	"					
4.10	001126·30	"	"					
4.20	001128·20	"	"	0·0778068				
					TRIAL 5.—			
11.15				0·0795293				
11.20	Motor	driven		0·0796003	No	load		
11.25				0·0796720				
					TRIAL 6.—			
12.0				0·0796720				
12.5	Motor	driven		0·0797401	No	load		
12.10				0·0798047				

DIX III.

SMALL ENGINE "L."

Observer, Mr. CLERK.				Mr. HAYWARD.	Professor ASHCROFT.	Captain SANKEY.	Mr. HAYWARD.
Time.	Anemo-meter.	Baro-meter.	Air Temperature.	Indicator-diagrams taken at	Exhaust Gases Samples taken at		Junker Calorimeter Gas tested at
28 DEC., 1904.							
4.55	0.0223050	Ins. of Mercury. 30.06	° F. 55.7	Time. 4.45	Time. 4.43		Time.
5.05	0.0225410	"	56.9	4.53	5.10		
5.15	0.0227790	"	"	5.2	5.14		
5.25	0.0220165	"	"	5.12			
5.30	0.0231295	"	56.4	5.21			
				5.26			
29 DEC., 1904.							
11.32							
11.34		30.00	55.2				
11.40	0.0282040						
11.50	0.0284385						
12.0	0.0286710	30.00	56.0				
12.10	0.0289080	"	56.0		11.37		10.30
12.20	0.0291420			11.40	11.42		11.30
12.30	0.0293720	29.99	56.2	11.52	12.20		
12.40	0.0296040	"	"	12.5			
12.50	0.0298320	29.99	55.8	12.24			
				12.36			
29 DEC., 1904.							
2.30	0.0324080	29.96	56.5				
2.40	0.0326730						
2.50	0.0329370						
3.0	0.0331920	29.96	56.4	2.37	2.27		
3.10	0.0334580			2.53	2.33		
3.20	0.0337180	29.95	56.2	3.7	2.54		
3.30	0.0339860	29.95	56.8	3.22	3.33		
29 DEC., 1904.							
3.50	0.0345350	29.95	57.0	4.25			
4.0	0.0348210						
4.10	0.0351140						
4.20	0.0354110	29.95	56.8				
30 DEC., 1904.							
11.15	0.0360180						
11.20	0.0361640	29.47	51.6				
11.25	0.0363080						
11.30	0.0364580						
30 DEC., 1904.							
12.0	0.0370670	"	50.4				
12.50	0.0372070						
12.10	0.0373350						

See Figs. 11 and 12, Plate 5.
Jacket-water, weight and temperature, exhaust calorimeter water, weight and temperature, temperature of exhaust gases . . .

11.0
11.15
11.30

TABLES VI.—TRIALS OF

Observer, Professor DALBY.						
Time.	Gas Meter.	Temperature.	Pressure.	Counter.	W.	w.
		° F.	Inches of Water.		TRIAL 7.—	
12.30	0·0798047	197	6
12.35	002261·9	48·0	1·6			
12.45	002317·4					
12.50	..	48·5	1·8	0·0802115	197	6
12.57	002382·8					
TRIAL 8, ABANDONED						
					TRIAL 9.—	
3.0	0·0808307	190	6
3.5	002718·2	48·0	1·4			
3.15	002772·6					
3.20	0·0812360	190	6
3.25	002826·8	48·0	1·4			
3.35	002881·2					
3.40	0·0816441	190	6
3.45	002935·2	48·0	1·4			
3.55	002989·7	48·0	1·4			
4.0	0·0820524	190	6
					TRIAL 10.—	
4.15	0·0820524	No	load
4.20	003047·55	48·0	2·2			
4.35	0·0824785		
4.40	003063·24					
4.55	0·0829048		
5.0	003079·10					
5.15	0·0833307		
5.20	003094·80					
					TRIAL 11.—	
10.50				0·0833307		
11.00	Motor driven			0·0835253	No	load
11.10				0·0837338		
11.20						
11.30				0·0841546		
11.40				0·0843793		
11.50						
12.00				0·0848243		
12.10				0·0850543		
12.20				0·0852630		
12.30				0·0854755		
12.40				0·0856860		
12.50				0·0858061		

MIDDLE ENGINE "R."

Observer, Mr. CLERK.				Mr. HAYWARD.	Professor ASHCROFT.	Captain SANKEY.	Mr. HAYWARD.
Time.	Anemometer	Barometer.	Air Temperature.	Indicator Diagrams taken at	Exhaust Gases Samples taken at		Junker Calorimeter. Gas tested at
30 DEC., 1904.				Time.	Time.		Time.
12.30	0·0375900	29·52	48·7				
12.40	0·0377740	...	48·5	12.38	12.15		
12.50	0·0379540	29·50	48·6		12.20		11.0
1.10	0·0383150			12.49			
OWING TO PRE-IGNITION OCCURRING.							
30 DEC., 1904.							
3.0	0·0392550	29·63	48·4				
3.10	0·0394440			3.05	2.17		
3.15	0·0395350			3.15	2.55		
3.20	0·0396300				3·00		
3.30	0·0398200	..	48·3	3.25			
3.40	0·0400080			3.35	3.30		
3.50	0·0401920						
4.0	0·0403770	29·69	48·4	3.46	3.50		
				3.54			
30 DEC., 1904.							
4.15	0·0407000	29·69	49·0				
4.25	0·0409280			4.20			
4.35	0·0411600			4.31			
4.45	0·0413910	..	48·8				
4.55	0·0416170			4.42			
5.5	0·0418500		48·8	4.50			
5.15	0·0420760	29·69		5.0			
				5.16			
31 DEC., 1904.							
10.5	0·0425620	29·99	44·4				
11.0	0·0427760						
11.10	0·0429970	..	44·4				
11.20	0·0432220						
11.30	0·0434460						
11.40	0·0436760	..	44·7				
11.50	0·0439070						
12.0	0·0441460	29·99	44·8				
12.20							
12.45	0·0451760	..	45·2				
						Jacket-water, weight and temperatures, exhaust calorimeter water, weight and temperature, temperature of exhaust gases	10.45

TABLES VI.—TRIALS OF

Observer, Professor DALBY.						
Time.	Gas Meter.	Temperature	Pressure.	Counter.	W.	w.
		° F.	Inches of Water.		Lbs.	Lbs.
TRIAL 12.—						
1.0	Motor	driven		0·0859419	No	load
1.10				0·0861181		
1.20				0·0862944		
TRIAL 13.—						
1.25	Motor	driven		0·0862944	No	load
1.30				0·0863690		
1.35				0·0864369		
TRIAL 14.—						
3.0	004212·6	47	1·6	0·0902165	112	17
3.5						
3.15	004261·7	47	1·6	0·0905225	112	18
3.20						
3.30	004307·9	47	1·6	0·0908283	112	17
3.35						
3.45	004356·3	47	1·8	0·0911346	112	17
3.50						
4.0	004405·8	47	1·7	0·0914401	112	17
4.5						
TRIAL 20.—						
1.15	92·5			0·1093390	190	2
1.20						
1.21					0·1094393	190
1.25	120·5			0·1095399	190	2
1.28					0·1095996	190
TRIAL 21.—						
	4 c. ft. of gas per minute				112	17

MIDDLE ENGINE "R."—continued.

Observer, Mr. CLERK.				Mr. HAYWARD.	Professor ASHCROFT.	Captain SANKKY.	Mr. HAYWARD.
Time.	Anemo-meter.	Baro-meter.	Air Temp.	Indicator Diagrams taken at	Exhaust Gases Samples taken at		Junker Calorimeter. Gas tested at
		Inches of Mercury.	° F.	Time.	Time.		Time.
31 DEC., 1904.							
1.0	0.0454620	30.00	45.1				
1.10	0.0456610						
1.20	0.0458580		45.0				
31 DEC., 1904.							
1.25	0.0459530						
1.30	0.0460450						
1.35	0.0461380						
2 JAN., 1905.							
		Professor DALBY.					
3.10	0.0530038	30.20	49.2		3.5		10.40
3.25	0.0533085	30.20	48.8				
3.40	0.0536189	30.20	48.6	3.33	3.34		
				3.38			
				3.51			
3.55	0.0539281	30.20	48.4	3.56			
				4.0			
4.10	0.0542358		49.8				
4 JAN., 1905.							
	Exhaust Temperature.						
1.15	760	29.71	54.0				
1.17	761						
1.19	761						
1.21	761						
1.23	761						
1.25	764						
1.27	761						
4 JAN., 1905.							

Figs. 11 and 12, Plate 1.

TABLES VI.—TRIALS OF

Observer, Professor DALBY.						
Time.	Gas Meter.	Temperature	Pressure.	Counter.	W.	w.
		° F.	Inches of Water.		Lbs.	Lbs.
11.0				0·0947700	TRIAL 15.— 572	15.— 50
11.5	007491·0	52	1·6			
11.10				0·0949314	572	50
11.15	007616·5	52	1·4			
11.20				0·0950938	572	55
11.25	007744·9	52	1·4			
11.30				0·0952566	572	50
11.35	007873·0	52	1·4			
11.40				0·0954205	572	52
11.45	008002·1	53	1·4			
11.50				0·0955844	572	45
11.55	008135·1	53	1·4			
12.0				0·0957550	572	55
12.5	008266·9					
12.15				0·0957550	TRIAL 16.— No	load
12.20	008305·0	53	1·6			
12.35				0·0961076		
12.40	008339·2					
1.0	008369·7	53	1·6			
1.15				0·0968114		
1.20	008400·2	53	1·6			
3.0	008972·0	51	1·4		TRIAL 17.—	
3.5				0·0968114	270·5	7
3.15	009079·0					
3.20				0·0970684	270·5	8
3.30	009193·5					
3.35				0·0973272	270·5	6
3.45	009308·1	51	2·0			
3.50				0·0975854	270·5	6
4.0	009422·9	51	2·4			
4.5				0·0978435	270·5	5
10.50				0·0978434	TRIAL 18.—	
10.55				0·0979097	No	load
11.0	Motor	driven		0·0979789		
11.5				0·0980487		
11.15				0·0981891		
11.25				0·0983327		
11.35				0·0984762		
11.45				0·0986225		
11.50				0·0986966	TRIAL 19.—	
12.0	Motor	driven		0·0986966	No	load
12.10				0·0988525		
12.20				0·0990079		
12.30				0·0991635		

LARGE ENGINE "X."

Observer, Mr. CLERK.				Mr. HAYWARD.	Professor ASHCROFT.	Captain SANKEY.	Mr. HAYWARD.
Time.	Anemo-meter.	Baro-meter.	Air Temp.	Indicator Diagrams taken at	Exhaust Gases Samples taken at		Junker Calorimeter. Gas tested at
		Inches of Mercury.	° F.	Time.	Time.		Time.
3 JAN., 1905.							
11.0	0.0556200	30.00	56.8				
11.10	0.0557970			11.5	11.5		10.45
11.20	0.0559760		57.8	11.13	11.20		10.50
11.30	0.0561550			11.24	11.30		
11.40	0.0563340		56.5	11.33			
11.50	0.0565160			11.45			
12.0	0.0567040	30.00	56.2	11.54	11.50		
12.5	0.0567950						
3 JAN., 1905.							
12.15	0.0570040	29.99	55.6				
12.25	0.0572210		52.2				
12.35	0.0574430		55.0				
12.45	0.0576725		54.7	12.41			
12.55	0.0578960		54.6	12.48			
1.5	0.0581203		54.3	12.51			
1.15	0.0583465	29.99	54.0	12.58			
1.30	0.0586880		53.8				
3 JAN., 1905.							
3.5	0.0607330	29.93	53.4		3.0		
3.15	0.0609470		53.4	3.12	3.5		
3.25	0.0611470		53.6	3.20			
3.35	0.0613560		53.4	3.30			
3.45	0.0615750		53.6	3.40	3.42		
3.55	0.0617780	29.94	53.6	3.50			
4.5	0.0619880		53.4	3.56			
4 JAN., 1905.							
10.50	0.0621810	29.77	52.4				
11.0	0.0623620						
11.10	0.0625500		53.4				
11.20	0.0627440		54.4				
11.30	0.0629380		55.1				
11.50	0.0633440	29.75	55.0				
3 JAN., 1905.							
12.0	0.0635510		55.1				
12.10	0.0637660		55.6				
12.20	0.0639780		55.5				
12.30	0.0641940	29.74					

Figs. 11 and 12, Plate 5.

TABLE VII.—DATA FOR HEAT ACCOUNTS

(The numbers representing Heat-Units in this Table stand for British Thermal

Designation of Engine	
Brake Horse-Power	
Test Number	
Line.	
1	Difference between wet- and dry-bulb thermometer d
2	Glaisher's factor f
3	Dew point ($t_1 - df$) t
4	Temperature of air by dry bulb (Plate 6) t_1
5	Pressure of steam at dew point. Lbs. per sq. inch. From steam tables p
6	Atmospheric pressure, $0.491 \times$ height of barometer, from Plate 6 p_1
7	Pressure of dry air at anemometer $p_1 - p = p_0$
8	Volume of 1 lb. of dry air in cubic feet at } $\frac{53.18 (t_1 + 461)}{p_0 \times 144} = v$ temp. t_1 and pressure p_0 , calculated from
9	Volume of air entering per hour measured by anemometer (Plate 6). V
10	Weight of dry air entering per hour, in lbs. $\frac{V}{v} = w$
11	Weight of steam associated with 1 lb. of dry } $\frac{v}{\text{spec. vol. of steam}} = w_1$ air at dew point
12	Weight of steam brought in by air per hour $w \times w_1 = w_2$
13	Total heat of 1 lb. of steam at pressure p from Tables
14	Heat brought in by steam in air per hour
15	Heat brought in by dry air per hour $0.24 \times w (t_1 - 32)$
16	Temperature of exhaust-gases leaving calorimeter t_0
17	Weight of steam in each lb. of exhaust-gases saturated at t_0 w_3
18	Total heat of steam in each lb. of exhaust gases, from Fig. 5 h
19	Weight of water produced per hour by combustion of gases } deduced from observations made with Junker calorimeter } w_4
20	Density of gas at 14.7 lbs. per sq. in. and 32° F. 0.038
21	Cubic feet of gas supplied per hour at temp. } and pressure of observation } G
22	Weight of gas supplied per hour assumed to be dry at entry. Lbs. w_5
23	Total weight of dry exhaust-gas per hour $(w + w_5 - w_4) = w_6$
24	Weight of steam carried away per hour by exhaust-gases $w_3 \times w_6 = w_7$
25	Observed weight of cooling water leaving } calorimeter per hour (Plate 6). } w_8
26	Weight of cooling-water entering } calorimeter per hour } $w_8 + w_7 - w_2 - w_4 = w_9$
27	Heat brought in by water entering calorimeter $w_9 \{t_2 - 32\}$
28	Heat brought into cylinder by gas $w_5 \{t_3 - 32\} \times 0.508$
29	Heat brought in by jacket-water. Weight of } water observed per hour = w_{10} (See Plate 6) } $w_{10} (t_4 - 32)$
30	Heat carried away by exhaust-gases per hour $w_6 \{t_0 - 32\} \times 0.239$
31	Heat carried away by steam in exhaust-gases $w_7 \times h$
32	Heat carried away by calorimeter-water per hour $w_8 \times \{t_5 - 32\}$
33	Heat carried away by jacket-water per hour $w_{10} \{t_6 - 32\}$

REDUCED FROM THE OBSERVATIONS RECORDED.

Units per hour reckoned from 32° F. All temperatures are in degrees Fahrenheit.)

Line.	L			R			X		
	0	2·87	5·2	0	10·8	20·9	0	27·9	52·7
	No Load.	‡ Load.	Full Load.	No Load.	‡ Load.	Full Load.	No Load.	‡ Load.	Full Load.
	4	3	2	10	14	9	16	17	15
1	Say 3°	Say 3°	Say 3°	Say 3°	Say 3°	Say 3°	2·9°	2·4°	3°
2	1·9	1·9	1·9	2·1	2·1	2·1	2·0	2·0	1·9
3	51·3	50·8	50·3	42·6	42·7	42·1	49·2	48·7	51·1
4	57·0	56·5	56·0	48·9	49·0	48·4	55·0	53·5	56·8
5	0·186	0·182	0·179	0·134	0·135	0·132	0·172	0·169	0·184
6	14·72	14·72	14·74	14·6	14·84	14·58	14·74	14·71	14·74
7	14·53	14·54	14·56	14·47	14·70	14·45	14·55	14·54	14·56
8	13·16	13·14	13·11	13·01	12·81	13·02	13·09	13·07	13·13
9	1,162	939	833	3,990	3,548	3,209	9,090	8,635	7,664
10	88·3	71·46	63·54	306·7	277·0	246·5	694·4	660·6	583·7
11	0·00782	0·00768	0·00753	0·00561	0·00564	0·00554	0·00722	0·00708	0·00776
12	0·690	0·549	0·478	1·72	1·56	1·37	5·01	4·67	4·53
13	1,097	1,097	1,097	1,095	1,095	1,095	1,097	1,096	1,097
14	757	602	524	1,883	1,708	1,501	5,510	5,130	4,970
15	530	420	366	1,244	1,130	970	3,833	3,409	3,474
16	71·3	92·6	88·1	69·3	88·3	91	83	98·9	87·6
17	0·0162	0·0333	0·0287	0·0151	0·0289	0·0316	0·025	0·041	0·028
18	17·9	37·0	31·9	16·6	32·0	35·1	27·8	45·6	31·3
19	0·775	2·94	4·53	2·75	11·6	17·9	5·73	26·7	45·0
20	0·038	0·038	0·038	0·038	0·038	0·038	0·038	0·038	0·038
21	15	56·8	87·7	50·8	199·8	331	100	466	785
22	0·56	2·09	3·23	1·88	7·48	12·2	3·68	17·2	28·9
23	88·1	70·6	62·3	305·9	273·0	241·2	692·5	651·1	568·5
24	1·43	2·35	1·79	4·62	7·89	7·63	17·31	26·70	15·92
25	352·7	386·7	584	366·2	1,105	1,807	164·7	2,178	4,287
26	352·6	385·6	580·7	366·3	109·9	1,795	171·3	2,173	4,253
27	9,308	10,410	14,980	4,834	16,920	20,640	2,398	26,730	47,630
28	6	23	36	15	57	99	39	166	300
29	80	2,727	5,701	0	5,063	7,000	935	8,148	12,290
30	827	1,024	836	2,727	3,674	3,400	8,608	10,400	7,554
31	1,577	2,615	1,988	5,078	8,736	8,466	19,250	29,700	17,790
32	13,470	22,390	35,610	13,330	61,870	107,000	8,400	139,600	257,700
33	344	11,470	17,370	0	37,110	62,060	7,150	72,900	124,700

TABLE VIII.—HEAT ACCOUNTS.
The numbers in this Table stand for British Thermal Units per Hour, reckoned from 32° F.

Designation of Engine	L			R				X		
	0	2·87	5·2	0	10·8	20·9	0	27·9	52·7	
Brake Horse Power	No Load.	Half Load.	Full Load.	No Load.	Half Load.	Full Load.	No Load.	Half Load.	Full Load.	
Test Number	4	3	2	10	14	9	16	17	15	
<i>Heat brought in per hour.</i>										
1. By combustion of gases. Lower calorific value	8,490	32,260	49,630	28,800	117,200	187,700	57,400	267,500	450,600	
2. " total heat of water produced by combustion	839	3,070	4,910	3,150	12,800	20,500	6,200	28,900	48,700	
3. " moisture in air	757	602	524	1,883	1,708	1,501	5,510	5,130	4,970	
4. " dry air	530	420	366	1,244	1,130	970	3,883	3,409	3,474	
5. " calorimeter water-supply	9,308	10,410	14,980	4,834	16,920	20,640	2,398	26,730	47,630	
6. " entering gas	6	23	36	15	57	99	39	166	300	
7. " jacket-water supply	80	2,727	5,701	..	5,063	7,000	935	8,148	12,290	
Total	20,000	49,512	76,147	39,926	154,878	238,410	76,315	339,983	567,964	
<i>Heat carried away per hour.</i>										
8. By exhaust-gases	827	1,024	836	2,727	3,674	3,400	8,608	10,400	7,554	
9. " steam in exhaust-gases	1,577	2,615	1,988	5,078	8,736	8,466	19,250	29,700	17,790	
10. " calorimeter-water	13,470	22,390	35,510	13,330	61,870	107,000	8,400	139,600	257,700	
11. " jacket-water	344	11,470	17,370	..	37,110	62,060	7,150	72,900	124,700	
12. Heat equivalent of B.H.P.	By diff.	7,300	13,230	By diff.	27,510	53,180	By diff.	71,000	134,100	
13. Radiation	3,792	3,792	3,792	18,791	18,791	18,791	32,907	32,907	32,907	
14. Unaccounted for	20,010	48,591	72,726	39,926	157,711	252,897	76,315	356,507	574,751	
Total	+921	+3,421	..	-2,833	-14,487	..	-16,524	-6,787	

TABLE IX.—HEAT BALANCE-SHEET.
(In British Thermal Units per Hour.)

Designation of engine .	L			R			X		
	0	2·87	5·2	0	10·8	20·9	0	27·9	52·7
Brake horse-power . .	No Load.	Half Load.	Full Load.	No Load.	Half Load.	Full Load.	No Load.	Half Load.	Full Load.
Test number . . .	4	3	2	10	14	9	16	17	15
Heat of combustion. Lower calorific value. }	8,490	32,260	49,630	28,800	117,200	187,700	57,400	267,500	450,600
Exhaust-waste . . .	4,434	11,504	17,518	10,009	41,665	75,156	18,278	115,365	177,970
Jacket-waste . . .	264	8,743	11,669	0	32,047	55,060	6,215	64,752	112,410
Radiation	3,792	3,792	3,792	18,791	18,791	18,791	32,907	32,907	32,907
Total waste	8,490	24,039	32,979	28,800	92,503	149,007	57,400	213,024	323,287
Heat converted into work equivalent of brake-HP.	7,300	13,230	..	27,530	53,180	..	71,000	134,100
Total accounted for	31,339	46,209	..	120,033	202,187	..	284,024	457,387
Unaccounted for	+921	+3,421	..	-2,833	-14,487	..	-16,524	-6,787

TABLE X.—HEAT BALANCE-SHEET.
In per cent. of the heat of combustion of the gas.

Designation of engine . .	L			R			X		
	0	2·87	5·2	0	10·8	20·9	0	27·9	52·7
Brake horse-power . . .			Full Load.	No Load.	Half Load.	Full Load.	No Load.	Half Load.	Full Load.
	No Load.								
Test number	4	3	2	10	14	9	16	17	15
Heat of combustion. Lower caloric value	100	100	100	100	100	100	100	100	100
Exhaust-waste	52·2	35·7	35·3	34·8	35·6	40·0	31·8	43·1	39·5
Jacket-waste	3·1	27·1	23·5	0·0	27·4	29·3	10·8	24·2	25·0
Radiation	44·7	11·8	7·6	65·2	16·0	10·0	57·3	12·3	7·3
Total waste	74·5	66·5	..	78·9	79·4	..	79·6	71·7
Heat converted into work equivalent of brake-H.P. }	0	22·6	26·7	0	23·5	28·3	0	26·6	29·8
Total accounted for	97·1	93·1	..	102·4	107·7	..	106·2	101·5
Unaccounted for	+2·9	+6·9	..	-2·4	-7·7	..	-6·2	-1·5

EFFICIENCY OF INTERNAL-COMBUSTION ENGINES

GENERAL SUMMARY OF RESULTS OF TRIALS AT ASHTON, TABLE X.

PLATE 6.

EXPERIMENT NO	I. H. P.				B. H. P.				GAS SUPPLY.						ANEMOMETER.						JACKET WATER				EXHAUST CALORIMETER				ANALYSIS OF EXH. GASES				EXP NO																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																						
	MEAN EFFECTIVE PRESSURE.	EXPLOSIONS MINUTE.	ENGINE CONSTANT.	I. H. P.	GAS PER I. H. P. PER HOUR.	NETT BRAKE LOAD.	REVS. PER MINUTE.	BRAKE WHEEL CIRCUM- FERENCE.	B. H. P.	GAS PER B. H. P. PER HOUR	MECH. EFF.	CUB. FT PER HOUR.	PRESS. AT METER.	TEMP. AT METER.	CALORIFIC VALUE AT WORKING TEMP. AND PRESS.		B. TH. U. SUPPLIED PER HOUR CALCULATED ON.		REVS. PER HOUR.	TEMP. OF AIR.	BAROMETER. INCHES OF MERCURY.	CONSTANT OF CALIBRATION	CUB. FT OF AIR SUPPLIED PER HOUR.	TOTAL CUB. FT OF MIXTURE PER HOUR.	PROPORTION OF AIR TO GAS.	PROPORTION OF AIR TO GAS IN EXPLOSIVE CHARGE.	WEIGHT OF WATER PER HOUR.	TEMP. ENTERING	TEMP. LEAVING.	B. TH. U. PER HOUR.	WEIGHT OF WATER PER HOUR	TEMP ENTERING		TEMP LEAVING	RISE OF TEMP.	B. TH. U. PER HOUR.	TEMP OF EXHAUST GASES.	CO ₂	O ₂	N ₂	CORRES- PONDING AIR SUPPLY																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																														
															HIGHER B. TH. U.	LOWER B. TH. U.	HIGHER VALUE.	LOWER VALUE.																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																					
	LBS. S ²				CUBIC FT	LBS.		FEET		CUBIC FT		CORRECTED FROM FIG. 3.	INCHES OF WATER.	F°						F°		CUB. FEET PER REV. OF POINTER.					LBS.	F°	F°			F°	F°	F°	F°																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																				
NO 1. DEC. 28. PRELIMINARY TRIAL. ENGINE L.	79.3	119.3	0.0006	5.68	15.28	65	266	9.198	4.82	18	.849	86.8							14120	56.5	30.06	.0595	840	926.8	9.68	8.58	147								91.5	5.32	9.88	84.8	9.85	1																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																															
NO 2. DEC. 29. FULL LOAD. ENGINE L.	76.3	125	"	5.72	15.33	72.3	258.9	MEAN OF TWO. 9.195	5.2	16.87	.909	87.7	1.65	54°	622	566	54,540	49,630	14000	56.0	30.00	.0595	833	920.7	9.50	9.15	159.7	67.7	110.8	11680	584	57.8	92.8	35	20440	88.08	5.23	9.43	85.34	9.75	2																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																														
NO 3. DEC. 29. 1/2 LOAD. ENGINE L.	80.5	74.6	"	3.60	15.78	37.5	275.1	"	2.87	19.8	.797	56.8	1.6	54°	"	"	35,330	22,260	15780	56.5	29.96	.0595	939	995.8	16.5	8.19	98.8	59.6	148.1	8746	386.7	59	89.9	30.9	11949	82.6	2.55	14.95	82.50	18.5	3																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																														
NO 4. DEC. 29 RADIATION TEST. ENGINE L.	82.6	19.3	"	.96	15.62	NO LOAD.	291					15.0	2.4	53.8°	"	"	9,329	8,490	19520	57.0	29.95	.0595	1162	1177	77.5	9.41	3.0	58.7		852.7	58.4	10.2	11.8	4162	71.3					4																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																															
NO 5. DEC. 30. CALIBRATION OF ANEM. ENGINE L.	ENGINE DRIVEN BY MOTOR						285												17600	51.6	29.17	.0595					116	43.5	59.5	1856										5																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																															
NO 6. DEC. 30. CALIBRATION OF ANEM. ENGINE L.	"	"	"	"	"	"	263												16080	50.4	29.47	.0595					116	43.5	57.0	1566											6																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																														
NO 7. DEC. 30. FULL LOAD. ENGINE R.	96.7	97.95	0.00274	26.0	12.92	191	203.4	18.438	21.7	15.98	.835	336	1.7	18°	629	567	211,400	190,500	10920	48.6	29.5	.286	3123	3459	9.73	8.92	713.1	13.6	155.8	60930	2784	42.7	75.0	32.3	89920	74.0	4.7	11.2	84.1	11.1	7																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																														
NO 8. DEC. 30. FULL LOAD. ENGINE R.	TUBE COOLED DOWN AND TRIAL RESTARTED BUT PRE-IGNITION OCCURRED AGAIN WITHIN 5 MINUTES OF STARTING. TEST STOPPED AND INDICATOR CONNECTION CHANGED.																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																						</

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