APPENDIX II.

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 THE MECHANICAL AND THERMAL EFFICIENCY
OF A PETROL ENGINE.

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The experiments described in this paper were made in the Engineering Laboratory of Cambridge University with the permission of Professor Hopkinson. The engine used was a 16-20 four-cylinder Daimler, kindly lent by the manufacturers. The following are its leading dimensions :—

Diameter of cylinders	3·56 inches (90 mm.).
Stroke	5·11 inches (130 mm.).
Compression ratio	3·85.

The object of the experiments was to find the mechanical and thermal efficiency of the engine under various conditions, and to determine under what conditions the best efficiency might be obtained. As the engine works on the Otto cycle, the complete indicator diagram consists of two loops, the larger one being due to the difference between the compression and expansion lines and the smaller to the difference between the suction and exhaust lines. In this paper the expression "indicated horse-power" will be taken to mean the area of the larger loop multiplied by a suitable constant; and the expression "pumping losses" will mean the area of the smaller loop to the same scale. The ratio of B.H.P. to I.H.P. will be called the mechanical efficiency, and the difference between I.H.P. and B.H.P. the mechanical losses. The difference between the mechanical losses and the pumping losses will be called the friction losses.

Mechanical Efficiency.

An article by Professor Hopkinson, published in "Engineering," February 12th, 1907, gave the general characteristics of the engine

when running under ordinary conditions. In this paper a curve is given of the mechanical losses at various speeds; it was obtained in the following manner:—The engine was run with only one cylinder firing, and the power developed in that cylinder was measured by means of an indicator designed by Professor Hopkinson, and described at a meeting of the Institution of Mechanical Engineers on October 18th, 1907. This power was taken as the mechanical loss in the engine. The speed was varied by altering the timing of the spark. In order to reduce as far as possible the pumping losses in the three idle cylinders, two of them were disconnected from the induction pipe, and drew in air at atmospheric pressure. This method was used up to about 800 R.P.M., beyond which speed it was unreliable, owing chiefly to the vibration of the engine and the inertia of the indicator rig. To obtain the top point of the curve, the maximum speed at which the engine would run with only one cylinder firing was determined, and it was assumed that, when running at full load at this speed, the mechanical losses would be equal to the I.H.P. of one cylinder, *i.e.*, equal to one-third of the B.H.P.; or in other words, that at this speed the mechanical efficiency would be 75 per cent.

The following alternative method of finding the mechanical efficiency of the engine was devised by the author. The engine was fitted with a Prony brake, with a 30-pound weight hanging on an arm at a radius of $42\frac{1}{2}$ inches. The weight was partly supported by a spring balance, and the brake load measured by the change in the reading of the balance. The engine was first run at full load at some particular speed, and the brake load carefully noted; one cylinder was then stopped firing, and the brake was adjusted till the speed was the same as before; the change in the brake load was then, subject to certain corrections, a measure of the I.H.P. of that cylinder. This was repeated for each cylinder, and the I.H.P. of the whole engine was thus found.

Let I be the I.H.P. of any cylinder :

Let P_1 be the pumping losses in the cylinder when that cylinder is firing :

Let P_2 be the pumping losses in the cylinder when that cylinder is not firing :

Let D be the area reduced to H.P., between the compression and expansion lines of a cylinder which is not firing :

Let B be the B.H.P. with all four cylinders firing :

Let b be the B.H.P. with three cylinders firing :

Let F be the horse-power absorbed in mechanical friction, exclusive of pumping.

Owing to a variety of causes, the I.H.P. of each cylinder is not the same; this does not affect the formulæ, but may be noticed in the table following, where b_1, b_2, b_3, b_4 , are the various values of b .

Then we have :

$$B = 4I - 4P_1 - F$$

$$b = 3I - 3P_1 - P_2 - D - F.$$

FIG. 6.

Subtracting b from B separately for each cylinder, and adding the results, we have :

$$4I = 4(B - b) - 4(D + P_2 - P_1).$$

Now, if we assume that $(P_1 - P_2)$ and D may be neglected, we have :

$$4I = 4(B - b).$$

But in order to get more accurate determinations of the I.H.P. the values of P_1, P_2 , and D were measured at three speeds with Professor Hopkinson's indicator, using a very light spring, and the true I.H.P. calculated by means of the above formula. These diagrams were not photographed, but were observed in a telescope, and plotted on squared paper. A pair is reproduced in Fig. 6.

TABLE I.

Speed	720	1000	1220
Temperature of jacket water	212° F.	212° F.	212° F.
B.H.P. with four cylinders firing (B)	11.80	14.72	17.10
Change in B.H.P. on stopping No. 1 (B - b1)	3.65	4.05	5.28
Change in B.H.P. on stopping No. 2 (B - b2)	3.76	4.89	5.76
Change in B.H.P. on stopping No. 3 (B - b3)	3.28	4.72	5.52
Change in B.H.P. on stopping No. 4 (B - b4)	2.92	4.72	5.18
Approximate I.H.P. (4 (B - b))	13.62	18.38	21.74
4P1 in H.P. (from diagram)	0.38	0.64	1.42
4P2 in H.P. ,,	0.46	1.36	1.21
4D in H.P. ,,	0.62	0.77	0.93
Correction [4 (D + P2 - P1)]	0.70	1.47	0.72
Corrected I.H.P.	12.92	16.91	21.02
Pumping losses	0.38	0.64	1.42
Friction losses (by difference) (F)	0.74	1.55	2.50

In order to secure steady running, the mixture in these trials was made stronger than the normal, and the spark was not quite fully advanced; this reduces the H.P. below its normal value, but does not affect the losses which it is the purpose of this trial to measure.

All the values obtained for mechanical losses are shown in Fig. 7. Points marked with a round dot are those obtained by the first or one-cylinder method. Curve B. is that already published (*a*), and represents the losses with the jacket water at about 150° F. Curve A. gives the losses when the jackets are flooded with cold water, and curve C. with the jacket water boiling. The effect of both speed and temperature on the losses is clearly indicated by these curves. Points marked with a cross are those obtained by the three-cylinder method, but to which the correction described above

(*a*) "Engineering," February 8th, 1907.

has not been applied. Points not marked with any temperature were obtained with boiling jacket water. It will be noticed that the agreement between the two methods is extremely close, but in the author's opinion this is partly fictitious, as the friction losses when using the three-cylinder method are sure to be less than when using the one-cylinder method, owing to the higher temperature of the cylinder walls themselves, though the jacket water may be at the same temperature in both cases. In the curves, this is compensated for by the fact that the three-cylinder method overestimates the losses to a greater extent than the one-cylinder method.

FIG. 7.

The three corrected values for the losses given in Table I. are shown in curve D., and these are the values used to determine the I.H.P. in the latter part of this paper.

These results show that the author's approximate method is at least as accurate as the one-cylinder method, and that no apparatus beyond that required for an ordinary brake test is necessary. Further, there is reason to suppose that the value of D., the difference between the compression and expansion curves of a cylinder which is not firing, is not very far from constant in engines of this type running at about the same mean piston speed, *i.e.*, 1,000 feet

per minute, and with a similar compression pressure. In this engine, its value in pounds per square inch is 3.0 at 1,220 and at 1,000 R.P.M., and 3.35 at 720 R.P.M. The difference between P1 and P2 is usually small, particularly if the exhaust is almost directly into the atmosphere so that the inertia of the gases in the exhaust pipe is not large. This condition was fulfilled in the trials at 720 R.P.M. and 1,220 R.P.M., but during the trial at 1,000 R.P.M. the exhaust passed through a long pipe. In this way, by determining the I.H.P. by the author's approximate method and applying a correction of say 3 pounds per square inch on each piston during one stroke in each cycle, a very approximate estimate of the mechanical efficiency may be obtained.

Calculated as percentages of the I.H.P., the final figures work out as follows:—

TABLE II.

Speed	720	1000	1220
Friction loss	5.7	9.2	11.4
Pumping loss.....	2.9	3.8	6.8
B.H.P.	91.4	87.0	81.8
I.H.P.	100.0	100.0	100.0

The lubrication was by splash only, and the amount of lubricant present in the crank chamber was estimated by observing whether the exhaust was smoking or not.

Careful tests were made at 720 R.P.M. to find out whether the amount of oil had any effect on the mechanical losses or I.H.P., and it was found that whether the exhaust was extremely smoky or entirely clear there was no change in either. However, at 1,220 R.P.M., the amount of oil seemed to have a distinct effect on the mechanical losses, and at this speed particular care was taken to have a normal amount of oil present.

It will be noticed that the pumping losses account for about one-third of the mechanical losses at all speeds. This is, of course, with the throttle full open; but if the engine be throttled the losses become as follows at 720 R.P.M.

TABLE III.

	Full Load.	Half Load.	No Load.
Suction pressure, pounds per square inch ..	0·60	4·40	8·00
Friction loss in H.P.	0·74	0·74	0·74
Pumping loss in H.P.	0·36	0·58	1·45
B.H.P.	14·50	7·40	0·60
I.H.P.	15·60	8·72	2·79

This increase in the pumping loss is the chief cause of the great falling off in the efficiency of such engines at light loads.

This table also shows that great care must be taken to separate the pumping loss from the mechanical loss when the I.H.P. is obtained by adding the mechanical loss to the B.H.P.

Thermal Efficiency.

The investigation into the thermal efficiency of the engine is concerned almost entirely with the effect of varying the strength of mixture on the performance of the engine.

Fig. 8 is a diagrammatic sketch of the carburetter. In its normal setting the adjustable air supply was entirely closed, and the throttle B was full open. To strengthen the mixture the throttle B was more or less closed; the effect of this was to increase the suction at the jet, and to cause the flow of petrol to be increased. To weaken the mixture, one or more holes of the adjustable air supply were uncovered. These changes in the adjustment of the carburetter had no effect whatever on the major portion of the air drawn in through the main air inlet; and they made very little difference to the pressure in the inlet pipe, which was read by a mercury gauge. So it is sufficiently accurate to say that with the main throttle A in any given position, the amount of air drawn in per stroke is constant, and that the strength of mixture is directly proportional to the amount of petrol consumed per stroke.

The fuel used was Pratt's motor spirit, of density 0·715 to 0·720. Its calorific value was determined with the following apparatus: A very steady stream of air was passed from a small gasometer to the

bottom of a glass tube about $\frac{3}{4}$ inch diameter and 16 inches long, closed at the bottom and held nearly but not quite vertical. Petrol was admitted through a capillary tube at the top of this tube and allowed to fall on to its surface: the ascending current of air absorbed the petrol and the mixture passed out at the top of the tube to the Bunsen burner of a Junker calorimeter. The air was slightly warmed before entering the apparatus, the whole of which was steam jacketed: due allowance was made in the results for all heat supplied in this way. The petrol could be fed into the capillary tube mentioned above either from a measuring bulb holding 40 cc.

FIG. 8.

or from a flask holding about 250 cc. and which could be removed and refilled while the supply was being taken from the measuring bulb. The apparatus was started with petrol from the flask, and when everything was in a steady state the measured quantity was passed through from the bulb and the heat taken up by the Junker calorimeter was noted. Various experiments were made with rates of burning varying from half to the full capacity of the calorimeter. The mean of seven or eight readings gave the lower calorific value as 19,000 B.Th.U. per pound; the higher value being 20,200 B.Th.U. per pound. A sample was submitted to Dr. Fenton, of the

Chemical Department, Cambridge University, for analysis, and its composition was found to be

Hydrogen	14.86
Carbon	84.66
	99.52

FIG. 9.

For the experiments here described the Daimler engine was fitted with a Prony brake, speed indicator, revolution counter, and a graduated gauge on the petrol tank.

The jacket water was always kept boiling, and before beginning each test the timing of the spark was carefully adjusted to give maximum power. Ignition was by accumulator and coil. At the

beginning and the end of each test the height of the petrol in the tank and the reading of the revolution counter were carefully noted; a stop-watch was also started and stopped.

The speed was kept as steady as possible during the test, and the

FIG. 10.

brake load frequently read. Tests usually lasted about ten minutes. In some cases samples of the exhaust gases were taken, and were afterwards analysed. From the readings above described, the mean speed B.H.P. and petrol per 1,000 revolutions can at once be calculated and the I.H.P. found by adding the losses already found

to the B.H.P. Series of tests were made with the throttle full open at 720, 1,000, and 1,220 R.P.M., also at 720 R.P.M. at half-load and at almost no load. In each series of tests the mixture was varied over very wide limits by altering the adjustment of the carburetter.

Curves were plotted for each of those series, showing the relation between (1) horse-power and strength of mixture; and (2) thermal efficiency and strength of mixture. These curves are remarkably

FIG. 11.

similar, and those relating to 720 and 1,220 R.P.M. at full load and 720 R.P.M. at half and almost no load are reproduced in Figs. 9, 10, 11 and 12.

The analyses of exhaust gases are marked on the curves opposite the points to which they refer. The gases were collected and analysed over mercury to avoid the absorption of carbon dioxide by water. The carbon dioxide, oxygen and carbon monoxide were

determined in each case by the ordinary methods of volumetric gas analysis. In some cases the hydrogen was also estimated, and the remaining gases exploded with oxygen to detect the presence of hydro-carbons. The water produced by the explosion in the cylinder cannot be directly measured, but can be inferred from the amount

FIG. 12.

of oxygen unaccounted for: the amount of oxygen being calculated from the amount of nitrogen present.

The general features of these curves connecting power and thermal efficiency with strength of mixture have been fully explained in the paper read by Professor Hopkinson and the Author before Section G. of the British Association, 1907, and also in a paper by

Mr. Dugald Clerk on "The Principles of Carburetting," read before the Institution of Automobile Engineers in December, 1907.* It is therefore unnecessary to discuss them fully here, but it may be pointed out how the phenomena recur in exactly the same way at all speeds and loads, there being a very flat maximum of power

FIG. 13.

extending much beyond the point at which the petrol is all burnt, and a sharp maximum of thermal efficiency coinciding very nearly with the point when combustion is just complete.

* Proc. I.I.A.E., Vol. II.

In these concluding remarks it will be assumed that the carburettor is so set that the thermal efficiency is a maximum.

Fig. 13 gives curves showing the relation between (1) thermal efficiency and speed; and (2) thermal efficiency and power. The thermal efficiency reckoned on I.H.P. naturally increases with the speed owing to decreasing loss of heat to the cylinder walls; owing, however, to the falling off in the mechanical efficiency, the thermal

FIG. 14.

efficiency reckoned on the B.H.P. reaches a maximum at about 1,000 R.P.M. As the engine develops maximum brake power at 1,150 R.P.M., it appears that from 1,000 to 1,200 R.P.M. would

be the most economical speed at which to run the engine. This corresponds to a mean piston speed of from 870 to 1,020 feet per minute.

The theoretical thermal efficiency of an engine using the air cycle with a compression ratio of 3.85 is 41.5 per cent., so that the relative efficiency of this engine, reckoned on the I.H.P. at 1,200 R.P.M., is 67.5 per cent., and reckoned on the B.H.P. at 1,000 R.P.M. is 56.5 per cent.

Considering the thermal efficiency of the engine at 720 R.P.M. with various loads, it is apparent that a moderate amount of throttling has very little effect on the efficiency reckoned on the I.H.P., and that its effect on the efficiency reckoned on the B.H.P. is almost entirely due to the increase in pumping losses. At less than half load the efficiencies fall off more rapidly.

The same results in a different form are shown in Fig. 14 in a curve showing the relation between petrol consumption per minute and H.P. This is derived from the previous curve, and gives for I.H.P. a straight line law which is analogous to the "Willans" line for a steam engine.