SOME FACTORS AFFECTING THE POWER, TORQUE, AND ECONOMY OF TWO-CYCLE ENGINES

THESIS FOR THE DEGREE OF M. S.

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1933

THESIS

Gas + oil engines

Mechanical engineering

SOME FACTORS AFFECTING
THE
POWER, TORQUE, AND ECONOMY OF TWO CYCLE ENGINES

THESIS

SUBMITTED AS PART OF THE REQUIREMENTS

FOR THE MASTER OF SCIENCE DEGREE

BY

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THESIS

ACKNOWLEDGMENT

We wish to express our appreciation to Professor G. W. Hobbs and the Evinrude Outboard Motors Corporation for their assistance and cooperation.

OBJECT

The object of this work is to study the characteristics of a two-cycle engine and to improve the power, torque, and economy without sacrificing low speed and starting qualities.

INTRODUCTION

An ever increasing interest is being shown in the two-cycle engine in both the marine and the automotive industries. As yet the new developments in the automotive field have been few; but several of the larger concerns have been working for some time on this type. The most outstanding advances have been made in the marine field since the advent of the three-port two-cycle engine of this type in 1920. One of the more recent developments is the rotary valve which makes possible higher speeds and greater power.

The most obvious advantage of the two-cycle over the four-cycle engine is that a power impulse is delivered at every revolution instead of at every other revolution. This gives a considerable reduction in weight per H. P. output; but this advantage is somewhat offset by the low economy obtained. There is no inherent reason(as far as the ideal cycle is concerned) why the two-stroke cycle should be less efficient than the four-stroke cycle; however, certain features of design contribute to cause this condition. The more important of these features follow:

1. Because of the fact that the exhaust and intake events occur in the same part of the cycle, there is considerable mixing of the fresh charge with the burned gas. This slows up the combustion process, causing burning to take place for a considerable part of the expansion stroke. In effect, the ratio of expansion is decreased, the gas being under considerable pressure when the exhaust port is uncovered. In passing, it may be noted that it is this mixing of fresh and burned gases that permits the use of unusually high compression ratios without detonation.

2. Near the end of the expansion stroke there is a period when both the inlet and the exhaust ports are open at the same time. The fresh charge then has an opportunity to escape through the exhaust port. This loss is increased by an increase in the crankcase pressure or by a decrease in back pressure on the exhaust.

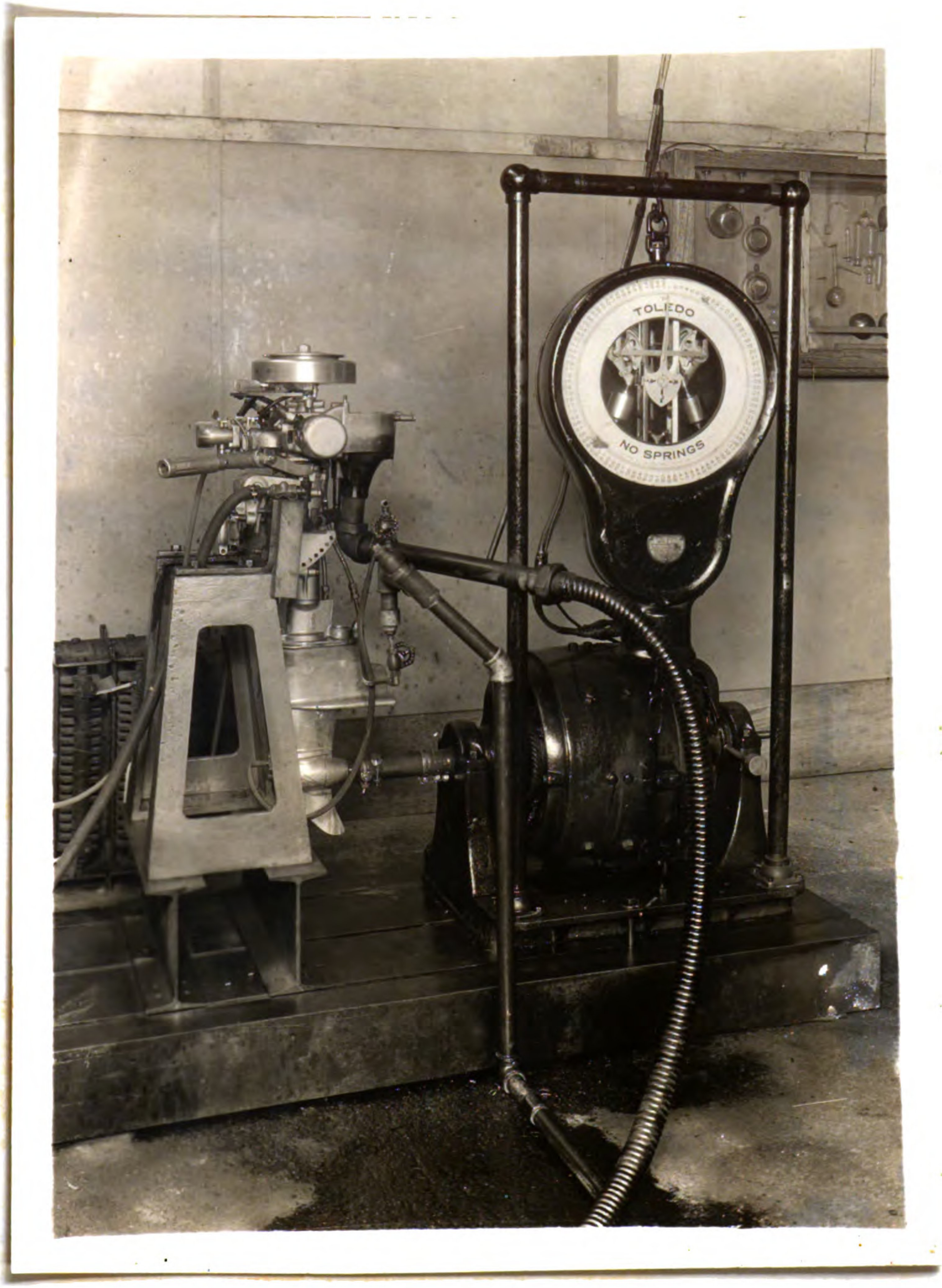
3. Obviously the pressure in the cylinder at the beginning of the intake event(the exhaust port being open) is greater than atmospheric; and it is necessary to give the charge an initial compression. This increases the pumping(friction) losses and decreases the volumetric efficiency.

It would seem that any advance in the design of two-cycle engines lies in the direction of improving upon some of the above-mentioned conditions.

APPARATUS AND SET-UP

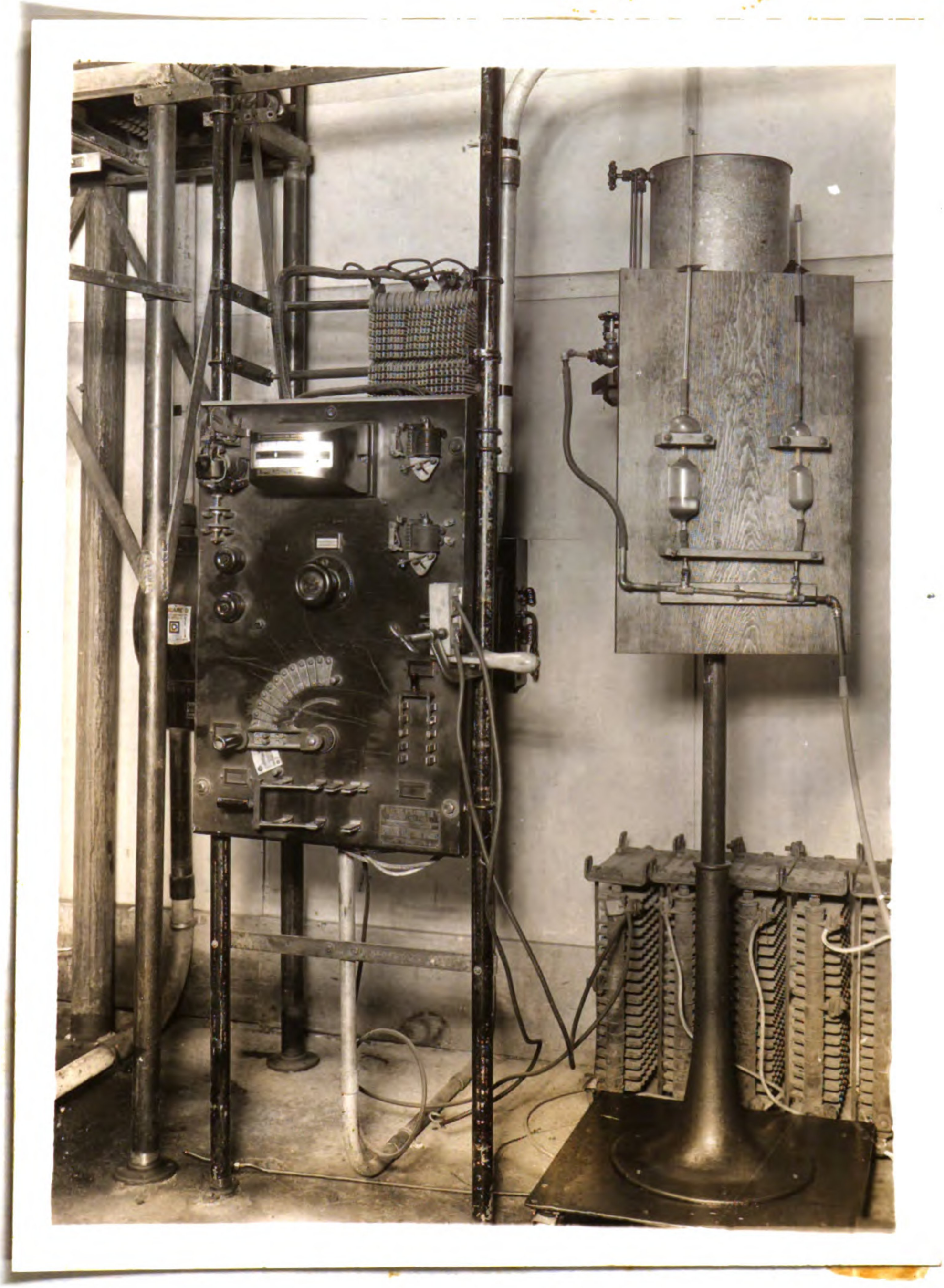
An Evinrude "Fleetwin" outboard engine was used for the test. This engine is of the two cylinder opposed type, both cylinders firing at once. It is provided with a disc rotary valve to control the admission of the charge. The engine proper is vertical, the upper end of the crankshaft carrying the flywheel which houses the magneto. The bottom end of the crankshaft is connected to the lower gear housing where two straight bevel gears transmit the power to the propeller shaft. The carburetor is of the conventional type with high speed and low speed jets.

The engine was mounted on a 2x8 inch plank(see page 4) supported by two cast iron legs which were bolted to the dynamometer bed. The normal path of the water - in at the gear housing, through the centrifugal pump, through the cylinders, into the muffler, and out - was altered to permit taking a heat balance. The pump was removed, and water from the service main was forced through the bottom unit; the line connecting the bottom unit to the cylinders was disconnected, and water was forced through the cylinders and muffler and thence to the weighing tank.



VIEW SHOWING SET-UP

OF APPARATUS



VIEW SHOWING SWITCHBOARD

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In order to determine the spark advance, a protractor was mounted on the engine directly below the spark lever.

The regular exhaust was removed and a special pipe coupling was adapted to the muffler. A flexible metal hose from this pipe discharged the gas to the main exhaust line. The vibration of the engine exhaust line made it difficult to obtain readings with a thermometer; so a chromel-alumel thermocouple was used in conjunction with a pyrometer.

The propeller was removed and the shaft was directly connected to an electric dynamometer. Considerable difficulty was experienced at this point, mention of which should not be irrelevant. Four aluminum spiders(three arm) were used in making a double universal coupling - aluminum being used to keep the radial load on the propeller shaft bearing as low as possible. One was mounted on the dynamometer shaft, and one on the engine shaft. The other two were keyed to the ends of a 5/8 inch shaft about 10 inches long, and the spiders were then connected through flexible fibre discs. It was found that this coupling gave too much torsional rigidity. The engine had a very large torque variation caused by the light flywheel used and by the fact that both cylinders fired simultaneously. This uneven torque, working against the inertia of the heavy armature resulted in the too-frequent shearing of keys and pins in the engine and coupling.

The remedy was simple and effective - a 10 inch length of $1\frac{1}{4}$ inch air hose was clamped to the two shafts, giving plenty

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plenty of torsional and lateral flexibility.

The dynamometer used was a 15 H. P. Sprague electric with a top speed of 4000 R. P. M. This dynamometer is of the conventional type, the load being measured directly by means of a set of Toledo scales. It was found necessary to connect a large bank of iron grid resistance and also a variable carbon pile rheostat in series with the line so that sufficiently low motoring speeds could be obtained.

PROCEDURE

All of the major instruments used were checked to insure their being in good condition. The scales were tested over the range used by means of weights. The dynamometer was checked by motoring it free; and it was found that the armature reaction exactly balanced the windage and friction. The tachometer used was checked against a calibrated Weston tachometer. The accuracy of the Orsat readings was indicated by the close grouping of the points on Lockwood's chart for the combustion of hydrocarbon fuels.

The engine, as received, was "run in" for four hours using 1½ pints of oil per gallon of gas. Some "Pyroil" graphitic lubricant was added to the fuel to insure adequate lubrication during this period.

The first run was a complete heat balance test on the original Evinrude pistons(set No. 1). The fuel was mixed

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according to the manufacturer's specifications(one pint of S. A. E. No. 40 oil per gallon of gas). It was metered to the carburetor through a 105 cc. burette from an over-head tank. The amount of cooling water was determined by weighing in a large tank; and the inlet and outlet temperatures were taken close to the engine. Exhaust gas temperatures and samples were taken close to the muffler. An Orsat apparatus of the bubbling type was used to make the gas analysis. The whole test was conducted with wide-open throttle, the spark and throttle being set for maximum power. The engine was carefully warmed up and adjusted before each test, and sufficient readings were taken to insure their accuracy.

Two extra pairs of pistons were supplied by the Evinrude Corporation which were identical with the originals excepting that the heads were blank. On one pair the heads were built up by welding so that the tops machined up 0.115 inches higher than the originals. The baffles were then milled out on this latter pair, a very light cut being taken. These were dressed down by hand until the clearance volumes checked within 0.5 c.c., these volumes being determined by assembling the engine and filling the clearance space with water. A complete power and economy test was then made on this pair(No. 2). The friction hp. was also checked by motoring with wide open throttle.

These pistons were removed and milled down, lowering the compression ratio and a similar test was then made on this pair(No. 3). Pistons No. 4 and 5 were handled in the same manner, an exhaust analysis being taken on No. 5. The other blanks(No. 6) were milled out to try the effect of a higher compression ratio with the same height of baffle(as compared with the originals).

The compression ratios with these special pistons were considerably higher than that of the original(No. 1) set, running as high as 7.95 on No. 2 set. In order to make certain that the engine was not losing power because of detonation, Ethyl fluid was mixed with the fuel in the burette. The power with and without the fluid checked exactly, proving the absence of detonation. As was mentioned before, the first test was made on the original pistons; and after all other tests had been run, it was repeated, using different speeds. The results(b.hp.) of both tests fell exactly on the same line, indicating that the engine remained in good operating condition and also that the testing procedure was consistent.

An attempt was made to secure the compression pressures with the different pistons by using a Holsaple gauge. As far as absolute values were concerned, the results were unsatisfactory because of the high engine speed and also because of difficulties with the rubber valve used in the gauge.

Very little trouble was experienced with the engine

during any of the tests. At one time the engine developed a tendency to alternately slow down and speed up. This trouble was traced to the magneto, where it was found that the breaker point facing had dropped from the breaker arm.

DISCUSSION

Heat Balance

The heat which is supplied by the fuel is roughly divided into five parts. These five divisions are b.hp., friction hp., loss to cooling water, radiation loss, and exhaust loss. For want of a better analysis of the problem, the friction loss is considered as separate from the loss to the cooling water. This must bring in some error since a large per cent of the total friction loss is due to piston friction which must show up as heat. At least a part of this heat must go to the cooling water. Some slight error is also introduced in assuming that the friction hp., as determined by motoring, is the same as that when the engine is running under load. "It is interesting to note that the frictional hp., as determined by subtracting the b.hp. from the i.hp. (the actual friction hp.), may be as much as 100 per cent greater than that determined by motoring."* This may be accounted for in part by the increase in rubbing forces because of higher cylinder pressures and in part by the better lubrication

under motoring conditions. The above-mentioned errors are not peculiar to this test, but are inherent in all heat balance tests.

* Automotive and Aircraft Engines - Judge.

The heat balance was taken with No. 1 pistons(Fig. 1) having a compression ratio of 4.22:1.

Figure 1

The most striking thing about this heat balance(to one who is familiar with a four-cycle balance) is that the loss to the exhaust is very high. This is accounted for chiefly by the fact that a large part of the charge is lost in the exhaust without being burned. (See discussion below). The heat energy contained in this unburned charge is included in the exhaust loss.

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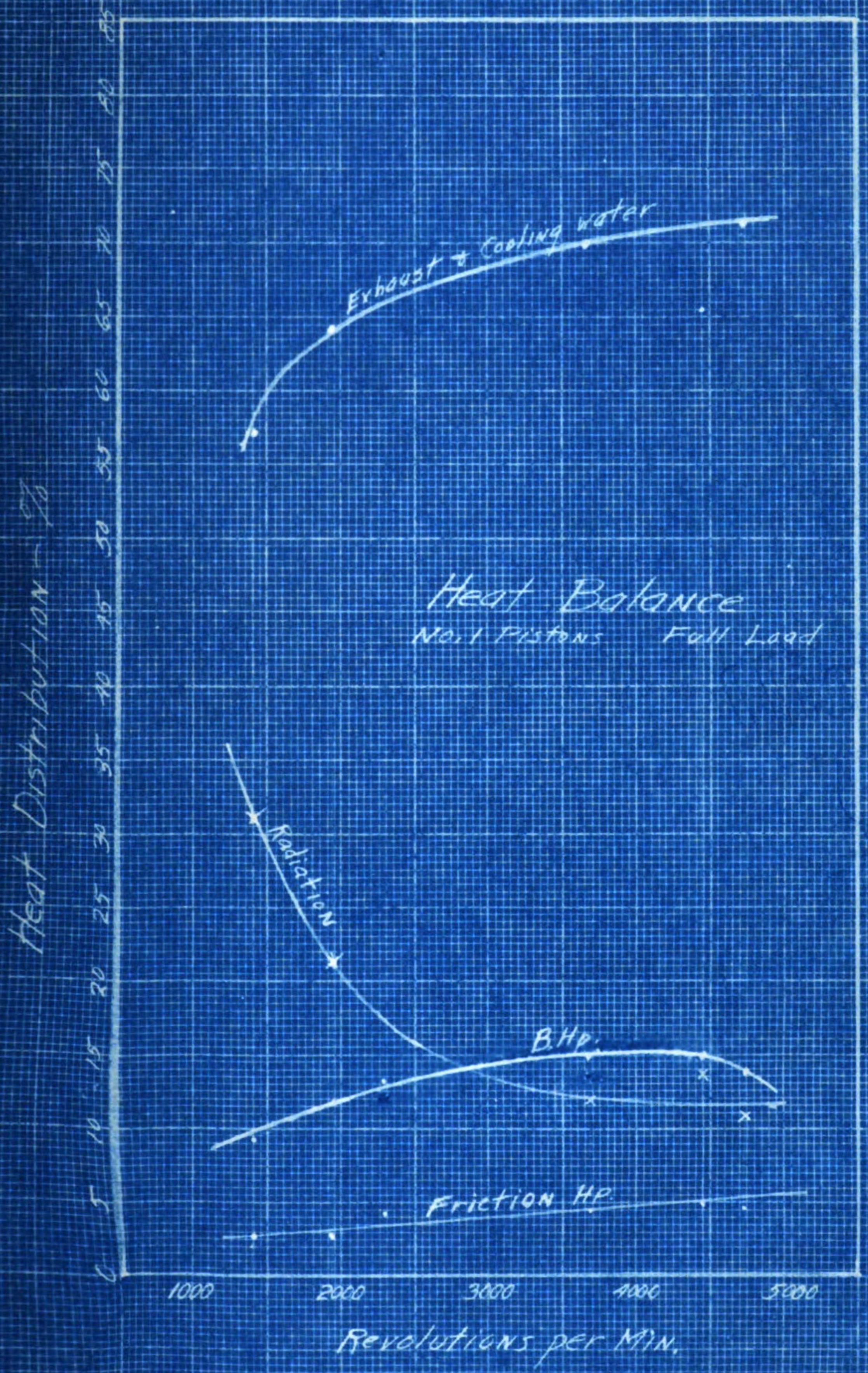


A. W. Judge, an English authority on internal combustion engines, gives a method for figuring, approximately, the amount of the fuel which escapes in the exhaust. His method is based on an exhaust gas analysis which includes the percentages of methane(CH_4) and hydrogen. No facilities were available for checking for these gases; but no serious error should be introduced by assuming the same values(for CH_4 and H) as were used in his computations. The analysis follows:

N_2 in exh. gas = 100 - (17+3) = 80%(by vol.)

(3% is the assumed value of CH_4+ H_2; and 17% is CO_2+ O_2+ CO from test)

O_2 in the fresh charge corresponding to 80 vols. of N_2 = 80 x .266 = 21.3 vols.

(.266 is the ratio of O_2/N_2 in the air)

The average O_2 in the exhaust is 7.5%; and since the proportion of O_2 indicates the proportion of charge:

$$\frac{\text{Vol. of charge escaping}}{\text{Total vol. of charge}} = \frac{7.5}{21.3} = 35.2\%$$

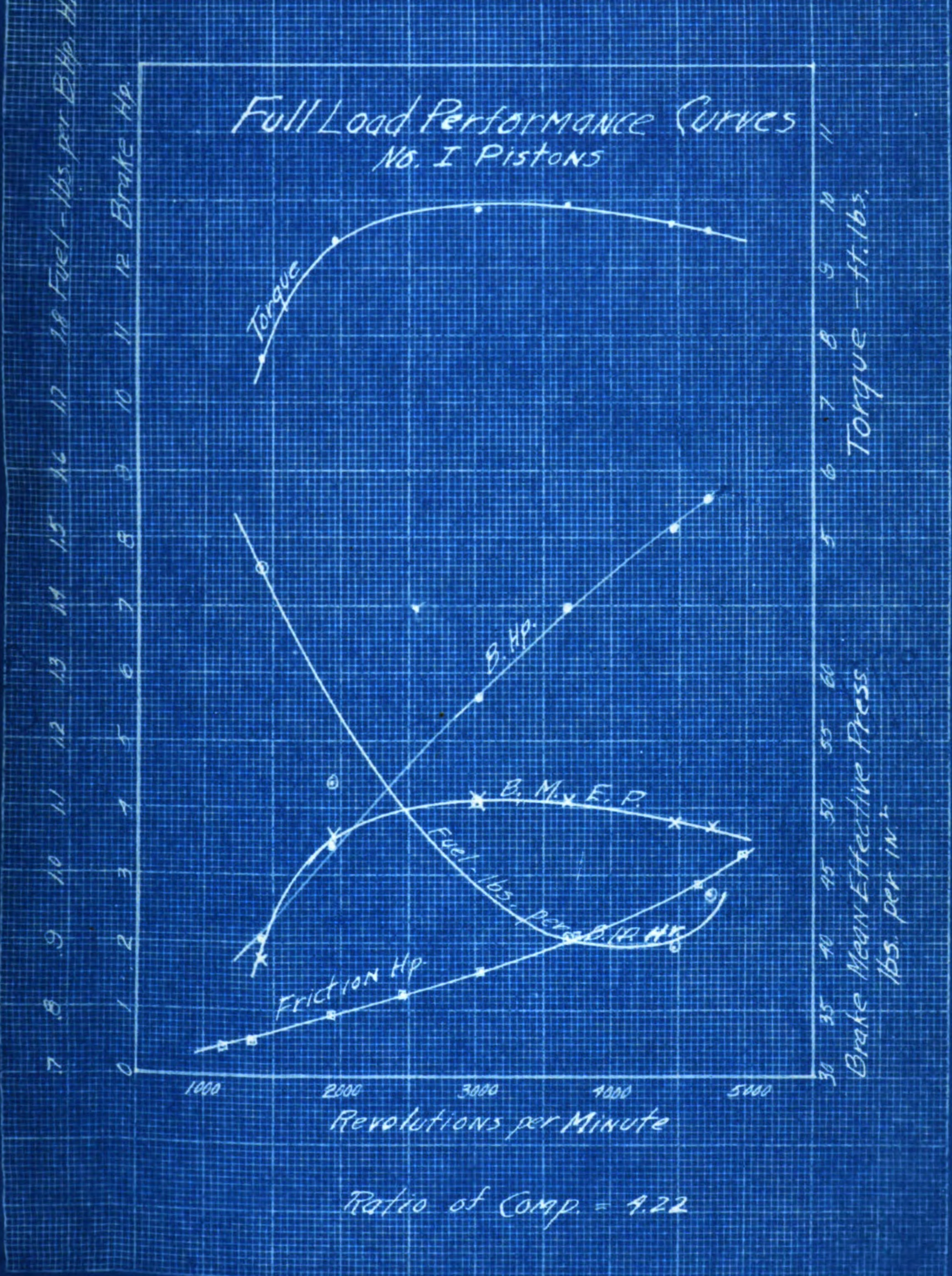
As a check on the above analysis:

It is well established that a 13:1 air fuel ratio in the cylinder gives maximum power; and since the carburetor was set for max. power, there must have been this ratio in the cylinders. For each pound of air there was 1/13 of a pound of gas in the charge; and the exhaust gas analysis showed 20.5:1 average air fuel ratio, indicating 1/20.5 pounds of gas per pound of air. Since only the burned charge is accounted for in the exhaust analysis, the difference, 1/13 - 1/20.5 or .0283 pounds of gas was not burned. The per cent of charge not burned would then be .0283 ÷ 1/13 or 36.4%, which is a fairly close check on the above analysis.

Judge, in his discussion of the variation of the charge loss with speed, shows the loss decreasing as the speed increases as is shown in Fig. 2-A.

Figure 2

"The diminution in the loss with increased speed is due to a falling off in the charge pressure in the crank-case and an increased terminal exhaust pressure in the cylinder ---


Full Load Performance Curves
No. I Pistons
Torque
B. Hp.
B. M. E. P.
Fuel - lbs. per B. Hp. Hr.
Friction Hp.
1000
2000
3000
4000
5000
Revolutions per Minute
Brake Hp.
Torque - ft. lbs.
Brake Mean Effective Press. lbs. per in.²
Ratio of Comp. = 4.22

due to increased back pressure".* Curve B shows the loss with a different valve timing giving a higher volumetric efficiency. "The timing of the port opening periods, also, has an appreciable effect upon the quantity of charge escaping; generally speaking, anything which increases the volumetric efficiency also increases the charge loss".* No such conditions were found in this test, the analysis indicating a fairly constant loss, the probable explanation being that the exhaust main was so large as to prevent the pressure from building up sufficiently to show a difference.

If the "net" fuel used, or the fuel actually burned, were used in figuring the heat balance, the percentages representing the b.hp.(thermal efficiency) would be considerably increased. "The net thermal efficiency values compare very favorably with the ordinary four-cycle values".*

The thermal efficiency(see Page 12) increases to a maximum of 16.5 per cent at 3800 R. P. M. The "net" thermal efficiency at this speed would be about 23 per cent, which is comparable to that of an average four-cycle engine, the maximum of which would be about 25 per cent. The maximum friction loss is about 5 per cent. It should be noted that the friction hp. includes the friction in the lower gear housing, making it appear unusually high. This loss should be a smaller per cent in a two-cycle than in a four-cycle because of the fact that the power

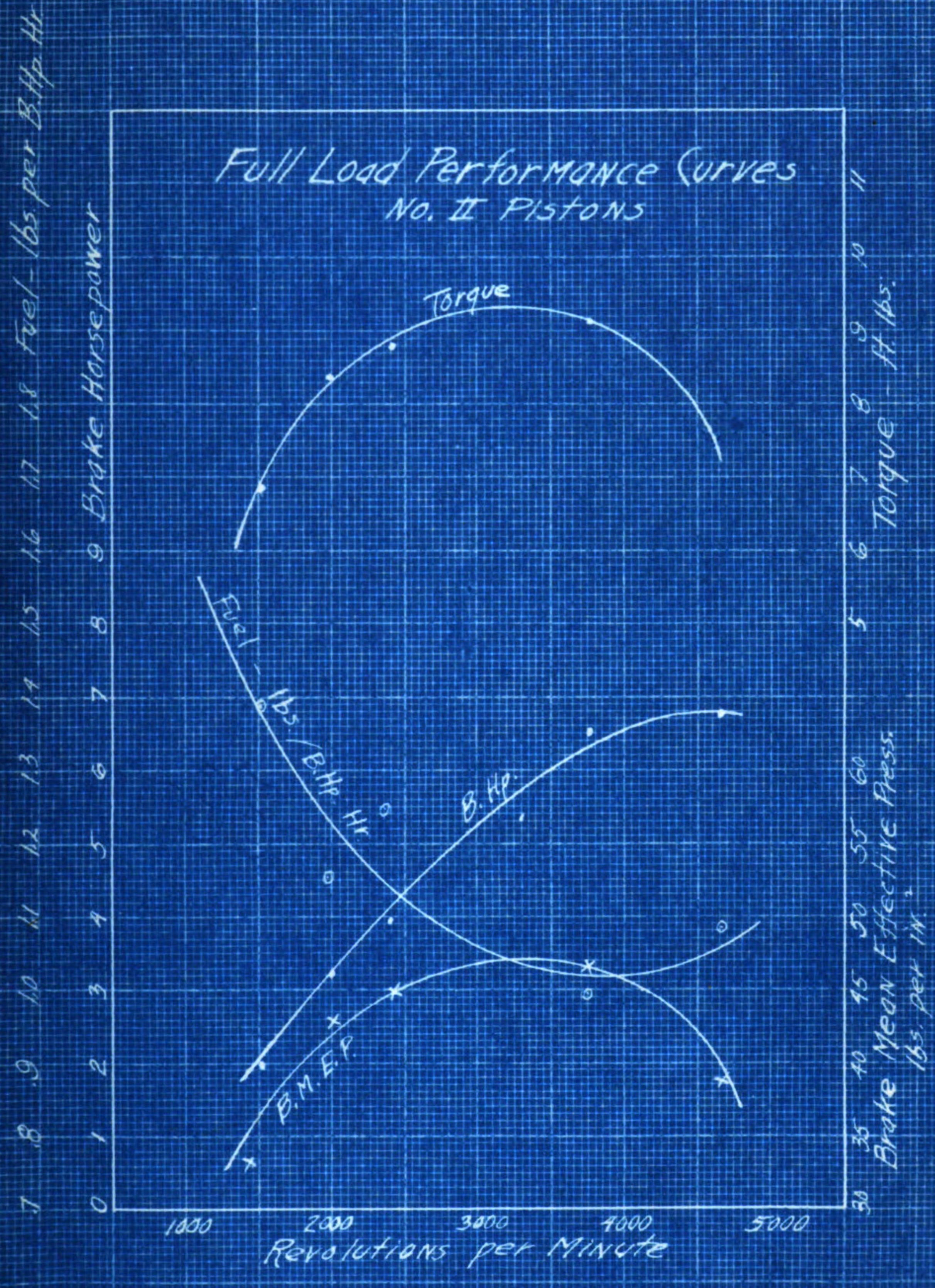
*Automobile and Aircraft Engines - Judge impulses are twice as frequent.

It was necessary to combine the exhaust and cooling water losses in the same curve because of the fact that with the set-up used any variation of the two losses could be obtained merely by changing the amount of water flowing. This combined loss increased with the speed, as would be expected, because of the increase in temperatures. The curve showing the radiation loss is not consistent with any reasoning which might be applied; it is included only to complete the balance.

The performance curves of No. 1 pistons are shown on Page 15. The maximum torque of 10 ft. lb. was developed between 3000 and 3500 R. P. M. The corresponding brake mean effective pressure was 50.5 lb. per sq. in. The maximum b. hp. developed was 8.6 at 4700 R. P. M.; at higher speeds the power fell off slightly. The fuel consumption was very high at low speeds, this being accounted for in part by the fact that a considerable amount of gasoline was sprayed from the carburetor intake. The minimum fuel consumption, corresponding to the thermal efficiency of 16.5 per cent, was .89 lb. per b. hp. hr.

NO. 2 Pistons

The results of the test on No. 2 pistons are shown on Page 18. This pair gave a ratio of compression of 8.0:1, almost double the ratio of the first pair. (See Fig. 3)



Ratio of Comp. = 8.0

Figure 3

It was found that these pistons had no advantages over the originals. The fuel consumption was considerably higher, never becoming less than 1-lb. per b.hp. hr. The torque and brake m.e.p. were lower, indicating either poor combustion or poor volumetric efficiency. It is probable that the very steep slope of the exhaust baffle caused too great a restriction to the flow of the exhaust gas. The absence of detonation when using ordinary fuel with the high compression ratio of this pair indicates that there was a great deal of mixing of the exhaust gas with the charge. The maximum torque and brake m.e.p. came at a higher speed than with No. 1 pistons which is characteristic of high compression engines. The ease of starting

did not seem to be materially affected by this increase in compression ratio, although a slightly increased pull was necessary due to the increase in compression pressure. The idling speed was from 50 to 100 R. P. M. higher than with No. 1 pistons. The results of this test indicate that the unfavorable shape of the piston head more than offset any increase in power and economy, which should have been obtained with the higher compression ratio.

No. 3 Pistons

For No. 3 pistons(Fig. 4) the inlet baffle was left as it was on No. 2; but the slope of the exhaust baffle was decreased, giving a compression ratio of 6.58:1. The performance, as shown on Page 21, was somewhat improved over No. 2 set, the maximum H. P. being increased by 5.9 per cent although it was still below that developed by No. 1 set. Inasmuch as this set(No. 3) was identical with No. 5 (which gave excellent results as to power and economy) except for the shape of the inlet baffle, it may safely be assumed that this feature was at fault. This is even more apparent in view of the fact that other things being equal, No. 3 set should have given better results because of its higher compression ratio. It is probable that the restricted inlet area caused excessive turbulence and mixing of the exhaust and fresh charge with consequent delayed burning and loss of power.

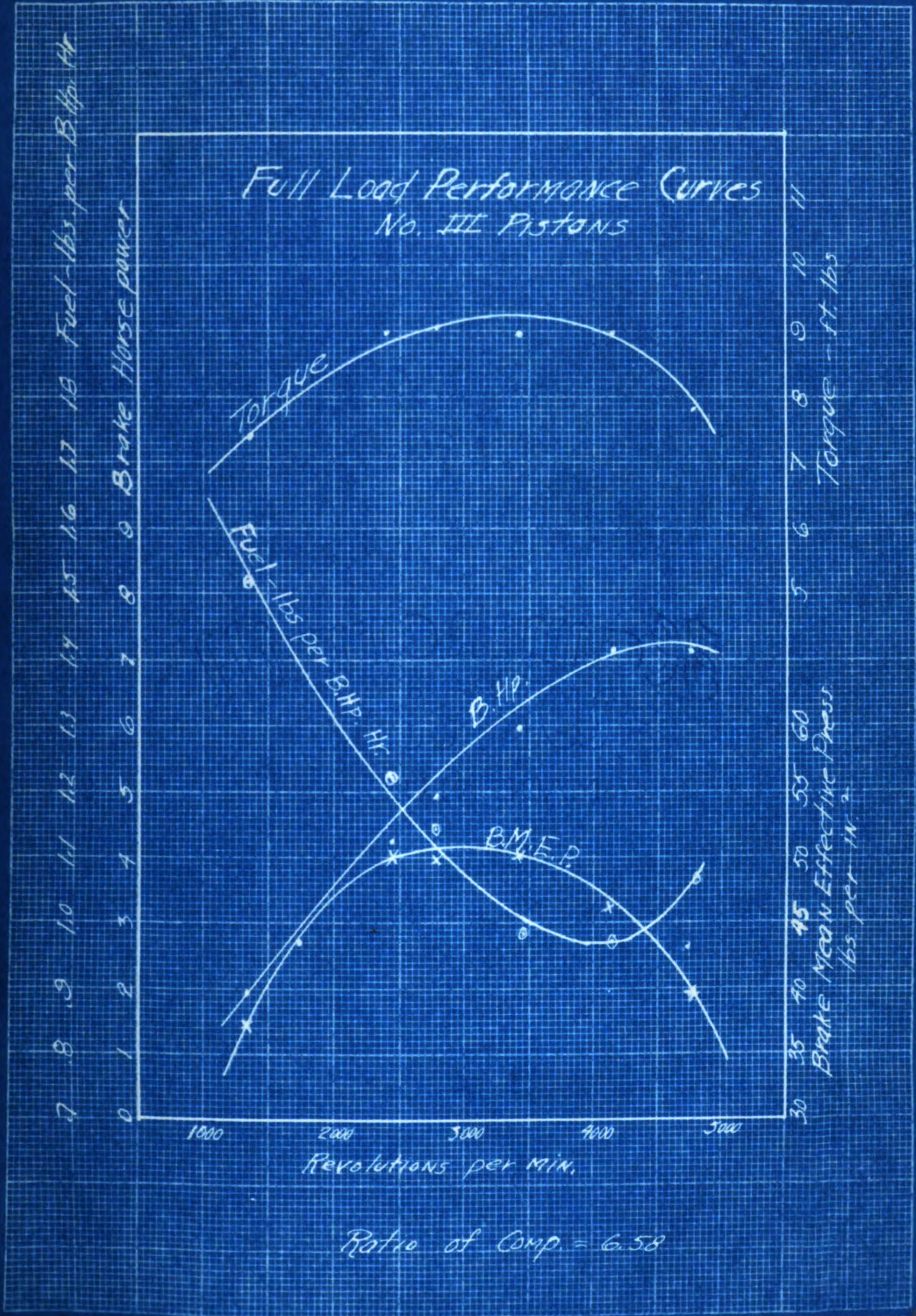
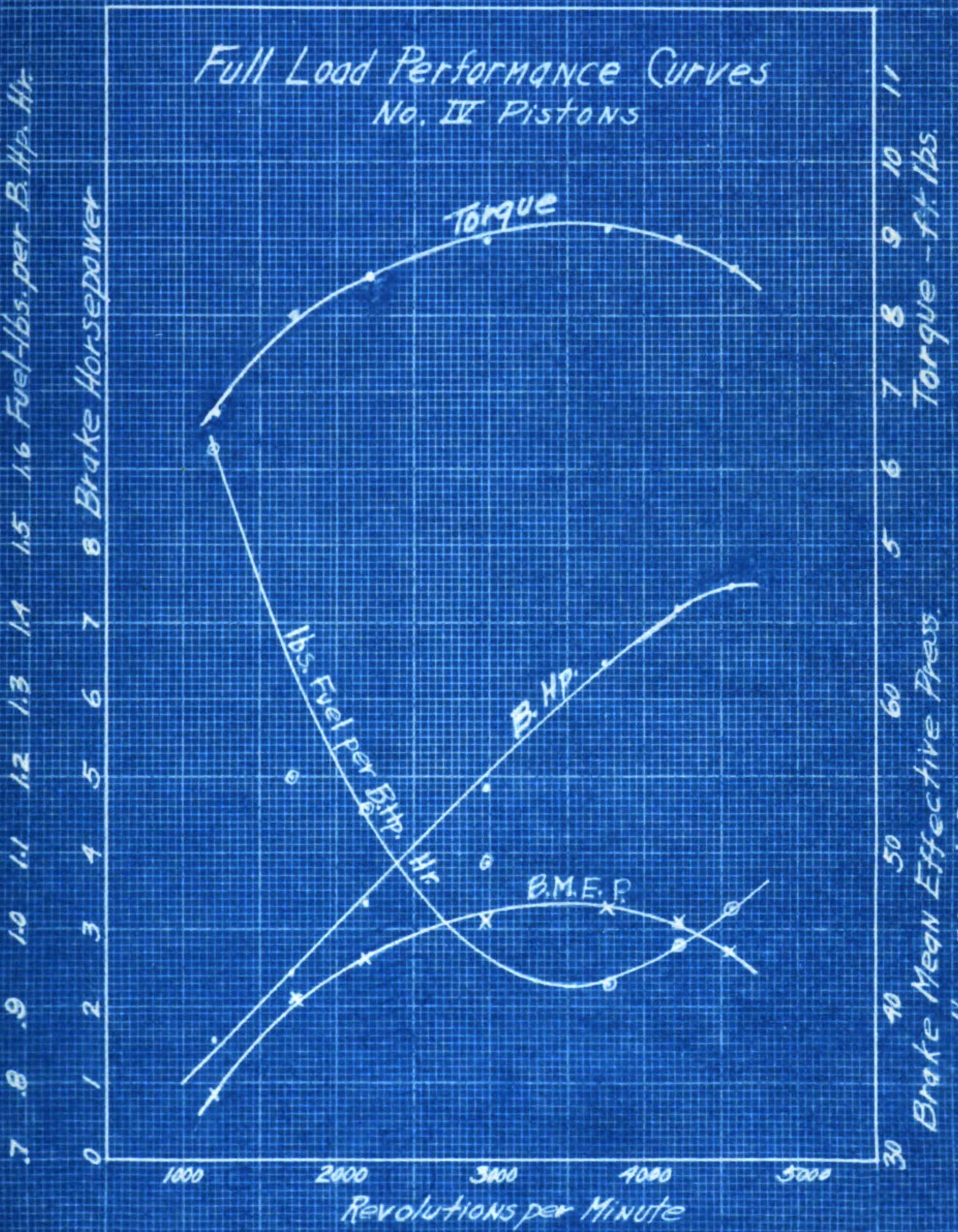

Full Load Performance Curves
No. III Pistons
Torque
Fuel-lbs per B.Hp. Hr.
B.Hp.
B.M.E.P.
Fuel-lbs. per B.Hp. Hr.
.7 .8 .9 1.0 1.1 1.2 1.3 1.4 1.5 1.6 1.7 1.8
Brake Horse power
0 1 2 3 4 5 6 7 8 9
Torque - ft. lbs
5 6 7 8 9 10 11
Brake Mean Effective Press
lbs per in.2
30 35 40 45 50 55 60
1000 2000 3000 4000 5000
Revolutions per min.
Ratio of Comp. = 6.58

Figure 4

No. 4 Pistons

The next set of pistons(Fig. 5) was designed with the intention of correcting the difficulties of the previous set. As is shown in the blue print, the inlet baffle was undercut a small amount, decreasing the compression ratio to 5.9:1. It was thought that the undercut baffle would direct the fresh charge up into the top of the cylinder away from the exhaust port, causing less loss of charge and less mixing of the gases. It was also expected that the slow speed running would be improved by having a fresher(unmixed) charge around the spark plug; however, the test showed no appreciable difference.



Ratio of Comp. = 5.90

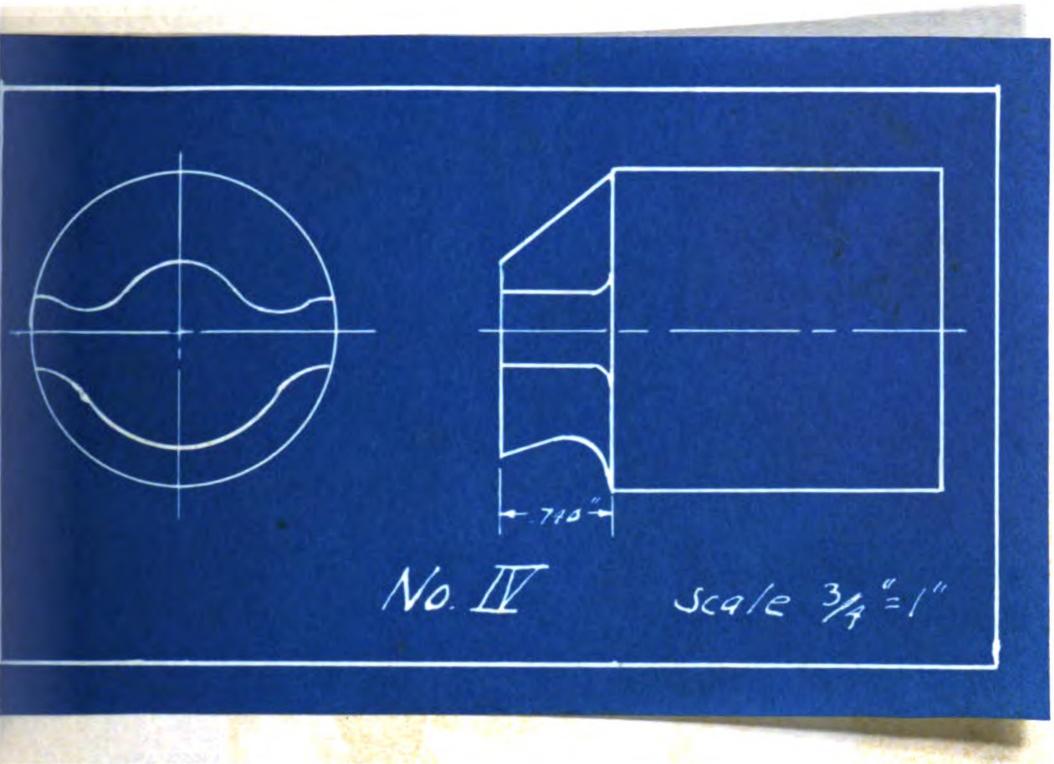
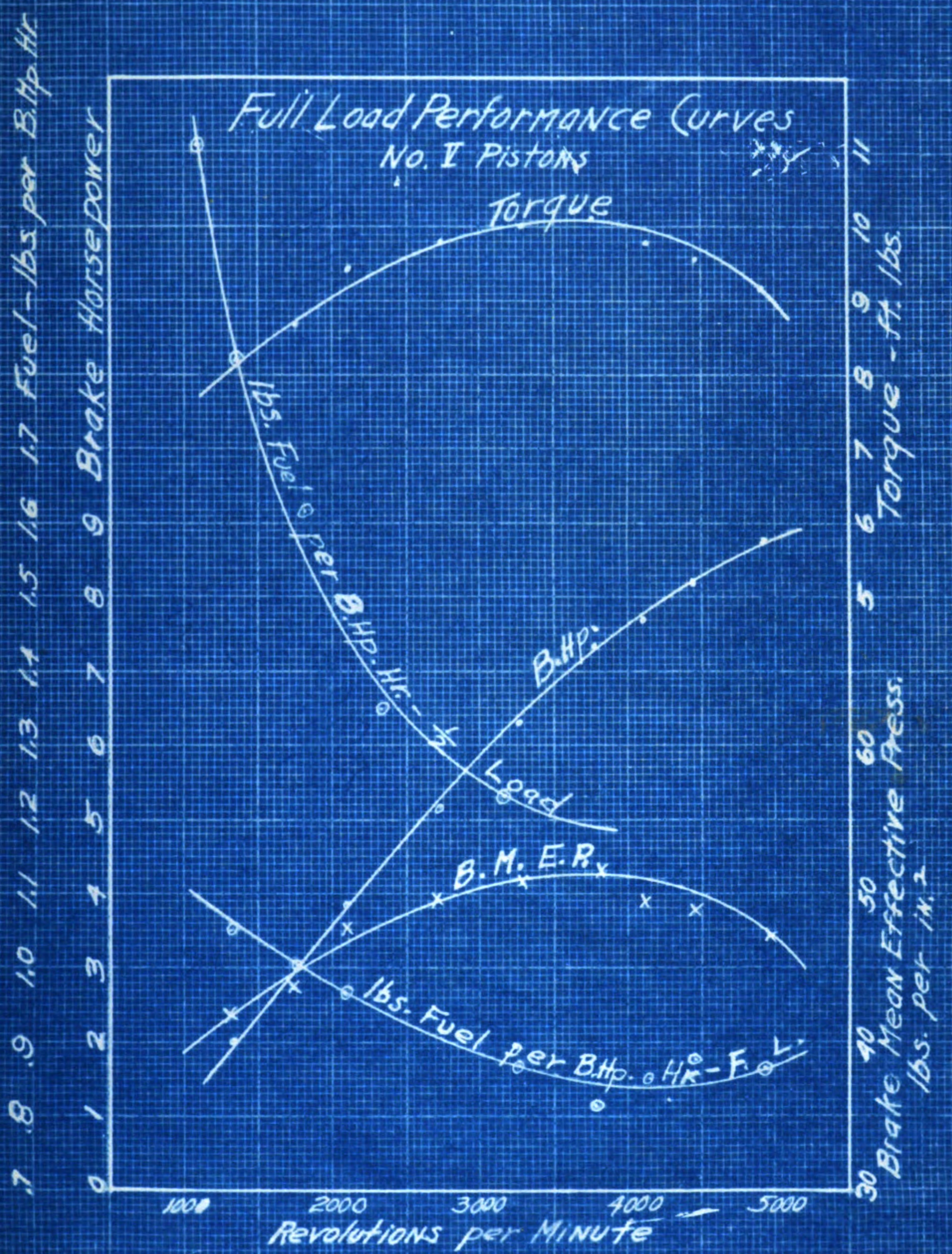


Figure 5

The b.hp. curve(Page 23) of this set is almost identical with that of No. 3 set over the greater part of the range. The economy was considerably improved in the speed range of 1500 to 3500 R. P. M. due probably to a smaller loss of charge in the exhaust. Unfortunately, no exhaust gas analysis was taken during this test, so the last-mentioned point cannot be definitely proved.

<u>No. 5 Pistons</u>

The best results as to power and economy were obtained with No. 5 pistons. These were identical with set No. 3 excepting that the inlet baffle was cut deeper, giving a



Ratio of Comp. = 5.58

larger passageway for the charge(see Fig. 6). The compression ratio of this set was 5.58:1.

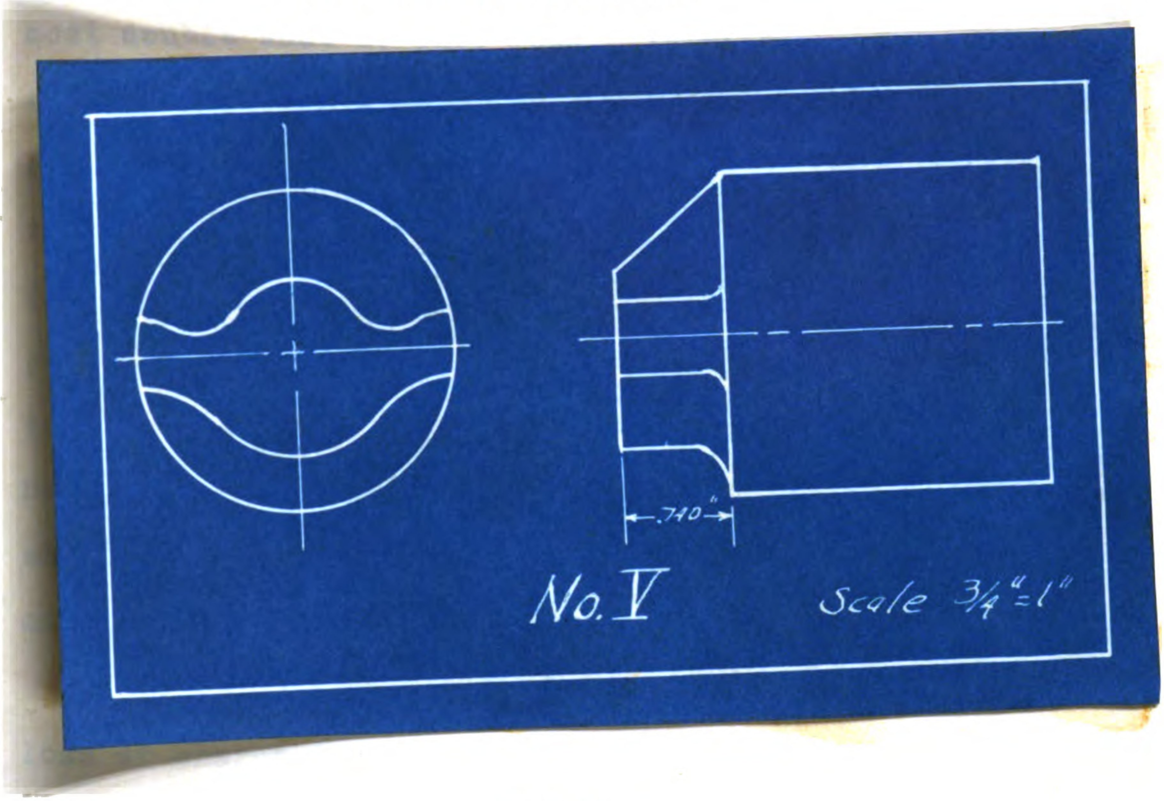


Figure 6

At 3400 R. P. M. these pistons developed 3.9 per cent more power than set No. 1. The most noticeable improvement was shown in the fuel consumption which showed a decrease of 7.0 per cent at the same speed. At lower speeds there was an even greater increase in fuel economy. The brake m.e.p. and the maximum torque were also slightly higher with No. 5 pistons and both occurred at a higher speed, as would be expected with the higher compression ratio.

A half load run was taken with these pistons, and it is interesting to note the extremely high fuel consumption.

At 3500 R. P. M. the consumption was 29 per cent greater; and at lower speeds it is still higher, increasing to almost double that at full load. It will be noticed that the half load run was discontinued at 3200 R. P. M. (see Page 25). The bearings became too hot at speeds higher than this because of the fact that the lubricating oil (mixed with the fuel) was cut down when the throttle was partially closed, while the lubrication demand remained the same as though the throttle were wide open. This is not to be considered as a criticism of the engine, because when in actual service with wide open throttle the motor only turns up about 3400 R. P. M.

From the exhaust gas analysis it was found that the loss of charge through the exhaust port was very nearly the same with both No. 1 and No. 5 sets of pistons. Therefore the increase in economy shown by No. 5 pistons must have been due solely to the higher compression ratio. Providing all other conditions are kept constant, the thermal efficiency increases with the ratio of compression. For an engine working on the Otto cycle, the efficiency can be expressed as $1 - (1/r)^{\gamma - 1}$ where γ is the ratio of the specific heats at constant pressure and at constant volume, respectively, and r is the ratio of compression. Evidently the efficiency increases as r increases. Another effect of a higher compression ratio is a decrease in the time of explosion, or the period of maximum temperature, during which most of the heat loss occurs.

<u>No. 6 Pistons</u>

The object of this test was to ascertain the effect of using a higher compression ratio with the same height of head as No. 1 set. The higher ratio (4.96) was obtained by giving the exhaust baffle a steeper slope.

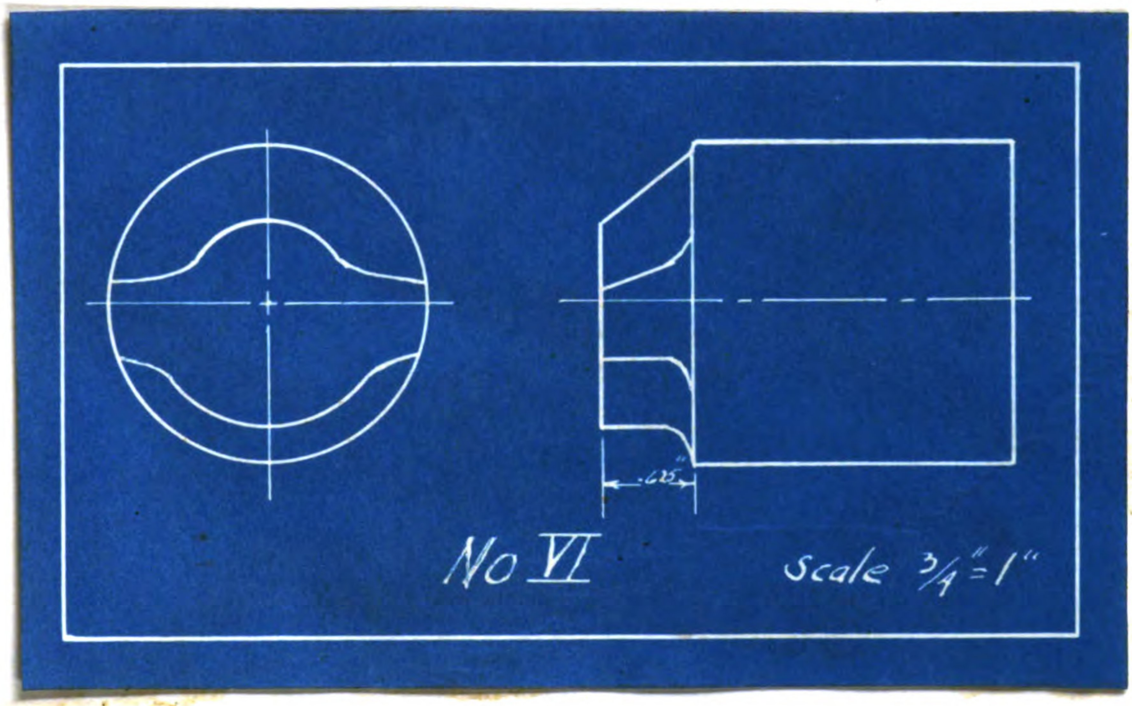
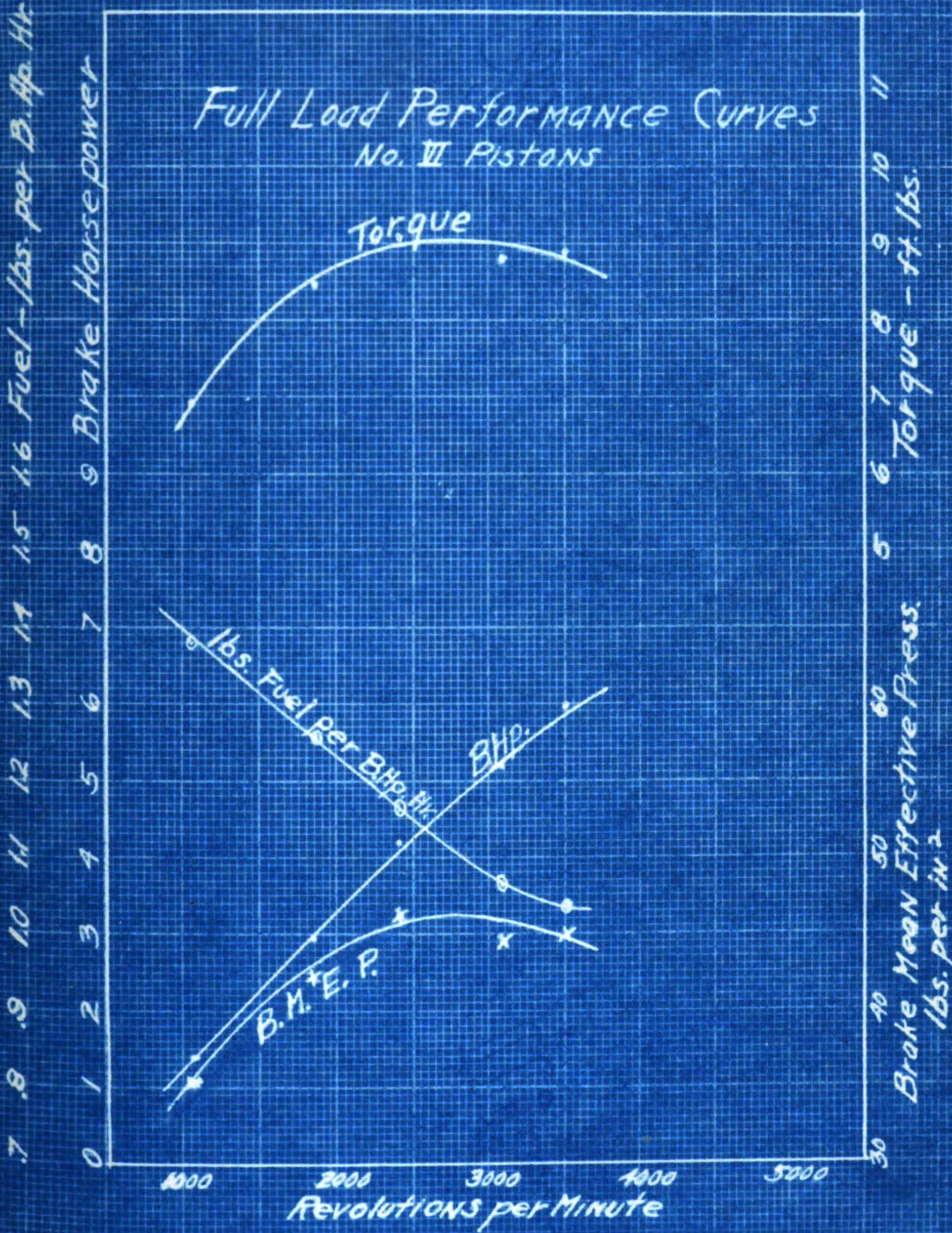


Figure 7

It was thought that perhaps the small clearance between the top of the cylinder and the heads of the higher pistons might have interfered with the flame propagation. The poor results obtained with this set tend to discredit this theory. The brake m.e.p., torque, and b.hp. were all very low; and



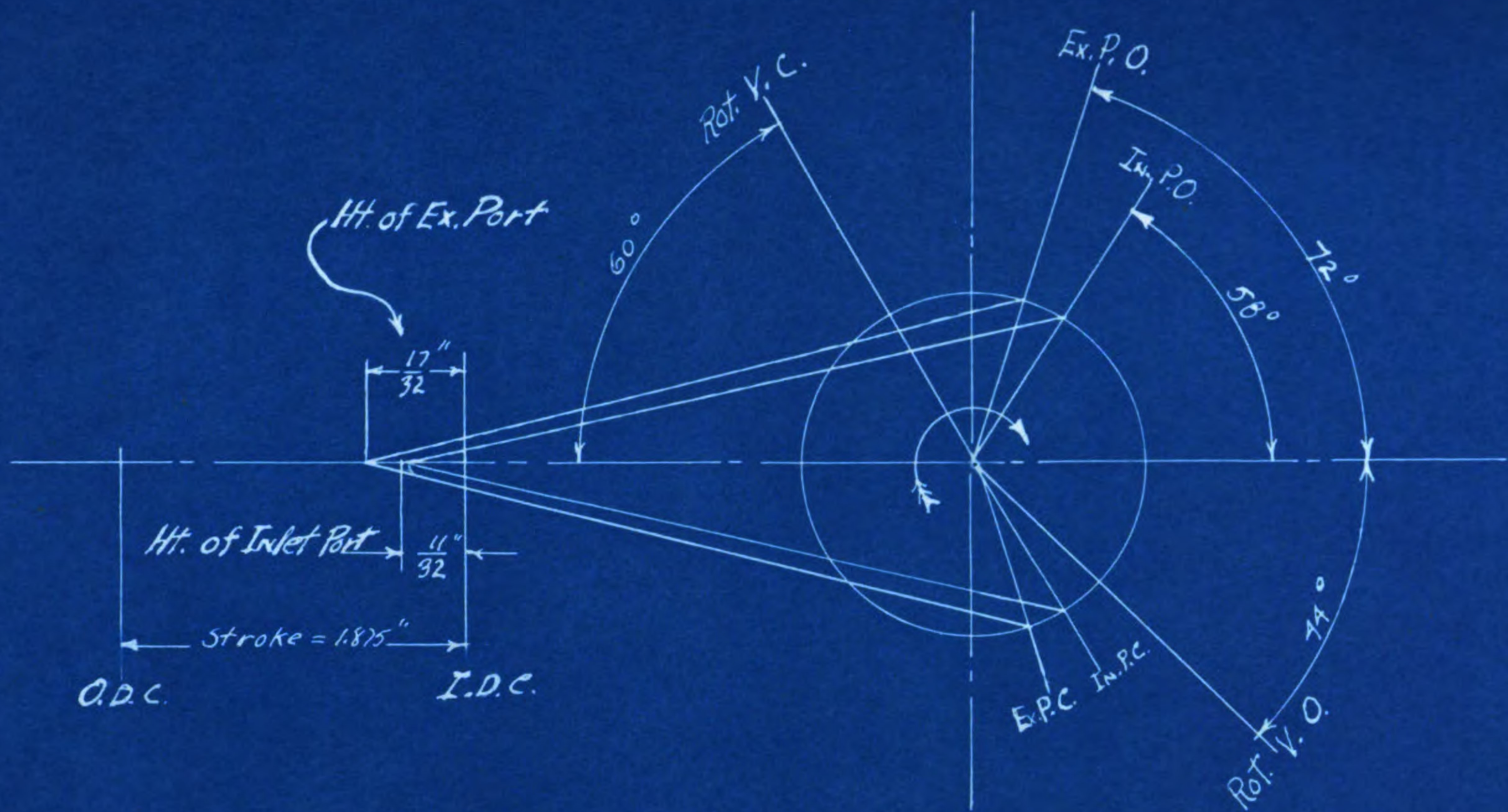
Ratio of Comp. = 4.96

the fuel consumption was high.

General

The comparative test of pistons No. 5 and 1, made with the engine installed on a row boat, verified the increase in power developed by No. 5 set. No. 1 set, when tested with wide open throttle, gave a maximum speed of 3170 R. P. M. No. 5 pistons were then substituted, and the R. P. M. was raised to 3330, an increase of 5 per cent. There was no noticeable difference in the starting qualities; and the idling speeds seemed to be very nearly the same.

The diagram showing the valve and port timing is shown on Page 31. It may be seen that the rotary valve opens 44 degrees after inner dead center; and the transfer port is not covered until 58 degrees after inner dead center. Therefore, during this 14 degree interval, there is no closed valve between the cylinder and the carburetor. After the pistons leave inner dead center, the volume inside the crankcase starts to increase, tending to make the pressure decrease. The pressure in the crankcase is also decreasing as the charge is being injected into the cylinder. Therefore, the ideal time for the rotary valve to open would be when the pressure in the crankcase reaches atmospheric. If the rotary valve opens late, part of the charge may be drawn back into the crankcase from the cylinder. If it opens too early, the pressure in the crankcase



Valve and Port Timing Diagram

Ratio $\frac{L}{R} = 3.92$

will be released and the maximum charge will not be injected into the cylinder. Thus, whether the valve opens too early or too late, the volumetric efficiency will be decreased; and a loss of power will result. The transfer port should close when the pressure in the cylinder is balanced by the pressure in the crankcase, plus the inertia effect of the charge in the transfer passage. The timing of the rotary valve and the ports must be carefully considered if the best performance is to be obtained.

CONCLUSION

1. In order to obtain maximum power with low compression ratios, the spark must be more advanced than for higher ratios. This is explained by the fact that the charge(with low ratios) burns more slowly.

2. The results of this test indicate that if the loss of unburned charge in the exhaust could be eliminated, the economy would compare favorably with that of four-cycle engines.

3. It is possible to use a higher ratio of compression without detonation with two-cycle engines than with four-cycle engines, because of the mixing of the charge with the exhaust gas.

4. Evidently the shape of the clearance space, or the shape of the piston head is a more important factor

than the compression ratio in determining engine performance. This is evidenced by the fact that pistons No. 2, 3, and 4 all gave higher ratios than No. 5 set; but, because of poorly designed baffles, all gave poorer results than No. 5 set. It would seem that the proper method of selecting a baffle shape would be to change the shape without changing the ratio until the best results were obtained.

5. The ideal method of increasing the compression ratio would be to form the cylinder head close to the exhaust baffle and leave most of the clearance volume around the inlet baffle. In this way the spark plug would be in the center of the compressed charge, and the flame travel would be at a minimum.

6. The ideal time for the rotary valve to open is when the pressure in the crankcase reaches atmospheric. The transfer port should close when the pressure in the cylinder is balanced by the pressure in the crankcase, plus the inertia effect of the charge in the transfer passage.

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Page 461.

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Chapter 7.

RESULTS OF HEAT BALANCE

(WITH NO. 1 PISTONS)

HEAT DISTRIBUTION

(B.T.U. / b.hp.hr.)

R.P.M.	B.HP.	Friction HP.	Exh. and Cooling H_2O	Radiation	Total
1398	2547	720	15,807	8,585	27,692
1920	2547	590	13,830	4,583	21,550
2270	2547	805			
3670	2547	720	11,980	1,883	17,130
4450	2547	827	11,240	2,316	16,930
4710	2547	858	13,380	1,995	18,780

HEAT DISTRIBUTION (%)

1398	9.2	2.7	57.1	31.0	100.0
1920	11.8	2.7	64.2	21.3	100.0
2270	13.2	4.2			
3670	14.9	4.2	69.0	11.9	100.0
4450	15.0	4.9	66.4	13.7	100.0
4710	13.6	4.6	71.2	10.6	100.0

RESULTS OF FULL LOAD RUN

WITH ORIGINAL PISTONS

(Comp. Ratio = 5.22:1)

R.P.M.	B.HP.	Fuel Consumption (lbs./b.hp.hr.)	B.M.E.P. (lbs./sq.in.)	Torque (ft.lbs.)
1398	2.03	1.455	38.6	7.64
1920	3.44	1.133	47.6	9.40
3020	5.62	1.108	50.3	9.85
3670	6.95	.902	50.5	9.95
4450	8.13	.892	48.8	9.65
4710	8.60	.978	48.4	9.55

RESULTS OF HALF LOAD RUN

WITH ORIGINAL PISTONS

R.P.M.	B.HP.	Torque (ft.lbs.)	Fuel (lbs./b.hp.hr.)	B.M.E.P. (lbs./sq.in.)
933	1.05	5.9	2.100	28.4
1245	1.52	6.4	1.560	32.4
1645	2.28	7.3	1.326	36.8
2540	3.44	7.1	1.343	36.0

RESULTS OF FULL LOAD RUN

WITH NO. 2 PISTONS

(Comp. Ratio = 7.95)

R.P.M.	B.HP.	Fuel Consumption (lbs./b.hp.hr.)	B.M.E.P. (lbs./sq.in.)	Torque (ft.lbs.)
1513	1.97	1.385	34.8	6.83
1980	3.18	1.145	42.8	8.40
2360	3.89	1.245	43.8	8.78
3750	6.52	.990	46.2	9.10
4640	6.74	1.085	38.5	7.60

RESULTS OF FULL LOAD RUN

WITH NO. 3 PISTONS

(Comp. Ratio = 6.51)

R.P.M.	B.HP.	Fuel Consumption (lbs./b.hp.hr.)	B.M.E.P. (lbs./sq.in.)	Torque (ft.Lbs.)
1350	1.88	1.510	37.0	7.30
2430	4.13	1.215	49.8	8.93
2805	4.80	1.134	49.0	8.98
3455	5.83	.971	49.8	8.88
4160	7.04	.962	45.8	8.89
4765	7.02	1.057	39.4	7.74

RESULTS OF FULL LOAD RUN

WITH NO. 4 PISTONS

(Comp. Ratio = 5.82)

R.P.M.	B.HP.	Fuel Consumption (lbs./b.hp.hr.)	B.M.E.P. (lbs./sq.in.)	Torque (ft.lbs.)
1180	1.52	1.627	34.2	6.75
1700	2.42	1.200	40.5	8.00
2175	3.37	1.160	43.0	8.50
2990	4.86	1.085	45.5	9.00
3750	6.48	.923	46.4	9.10
4230	7.20	.980	45.6	9.00
4565	7.50	1.029	43.7	8.60
4700	7.82	.970	44.0	8.65

RESULTS OF FULL LOAD RUN

WITH NO. 5 PISTONS

(Comp. Ratio = 5.58)

R.P.M.	B.HP.	Fuel Consumption (lbs./b.hp.hr.)	B.M.E.P. (lbs./sq.in.)	Torque (ft.lbs.)
1340	2.10	1.050	42.0	8.28
1740	2.89	1.008	44.0	8.70
2105	3.80	.978	48.0	9.50
2760	5.14	.980	49.8	9.80
3295	6.30	.875	51.0	10.00
3815	7.40	.814	51.8	10.20
4140	7.70	.855	49.5	9.80
4475	8.21	.884	49.0	9.60
4970	8.80	.863	47.2	9.25

RESULTS OF HALF LOAD RUN

WITH NO. 5 PISTONS

R.P.M.	B.HP.	Fuel Consumption (lbs./b.hp.hr.)	Torque (ft.lbs.)	B.M.E.P. (lbs./sq.in.)
1046	.56	2.090	4.10	20.8
1765	1.23	1.625	5.35	27.0
2372	1.94	1.345	6.28	31.7
3135	2.61	1.227	6.40	32.5

RESULTS OF FULL LOAD RUN

WITH NO. 6 PISTONS

(Comp. Ratio = 4.9)

R.P.M.	B.HP.	Fuel Consumption (lbs./b.hp.hr.)	B.M.E.P. (lbs./sq.in.)	Torque (ft.lbs.)
1046	1.38	1.380	35.1	6.90
1846	2.98	1.255	42.8	8.45
2383	4.15	1.170	46.3	9.10
3060	5.13	1.060	44.6	8.80
3500	5.98	1.033	45.0	8.90

COMPRESSION RATIOS

Piston No.	Average Clearance Vol. (Cu. in.)	Compression Ratio
1	1.66	4.22
2	.76	8.00
3	.96	6.58
4	1.10	5.90
5	1.17	5.58
6	1.36	4.96

Displacement = 5.37 cu. in.

RESULTS OF FRICTION TEST

R.P.M.	Friction HP.
1140	.48
1330	.58
1920	.90
2450	1.21
3040	1.55
3720	2.04
4650	2.88
4960	3.33

ENGINES CONSTANTS

Rotary Valve Opens 44 degrees after Inner D. C.

Rotary Valve Closes 60 degrees after Outer D. C.

Exhaust Ports Uncovered 72 degrees before Inner D. C.

Transfer Ports Covered 58 degrees after Inner D. C.

Exhaust Ports(4) .59 in. high x .50 in. wide.

Transfer Ports(4) .41 in. high x .50 in. wide.

Length of Stroke - 1.875 in.

Diameter of Cylinder Bore - 2.252 in.

Diameter of Piston(skirt) - 2.246 in.

Diameter of Piston(above ring grooves) - 2.241 in.

Length of Connecting Rod - 3.69 in.

SAMPLE COMPUTATIONS

Note: All sample computations are for the first run with No. 1 pistons. (1398 R.P.M.)

Horsepower

$$HP = \frac{2 \pi R W r}{33000} \times C.F.$$

$$= K \times WN \times C.F.$$

Where : K = dyn. constant = $\frac{1}{6000}$

$W = \frac{\text{net scale load}}{1.46}$

(1.46 is gear ratio, dyn: motor)

N = Motor R.P.M.

$$CF = \sqrt{\frac{T_o}{T_s}} \quad \frac{P_s}{P_o}$$

Where: T_o = 460 + room temp.

T_s = 520

P_s = 29.92

P_o = Barom. - Vapor Pressure)

$$CF = \sqrt{\frac{523}{520}} \times \frac{29.92}{28.72 - .15} = 1.05$$

$$HP = \frac{8.32 \times 1398 \times 1.05}{6000} = 2.03$$

Lbs. Fuel / b.hp.hr.

Gas is 62.0 Bé or 6.07 lb. per gal.

1 gal. = 231 cu. in.

1 cu. in. weighs $\frac{6.07}{2.31}$ = .0263 lb.

105 cc. weighs $\frac{105 \times .0263}{16.39}$ = .1685 lb.

In 105 cc. of mixture there are $\frac{8 \times 105}{9}$ = 93.4 cc. gas

105 cc. mixture contains .1498 lb. gas.

lbs. per hr. = $\frac{.1498 \times 3600}{\text{Time in Sec.}} = \frac{539}{T}$

time = 184.75 sec.

lbs. per hr. = $\frac{539}{184.75}$ = 2.915

lbs. per b.hp.hr. = $\frac{2.915}{2.03}$ = 1.455

TORQUE

$$T = \frac{63{,}000 \times HP}{R.P.M. \times 12} = \frac{63{,}000 \times 2.07}{1398 \times 12} = 7.64 \text{ ft. lbs.}$$

BRAKE MEAN EFFECTIVE PRESS.

$$\frac{P\ L\ A\ N.}{33{,}000} = \frac{2\,\pi\,W\,R\,N_2}{33{,}000}$$

Where: $L \times A = \frac{\text{Disp.}}{12}$

N_1 = Working strokes per min. per cyl.

$W \times R$ = Torque

N_2 = R.P.M.

$$P = \frac{2\pi T \times 12}{\text{Disp.}} \times \frac{\cancel{N}_2}{\cancel{N}_1}$$

$$P = 24\pi \times \frac{T}{D}$$

$$D = 2 \times \frac{\pi}{4} (2.25)^2 \times 1.875 = 13.88 \text{ cu. in.}$$

$$P = \frac{75.4}{13.88} \times 7.64 = 38.6 \text{ lb. per sq. in.}$$

HEAT BALANCE

Lbs. fuel per b.hp.hr. = 1.455

Air - fuel ratio = 17.4:1

Lbs. dry air per b.hp.hr. = 17.4 x 1.455 = 25.3

Total wt. of dry charge = 25.3 + 1.455 = 26.755

Moisture in charge(from psychrometric chart)

$$= \frac{18 \times 25.3}{7000} = .065$$

Total charge = 26.755 - .065 = 26.82 lbs.

Wt. moisture in exhaust from combustion

= 1.44 x 1.455 = 2.09

Wt. dry exh. gas = 26.755 - 2.09 = 24.665 lbs.

$$\therefore \frac{24.665}{26.82} = 92\% = \text{dryness of exh. (by wt.)}$$

Analysis of dry exh. (by wt.)

% Vol.		% Wt.
CO_2	6.85 x 44 = 301.0	10.25
O_2	6.75 x 32 = 216.0	7.36
CO	4.3 x 28 = 120.0	4.08
N_2	82.1 x 28 = 2300.0	78.30
	2937.0	99.99

B.T.U. / mol. from 0 degrees to T degrees F. (abs.)

From Polson: (for CO_2)

$$\text{B.T.U.} = T(5.165 + 1.95 \times 10^{-3}T - .2 \times 10^{-6}\, T^2)$$

$$= T(5.165 + 1.95 \times .788 - .2 \times 10^{-6} \times 622000)$$

$$= T(6.579)$$

(1) Avge. for exh. temp. = T(6.82)

(2) Similarly an average value was found for the inlet temps.

The difference (1) - (2) gives the B.T.U. per mol. over the range of temps.(exh. and inlet) substituted in the formulae.

Analysis of wet exh. gas(by wt.)		Heat / lb. over range B.T.U.	Heat / lb. (wet exh.)
H_2O	.080	94.5	7.56
CO_2	.0944	49.8	4.71
O_2	.0677	41.7	2.82
CO	.0376	47.5	1.79
N_2	.7210	47.5	34.20
	1.0007		51.08

Heat / b.hp.hr. to exh. from wet charge

$= 51.08 \times 26.82 \qquad = 1370$ B.T.U.

A.P.I. of oil at 69°F. = 28.5

A.P.I. corrected = 27.9

$$\text{Spec. gr.} = \frac{141.5}{131.5 + 27.9} = .899$$

Wt. of gal. of oil = 7.50 lbs.

Wt. of gal. of gas = 6.07 lbs.

$$\text{Ratio } \frac{\text{oil}}{\text{gas}} \text{ (by wt.)} = \frac{1}{8} \times \frac{7.5}{6.07} = .155$$

Heat loss to oil / b.hp.hr. = $.155 \times 1.455 \times 133$

= 27.8 B.T.U.

Loss of fuel through exh. port.

N_2 in exh. = 82.1 - 3 = 79.1% (by vol.)

(CH_4 & H_2 assumed to be 3%)

O_2 in fresh charge = $79.1 \times .266 = 21.1\%$

O_2 in exh. gas = 6.75%

$$\therefore \frac{\text{Vol. of charge escaping}}{\text{Tot. vol. of charge}} = \frac{6.75}{21.1} = 32.0\%$$

$.32 \times 27{,}692 = 8{,}850$ B.T.U./ b.hp.hr.

Heat loss to cooling water

$Q = (T_2 - T.) \times \text{wt.} (50 \text{ min.})$

$= (144 - 55.5)\ 106.5 = 9{,}430$ B.T.U.

Heat / hr. = $60 \times 9{,}430 = 11{,}300$ B.T.U.

$$\text{B.T.U. / b.hp.hr.} = \frac{11{,}300}{2.03} = 5{,}560$$

Total loss to exhaust and Cooling Water

$= 1370 + 27.8 + 8,850 + 5,560 = 15,807$ B.T.U./b.hp.hr.

COMPRESSION RATIO

Disp. $= \frac{\pi}{4}(2.25)^2 \times 1.33 = 5.37$ cu. in.

1.33 = stroke above exh. port.

Clear. Vol. = 1.66 cu. in.

$$\text{Ratio of Comp.} = \frac{\text{Disp. \& Cl. Vol.}}{\text{Cl. Vol.}}$$

$$= \frac{5.37 - 1.66}{1.66}$$

$$= 4.22$$

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