THE PERFORMANCE OF A DIESEL ENGINE USING BENZOL AS A FUEL

by

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in Mechanical Engineering
from the
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Acceptance:

Instructor in Charge of Thesis.

Cambridge, Massachusetts

June, 1945

Professor George W. Swett

Secretary of the Faculty

Massachusetts Institute of Technology

Dear Sir:

In partial fulfillment of the requirements for the degree of Bachelor of Science in Mechanical Engineering, we hereby submit a thesis entitled "The Performance of a Diesel Engine Using Benzol as a Fuel."

Respectfully,

ACKNOWLEDGEMENTS

The authors wish to express their appreciation to Professor A. R. Rogowski for suggesting the problem and lending his guidance throughout the work, and to Messrs. Ku, Kano, Fardy, and particularly to Mr. Livengood of Sloane Laboratories for their valuable assistance and suggestions.

SYMBOLS

BMEP: Brake mean effective pressure, lb. per sq. in.

BSFC: Brake specific fuel consumption, lb. per brake horsepower hour

BHP: Brake horsepower

ISFC: Indicated specific fuel consumption, lb. per indicated horsepower hour

IHP: Indicated horsepower

F/A: Fuel-air ratio

P_i : Inlet pressure

P_e : Exhaust pressure

T_i : Inlet temperature

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INTRODUCTION

Compression ignition engines require fuel pumps so timed as to inject the solid fuel at a time when the temperature and pressure conditions in the cylinder are such that the fuel will immediately begin to burn. These pumps must deliver the fuel to the injection nozzle at a pressure considerably higher than that in the cylinder at the time of injection. Considering the fact that compression ignition engines use extremely high compression ratios, it may be seen that this pressure is high. Hence, such a pump, as well as being subject to the mechanical difficulties found in any high pressure pumps, must consume a considerable amount of power.

A compression ignition engine having the fuel delivered carburated through the intake would be subject to none of the difficulties or power losses attributed to the high pressure fuel pumps. By using benzol as a fuel it was hoped that some indication of the performance of an engine under such conditions could be obtained.

As far as could be ascertained, it is believed that no previous work has been done in this field. Diesel engines have been used which, after starting on liquid diesel oil, have continued to run on a mixture of injected diesel fuel and carburated sewer gas; but it has been found that these engines will not operate on the carburated sewer gas alone, but must have some diesel fuel injected to maintain combustion.

Benzol was chosen as the fuel to be used in this investigation, since it has a characteristically short delay period; it is comparatively volatile; and its compression-ignition characteristics are within the range of the engine used.

PURPOSE OF THE INVESTIGATION

The purpose of this investigation is to get some indication of the performance of a diesel engine using a carburated fuel, benzol, induced through the intake valve. A secondary purpose is to determine some of the mechanical difficulties that must be overcome in the engine for it to operate satisfactorily under the imposed conditions.

The performance is to be indicated by the following observations:

(1) The effect upon the brake horsepower and the brake specific fuel consumption by varying the fuel-air ratio, all other conditions remaining constant.

(2) The effect of throttling at various fuel-air ratios, all other conditions remaining constant.

(3) The high and low fuel-air ratio limits at various conditions for the fuel used.

(4) The effect of fuel-air ratio, and speed, upon the knock intensity. (obtained by ear)

APPARATUS

A. Engine

The compression-ignition engine used was a single cylinder engine with a Comet head. It was converted from a C.F.R. engine, and had a fixed compression ratio of 13.5, 3.25 inch bore, 4.50 inch stroke, and 37.4 cubic inch displacement.

A 1/16 inch annealed copper head gasket was used to withstand the high pressures encountered. A soft fiber gasket around the copper was used to seal the water-jacket.

A neon-tube stroboscope was used with a tachometer for speed measurements. A disk with 36 equally spaced, radial lines was attached to the flywheel. With this devise the exact engine speed could readily be determined.

B. Fuel System

The benzol was led from an overhead 6 gallon tank through a rotometer and then into a two piston, alternately acting, Bosch fuel pump. From there it was injected into the intake manifold through a spray nozzel. The fuel pump was driven by a 1 1/2 horsepower constant speed motor. The rotometer was used to measure the fuel flow.

C. Air System

The air system was provided with two 53 gallon capacity surge tanks to smooth out the pulsations. The throttle valve was between the two surge tanks. The intake manifold was jacketed with both steam and water connections for inlet temperature control.

D. Engine Indicator

The M.I.T. high-speed engine indicator was used for taking the indicator diagrams. Due to the danger of high pressures in the indicator pressure cylinder, the diagrams were not always completed at high pressures.

E. Electric Dynamometer

A D.C. dynamometer was used for both starting and loading. The engine power was absorbed by the dynamometer with a beam scale for measuring the engine torque. The speed of the engine was varied by varying the field current in the dynamometer.

F. Calibration

1) Fuel System

The rotometer was calibrated by weighing the amount of fuel flow at a steady reading of the meter for a given time interval.

2) Air System

The air flow was calibrated by measuring the pressure difference across a standard 0.5 inch orifice.

G. Load Measurement

The load measurement constants were:

$$BMEP = 4.245 \times BL$$

$$BHP = \frac{BL \times rpm}{5000}$$

PROCEDURE

The usual steel and asbestos gaskets used in the converted C.F.R. diesel were found to have a very short life, often not over one hour of running time when benzol was being burned in the engine. The benzol was a solvent for the composition covering of the gasket, and after this was removed, the steel would burn, usually leaving a passage for the gases into the water-jacket. Hence, after some experimentation, a solid annealed copper gasket with a minimum surface area was found to withstand the cylinder temperatures and pressures. (cf. Fig. 1).

Since the temperatures in the cylinder would burn the diesel fuel injection nozzel, it had to be removed. This, then, necessitated the finding of a satisfactory method of starting combustion; for it was found that the engine would not fire using benzol unless some other fuel were added to start ignition. Ether introduced into the air intake would ignite the benzol, but severe knocking resulted, and the accompanying high pressures made this method inadvisable. Finally it was found that diesel oil injected into the intake manifold near the intake valve would accomplish the purpose.

Due to the fact that it was difficult to start the engine and much time was required for the starting diesel fuel to burn entirely out of the cylinder, the performance of the engine is expressed in terms of brake quantities rather than in indicated quantities, since the engine would have to be stopped to measure the engine friction after each run.

With these modifications a series of tests for engine performance were than carried out.

Series I Tables I,II,III

1. 1000 rpm.
   F/A = 0.090, 0.110, 0.130, and 0.150
2. 1100 rpm.
   F/A = 0.090, 0.110, 0.130, and 0.150
3. 1200 rpm.
   F/A = 0.090, 0.110, 0.130, and 0.150

Series II Tables IV, V, VI, VII

1. P_i = 27.9 in. Hg
   1100 rpm. F/A = 0.07, 0.08, 0.09, and 0.11
2. P_i = 25.9 in. Hg
   1100 rpm, F.A = 0.07, 0.08, 0.09, 0.10, and 0.11
3. P_i = 23.9 in. Hg
   1100 rpm, F/A = o.o7, 0.08, 0.09, and 0.10
4. P_i= 21.9 in. Hg
   1100 rpm, F/A = 0.06, 0.07, 0.07, 0.08, and 0.09

Series III Tables VIII, IX

This series was a repetition of Series I, omitting the 1200 rpm ruh.

Throughout all of these runs observations were made as to the knock intensity and the fuel-air ratio limits.

All runs were corrected in the usual manner for variations in barometric pressure.

Owing to the limited range of the equipment and to the unpredictability of the operation of the engine, these tests were restricted and could not be extended too far. Runs did not lend themselves to duplication, but nevertheless, the range covered by these tests gave sufficient information to be studied. General trends in output and consumption characteristics were distinctly illustrated.

After the data showing effect of inlet pressure (Series II) were tabulated, a definite difference in range of fuel-air ratio for this series and Series I led to further investigation of fuel-air ratio limits at atmospheric inlet pressure. Series III shows the results of this investigation. This showed a continual variation of the data in Series I. After 1000 and 1100 rpm runs were taken and the difference in operation was defined, the M.I.T. indicator was attached to the engine, and the cards showed a number of possible reasons for the variation. It was also noted that after operating at these fuel-air ratios for an hour or so, the engine would instantaneously revert back to approximately the original range of fuel-air ratios of operation. Observations of the indicator cards and the fact that the engine reverted back to the original fuel-air ratios lead to belief that the trouble possibly lay in either wornout piston rings or defective valve operation.

The engine was torn down and it was found that the piston rings were in comparatively good condition. However, the exhaust valve was badly burned, and the valve stems showed signs of possible sticking. (cf. appendix). Also, the swirl chamber in the Comet head was found to be badly burned and beyond use for normal diesel operation. Due to this fact, and to the fact that the valves and valve seats would have to be reground for further tests, it was decided that the entire investigation must be concluded. There was not enough time to make all these repairs and to take enough data to substantiate any new set of operating conditions. The data already taken was enough to establish trends and general characteristics. Generally, data in Series I may be used for approximate numerical data, but Series II and Series III must be considered only with the fact of defective exhaust valve operation in mind.

DISCUSSION OF RESULTS AND CONCLUSIONS

The general characteristic of the curves of brake horsepower versus fuel-air ratios (cf. Fig. 2, 3, 4, 5, 6, 7,) is quite different from those obtained from a spark ignition engine or a compression ignition engine burning diesel fuel. The controlling factor for these curves is seen in comparing the indicator diagrams (cf. Fig. 15, 16, 17). These cards were taken at constant speed, 1200 rpm., and varying fuel-air ratios. The general trend shows that with a lean fuel-air ratio, combustion starts earlier and the maximum pressure in the cylinder is reached earlier before top dead center. This type of combustion requires more work by the piston in comparison, and therefore subtracts from the power output of the engine. In the richer fuel-air ratios the combustion starts later and the peak pressure is finally reached at or slightly after top dead center. (Cf. Fig. 17). The curves bear this fact out in that in the range of indicator cards available, the brake horsepower reached a maximum at the higher fuel-air ratios. Fig. 17 also indicates that a maximum fuel-air ratio running condition is approached, for if the fuel-air ratio were increased further, combustion would not take place before top dead center, and therefore there would be no firing.

Due to starting difficulties, friction horsepowers were not recorded under the different operating conditions. However, in the range of 1000 rpm to 1200 rpm, friction horsepowers varied from about 2 to 5. Were these data plotted for indicated horsepower rather than brake horsepower, the scale would only be moved higher, but the same shape would be retained.

The increase in power output in the lean fuel-air ratios was probably due to the fact of more uniform mixture of fuel and air, causing a definite flame front to develop, probably from a hot spot around the exhaust valve, and therefore a slower pressure rise and possibly peak pressure point after top center. Flame speed decreases with decreasing fuel-air ratio.

As all comparisons were to be made on a constant speed basis, brake horsepowers were plotted rather than mean effective pressures. At constant speed, these two quantites are directly proportional to each other.

The fuel consumption curves found on the same sheet with the brake horsepower curves are of characteristic shapes compared to the BHP curves. As BSFC is inversely proportional to BHP, we find that where BHP decreases, BSFC increases and vice versa.

The higher magnitude of the BSFC may be accredited to the fact that the heating value of benzol (17,330 btu per lb.) is less than that of octane or diesel fuel. Therefore, using benzol, fewer btu are put into the engine per pound of fuel. Output is therefore less per pound of fuel put in, and specific fuel consumption is greater.

The other characteristics of benzol are seen in Fig. 7 and 8. The general characteristics of these curves are similiar to those at atmospheric inlet pressure. It is noted, however, that the range of fuel-air ratios has shifted to a low region. This, it is believed, was caused by improper valve action. The details will be explained later in the discussion. Fuel consumption at varying inlet pressure remains characteristic of the brake horsepower curves. As inlet pressure decreases, less air is taken into the engine, and therefore less power is expected to be obtained. This trend is verified by the curves.

Although there are no plots of speed versus output by comparing the data, it is seen that brake horsepower remained relatively constant with varying speed. However, if the estimated friction load is taken into account, we see that as speed increases, indicated horsepower will also increase at constant fuel-air ratio. This is a typical characteristic of the operation of compression ignition engines. It was noted that as engine speed increased, the tendency of the engine to detonate was increased also. This is explained by the fact that the delay period of the fuel occupied a greater number of crank angle degrees at higher speeds. A more rapid pressure rise ~~speed~~ was experienced and burning started at a relatively higher temperature and pressure in the cylinder. (Cf. Fig. 17) All these facts lead to detonation in the compression ignition engine.

The curve of knock intensity versus fuel-air ratio (Cf. Fig. 9) is entirely an estimated trend. Data was taken by ear and may be considered as a general characteristic for all speeds investigated. Knocking was present for all conditions, and values of knocking intensity are compared in a relative sense.

The high knock intensity at rich fuel-air ratios may be accredited to a long delay period. In compression ignition, detonation is caused from long delay period giving rapid ~~rpaid~~ pressure rise and instantaneous explosion of fuel. The desired condition is to have as short a delay period as possible and comparatively early ignition with a slow pressure rise. With rich fuel-air ratios, there is more fuel in the mixture and more heat is absorbed in heating the mixture to the ignition point. This causes the long delay and resulting detonation, or knocking, as it is called. The indicator cards show this late burning and rapid pressure rise (Cf. Fig. 17).

The increased knock intensity at the lean fuel-air ratios is caused from the fact that the mixture is very light, and enough air is present to make a highly explosive mixture. As combustion is probably complete, the exhaust temperatures are higher with lean mixtures and a hot spot forms at the exhaust valve. This causes a sort of pre-ignition and explodes the mixture against the rapid motion of the piston considerably before top dead center. The knocking in this range, although increasing with decreased fuel-air ratio, is not as heavy or sharp a knock as was heard with the rich mixture.

At medium fuel-air ratios, knocking was at a minimum. The mixture was probably nearer the "Chemically correct" fuel-air ratio, and combustion was complete. Here, however, there is more fuel, and the tendency to explode the fuel is less. Probably the hot spot at the exhaust valve existed and started combustion, and a normal flame front developed from that point.

The delay period was short, and the flame front traveled evenly across the cylinder. Indicator cards bear this assumption by showing a slow pressure rise in the medium fuel-air ratio range (Cf. Fig. 16).

As would be expected, the throttling effect on knocking was to decrease it. The inlet pressure was decreased and therefore peak pressures were decreased. This decreased cylinder pressure, and therefore allowed slower burning of the fuel. Fig. 15 shows this fact with the early ignition point.

As previously mentioned, a variation in the range of operation was noted after the throttle runs were recorded. Further investigation showed that at atmospheric pressure the engine continued to run at this lower range of fuel-air ratios (Cf. Fig. 5 and 6). The engine was torn down and evidence of improper exhaust valve operation was found, which explains the shift in fuel-air ratios. A gummy formation on the valve stem indicated valve sticking. If the valve was sticking at any time, this meant that high temperature exhaust gases were passing or leaking past the valve. This accounted for the hot spot, and also caused severe burning of the valve and valve seat which in turn led to more leaking past the valve.

Assuming the above condition, this explanation follows. Gases leaking past the valve reduced the pressure in the cylinder, and therefore reduced the brake mean effective pressure. This lowered the brake horsepower, which is seen in the figures noted. The hot spot at the exhaust valve caused earlier combustion and therefore earlier peak pressure also leading to low brake horsepower.

Further reason to believe the trouble was with the exhaust valve is seen by comparing Fig. 13 and 14. In Fig. 13, the exhaust stroke shows a greater scatter at lower exhaust valve pressure than does Fig. 14. These two cards were taken under identical running conditions and at approximately the same time. Also note the relative peak pressures, motoring and firing. The one with leaking exhaust valve has the lower pressures.

It is assumed that conditions for data plotted on Fig. 2, 3, and 4 were without valve sticking. This should make higher pressures, more knocking, and greater scatter on compression stroke--all these factors were borne out by experiment.

It is safe to assume that the original high fuel-air ratios (Tables I, II, and III) are in the characteristic range for the benzol compression ignition engine. In considering the remainder of the data, the general trends may be thought of as typical, but numerical values must be discounted.

RESULTS

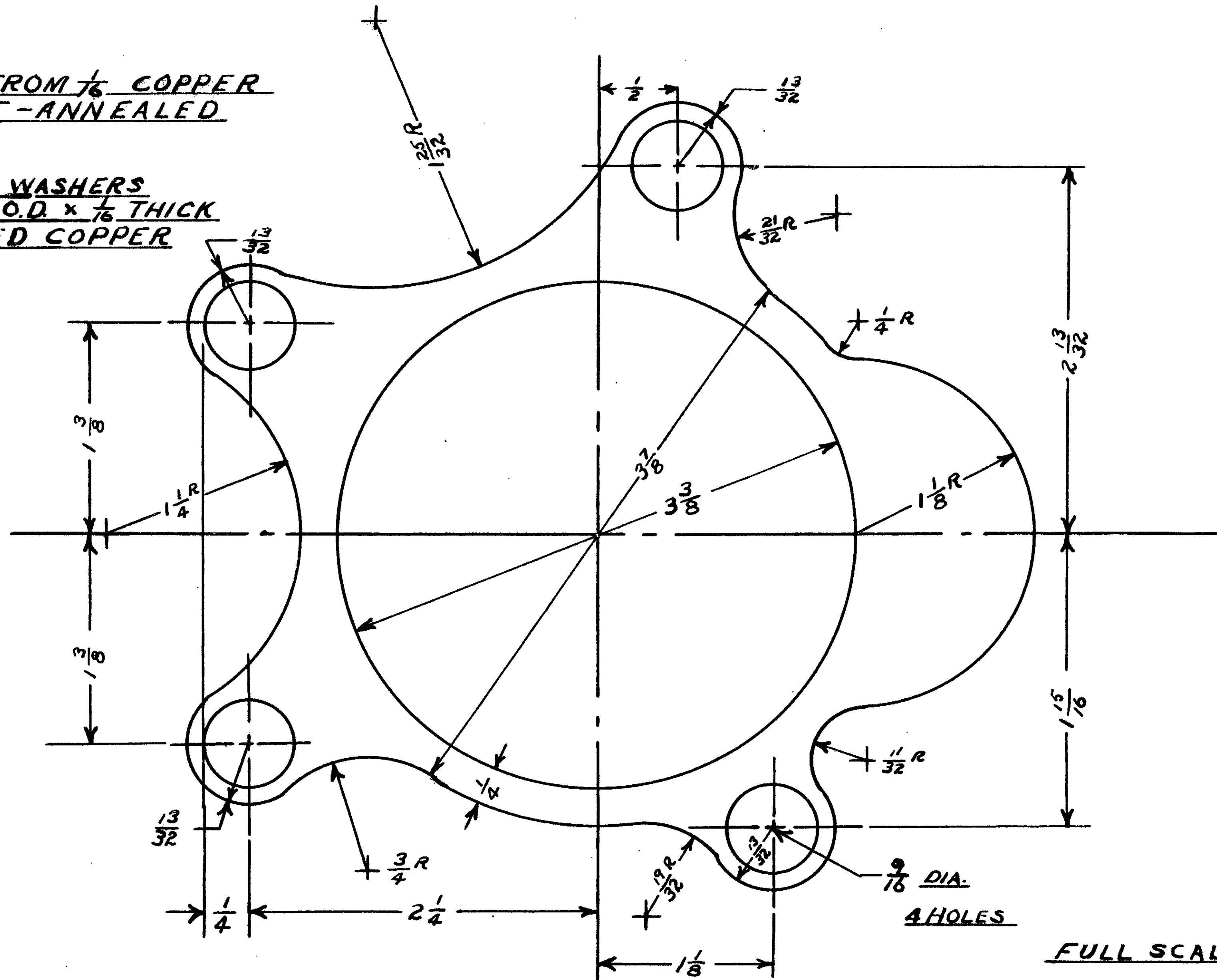
The results for the tests are presented in Fig. 2 to 9 inclusive.

Some representative indicator diagrams are given in Fig. 11 to 17 inclusive.

The data from the tests are given in the appendix.

NOTE:
CUT FROM $\frac{1}{16}$ COPPER
SHEET-ANNEALED

MAKE 2 WASHERS
$\frac{9}{16}$ I.D. × $\frac{13}{16}$ O.D. × $\frac{1}{16}$ THICK
ANNEALED COPPER



FULL SCALE

FIG. 1

HIGH PRESSURE GASKET
FOR C.F.R. DIESEL NO. 3
SLOAN LABORATORY - M.I.T.
T.R. HICKEY & T.I. STEPHENSON
MARCH 26, 1945

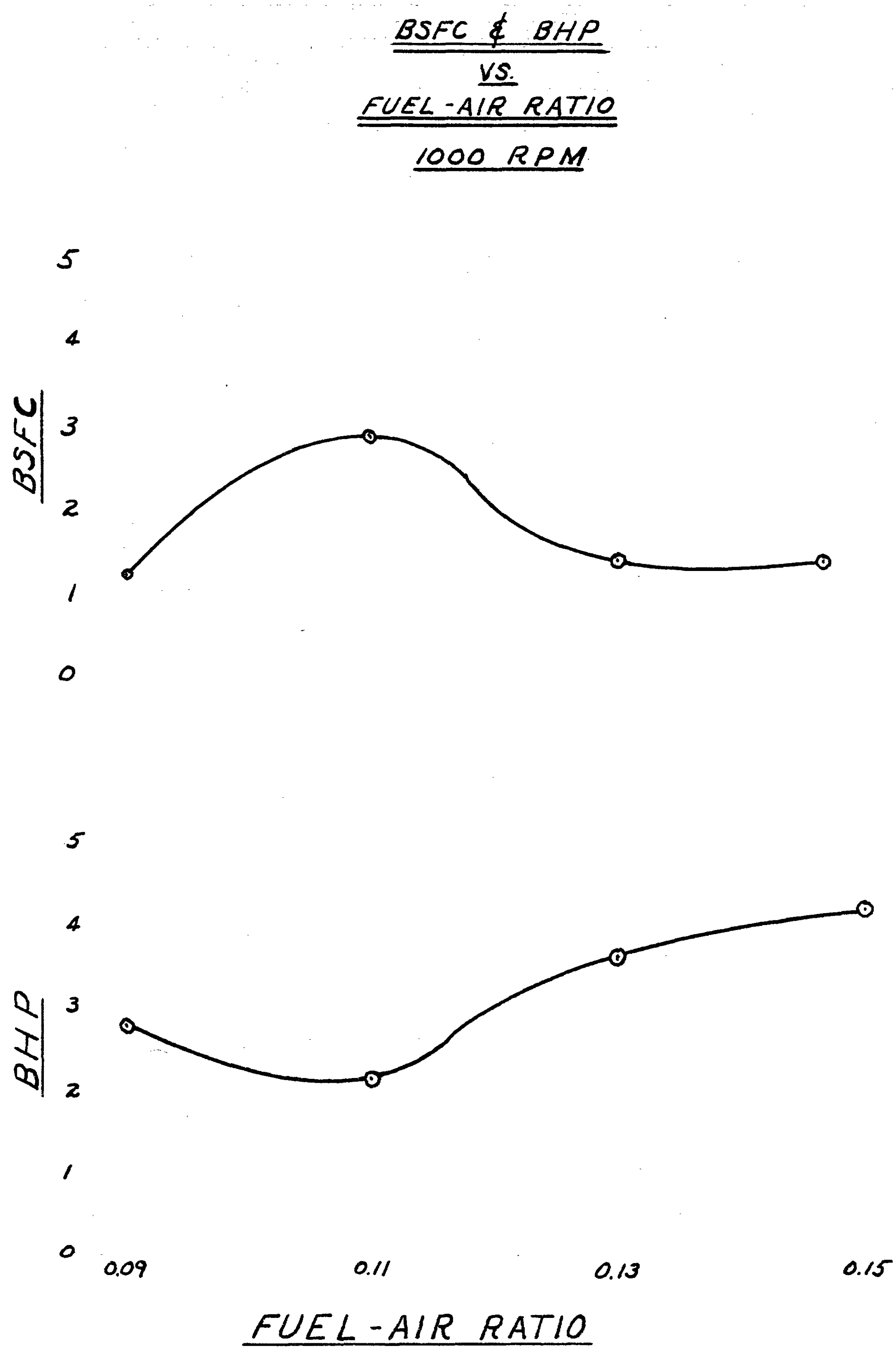

BSFC & BHP
VS.
FUEL-AIR RATIO
1000 RPM
5
4
3
2
1
0
BSFC
5
4
3
2
1
0
BHP
0.09
0.11
0.13
0.15
FUEL-AIR RATIO

Fig. 2

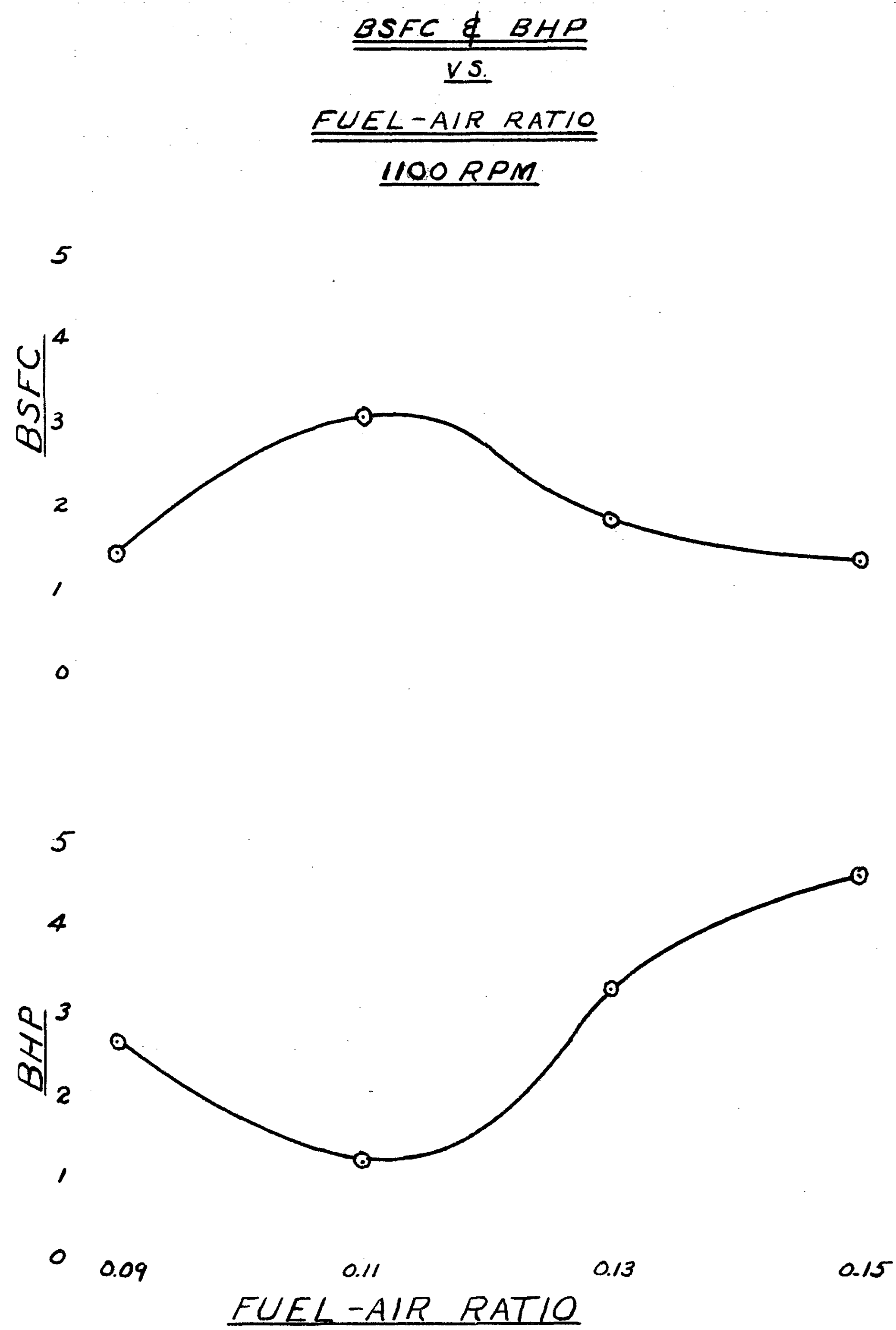

BSFC & BHP
VS.
FUEL-AIR RATIO
1100 RPM
BSFC
5
4
3
2
1
0
BHP
5
4
3
2
1
0
0.09
0.11
0.13
0.15
FUEL-AIR RATIO

Fig. 3

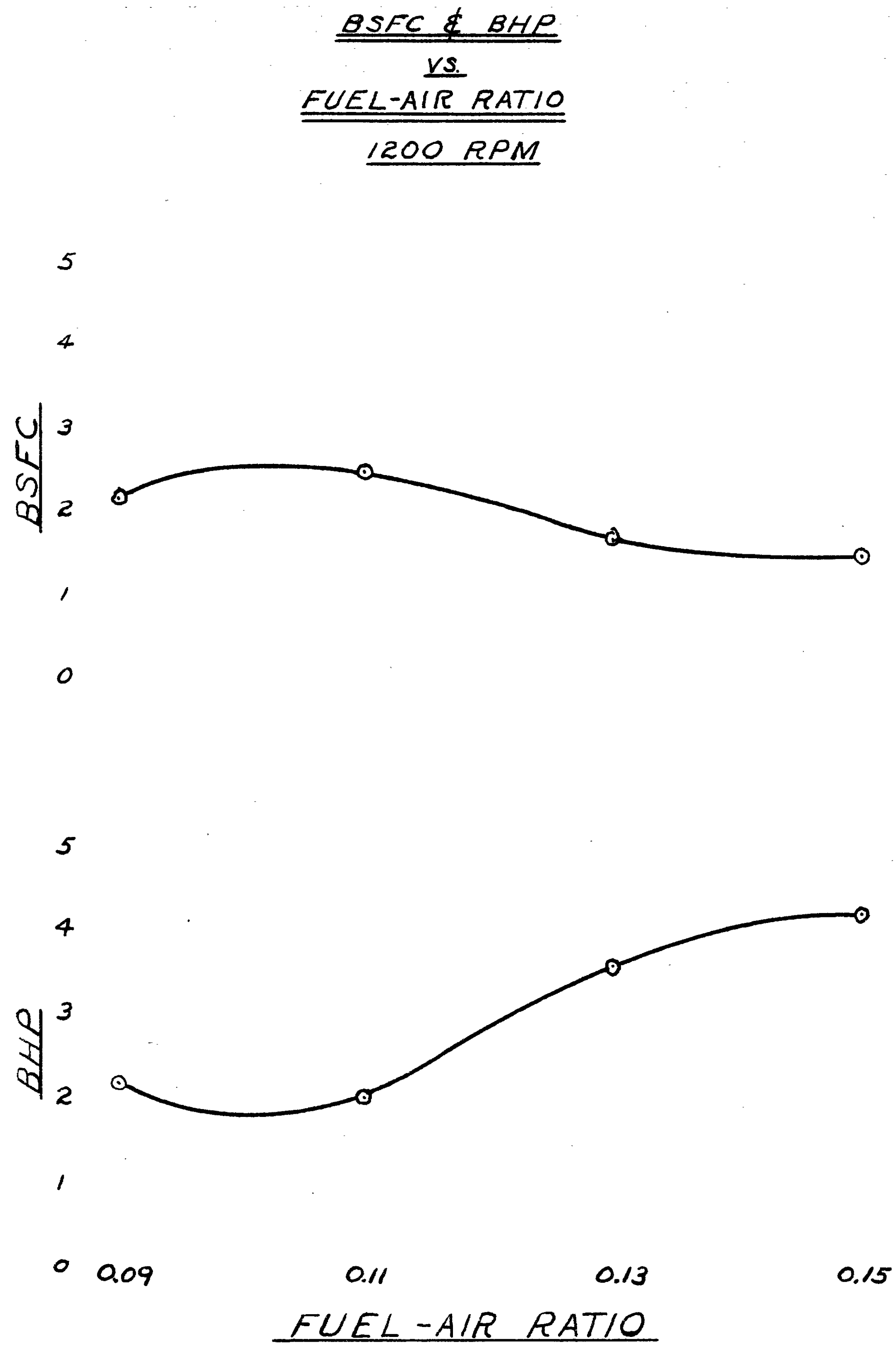

BSFC & BHP
VS.
FUEL-AIR RATIO
1200 RPM
BSFC
5
4
3
2
1
0
BHP
5
4
3
2
1
0
0.09
0.11
0.13
0.15
FUEL-AIR RATIO

Fig. 4

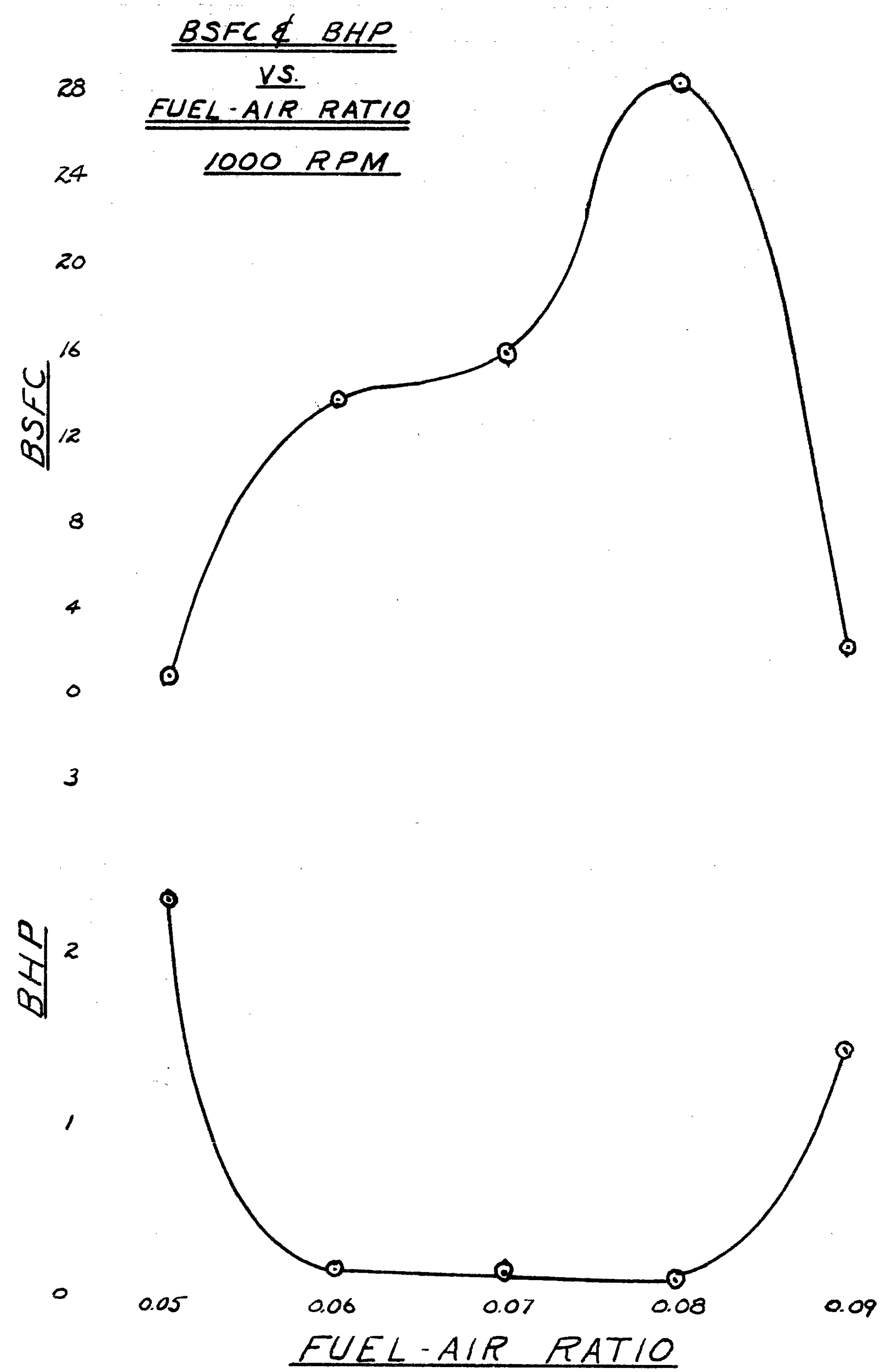

BSFC & BHP
VS.
FUEL-AIR RATIO
1000 RPM
BSFC
28
24
20
16
12
8
4
0
3
BHP
2
1
0
0.05
0.06
0.07
0.08
0.09
FUEL-AIR RATIO

Fig. 5

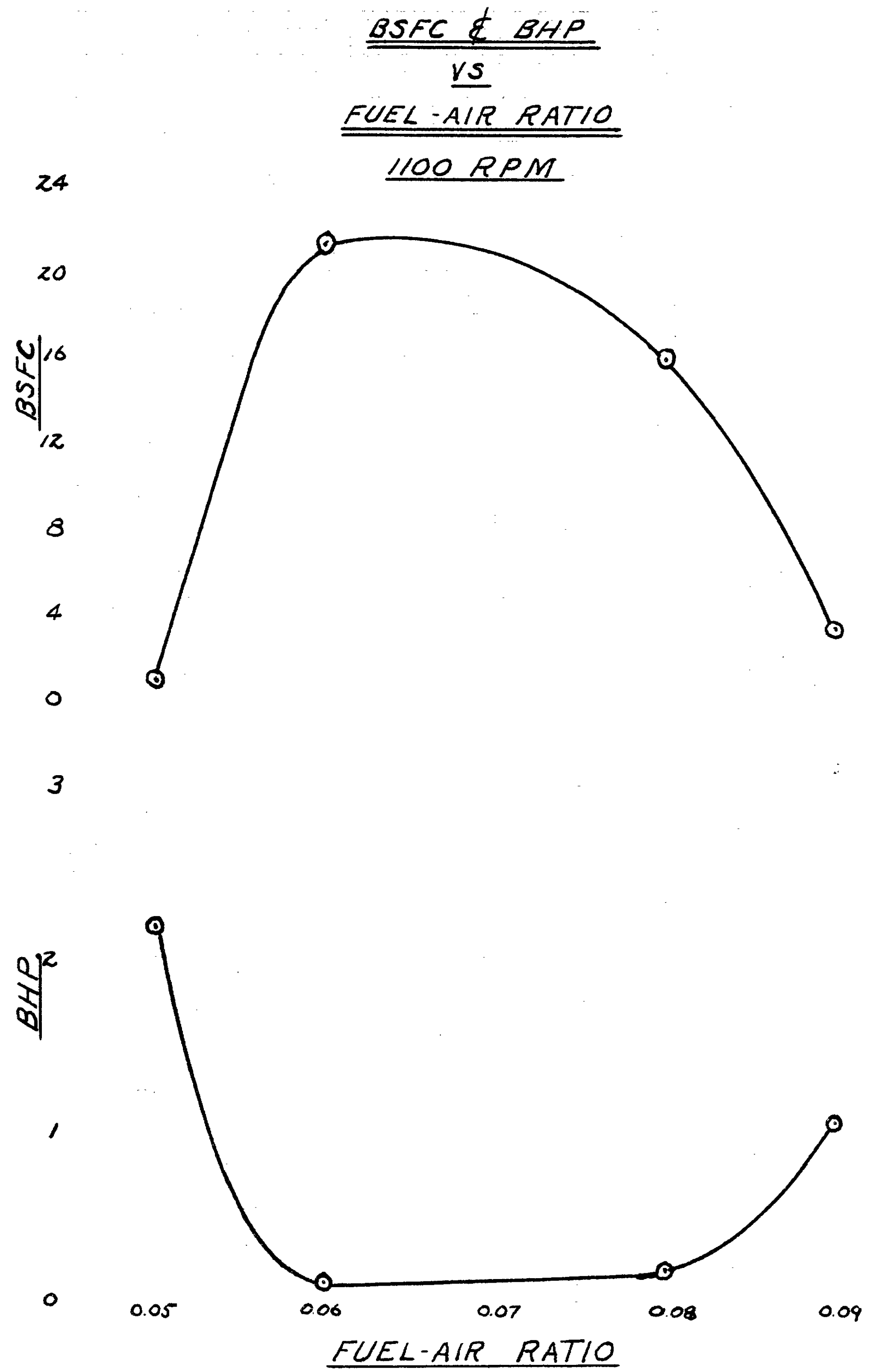
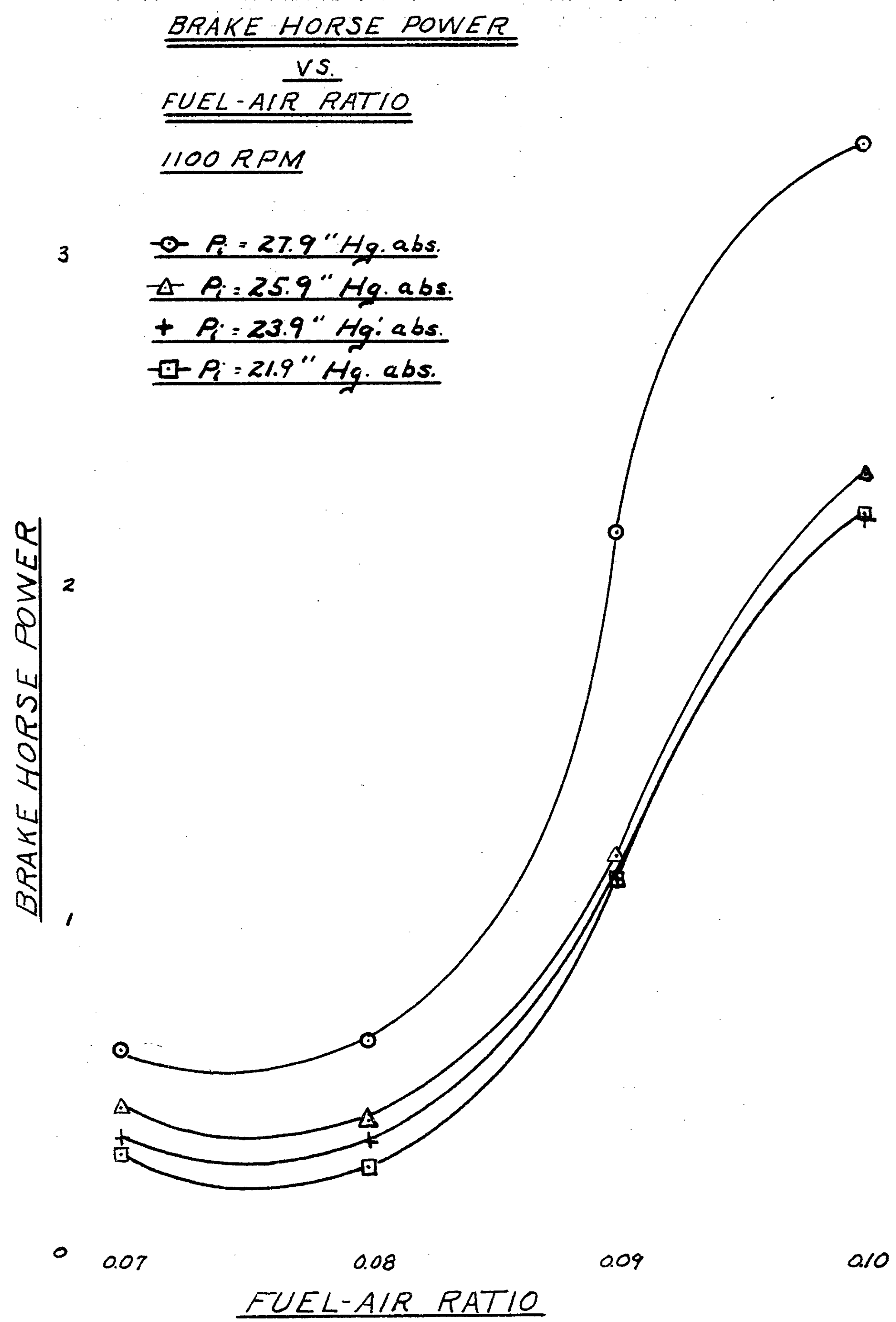

BSFC & BHP
VS
FUEL-AIR RATIO
1100 RPM
24
20
16
BSFC
12
8
4
0
3
2
BHP
1
0
0.05
0.06
0.07
0.08
0.09
FUEL-AIR RATIO

Fig. 6



BRAKE HORSE POWER
VS.
FUEL-AIR RATIO
1100 RPM
Pi = 27.9" Hg. abs.
Pi = 25.9" Hg. abs.
Pi = 23.9" Hg. abs.
Pi = 21.9" Hg. abs.
3
2
1
0
BRAKE HORSE POWER
0.07
0.08
0.09
0.10
FUEL-AIR RATIO

Fig. 7

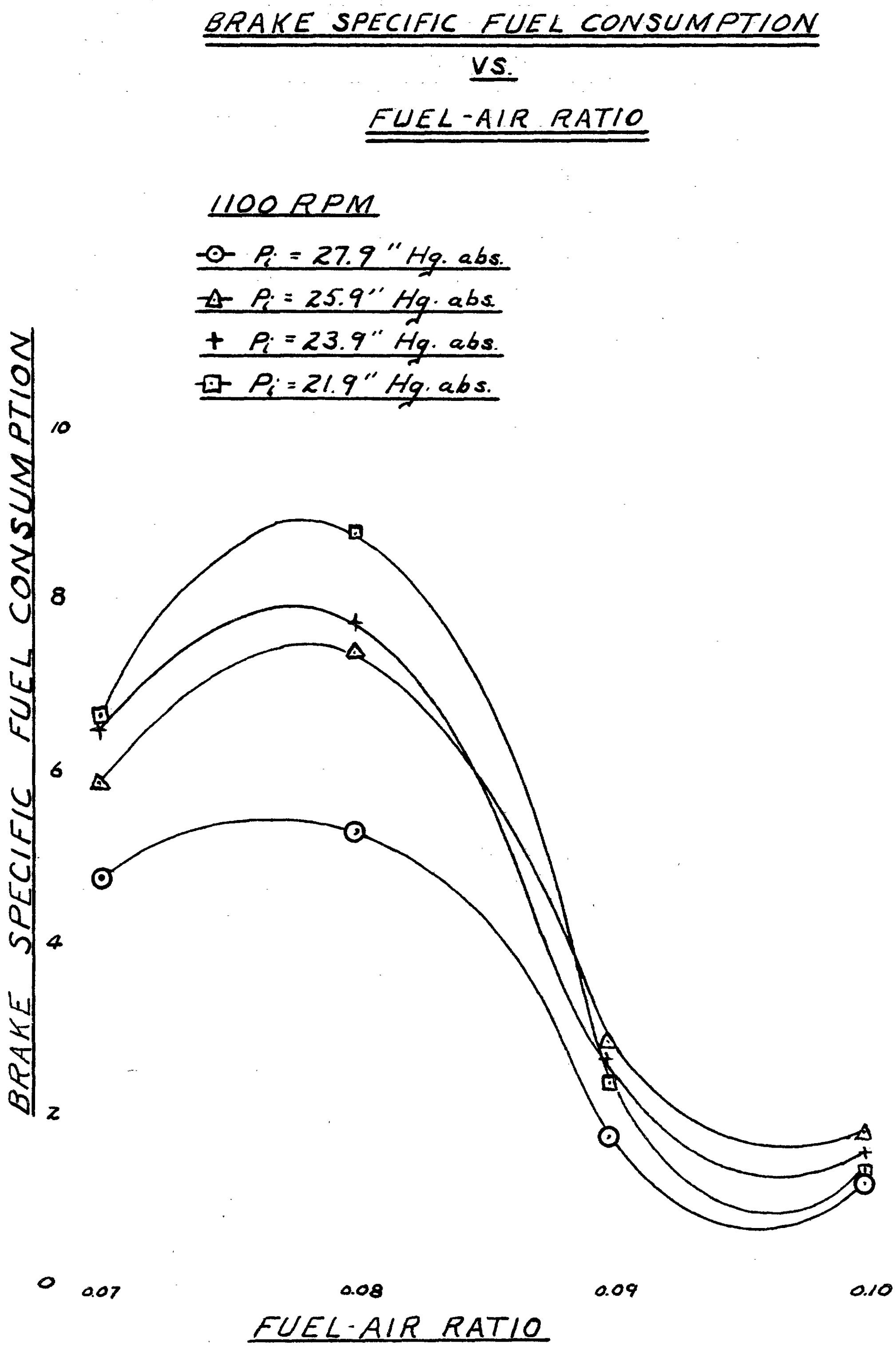

BRAKE SPECIFIC FUEL CONSUMPTION
VS.
FUEL-AIR RATIO
1100 RPM
P_i = 27.9 " Hg. abs.
P_i = 25.9" Hg. abs.
P_i = 23.9" Hg. abs.
P_i = 21.9" Hg. abs.
BRAKE SPECIFIC FUEL CONSUMPTION
10
8
6
4
2
0
0.07
0.08
0.09
0.10
FUEL-AIR RATIO

Fig. 8

KNOCK INTENSITY

VS

FUEL-AIR RATIO

NOTE:

OBSERVATIONS TAKEN BY EAR OVER ENTIRE RANGE OF INVESTIGATION

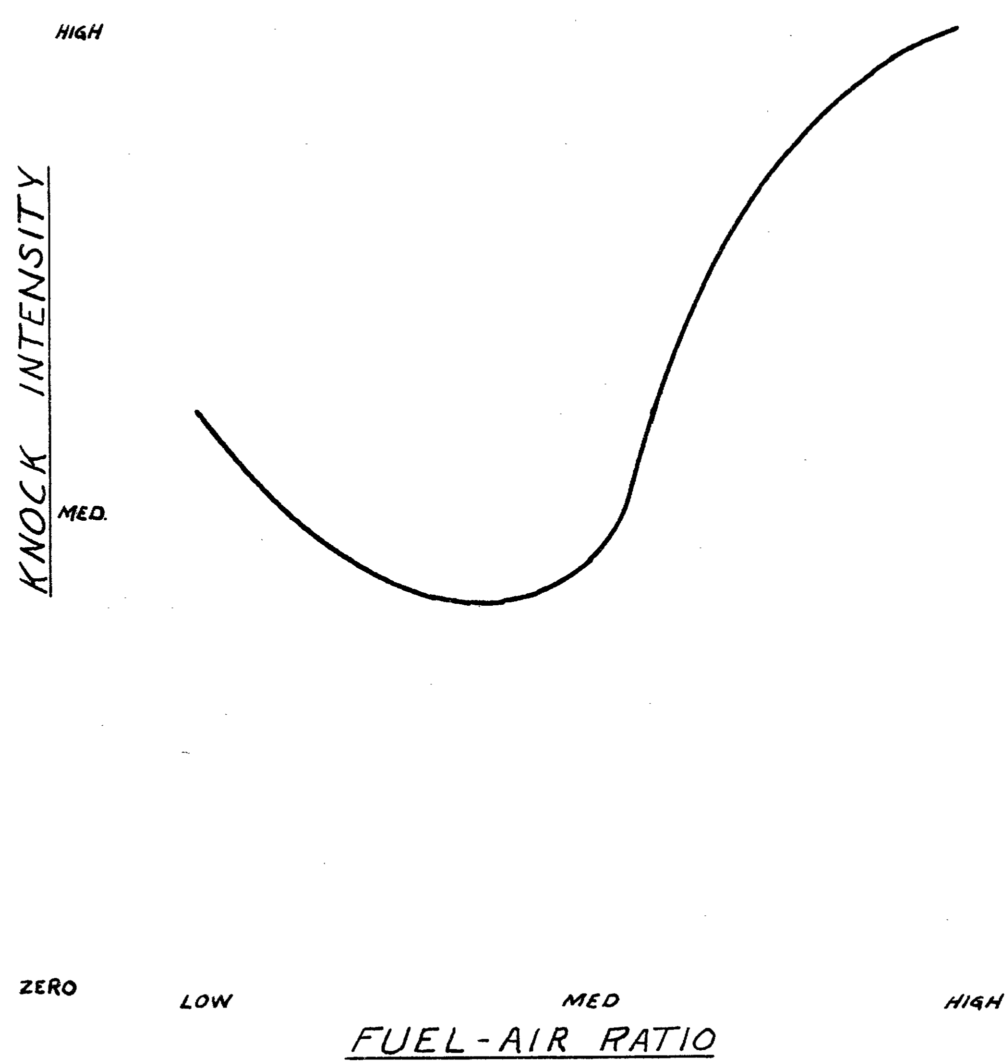


Fig. 9

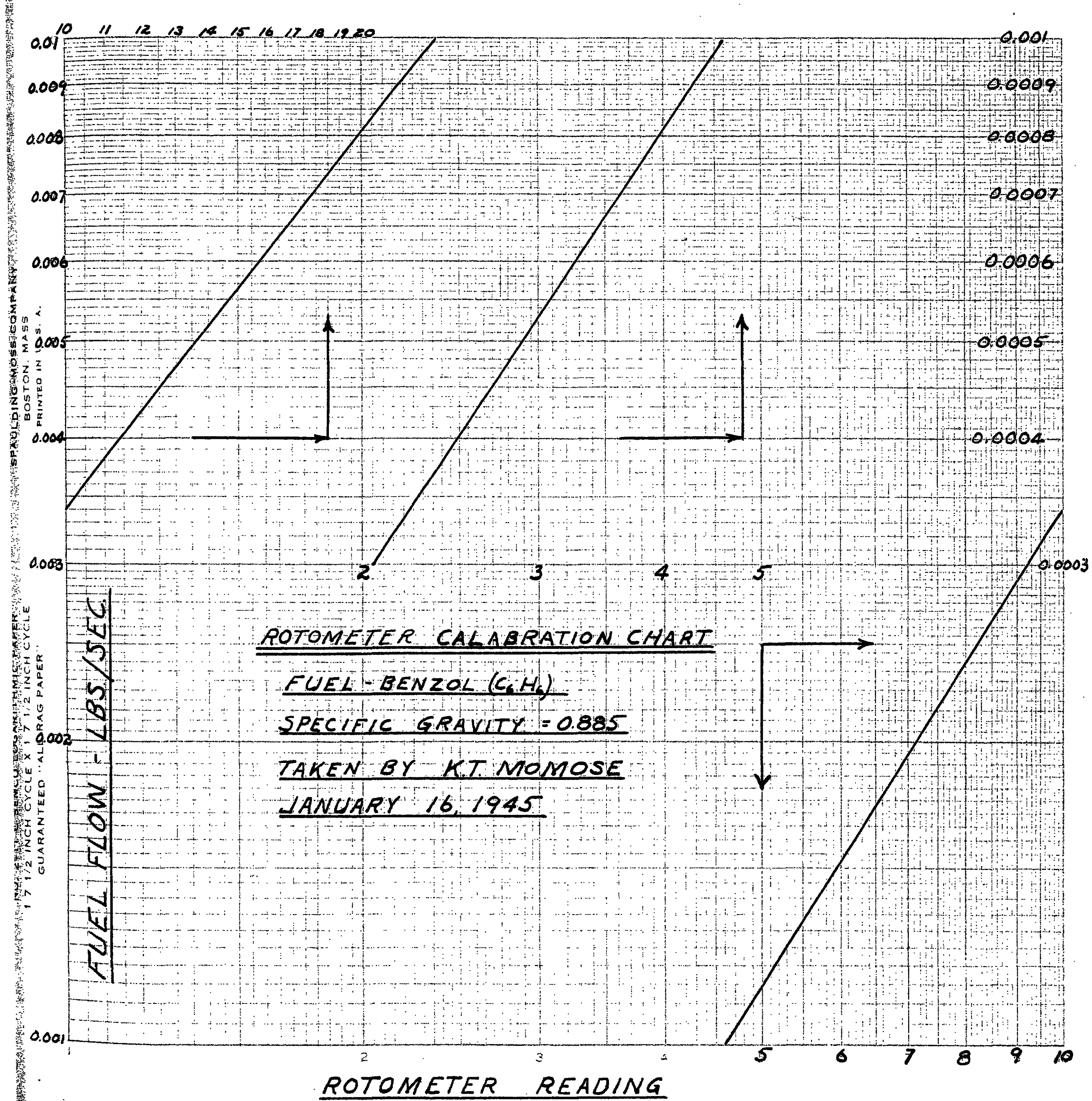

ROTOMETER CALABRATION CHART
FUEL - BENZOL (C_6H_6)
SPECIFIC GRAVITY = 0.885
TAKEN BY K.T. MOMOSE
JANUARY 16, 1945
FUEL FLOW - LBS/SEC
ROTOMETER READING
SPAULDING-MOSS COMPANY
BOSTON, MASS
PRINTED IN U. S. A.
GUARANTEED "ALL RAG PAPER"
0.01
0.009
0.008
0.007
0.006
0.005
0.004
0.003
0.002
0.001
0.0009
0.0008
0.0007
0.0006
0.0005
0.0004
0.0003
10 11 12 13 14 15 16 17 18 19 20
2 3 4 5
5 6 7 8 9 10

Fig. 10

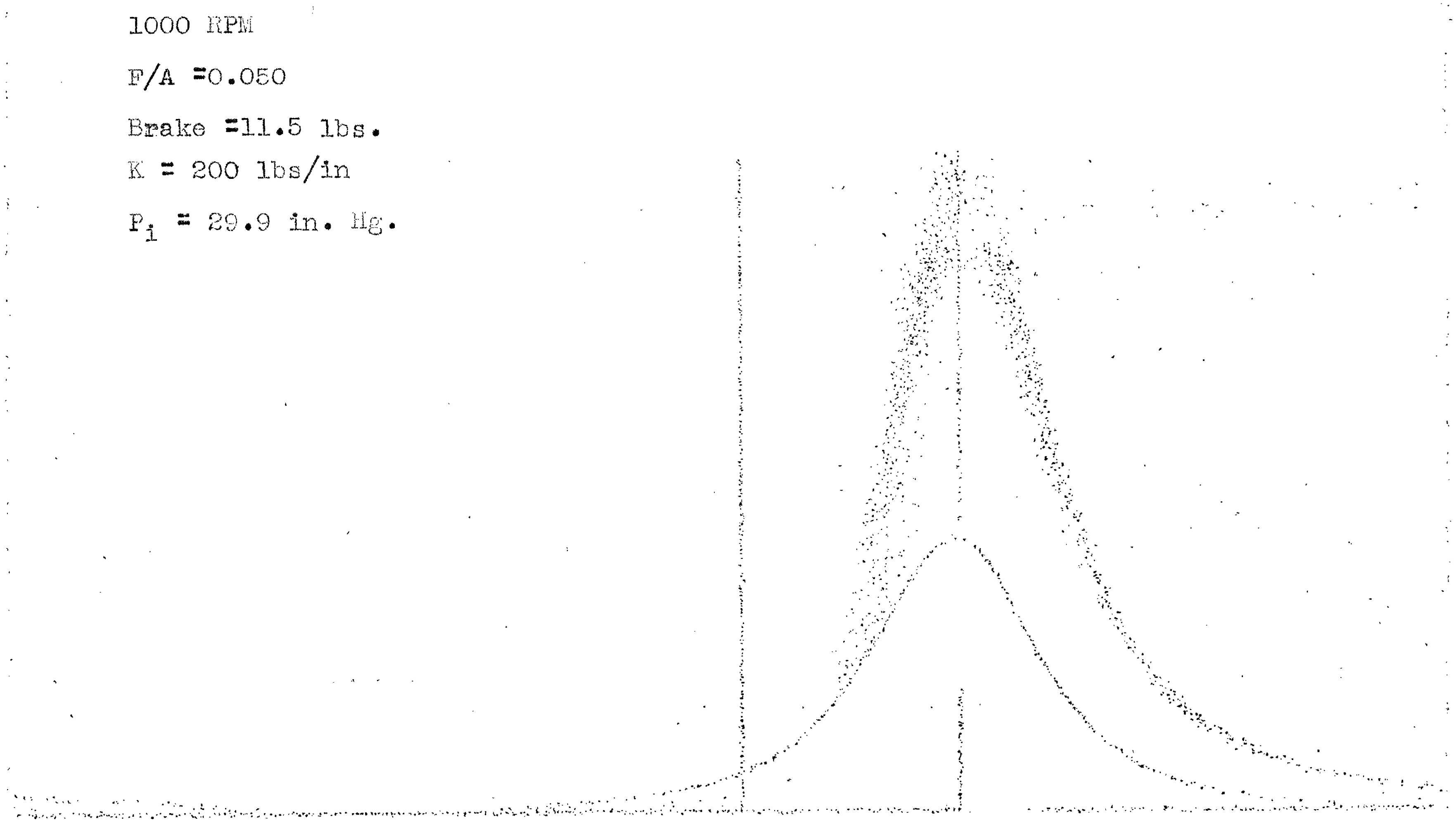


Fig. 11

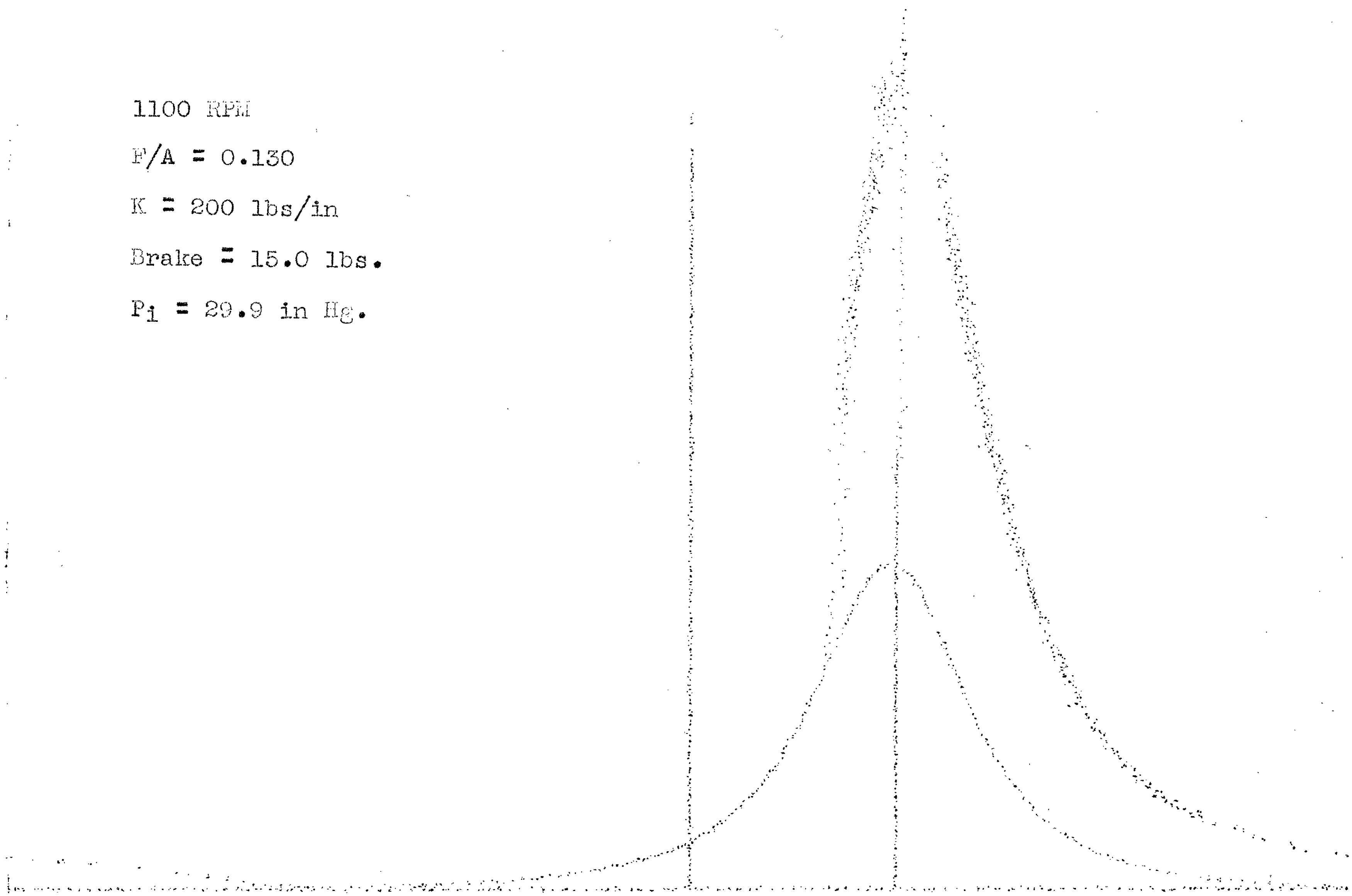


Fig. 12

CONDITION A

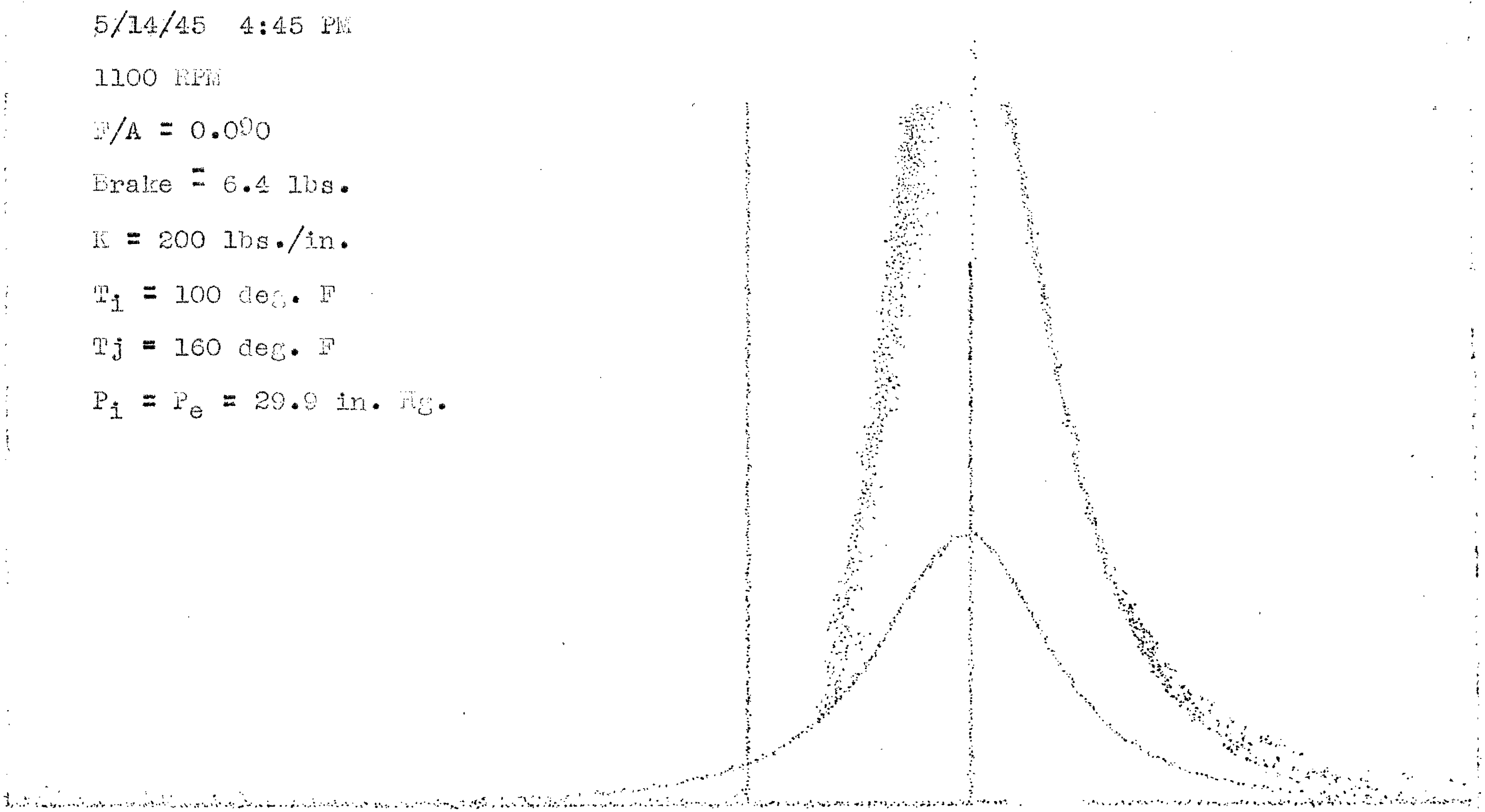


Fig. 13

CONDITION B

5/14/45 5:15 PM

1100 RPM

F/A = 0.090

Brake = 14.2 lbs.

K = 200 lbs./in.

T_i = 100 deg. F

T_j = 160 deg. F

P_i = P_e = 29.9 in.Hg.

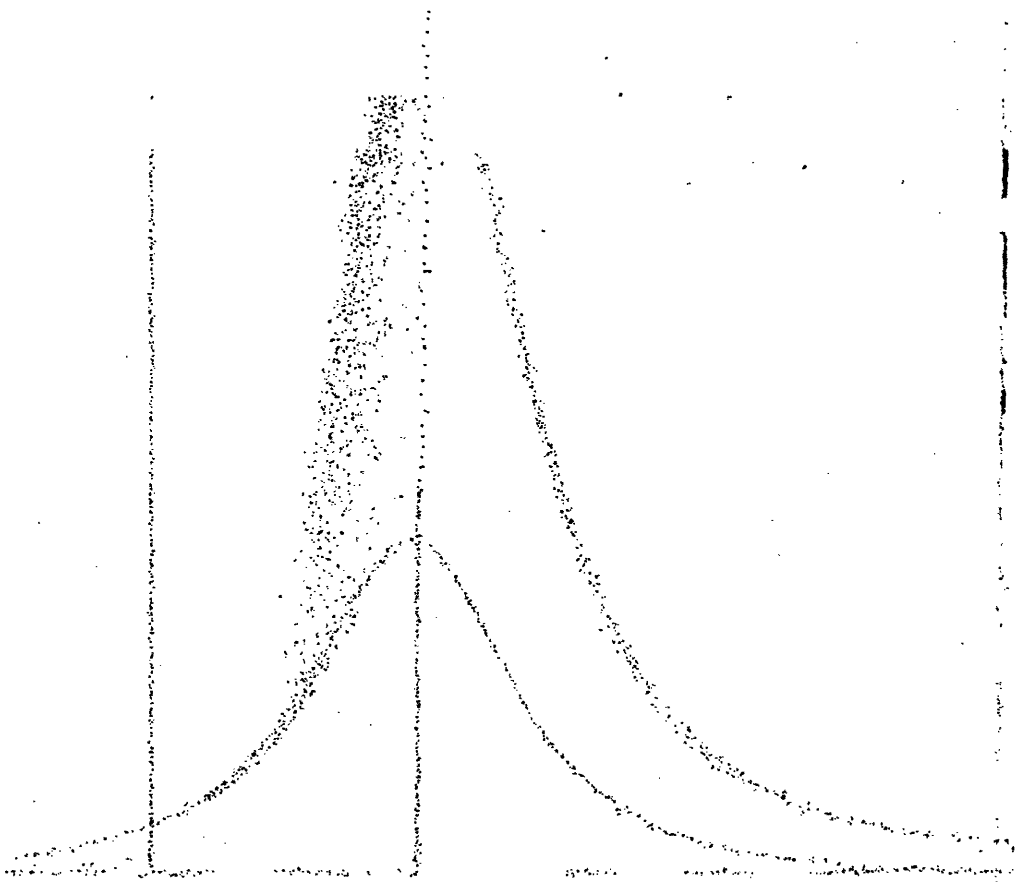


Fig. 14

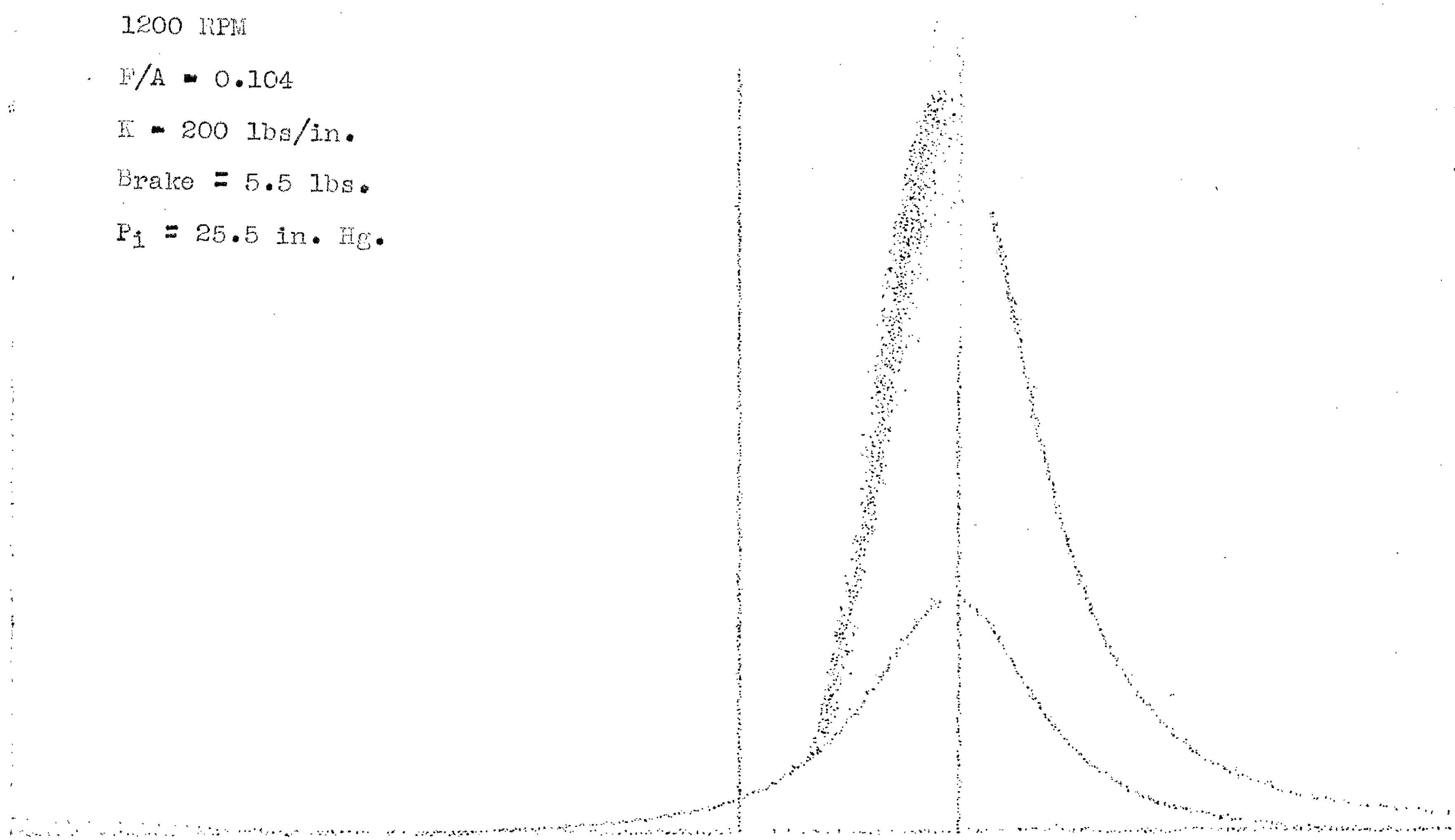


Fig. 15

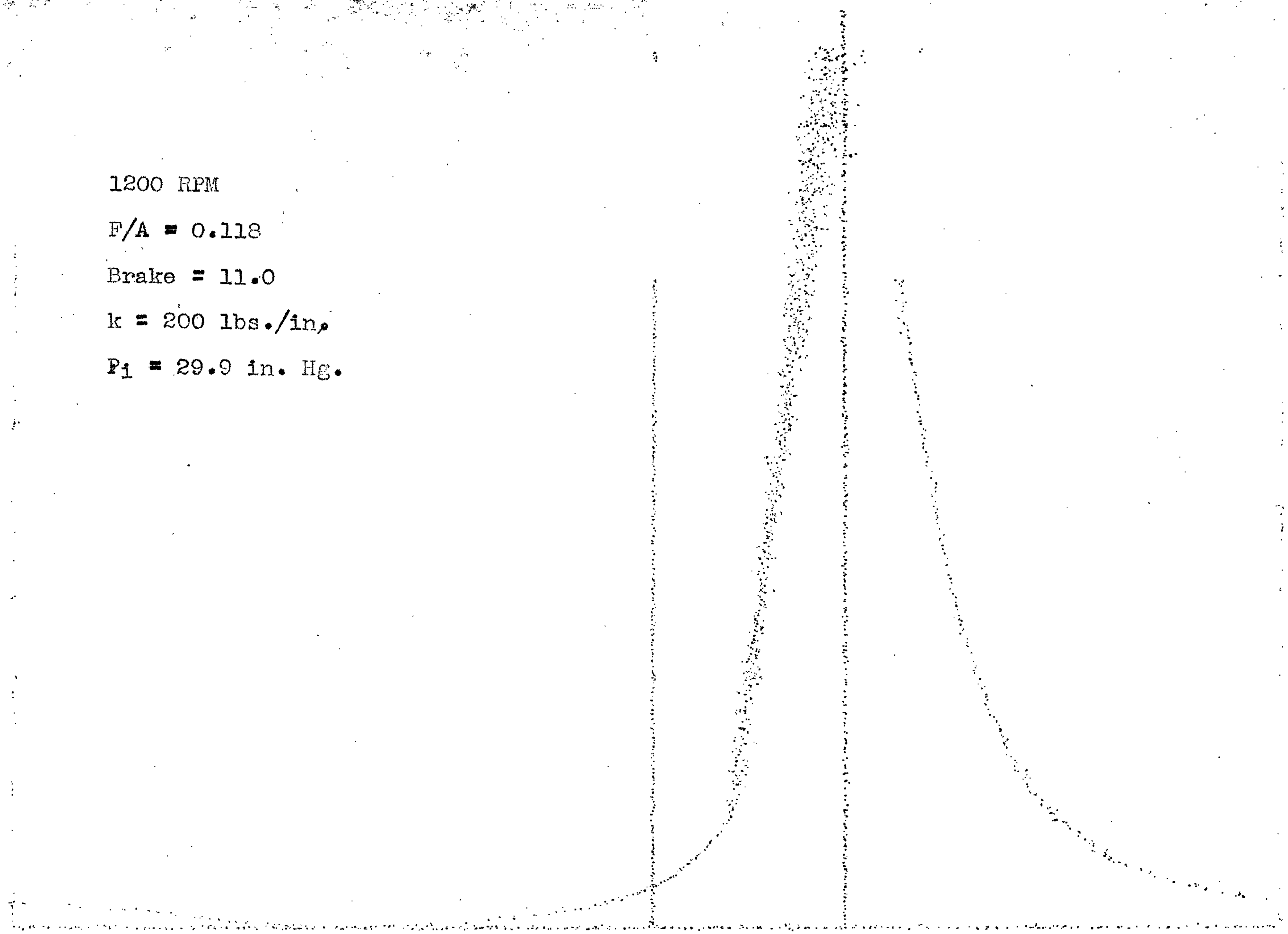


Fig. 16

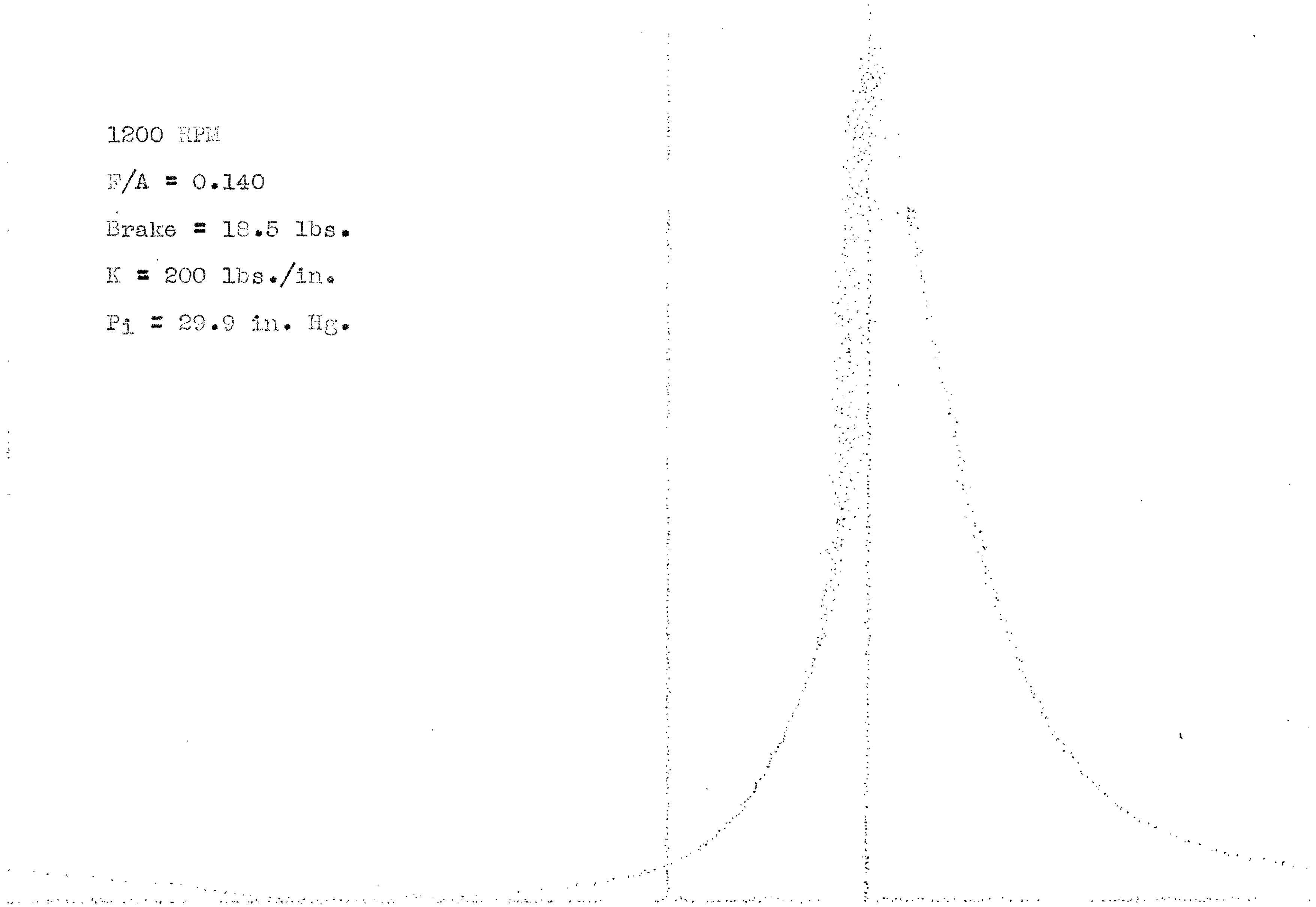


Fig. 17

SUGGESTIONS FOR FURTHER INVESTIGATION

It is obvious that no reliable observations will be made unless some care is taken to insure that the exhaust valve and its seat will not become burned, and the exhaust valve will not stick. Perhaps a water cooled valve will achieve this end.

From the operation of the engine it was observed that a small variation in the inlet temperature of the fuel mixture would cause a considerable variation in the engine performance. It is suggested that a system for controlling the inlet temperature within one degree be used.

To keep the benzol from condensing on the manifold walls, it is further suggested that the intake air be heated previous to mixing with the fuel.

Since the fuel used, benzol, shows extremely large fuel consumption at all conditions, it is suggested that other fuels be tried to see if this fuel consumption can be reduced. If the rate of pressure rise of the benzol be slowed down, more power will result, and therefore a smaller fuel consumption. Some dope might accomplish this purpose and give a more controlable output.

APPENDIX

PHYSICAL CONDITION OF ENGINE AFTER TESTS

After completion of the tests on the engine, it was torn down in order that the effects of the tests could be seen.

The piston and cylinder head were entirely free of carbon. This was attributed to the fact that benzol is an excellent solvent of most materials.

As was deduced from the indicator diagrams, the exhaust valve was badly burned. The valve seat was burned, but not so badly as the valve. The stem of the valve was coated with a hard sticky substance on its bearing surfaces. It was this observation that suggested that perhaps the valve had been sticking during operation.

The Comet head swirl chamber was so badly burned and eroded that the purchase of a new unit was necessary before the engine could again be used as a diesel engine. The damage to this unit was concentrated about three points. A large section of the lip of the channel leading from the spherical part of the chamber into the cylinder had burned off and there were two cracks, one major and one minor, on the remaining part of the lip. The two other damaged sections were around the circumference of the two holes at the top of the unit. Channels were burned radially outward from them.

The special copper gasket used for this test was found to have become distorted. The cylinder hole of the gasket had assumed a more nearly elliptical, rather than circular, shape. This distortion was due to the elongation of the thinner sections of the gasket. There could not have been any bending, since the gasket, with the exception of elongated parts, retained its original dimensions and position.

TABLE I

DATA AT 1000 RPM - CONSTANT INLET PRESSURE

$\frac{F}{A}$	BL	OIL TEMP.	JACKET TEMP.	P_i	P_e	T_i	AIR CONS.	FUEL CONS.	BHP	BSFC
0.090	17.3	138	158	14.7	14.7	100	0.0098	0.00089	3.46	0.92
0.090	1.0	140	156	14.7	14.7	120	0.0098	0.00090	0.20	16.20
0.090	10.6	170	156	14.7	14.7	100	0.0096	0.00096	2.12	1.47
0.110	8.9	173	154	14.7	14.7	100	0.0101	0.00111	1.97	2.23
0.110	13.1	170	150	14.7	14.7	120	0.0100	0.00110	2.02	1.51
0.110	5.1	138	150	14.7	14.7	100	0.0101	0.00112	1.02	3.95
0.130	17.8	138	150	14.7	14.7	100	0.0103	0.00134	3.57	1.35
0.130	19.5	150	158	14.7	14.7	120	0.0100	0.00130	3.95	1.18
0.130	15.8	175	150	14.7	14.7	100	0.0102	0.00133	3.16	1.52
0.150	20.2	168	150	14.7	14.7	100	0.0105	0.00156	4.04	1.39
0.150	22.0	150	150	14.7	14.7	120	0.0104	0.00156	4.40	1.28
0.150	20.4	138	150	14.7	14.7	100	0.0103	0.00154	4.08	1.36

TABLE II

DATA AT 1100 RPM - CONSTANT INLET PRESSURE

$\frac{F}{A}$	BL	OIL TEMP.	JACKET TEMP.	P_i	P_e	T_i	AIR CONS.	FUEL CONS.	BHP	BSFC
0.090	9.6	138	156	14.7	14.7	100	0.0108	0.00097	2.11	1.65
0.090	10.9	150	152	14.7	14.7	120	0.0105	0.00095	2.40	1.42
0.090	13.0	165	154	14.7	14.7	100	0.0107	0.00697	2.86	1.21
0.110	4.8	165	160	14.7	14.7	100	0.9100	0.00120	1.02	4.24
0.110	3.5	150	154	14.7	14.7	120	0.0107	0.00118	0.77	5.53
0.110	7.9	138	154	14.7	14.7	100	0.0110	0.00121	1.74	2.50
0.130	8.7	138	150	14.7	14.7	100	0.0114	0.00148	1.91	2.79
0.130	18.1	150	150	14.7	14.7	110	0.0114	0.00148	3.98	1.33
0.130	17.0	160	154	14.7	14.7	100	0.0113	0.00147	3.74	1.41
0.150	20.8	160	150	14.7	14.7	100	0.0114	0.00171	4.58	1.34
0.150	20.2	160	150	14.7	14.7	110	0.0114	0.00172	4.44	1.39
0.150	21.0	140	160	14.7	14.7	100	0.0114	0.00171	4.62	1.33

TABLE III

DATA AT 1200 RPM - CONSTANT INLET PRESSURE

$\frac{F}{A}$	BL	OIL TEMP.	JACKET TEMP.	P_i	P_e	T_i	AIR CONS.	FUEL CONS.	BHP	BSFC
0.090	5.0	140	158	14.7	14.7	100	0.0116	0.00104	1.20	3.12
0.090	12.9	150	156	14.7	14.7	110	0.0113	0.00107	3.10	1.24
0.110	9.3	150	152	14.7	14.7	100	0.0116	0.00128	2.23	2.07
0.110	9.5	150	154	14.7	14.7	110	0.6114	0.00126	2.28	1.98
0.110	6.1	140	150	14.7.	14.7	100	0.0119	0.00131	1.46	3.24
0.130	18.2	140	150	14.7	14.7	100	0.0123	0.00160	4.88	1.32
0.130	10.3	150	150	14.7	14.7	110	0.0114	0.00126	2.28	1.98
0.130	13.4	150	154	14.7	14.7	100	0.0121	0.00157	3.22	1.76
0.150	18.8	150	160	14.7	14.7	98	0.0121	0.00181	4.51	1.44

TABLE IV

DATA AT 1100 RPM - INLET PRESSURE 27.9 IN. HG.

$\frac{F}{A}$	BL	OIL TEMP.	JACKET TEMP.	P_i ("Hg)	P_e	T_i	AIR CONS.	FUEL CONS.	BHP	BSFC
0.070	2.8	143	158	27.9	14.7	100	0.0115	0.00081	0.62	4.73
0.080	2.9	143	160	27.9	14.7	100	0.0117	0.00094	0.64	5.28
0.090	9.8	143	158	27.9	14.7	100	0.0117	0.00105	2.16	1.75
0.100	15.1	143	155	27.9	14.7	100	0.0118	0.00118	3.32	1.28
0.110	16.1	143	154	27.9	14.7	100	0.0121	0.00133	3.55	1.35

TABLE V

DATA AT 1100 RPM - INLET PRESSURE 25.9 IN. HG

$\frac{F}{A}$	BL	OIL TEMP.	JACKET TEMP.	P_i ("Hg)	P_e	T_i	AIR CONS.	FUEL CONS.	BHP	BSFC
0.070	2.0	143	160	25.9	14.7	100	0.0102	0.00072	0.44	5.85
0.080	0.3	143	156	25.9	14.7	100	0.0102	0.00082	0.40	7.40
0.090	5.4	143	150	25.9	14.7	100	0.0103	0.00093	1.19	2.81
0.100	10.1	143	158	25.9	14.7	100	0.0107	0.00107	2.22	1.74
0.110	14.4	143	152	25.9	14.7	100	0.0107	0.00118	2.88	1.47

TABLE VI

DATA AT 1100 RPM - INLET PRESSURE 23.9 IN. HG.

$\frac{F}{A}$	BL	OIL TEMP.	JACKET TEMP.	P_i ("Hg)	P_e	T_i	AIR CONS.	FUEL CONS.	BHP	BSFC
0.070	1.6	135	150	23.9	14.7	100	0.0091	0.00064	0.35	6.50
0.080	1.5	135	154	23.9	14.7	100	0.0092	0.00073	0.33	7.74
0.090	5.1	136	156	23.9	14.7	100	0.0094	0.00084	1.12	2.70
0.100	10.0	138	150	23.9	14.7	100	0.0094	0.00094	2.20	1.53

TABLE VII

DATA AT 1100 RPM - INLET PRESSURE 21.9 IN. HG

$\frac{F}{A}$	BL	OIL TEMP.	JACKET TEMP.	P_i ("Hg)	P_e	T_i	AIR CONS.	FUEL CONS.	BHP	BSFC
0.060	6.2	138	152	21.9	14.7	100	0.0081	0.00050	1.49	1.20
0.070	1.4	138	156	21.9	14.7	100	0.0081	0.00057	0.31	6.60
0.080	1.2	138	150	21.9	14.7	100	0.0081	0.00065	0.26	8.80
0.090	5.1	138	154	21.9	14.7	100	0.0081	0.00073	1.12	2.33

TABLE VIII

DATA AT 1000 RPM - INLET PRESSURE CONSTANT

$\frac{F}{A}$	BL	OIL TEMP.	JACKET TEMP.	P_i	P_e	T_i	AIR CONS.	FUEL CONS.	BHP	BSFC
0.050	11.5	142	150	14.7	14.7	100	0.0106	0.00053	2.30	0.83
0.055	3.2	142	150	14.7	14.7	100	0.0103	0.00057	0.64	3.18
0.060	0.8	142	150	14.7	14.7	100	0.0102	0.00061	0.16	13.70
0.065	0.5	140	152	14.7	14.7	100	0.0102	0.00066	0.10	23.80
0.070	0.8	142	150	14.7	14.7	100	0.0101	0.00071	0.16	15.90
0.080	0.5	142	153	14.7	14.7	100	0.0099	0.00079	0.10	28.40
0.090	7.2	142	150	14.7	14.7	100	0.0099	0.00089	1.44	2.23

TABLE IX

DATA AT 1100 RPM - INLET PRESSURE CONSTANT

$\frac{F}{A}$	BL	OIL TEMP.	JACKET TEMP.	P_i	P_e	T_i	AIR CONS.	FUEL CONS.	BHP	BSFC
0.050	9.9	135	150	14.7	14.7	100	0.0117	0.00059	2.18	0.97
0.060	0.5	142	150	14.7	14.7	100	0.0111	0.00066	0.11	21.70
0.080	0.9	147	152	14.7	14.7	100	0.0110	0.00088	0.20	15.90
0.090	4.7	147	150	14.7	14.7	100	0.0110	0.00099	1.03	3.44
0.100	5.5	147	148	14.7	14.7	100	0.0111	0.00110	1.20	3.30

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