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THE MIXTURE REQUIREMENTS OF
AN INTERNAL COMBUSTION ENGINE
AT VARIOUS SPEEDS AND LOADS

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A Thesis Entitled

THE MIXTURE REQUIREMENTS OF AN INTERNAL COMBUSTION ENGINE AT VARIOUS SPEEDS AND LOADS

by

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in

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PREFACE

The investigation herein reported was conducted in the Sloan Automotive Laboratory, Massachusetts Institute of Technology, over the period February 10 to May 1, 1941.

A Ford V-8 eighty-five horsepower engine was used for all of the tests.

We thank Messrs. J.R.Diver, G.B.Wood, jr., C.F. Wood, and W.A.Leary for their many helpful suggestions.

We acknowledge our gratitude for the cooperation, instruction, and guidance of Professor C.F.Taylor, Assoc. Professor E.S.Taylor, and Asst. Professor A.R.Rogowski, of the Massachusetts Institute of Technology, who gave freely of their time and knowledge during the progress of this investigation.

The opinions and assertions contained herein are the private ones of the authors and are not to be construed as official or reflecting the views of the Navy Department or the naval service at large.

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PART I
INTRODUCTION

1. Purpose.

Valuable information on the steady-running mixture requirements of an internal combustion engine, as affected by speed and load, is contained in the report of the classic experiment conducted by Messrs. O. C. Berry and C. S. Kegerreis at the Engineering Experiment Station, Purdue University, in 1920. Since that time many advances have been made in the field of the internal combustion engine, both in engine and accessory design and in operating procedure. Gasoline fuels have been improved and considerably standardized. There is now available much more information on the nature, causes, and effects of detonation than in 1920. It was therefore deemed appropriate to check the conclusions of these experiments using a modern automobile engine and possibly more accurate equipment.

Of particular interest were determination of (a) the maximum economy fuel-air ratios for various speeds and loads, (b) the best-power fuel-air ratios for various speeds and loads, and their possible variations with speed and load, and (c) possible variation in the mixture ratio versus power ratio relation for maximum economy, with change in the basic speed.

2. Laboratory Data Required.

It was decided to make test runs at four speeds,

namely, 1000, 2000, 3000, and 4000 r.p.m. For each speed four series of runs were to be made with constant throttle settings such as to give full, and approximately three-quarters, one-half, and one-quarter power, respectively. This would then give sixteen series of runs, each at constant speed and at the appropriate throttle setting. For each series the fuel-air ratio would be varied from the lean to the rich limits of smooth running, and at arbitrarily selected fuel-air ratios the power output and the specific fuel consumption determined. In all cases the variables of the test would be fuel-air ratio, mean effective pressure, and specific fuel consumption. From such data it is possible to plot curves of mean effective pressure versus fuel-air ratio, and of specific fuel consumption versus fuel-air ratio. These curves may be plotted on both the brake power and the indicated power bases. From these basic results and derived curves the desired information, previously discussed, can be obtained.

3. Theoretical Considerations.

Mixture Distribution. The authors of reference (1) found that the fuel-air ratios for highest efficiency and for highest power are affected by (a) the dryness of the mixture, (b) the quality of the fuel used, and (c) distribution differences between cylinders. These

conclusions have been amply supported in subsequent practice and experiment. Effects of (a) and (c) could be eliminated by the use of a completely dry and homogeneous mixture, and it is believed that such was the case in the investigation here reported. While a dry and homogeneous mixture does not necessarily eliminate any difference in the amount of mixture supplied to the individual cylinders, it should eliminate differences in the quality of the mixture supplied to the various cylinders, and the latter effect alone is of interest in an investigation of this nature.

Detonation Control. It is well known that detonation usually affects power output, the normal effect, with detonation of sufficient intensity, being a reduction in power. It was therefore considered desirable that, if possible, all chance of detonation be eliminated. Reference (3) indicates that a C.F.R. engine, operating under conditions similar to those of the test runs most conducive to detonation, will not detonate with compression ratios of less than 8.5 when 100 octane gasoline is used. Also under these conditions, a C.F.R. engine with compression ratio of 6.3 (that of the engine here involved) will not detonate when fuels of higher than 81 octane number are used. Therefore 100 octane gasoline was used throughout the investigation, and the authors believe that detonation did not occur in any of the test runs.

Thermal and Volumetric Efficiencies. If a series of test runs is made at constant speed and constant throttle setting but at various fuel-air ratios, for proper comparison of the runs both the thermal efficiency and the volumetric efficiency should depend only upon fuel-air ratio and spark advance. This will be the case if the values of all other engine variables are maintained constant. Reference to the data sheets indicates thatsensibly constant values were maintained during each series of runs for coolant inlet temperature, coolant outlet temperature, lubricating oil temperature, mixture inlet temperature, mixture inlet pressure, and exhaust pressure.

Dynamic effects in the induction system will influence volumetric efficiency, but with constant values of speed, throttle setting, and inlet temperature, dynamic effects in the induction system will be constant.

Dynamic effects in the exhaust system will also influence volumetric efficiency but with constant values of all engine variables except fuel-air ratio and spark advance, these effects in the exhaust system will depend only upon fuel-air ratio and spark advance. Since these latter variables will have little effect upon the exhaust temperature, changes in the dynamics of the exhaust system will be negligible.

<u>Friction Horsepower</u>. If the friction power is properly determined for the particular series of runs in question the friction power may be added to the brake power to give indicated power and the results of the series may be compared on an indicated basis. Likewise, series of runs at different speeds and different throttle settings may then be compared on an indicated basis.

The shortcomings of the motoring method of obtaining friction horsepower were recognized. However, because of the prohibitive inconvenience of any more accurate method, the motoring method was used throughout this investigation, in accordance with common practice.

<u>Spark Setting</u>. In an internal combustion engine the time required for combustion results in a loss of area of the indicator diagram with a corresponding loss in output and efficiency. The time of combustion is a function of speed, load, and fuel-air ratio, which are the primary variables in these tests. It is a function also of the pressure, temperature, and exhaust-gas dilution of the fresh charge and several other variables, all of which vary with speed, load, and fuel-air ratio. The variation of time of combustion in these tests was consequently of considerable magnitude.

Loss of output and efficiency due to combustion

time is minimum at best power spark advance. It appears logical to employ this best power spark advance for each experimental point in order to place all experimental data on the most rational basis for comparison. This practice was followed throughout the tests.

Although modern operating practice is not to use best power spark advance under all conditions, when the spark is retarded from this optimum setting it is retarded only sufficiently to limit detonation. Inasmuch as detonation was controlled in these tests by use of 100 octane gasoline, such deviation from the best power setting was not necessary and it is believed that the practice followed herein represents the mode of operation which is most desirable.

4. Laboratory Equipment and its Operation.

Provisions necessary. In conducting the tests it was necessary to provide means of accurately measuring (a) the rate of fuel consumption, (b) the rate of air consumption, (c) the power output, and (d) the friction power. Constant values had to be maintained for (a) the desired speed, (b) the desired throttle setting, (c) the inlet pressure, (d) the exhaust pressure, (e) the inlet temperature, (f) the inlet and outlet coolant temperatures, and (g) the lubricating oil temperature.

Engine. All of the tests were made with a Ford V-8 engine, model of 1935, bore 3-1/16 inches, stroke 3-3/4 inches, displacement 221 cubic inches, compression ratio 6.3, rated at 85 horsepower at 3800 r.p.m. It was equipped as furnished by the manufacturer except where modified as indicated below.

Carburetion. Instead of the carburetor supplied with the engine a large steam-jacketed mixing tank was used. The tank was internally baffled and had a capacity of approximately nine cubic feet. Air was drawn through the tank into the engine. Fuel was discharged into the tank through an adjustable valve and steam-jacketed passage. In this tank the mixture had ample opportunity to become homogeneous, and the steam jacketing allowed the inlet temperature to be maintained at a value which would insure dryness. The fuel valve permitted adjustment of the fuel-air ratio. A sketch of the tank and induction system is shown in Fig. 2.

Ignition. The standard ignition system was used, but the distributor was modified to permit manual adjustment of the spark setting. On the forward end of the engine was attached a disk containing a grounded neon light behind a radial slot. When a graduated arc, secured to but insulated from the frame, was connected to one of the spark plugs, and the disk properly syn-

chronized, the neon light would flash at such a point as to indicate the actua l spark advance. Thus the spark setting was accurately indicated while the engine was running and there was provided a means of making and indicating desired changes.

Fuel Measurement. The fuel system is shown in Fig. 2. While measuring flow rate, fuel was supplied to the mixing tank from a graduated burette. The time for consumption of a volume of gasoline as indicated by the burette was measured by means of an electric stop watch. Upon completion of a timed run, the supply of fuel in the measuring burette was replenished by proper manipulation of the three-way valve. A more detailed description of the fuel system is contained in the appendix.

Air measurement. The rate of air consumption was measured by means of a graduated set of calibrated orifices. Prior to entering the mixing tank the air passed through an air barrel in the entering end of which orifice plates could be mounted. An inclined alcohol manometer indicated the pressure drop between the inside of the barrel and the atmosphere. Calibration charts furnished with the orifices showed the time rate of air flow versus pressure difference.

Power Measurement. Load or motoring power was applied to the engine by means of an electric dynamom-

eter. The stator of the dynamometer was linked to a Fairbanks beam balance, on which restraining force was measured. The dynamometer was manufactured by the General Electric Company, and was rated at 885 amperes at 250 volts (300 horsepower).

Speed Control. Speed was controlled by varying the load. This could be done by varying the armature resistance and the field resistance. Fine control was obtained by a vernier in the field rheostat. Speed was indicated by means of a mechanical tachometer and counter. However, the speed was accurately indicated, for any even hundred r.p.m., by a stroboscope and disk on the crankshaft. The tachometer was used for a rough indication and the stroboscope and field vernier were used for accurate control.

Throttling. Throttling was accomplished by placing a brass plate, containing an orifice of a selected size, in the flange connection between the induction pipe from the mixing tank and the inlet manifold of the engine. This location was chosen to limit to as small a section as possible the low pressures of the inlet manifold, thus minimizing the effects and possibilities of leaks in the induction system.

Inlet Pressure. An adjustable gate valve was placed in the air line between the air barrel and the

mixing tank. This valve was manipulated to maintain the absolute pressure in the mixing tank at a value slightly below that of the lowest barometric pressure expected. This value was 710 millimeters of mercury, which corresponds to an average altitude (for standard atmosphere) of about 2000 feet above sea level. Mixing tank pressure was indicated on a water manometer. The valve gave good control over this small pressure difference and required only occasional attention.

Exhaust Pressure. The exhaust pressure was maintained at an absolute pressure slightly above the highest barometric pressure expected. Pressure was indicated on a manometer and controlled by means of an adjustable valve between the engine and the laboratory exhaust suction line. This valve required only occasional attention.

Inlet Temperature. The inlet temperature was controlled by regulating the amount of steam entering the jacketing space of the mixing tank. Mixture inlet temperature was indicated by a thermometer which was located in the induction pipe between the mixing tank and the inlet manifold.

Coolant Temperatures. The water pumps supplied with the engine were left intact, but instead of the radiator a water reservoir tank of about ten gallons

capacity was used, and adjustable thermostats were placed in the discharge lines. The thermostats gave very accurate control of the temperature of the outlet water. The inlet water temperature was controlled by manually regulating the amount of cold water from the laboratory mains that entered the reservoir, a like amount of warm water overflowing to the drains. This gave very accurate control of the water inlet temperature. A sketch of the cooling system is shown in Fig. 3, and the system is more fully described in the appendix.

Lubricating Oil Temperature. It was found that the oil temperature varied over a range of only a few degrees for a series of runs at any particular speed and throttle setting. At the higher powers it was necessary to keep the coolant temperatures at lower values to prevent the temperature of the oil from exceeding a safe value. At the highest powers it was necessary to direct the blast from one or two portable blowers onto the crankcase.

5. Laboratory Procedure.

Preliminary Runs. In order to determine the sizes of the throttling orifices that would give the desired power ratios it was necessary to make orifice calibration runs. For these runs a set of orifices were prepared. The sizes of these orifices were so selected

that the one set would produce from less than one-quarter power to full power at each of the four test speeds with approximately equal increments of power ratio. With this set of calibration orifices a series of runs was made at each of the test speeds. For each run the fuel-air ratio and spark were set at values to give approximately best power. From this data it was possible to plot curves of throttle orifice diameter versus "best power" ratio for each of the test speeds. From these curves were obtained the orifice sizes necessary to give the power ratios desired for the tests.

Method of Making Record Runs. For a series of test runs the proper throttle orifice was put in place and the engine brought up to the proper speed. Before taking any data, temperatures were allowed to stabilize and the proper adjustments were made to the inlet and exhaust pressures.

Conditions for a run were established by arbitrarily setting the fuel-air ratio through adjustment of the fuel needle valve. Before the taking of data for each run, the spark was set to produce optimum power as indicated by the brake load. When the fuel-air ratio was changed to establish conditions for the subsequent run, sufficient time was allowed for the engine to settle down to the new fuel-air ratio and the spark was set

for optimum power before data was taken. The speed was at all times maintained constant by an observer who had this duty alone.

The fuel-air ratio was varied back and forth between the limits of smooth running. Near the peak of the power versus fuel-air ratio curve the points were taken closer together than in the definitely lean or rich portions of the curve. The fuel-air ratio was arbitrarily changed in either the rich or the lean direction as appeared desirable. In almost all series of runs large changes in the fuel-air ratio were at some time made, but it was found that neither the direction nor the amount of this change affected either the regularity of the measured data or the smoothness of the resultant curves. Runs were continued until the rich and lean limits of smooth running had been reached and points sufficient to give a good curve had been secured.

As each point was obtained it was entered on a laboratory plot of brake load versus fuel-air ratio. This plot indicated when additional runs were necessary to fill in gaps in the curve and when check runs might be advisable, so that these runs could be made while the proper throttle orifice was in place and the operating temperatures and pressures were at the proper values.

Friction Power Measurement. When the last of a series of runs had been completed the ignition switch was cut and the engine was immediately motored to determine the friction power for the series of runs. Because of the constancy of engine operating temperatures throughout a series of runs, the friction power, as measured by the motoring method, was the same for all runs of a series.

Determination of Best Power Spark Setting. In the preliminary runs best power spark setting was determined by holding the load constant and noting the effect of a two degree change in spark setting on speed as indicated by observation of the stroboscopic disk. When a spark setting was found such that a change in either direction resulted in a reduction of speed, it was assumed that best power was being produced.

The dynamometer field was separately excited and liable to fluctuate with any sharp change in the load on the electric system. This fact made possible false indications of the effect of change in spark advance on speed. For this reason the method of setting spark advance during the test runs was changed to the more certain one of actually measuring power output at each spark setting. As many as three passes back and forth

over the best power setting were made, changing the setting in one, two, or three degree increments as appeared desirable. Since the speed was maintained constant during this procedure, tabulation of the brake loads at each spark setting enabled the observer to select the exact spark setting to give best power. This procedure was followed prior to the taking of data for each run.

Lean and Rich Limits Defined. From previous experience and preliminary running of the engine it was known that the limits of smooth running for rich and for lean fuel-air ratios could be extended by use of the proper spark gap in each case. Inasmuch as spark setting is not adjustable in the case of an engine in operation, and in view of the fact that onefixed spark gap setting will provide satisfactory ignition over the most useful range of fuel-air ratios at all speeds and loads, a fixed value of the gap setting was maintained throughout the tests. This gap was .025 inch, the value recommended by the manufacturer for normal operation.

Preliminary operation of the engine showed that even an occasional misfire would so disturb the speed control and so jeopardize the obtaining of a correct brake arm reading, as to place in doubt the accuracy of the data obtained on any run in which missing occurred. Of the two factors mentioned, the effect upon speed was

the most important. Since not only the value of the fuel-air ratio but also the equilibrium value of every temperature and pressure varied with speed, it was found imperative that speed be maintained absolutely constant at the desired value, not only while taking data on a record run, but also at all times during the progress of a series of runs.

In view of the critical effect of even minor speed fluctuations no runs were made at values of the fuel-air ratio for which running was not sufficiently smooth to ensure the obtaining of accurate readings. Therefore, the limits of smooth running at both ends of the useful range of fuel-air ratios, as found in these tests, are truly the limits of smooth running. In each case the limit is such that if the mixture ratio is enriched (or leaned) theslightest amount, running of the engine will become so erratic as to make questionable the accuracy of data.

Values of Coolant and Oil Temperatures. It was not possible to keep coolant inlet and outlet temperatures and oil temperature at the same value for the various series of runs. However, these temperatures were maintained at sensibly constant values for all runs of a series at a particular speed and particular throttle setting, and were carried at values reasonably

close to those which one might expect in the operation of a modern engine at the outputs in question.

Determination of Fuel Specific Gravity. The main fuel barrel was refilled from time to time during the course of the investigation. At each refilling a sample of the gasoline was drawn off over water, its temperature measured, and its specific gravity determined by means of a hydrometer. It was found, throughout the period of the test runs, that the specific gravity of the fuel remained constant when referred to a standard temperature of 73° Fahrenheit, and the specific gravity-temperature relationship for the fuel used was determined experimentally to be:

$$\text{SPECIFIC GRAVITY} = 0.699 - 0.00046(^\circ F - 73)$$

The temperature of the fuel was then taken for each run, the specific gravity calculated from the above formula, and this value then used for the determination of the fuel rate.

Photograph on following page.

A. Dynamometer control panel.

B. Beam balance.

C. Fuel measuring burette and accumulator tank.

D. Air measuring orifice in air barrel.

E. Fuel thermometer.

F. Mixture inlet thermometer.

G. Air inlet check valve.

PLATE I

General view of apparatus

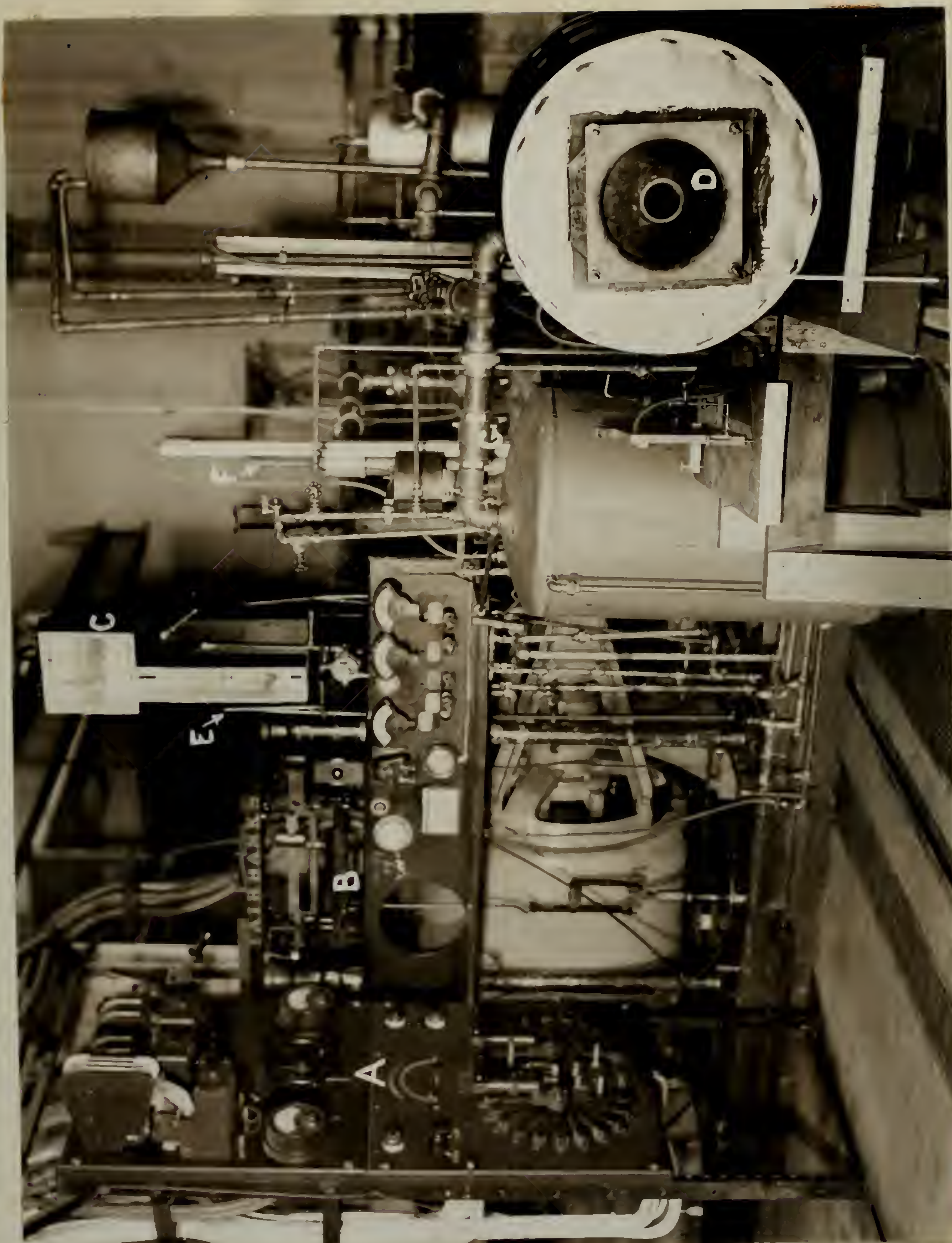


PLATE I

General view of apparatus

Photograph on following page.

A. Engine control panel showing, left to right, top row: ignition switch, oil temperature gage, water temperature gage (left bank), water temperature gage (right bank; bottom row: fuel pressure gage, cooling water valves.

B. Mixing tank.

C. Air barrel.

D. Air Barrel manometer.

E. Fuel needle valve. Line from this valve to mixing tank shows steam jacketing.

F. Inlet air pressure control valve.

G. Mixing tank pressure relief valve.

H. Three-way fuel valve.

I. Exhaust pressure control valve.

J. Fuel thermometer.

K. Mixture inlet thermometer.

L. Fuel sampling line.

PLATE II

Front view of apparatus

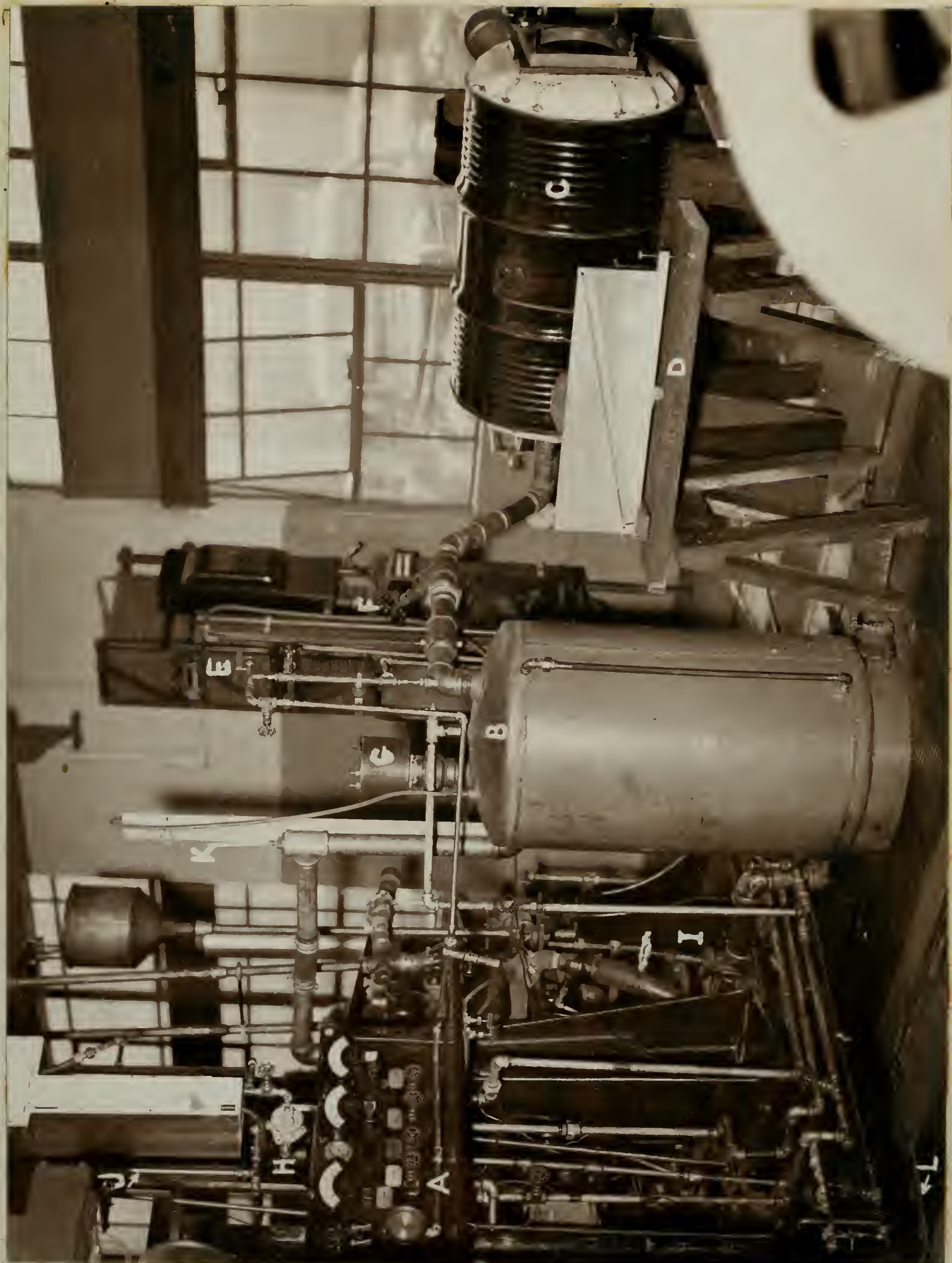


PLATE II

Front view of apparatus

Photograph on following page.

A. Engine.

B. Throttling orifice flange.

C. Water outlet thermostats.

D. Water reservoir tank.

E. Dynamometer reduction gear.

F. Fuel pump and motor.

G. Fuel pressure regulating valve.

H. Exhaust pressure manometer.

I. Inlet pressure manometer.

J. Electric lead from spark plug to spark protractor.

PLATE III

Rear view of apparatus

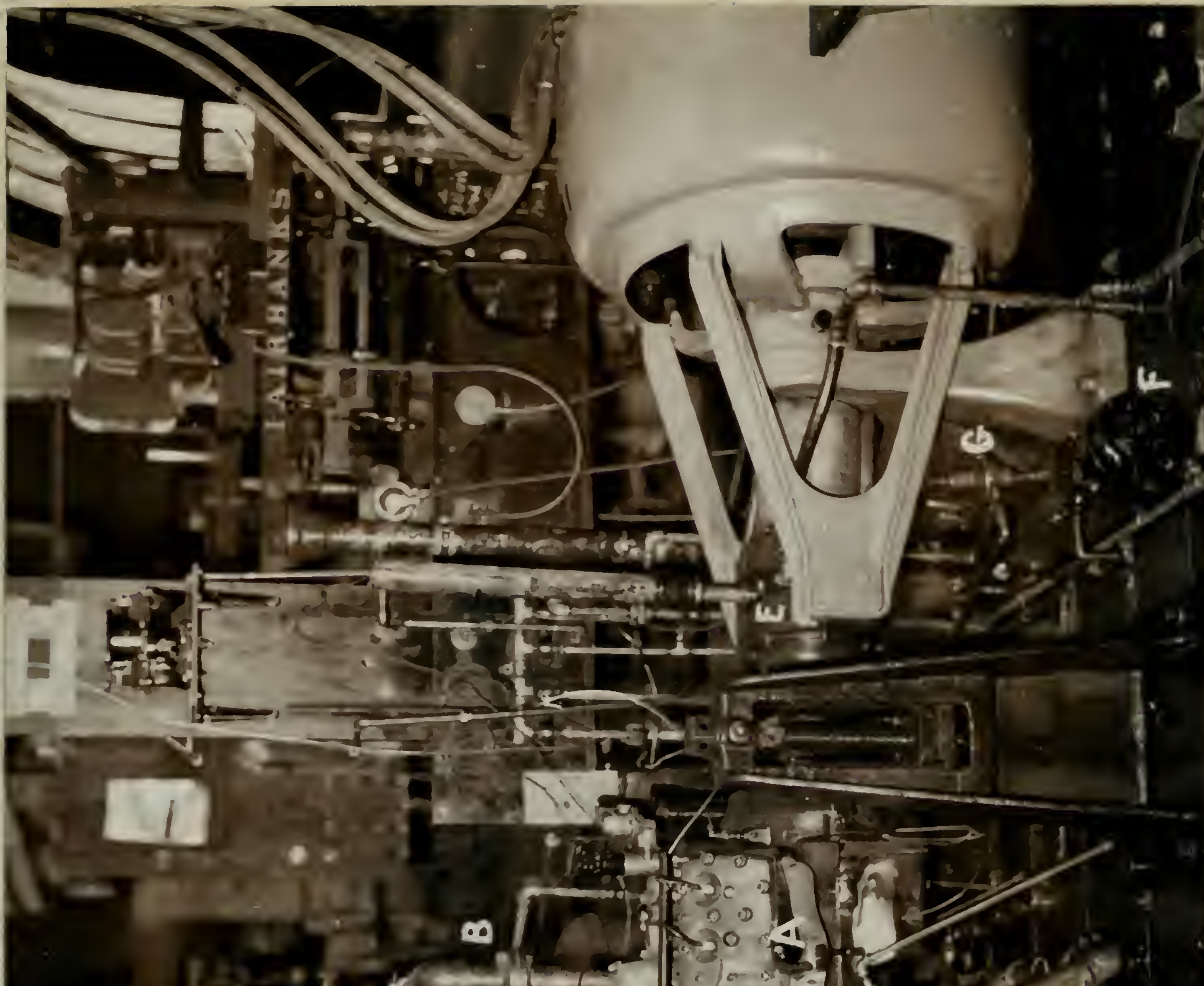


PLATE III

Rear view of apparatus

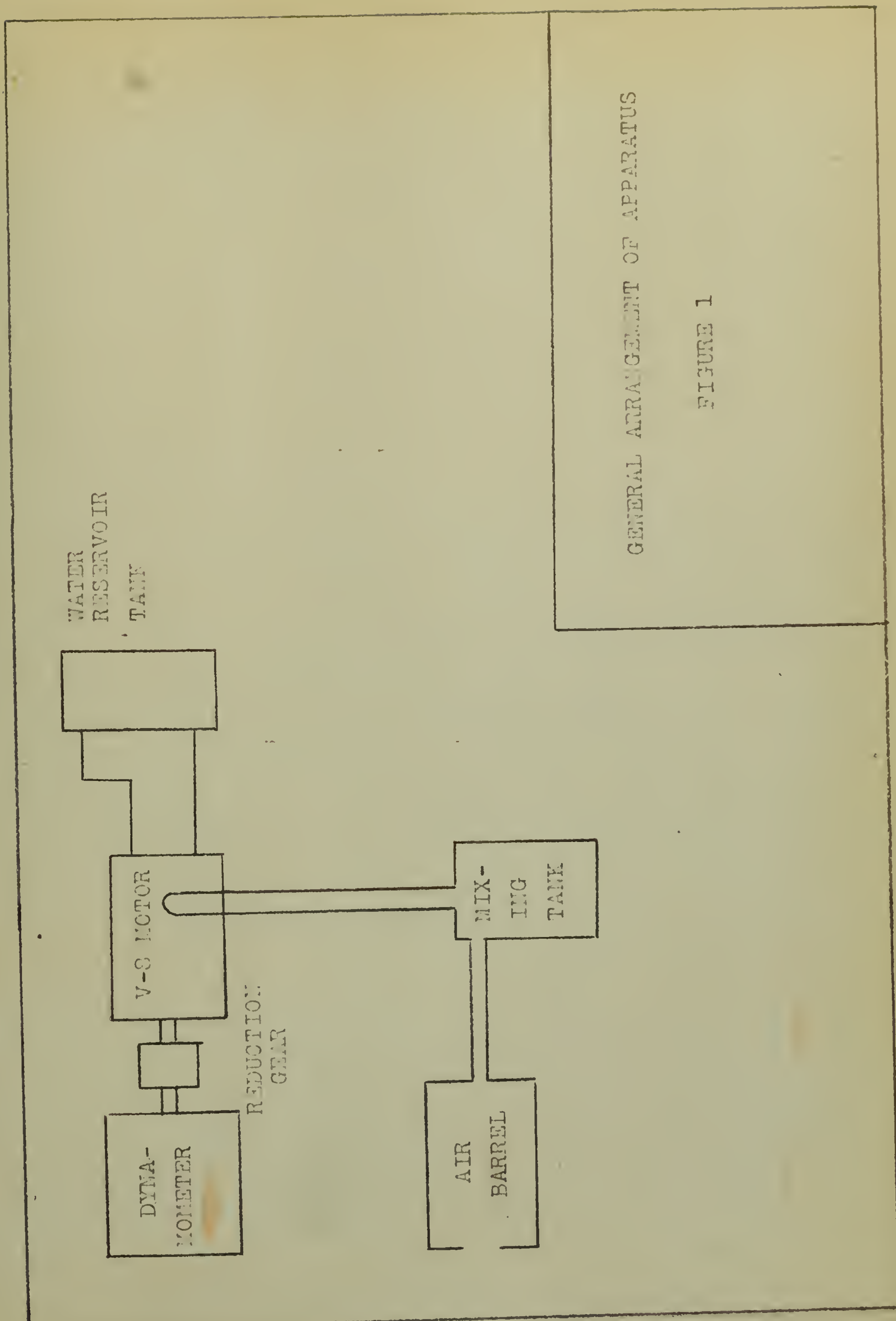
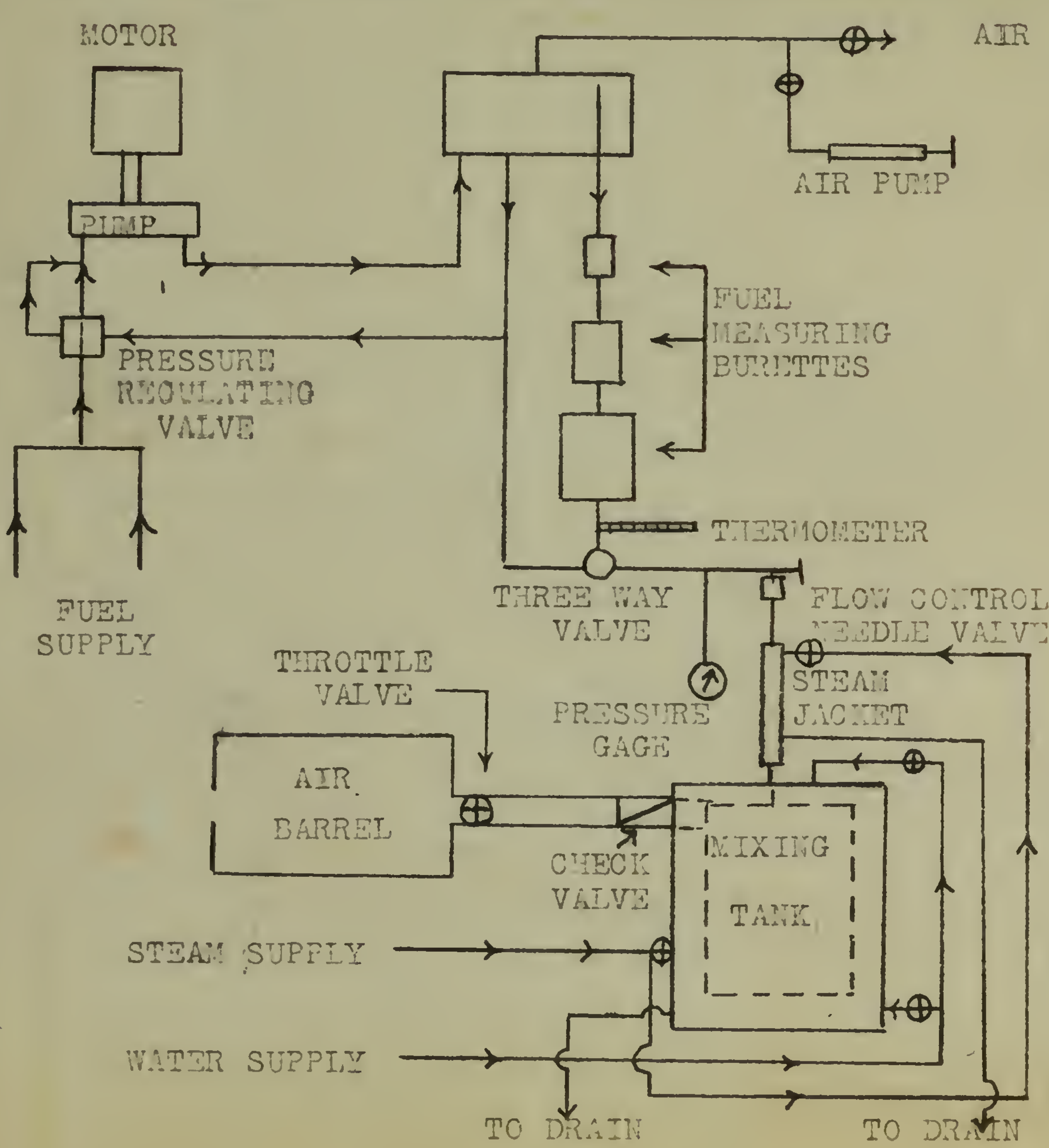

GENERAL ARRANGEMENT OF APPARATUS
FIGURE 1
WATER RESERVOIR TANK
V-8 MOTOR
MIX-ING TANK
REDUCTION GEAR
DYNA-MOMETER
AIR BARREL

FIGURE 1



CARBURETION SYSTEM

Showing fuel flow control and measuring apparatus

FIGURE 2

FIGURE 2

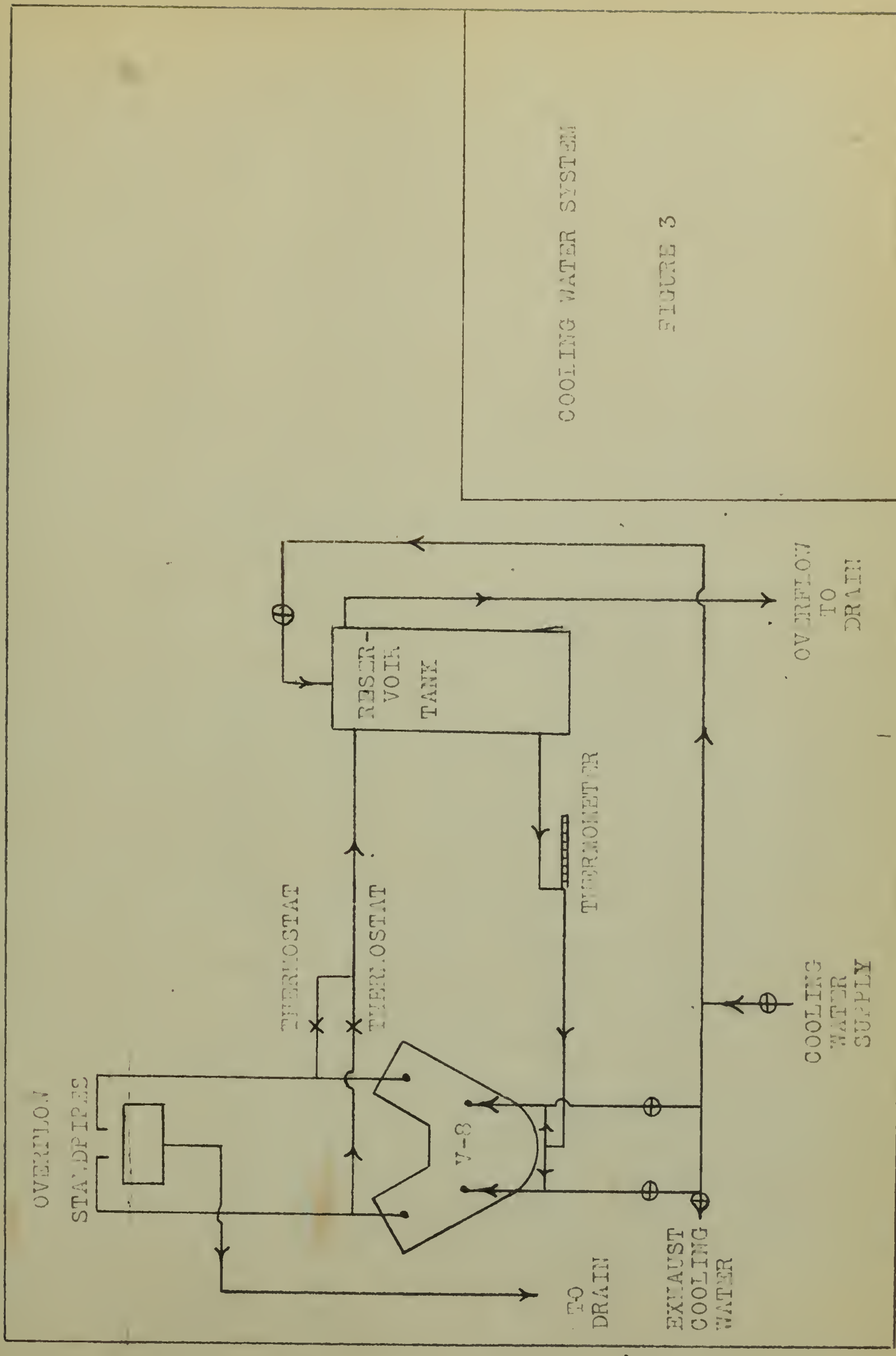
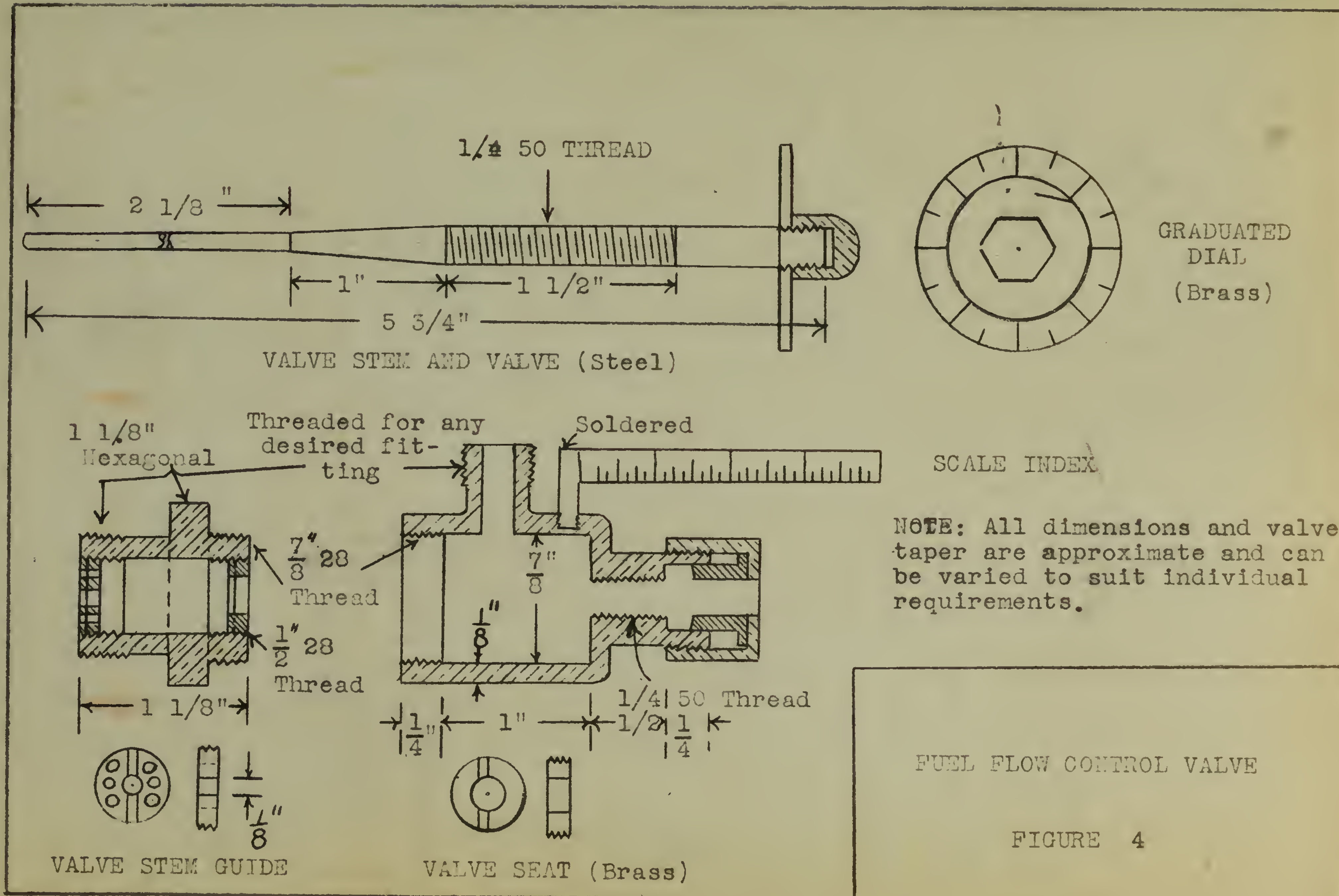
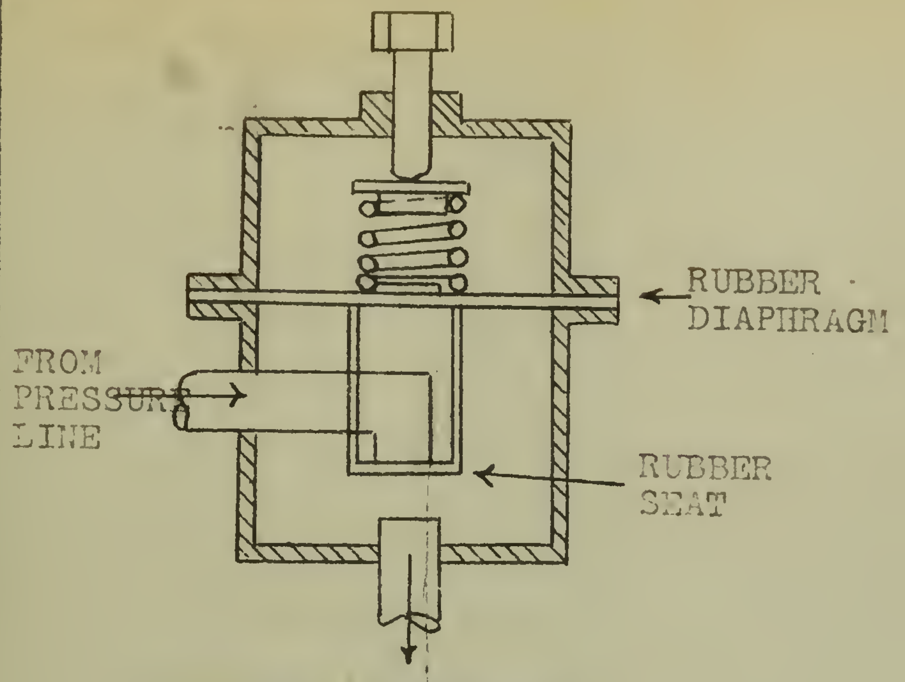

COOLING WATER SYSTEM
FIGURE 3
OVERFLOW TO DRAIN
RESER-VOIR TANK
THERMOMETER
THERMOSTAT
THERMOSTAT
OVERFLOW STANDPIPES
V-8
TO DRAIN
EXHAUST COOLING WATER
COOLING WATER SUPPLY

FIGURE 3

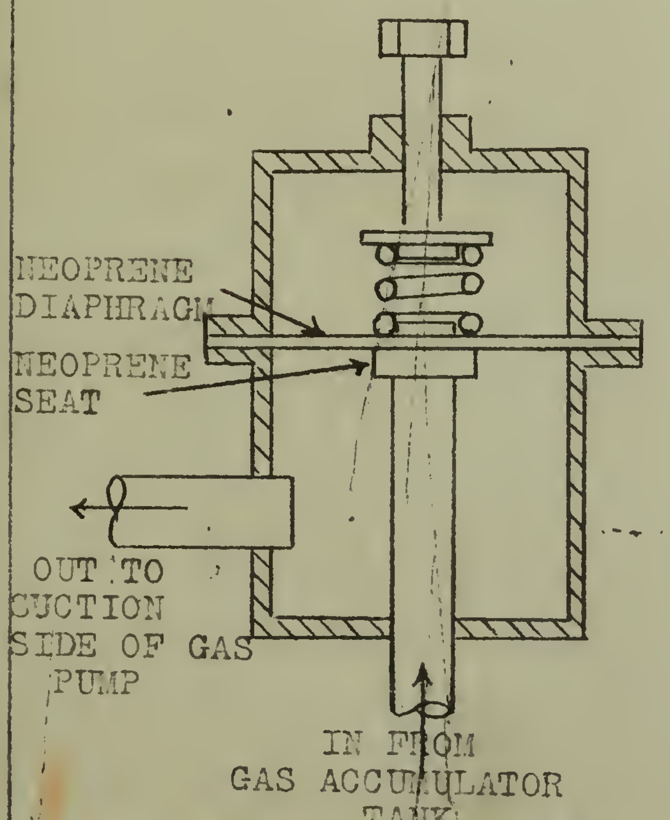


1/4 50 THREAD
2 1/8"
1"
1 1/2"
5 3/4"
VALVE STEM AND VALVE (Steel)
GRADUATED
DIAL
(Brass)
1 1/8"
Hexagonal
Threaded for any
desired fit-
ting
Soldered
SCALE INDEX
7/8" 28
Thread
1/2" 28
Thread
7/8"
1/8"
1 1/8"
1/4"
1"
1/2
1/4 50 Thread
1/4
1/8"
VALVE STEM GUIDE
VALVE SEAT (Brass)
NOTE: All dimensions and valve taper are approximate and can be varied to suit individual requirements.
FUEL FLOW CONTROL VALVE
FIGURE 4

FIGURE 4



ORIGINAL REDUCING VALVE DESIGN



MODIFIED PRESSURE CONTROL VALVE

PRESSURE CONTROL VALVE

Showing modification of original reducing valve design to give accurate control of pressure in fuel system.

FIGURE 5

PART II
RESULTS

1. Accuracy.

Fuel Measurement. The rate of fuel consumption was determined by measuring the time for a known quantity of fuel to be consumed. As the fuel level was lowered in the measuring burette the watch was started and stopped when the level passed calibration marks on the burette necks. These necks were of small diameter and repeated tests showed the error in measurement of fuel quantity to be negligible.

The burette volumes were checked independently by two different observers on different days. One observer measured volumes by filling the bulbs with hundred octane gasoline whose specific gravity was measured. The burettes were then weighed before and after addition of fuel. This was done both filling and emptying to take into account the wetting of the glass surfaces. The second observer checked these results by filling the bulbs with distilled water from an accurately calibrated graduate. These independent determinations checked to within 0.1 percent.

The brake reading for each run was taken after all conditions had stabilized, and with the fuel level at its normal point in the accumulator tank. During

the measurement of rate of fuel consumption the actual head of gasoline on the discharge line to the mixing tank was decreasing so that the rate of fuel flow was less and the measured mixture ratio leaner than at the time of power measurement. Investigation of this error established that the maximum change in head amounted to about 0.3 pound per square inch (the average difference during the course of a run being somewhat less than this). Accordingly fuel flow was measured at high and low rates when the pressure was that used in the test runs, and again at the same needle valve settings when the fuel pressure had been reduced by 0.3 pound per square inch. This test showed a maximum error of 2 percent which is within the limits of experimental accuracy. The authors therefore believe that this error does not affect the validity of the results.

Preliminary running established best-power fuel-air ratios at about 0.065 to 0.070, values unexpectedly low. The authors at first suspected the accuracy of timing, and made tests of the stop-watch and of the timing procedure described elsewhere. Finding no important error there, the authors suspected the possibility of fuel leakage through the three-way valve, although this did not seem probable with the maximum pressure differential across the valve only 0.3 pound per square inch. However, an additional gate valve was installed

in the fuel line adjacent to the three-way valve and throughout the period of running occasional runs were immediately repeated with both gate and three-way valves closed, thus insuring fuel supply from only the calibrated burette. No error from this source was found.

The question of inaccuracies resulting from the small amounts of fuel measured in the tests was investigated by preliminary runs in which varying volumes of fuel up to the full capacity of the burette were used. Regardless of the volume used there was no change in the measured fuel rate.

Time Measurements. Time was measured by an electric stop-watch whose accuracy was carefully checked. Readings could be estimated to 0.01 second and errors from this source are considered negligible.

Air Flow Measurements. Air flow was measured by admitting air through one of a series of sharp-edged calibrated orifices long in use in the Sloan Automotive Laboratory for this purpose. Pressure drop from atmosphere to the inside of the air barrel was measured by an alcohol manometer with a slant height of ten centimeters for each inch rise, thus increasing the sensitivity about four times, and permitting the pressure drop to be read accurately to 0.01 inch of alcohol.

It is believed by the authors that the accuracy of calibration of the air measuring orifices was considerably greater than the accuracy with which the manometer could be read. An error of 0.01 inch in manometer reading would produce a maximum error in calculated fuel-air ratio of about 0.1 percent.

After the induction system was set up it was tested under air pressure as one unit from the air orifice to the throttle orifice. This included all of the system except the intake manifold and six inches of pipe bolted to it. All seams, joints, and connections were painted with soapy water to disclose several small leaks which were then eliminated. Under a pressure of ten inches of mercury there was a drop of 0.1 inch in ten minutes. This pressure difference was about five times as great as any imposed during the experiment.

The portion of the system from the throttle orifice to the intake valves was made as airtight as the rest of the system, but not tested under pressure.

Since any air leakage would have produced measured fuel-air ratios richer than actually existing, and since all data of this experiment indicate maximum power at mixtures much leaner than found by previous investigators, (Ref. (1)), it is believed that the air leakage in this experiment was negligible.

Temperature Measurement. All thermometers could be read to within one degree. Water outlet and oil temperatures were measured with bulb thermometers, all other temperatures by mercury-in-glass thermometers placed with their bulbs centered in the fluid streams to be measured.

Speed Control. Speed was controlled at all times by an experienced observer using a vernier rheostat in the dynamometer field. Speed variation was observed by means of a stroboscope and striped disk on the engine shaft, so divided that its pattern remained visibly stationary at every even hundred r.p.m. Speed was checked as necessary by use of the electric stop-watch and a revolution counter. It was possible to control speed so accurately that errors from speed changes were negligible.

Power Measurement. The dynamometer had a power absorbing capacity of about three hundred horsepower; much more than required. Pedestal bearings were carefully freed so that static friction of the dynamometer stator was less than could be measured by the beam balance. Windage loss of the dynamometer was considered negligible. The beam balance could be read to within one tenth of a pound under smooth running conditions

with an experienced observer controlling speed. At rich and lean limits of running, motion of the scale arm was erratic and no data were taken beyond or below fuel-air ratios where accuracy of balance readings became less than described above.

Detonation Power Loss. Reference (3) indicates that under the most severe conditions of the test, 100 octane gasoline should not detonate. As discussed in the introduction, it is believed that there was no detonation under any conditions of the investigation.

Throttle Opening. As discussed elsewhere, throttle opening was constant, the actual throttle consisting of a hole drilled and carefully machined in a brass disk bolted between gaskets and the flanges at the juncture of the inlet pipe and the engine intake manifold. This system was similar to that employed in the experiments of reference (1).

Friction Horsepower. This was determined by motoring the engine. In the shortest possible time after cutting the ignition, the engine was motored by the dynamometer, and the beam balance read. This reading was obtained in each case about one minute after the ignition switch had been opened. Temperatures and pressures were held as closely as possible to values existing during the preceding run. The regularity

of friction horsepower measurements is shown by the curves of Fig. 34.

Water Jacket Temperature Control. Preliminary running established the fact that it would be necessary to exercise careful control of jacket temperatures. When outlet temperatures and rate of circulation of cooling water were controlled by automatic thermostats, with no control being exercised over water inlet temperature, the variation in friction horsepower, and consequently in speed, was so great and so erratic that accurate speed control was impossible. The cooling system was then modified to produce the following conditions: (a) outlet temperature maintained constant by automatic thermostat, (b) inlet temperature maintained constant by manual control, (c) rate of circulation of cooling water varied by means of thermostats as required by variation in amount of heat rejection. It is considered that errors from variation in friction horsepower and speed produced by the cooling system were a minimum under these conditions.

As the recorded data indicate, the water inlet temperature was maintained within four degrees of a predetermined constant value. When operating conditions were radically changed (usually in connection with a radical change in fuel-air ratio), accurate control of water inlet temperature was temporarily lost. At such

times the operations of obtaining best-power spark setting and of making test runs were discontinued until an equilibrium condition had been reestablished in the cooling system.

At the lower speeds the thermostats functioned very satisfactorily in maintaining constant outlet temperature and in varying the rate of circulation as required. At 4000 r.p.m. it was necessary to adjust the thermostats to their lowest temperature setting to maintain lubricating oil temperature at a reasonable value. At these settings the sensitivities of the thermostats were reduced. The curves obtained at 4000 r.p.m. indicate that this condition did not have any detrimental effect.

Spark Setting. Spark advance was set within one degree of the correct best-power value in every case. Because of the small change in b.m.e.p. with spark advance near the correct value, this small variation of one degree introduced negligible error.

Overall Accuracy. It is believed that any accumulation of errors would have resulted in considerable scattering of observed points. The regularity of plotted points, as shown on the curves of this report, indicate that the experimental error was very small.

2. Presentation of Results.

The results of the investigation are presented in graphical form in figures 6 to 37, inclusive. The curves on the brake basis are first in order followed by curves on the indicated basis, with the exception that the curves of maximum economy fuel-air ratio versus brake power ratio and versus indicated power ratio, are the last two figures presented. The throttle orifice calibration and friction power curves follow the indicated basis general curves.

3. Discussion of Results.

Brake Power Basis. Examination of the curves of mean effective pressure versus fuel-air ratio shows that maximum power, for all speeds and loads, occurs at a substantially constant fuel-air ratio of about 0.07. In individual curves variation from this value may be accounted for by choice in fairing in the curves and experimental error. However, such variations as do exist are of small magnitude and indicate no systematic trends.

Theoretical analysis of the fuel-air cycle indicated that for a compression ratio of 6.3 maximum power occurs at a fuel-air ratio of 0.0715. The value found in the investigation is in close agreement, and the difference may be attributed to experimental error. The close agreement seems to support the belief that a dry

and homogeneous mixture was used and that distribution was good.

If the mixture ratio is leaned from that for best power, the power falls off at a greater rate than if the mixture is enriched.

It is also seen that the rich limit of smooth running decreases with increase in the reference speed, and that fuel-air ratio adjustment has more pronounced effect on the power output at 4000 r.p.m. than at the lower speeds. During the progress of the experimental runs it was also found that spark setting had more critical effect on power output at higher than at the lower speeds.

Figures 10 to 13 show that minimum specific fuel consumption always occurs at a value of fuel-air ratio leaner than that for best power. This minimum, for all speeds, occurs at leaner fuel-air ratios with increase in the power ratio, and the rate of change of the minimum point with change in the power ratio is about the same for all speeds. At zero power ratio (idling conditions) best power and minimum specific fuel consumption occur coincidentally at only the best power fuel-air ratio.

The increase in brake specific fuel consumption with throttling is due to the fact that at lowered power ratios friction power becomes a greater percentage of indicated power.

Fig. 18 shows curves of minimum brake specific fuel

consumption versus speed for four power ratios. It was obtained by interpolating as necessary between the curves of brake specific fuel consumption so as to give in all cases the same power ratio. It is interesting to note that brake specific fuel consumption is substantially constant for all speeds up to 3000 r.p.m. (piston speed 1875 feet per minute), beyond which point it increases with speed. This may be accounted for by the almost linear variation of both air capacity and friction power with speed up to about 3000 r.p.m., as found by previous investigators using this same engine.

This relation between air capacity and friction power may be shown as follows;

$$BSFC \sim \frac{\text{Fuel rate}}{IHP - FHP}$$

At constant F/A -- Fuel rate $\sim$ Air Capacity

Assuming that IHP $\sim$ AC

$$\text{then } BSFC \sim \frac{AC}{AC - FHP} = \frac{1}{1 - \frac{FHP}{AC}}$$

If AC $\sim$ Speed

and FHP $\sim$ Speed

$$\text{then } \frac{FHP}{AC} = K, \text{ and}$$

$$BSFC \sim \frac{1}{1 - K} = K'$$

Beyond 3000 r.p.m. air capacity increases at less than, and friction power at greater than the linear rate, accounting for the increase in brake specific fuel con-

sumption beyond this point. This relation between air capacity and friction power with change in speed is also shown by the curves of brake specific air consumption versus speed, Fig. 19. These latter curves are for best power fuel-air ratio and the data was interpolated where necessary so as to give constant power ratios.

The curves of brake specific fuel consumption versus brake mean effective pressure are shown in figures 14 to 17, inclusive, and curves for the same speed but different power ratios are shown on the same sheet. Tangents to these curves, as drawn, then indicate most economical operation at the speed in question. As determined by operating conditions at these points of tangency a curve of maximum economy fuel-air ratio versus power ratio was obtained. This curve is shown in Fig. 36. Since the points of tangency were so ill-defined and the selection of the points a matter of considerable personal choice, the tangent curves were displaced 5 percent in the direction of greater specific fuel consumption to obtain more certain intersections and consequent determination of operating conditions. The resultant curve, as so determined, is also shown in Fig. 36. The relative displacement of these two resultant curves shows that nearly maximum economy can be obtained over a fairly wide range of fuel-air ratios. This range is greatest at the low power ratios, decreasing as the pow-

er ratio increases.

Indicated Power Basis. The curves of indicated specific fuel consumption versus fuel-air ratio, figures 24 to 27, show higher consumptions for the throttled conditions. This difference is most apparent at 1000 r.p.m., decreasing with increase in speed to be almost absent at 4000 r.p.m. This trend is shown in Fig. 32 which presents curves of minimum indicated specific fuel consumption versus speed. The trend may be accounted for by the greater heat loss to the coolant when throttled due to increase in time of combustion of a diluted mixture, and also by the theoretical loss in efficiency with increased dilution as predicted by fuel-air cycle analysis. (Reference (2) page 41) Heat loss caused by increase in time of combustion is almost proportional to the reciprocal of the speed, and is much less pronounced at the high speeds.

This curve also shows a decrease in indicated specific fuel consumption with increase in speed. This trend is due to the decreased time available for heat loss per cycle with increase in speed, since combustion rate increases almost in proportion to speed. The net result is an increase in efficiency with increase in speed. This trend is also shown by the curves of indicated specific air consumption versus speed for four power ratios, Fig.

33. This curve shows indicated specific air consumption at best power fuel-air ratio. The data was interpolated as necessary so as to give constant power ratios.

This increase in efficiency with speed might be attributed to errors in the determination of friction power. However, assuming that indicated power is directly proportional to air capacity:-

$$IHP_{1000\ full} = 26.7$$

$$\therefore\ IHP_{4000\ full} = 26.7 \times \frac{.191}{.0545} = 93.5$$

$$BHP_{4000\ full} = 58.1$$

$$\therefore\ FHP_{4000\ full} = 93.5 - 58.1 = 35.4$$

$$FHP_{4000\ full}\ \text{as measured} = 56.7$$

Therefore the measurement of FHP would have been

$$\frac{56.7 - 35.4}{35.4} = \frac{21.3}{35.4} = 60\%\ \text{in error}$$

if it were true that IHP is directly proportional to air capacity. This error seems entirely unreasonable, particularly since the values of friction horsepower as found agree very closely with those found by previous investigators using the same engine, and the curves of friction power versus power ratio and speed, Fig. 34, show the determination to have been fairly consistent.

Both best power fuel-air ratio and change in indicated efficicency with speed closely follow theory which further shows that the mixture was dry and homogeneous and that combustion and distribution were good.

The curve of best economy mixture versus indicated power ratio is shown in Fig. 37. Only one curve is presented since no systematic variation with speed is indicated, and the scattering of points is considered to be due to experimental error.

It is to be supposed that the experimental error on the indicated basis will be larger than on the brake basis, since in reducing the data to the indicated basis the friction power must be used, the measurement of which introduces a source of additional error not present in the analysis of the data on the brake basis.

Theoretical Treatment of Trend in Indicated Efficiency . The trend in indicated efficiency may be explained on theoretical grounds as follows:-

$$\text{Ind. eff.} = \frac{33{,}000 \times 60}{778 \times \text{ISAC} \times F/A \times 18{,}900}$$

For the full power condition

$$\text{ISAC}_{4000 \text{ r.p.m.}} = 6.0$$

$$\text{ISAC}_{1000 \text{ r.p.m.}} = 7.4$$

$$\text{Ind. eff.}_{4000 \text{ r.p.m.}} = \frac{33{,}000 \times 60}{778 \times 6.0 \times .07 \times 18{,}900} = 32.1\%$$

$$\text{Ind. eff.}_{1000 \text{ r.p.m.}} = \frac{33{,}000 \times 60}{778 \times 7.4 \times .07 \times 18{,}900} = 26.0\%$$

Theoretical fuel-air cycle efficiency for F/A .07 and compression ratio 6.3 is 37.5%.

$Q \sim K \Delta T A (\rho s)^n$

where Q = heat transfer, BTU/min.

ΔT = average temperature difference

A = area exposed

ρ = density of gases

s = piston speed, ft./min.

n = empirical coefficient

A is constant and ΔT and ρ are sensibly constant.

To the first approximation, disregarding blowdown losses, etc., and assuming that the change in efficiency is due entirely to direct heat losses:-

Heat loss to exhaust, theoretical cycle,

$\sim (1 - .375) = .625$

Total heat loss, 4000 r.p.m., per cycle,

$\sim (1 - .321) = .679$

Total heat loss, 1000 r.p.m., per cycle,

$\sim (1 - .260) = .740$

Direct heat loss per cycle, 4000 r.p.m.

$\sim (.679 - .625) = .054$

Direct heat loss per cycle, 1000 r.p.m.

$\sim (.740 - .625) = .115$

So, $\frac{.054 \times 4}{.115} = (4/1)^n$

$n = .455$

This approximate value of the exponent "n" is reasonable and agrees with general theory and practice.

3. Conclusions.

(1) The fuel-air ratio for best power does not depend on speed or load, but is a constant for all speeds and loads.

(2) Best power occurs at a fuel-air ratio about equal to that of the fuel-air cycle for the same compression ratio (.07 for the engine of the test) when the mixture is dry and homogeneous and the distribution good.

(3) The fuel-air ratio for maximum economy depends on the power ratio and varies as is shown in Fig. 36. There was no indication that this relation changes with change in the reference speed.

(4) For this engine brake efficiencies are substantially constant for a given power ratio up to the speed to which air capacity and friction power vary in linear fashion with speed. Beyond this speed, where air capacity increases at less than and friction power at greater than the linear rate, brake specific fuel consumption for a given power ratio increases with increase in speed.

(5) Indicated efficiency decreases with throttling, this effect being most pronounced at low speeds and practically absent at speeds close to the rated.

(6) Indicated efficiency increases with increase in speed.

(7) The maximum economy fuel-air ratio as determined by indicated power ratio was shown not to vary with change in the reference speed.

(8) With a dry and homogeneous mixture and good distribution, indicated performance as affected by fuel-air ratio, power ratio, and speed may be closely predicted on theoretical grounds.

(9) An engine is more critical to proper adjustment of the fuel-air ratio and spark at higher than at lower speeds.

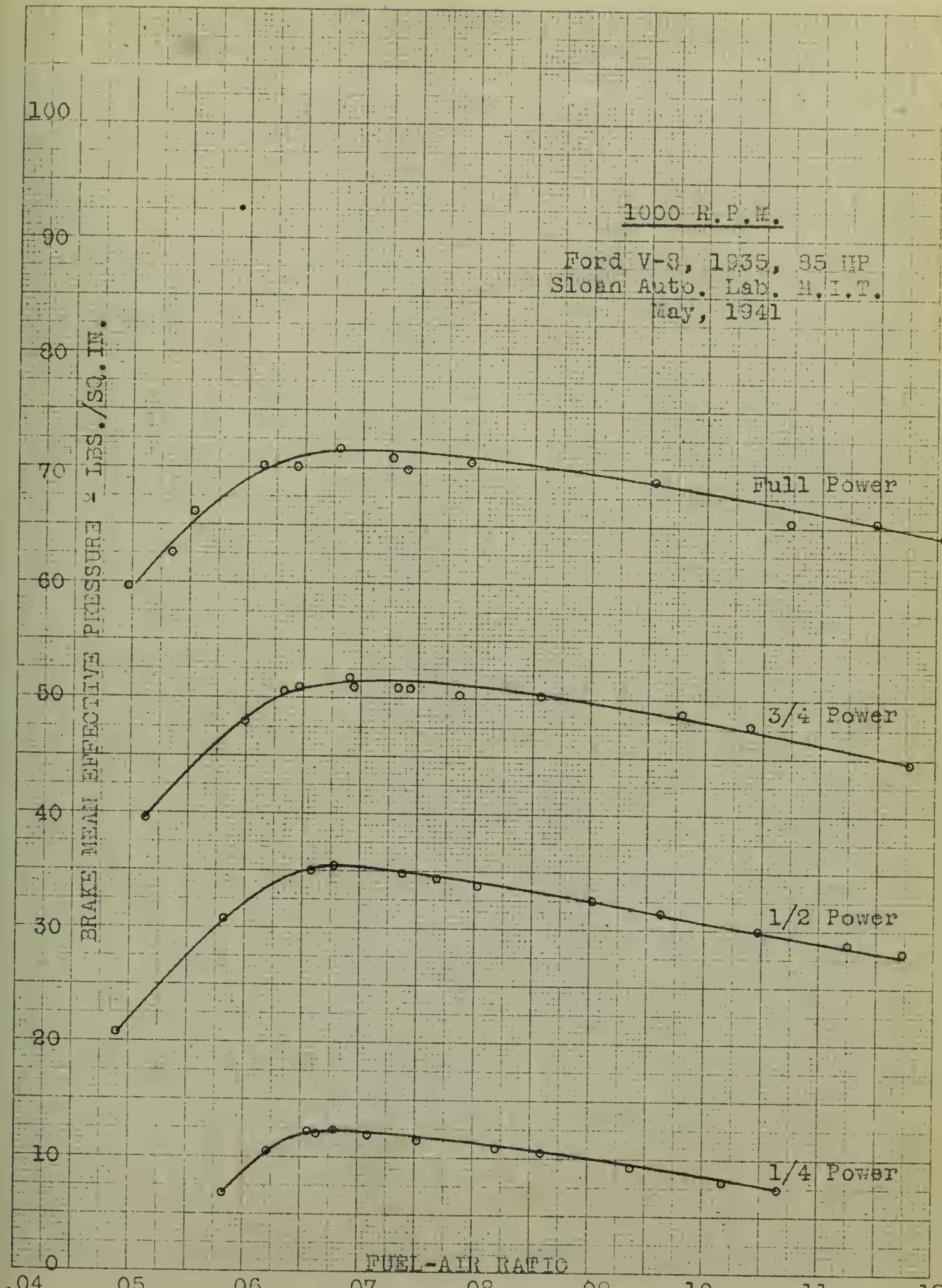


FIGURE 6
Variation of BMEP with F/A - 1000 R.P.M.

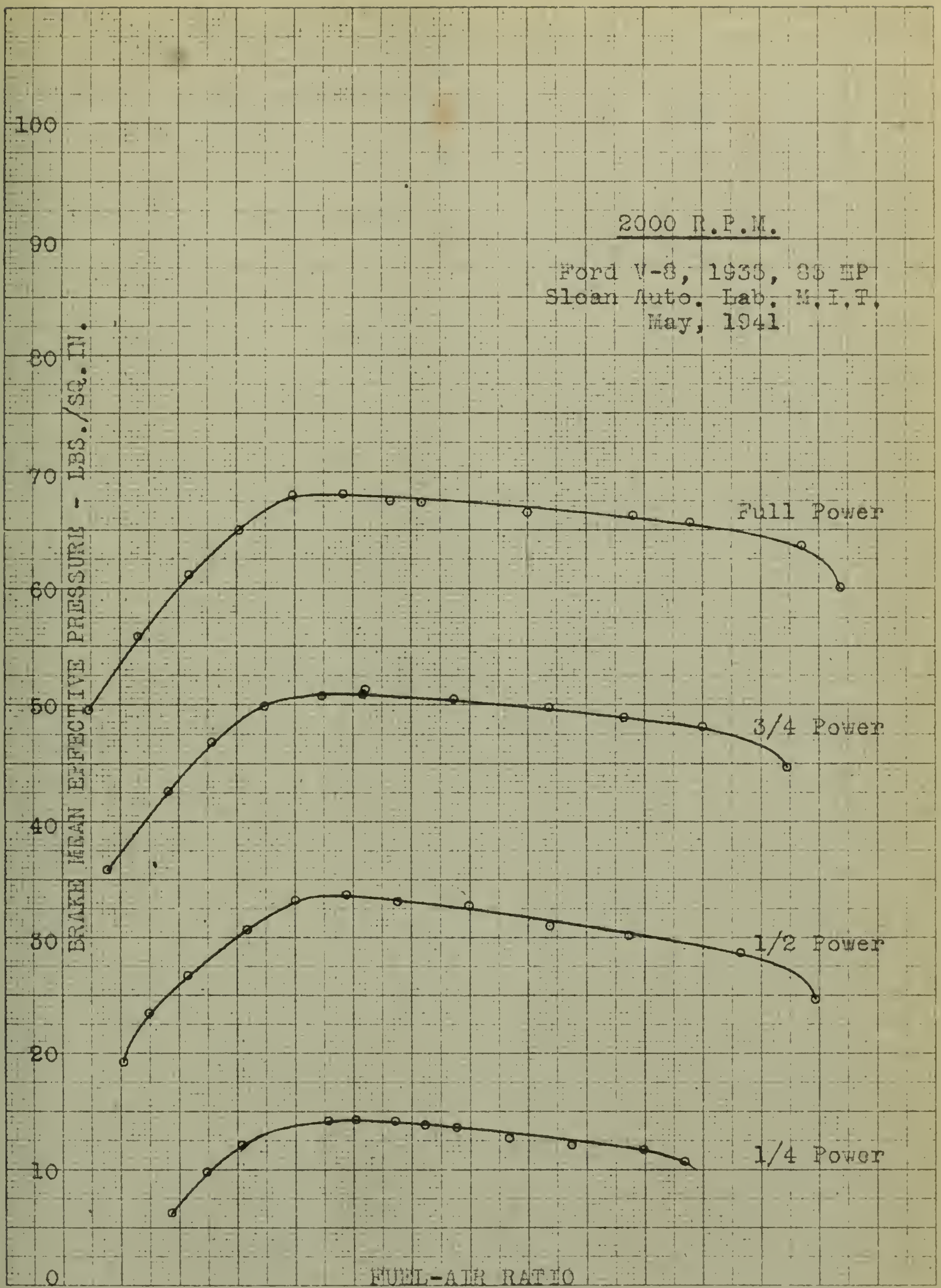


FIGURE 7
Variation of BMEP with F/A - 2000 R.P.M.

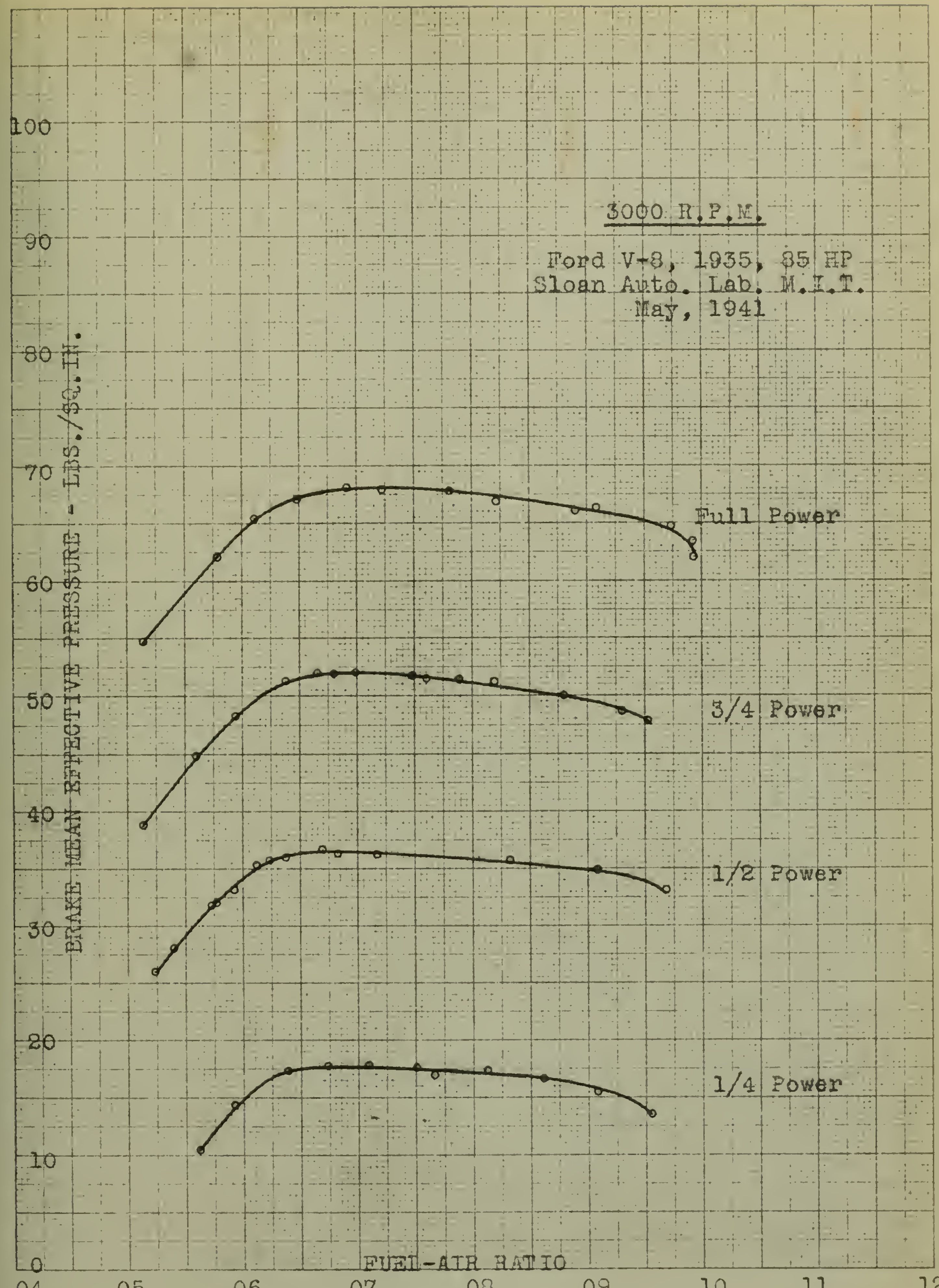


FIGURE 8
Variation of BMEP with F/A - 3000 R.P.M.

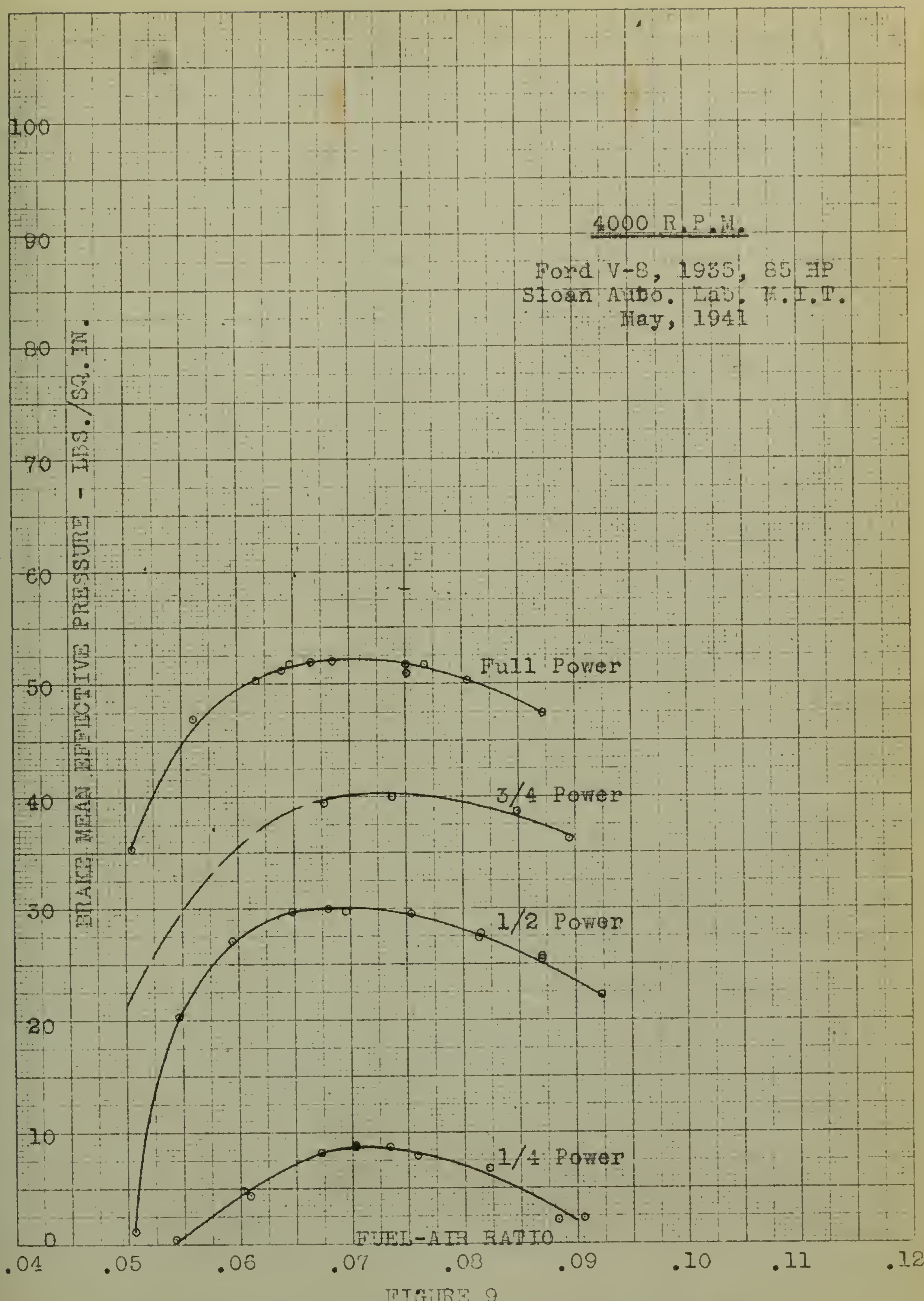


FIGURE 9
Variation of BMEP with F/A - 4000 R.P.M.

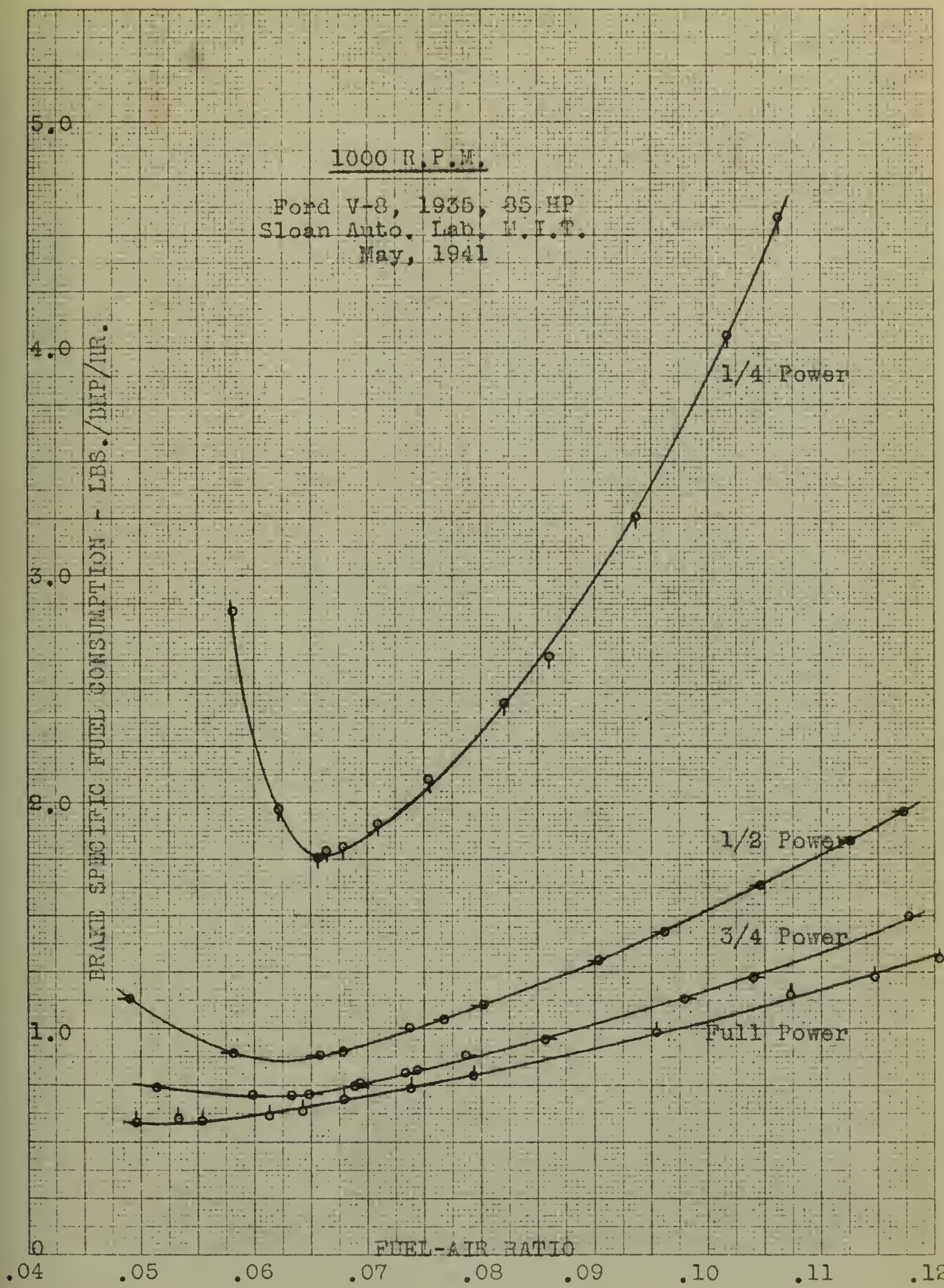


FIGURE 10

Variation of BSFC with F/A - 1000 R.P.M.

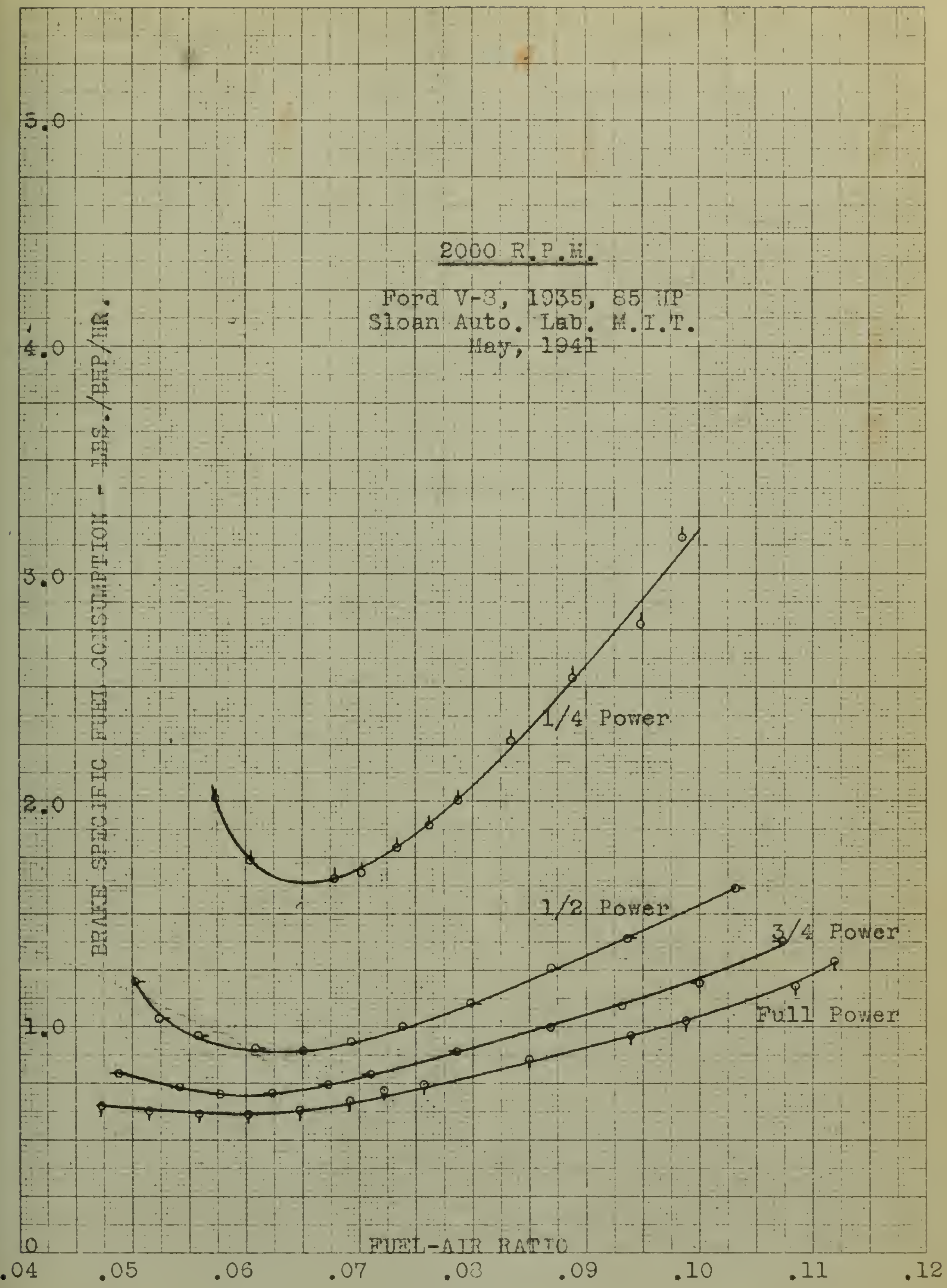


FIGURE 11
Variation of BSFC with F/A - 2000 R.P.M.

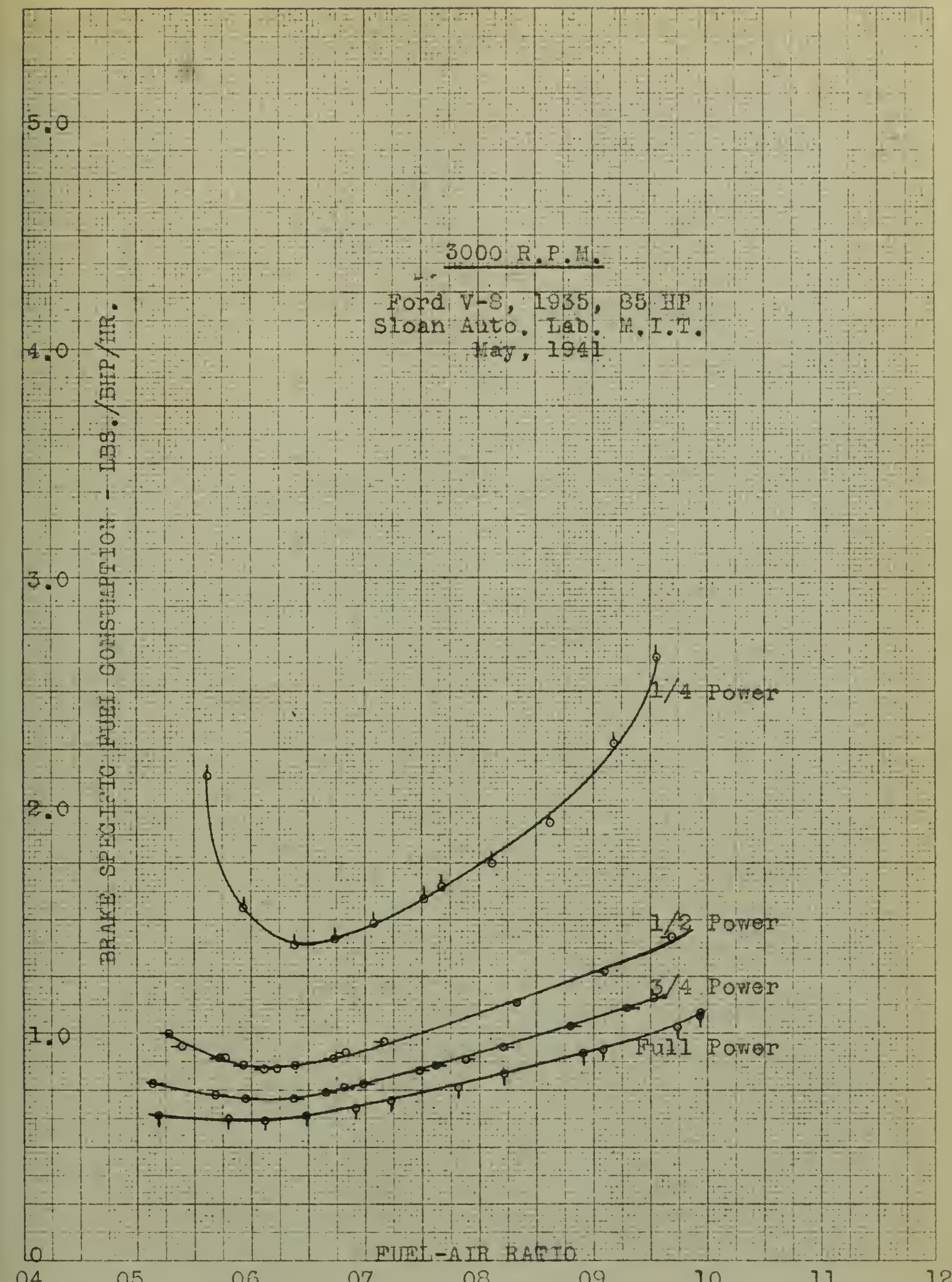


FIGURE 12

Variation of BSFC with F/A - 3000 R.P.M.

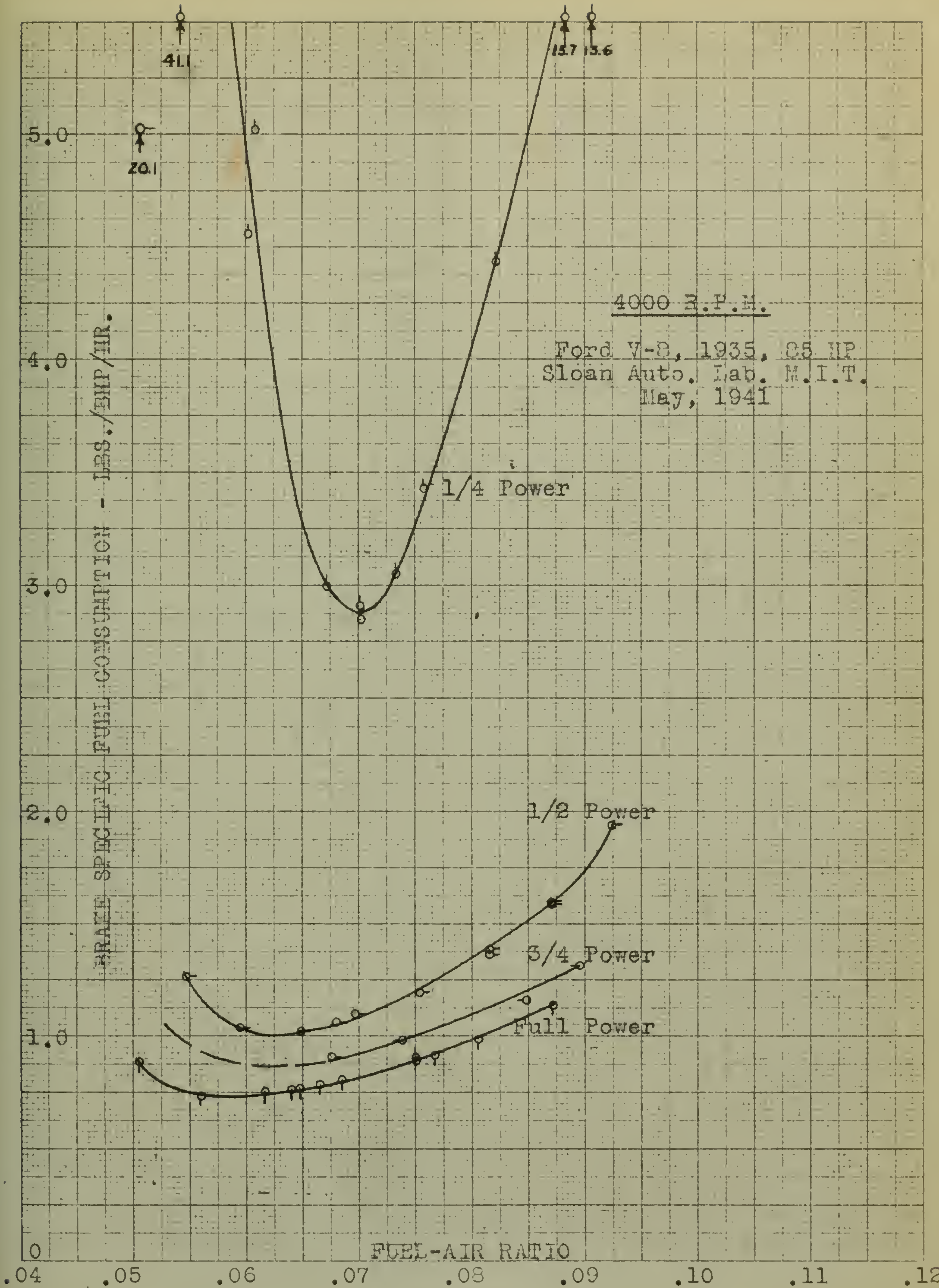


FIGURE 13
Variation of BSFC with F/A - 4000 R.P.M.

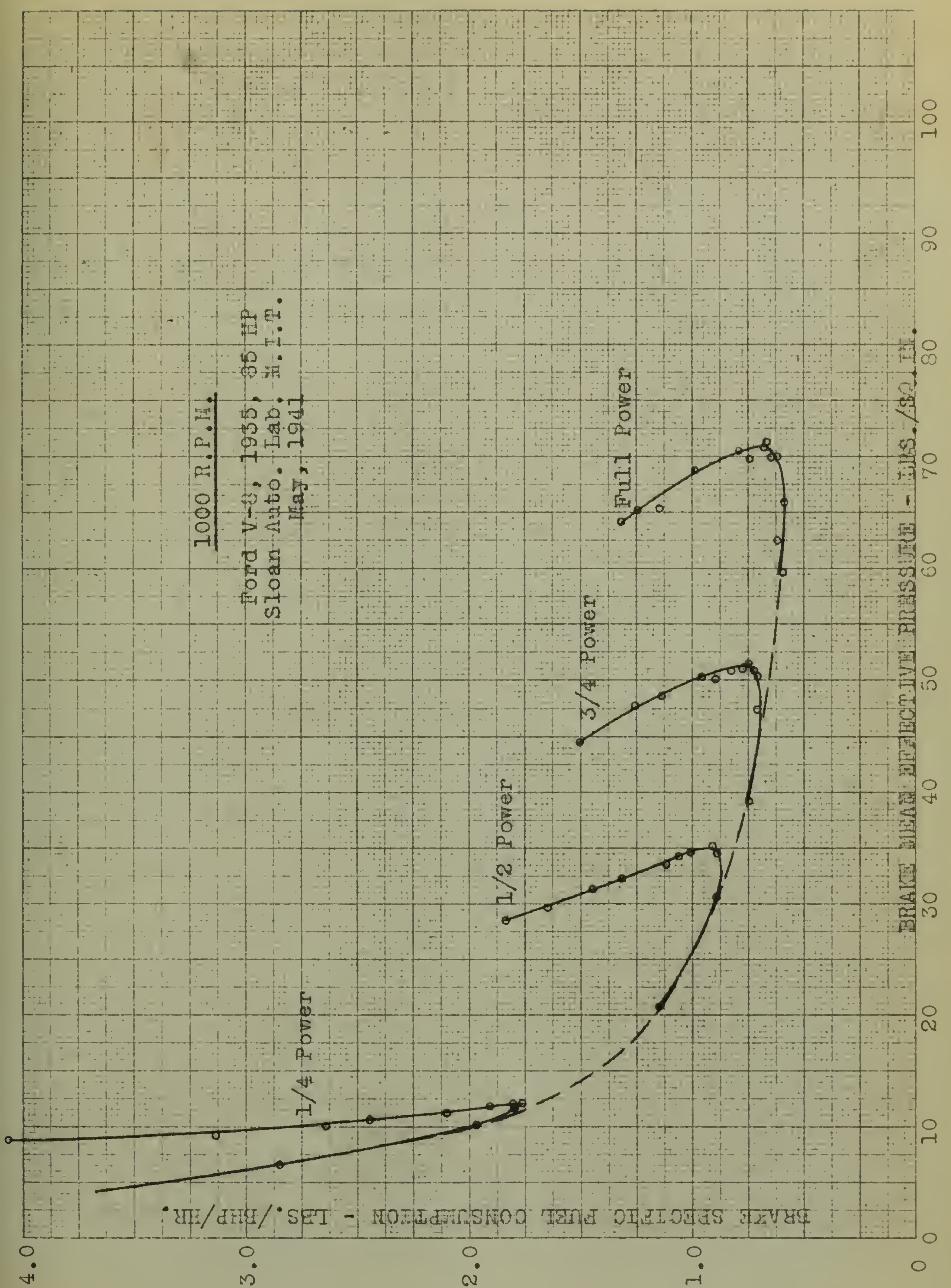


FIGURE14

Variation of BSFC with BMEP - 1000 R.P.M.

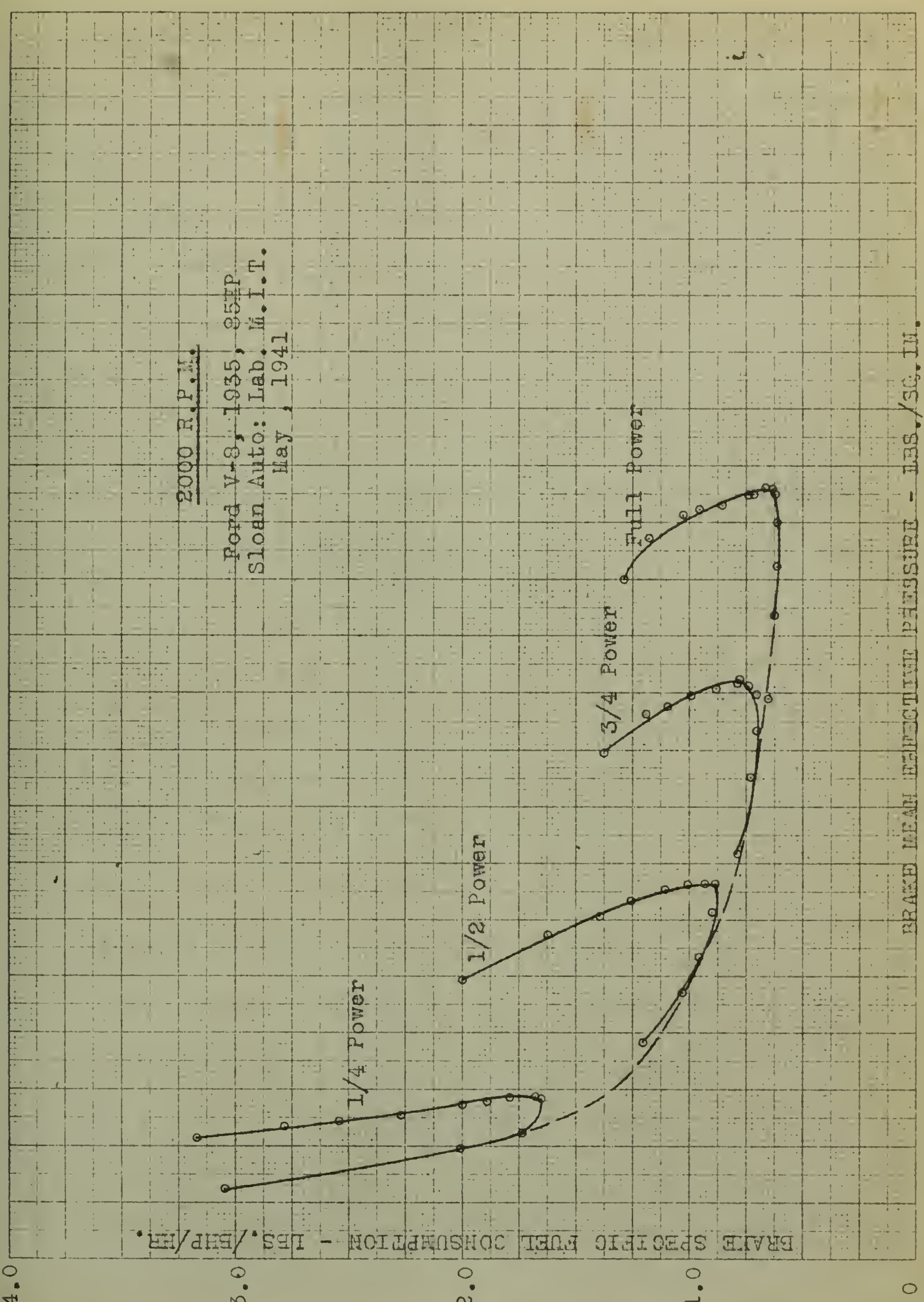


FIGURE 15
Variation of BSFC with BMEP - 2000 R.P.M.

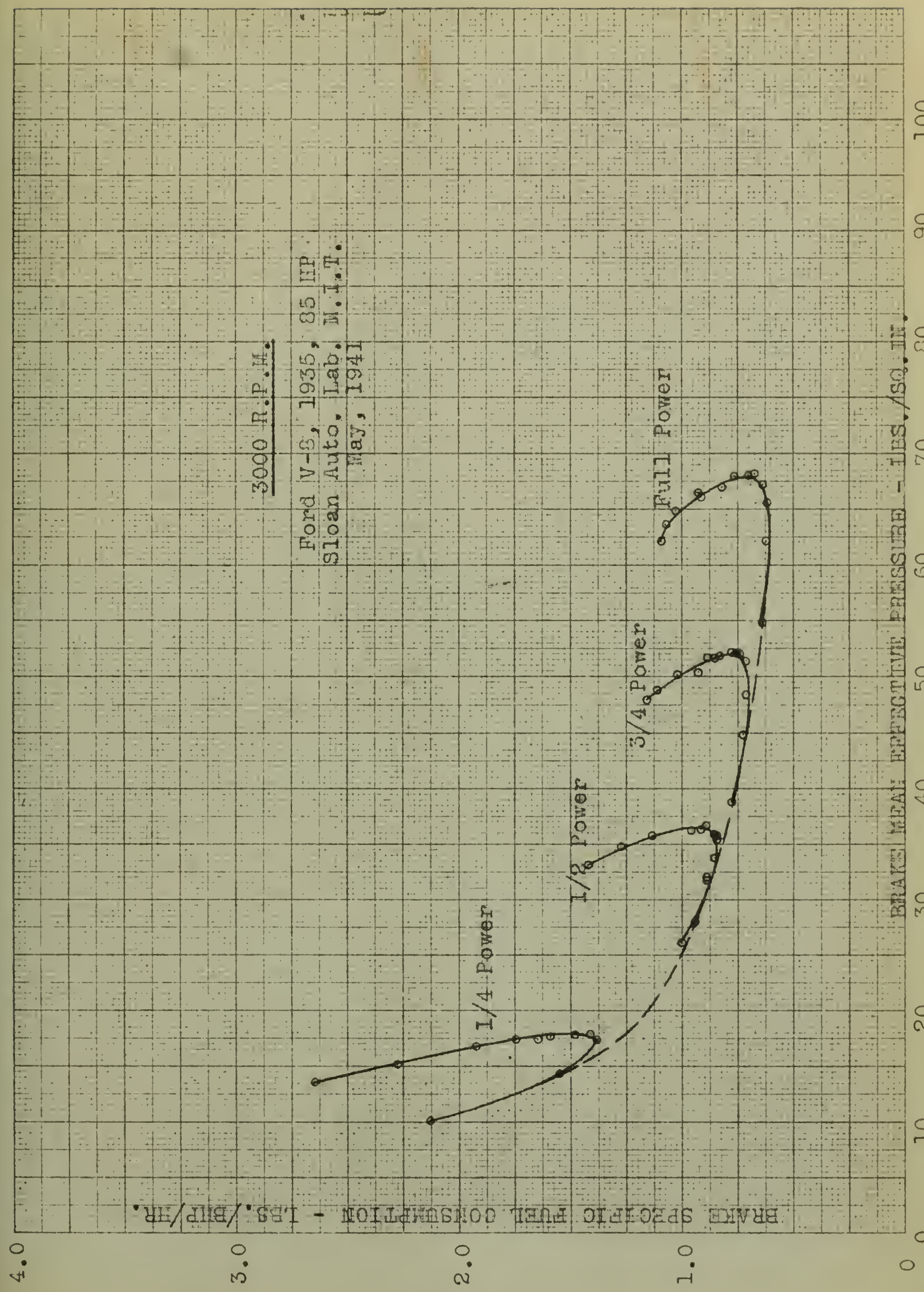


FIGURE 16
Variation of BSFC with BMEP - 3000 R.P.M.

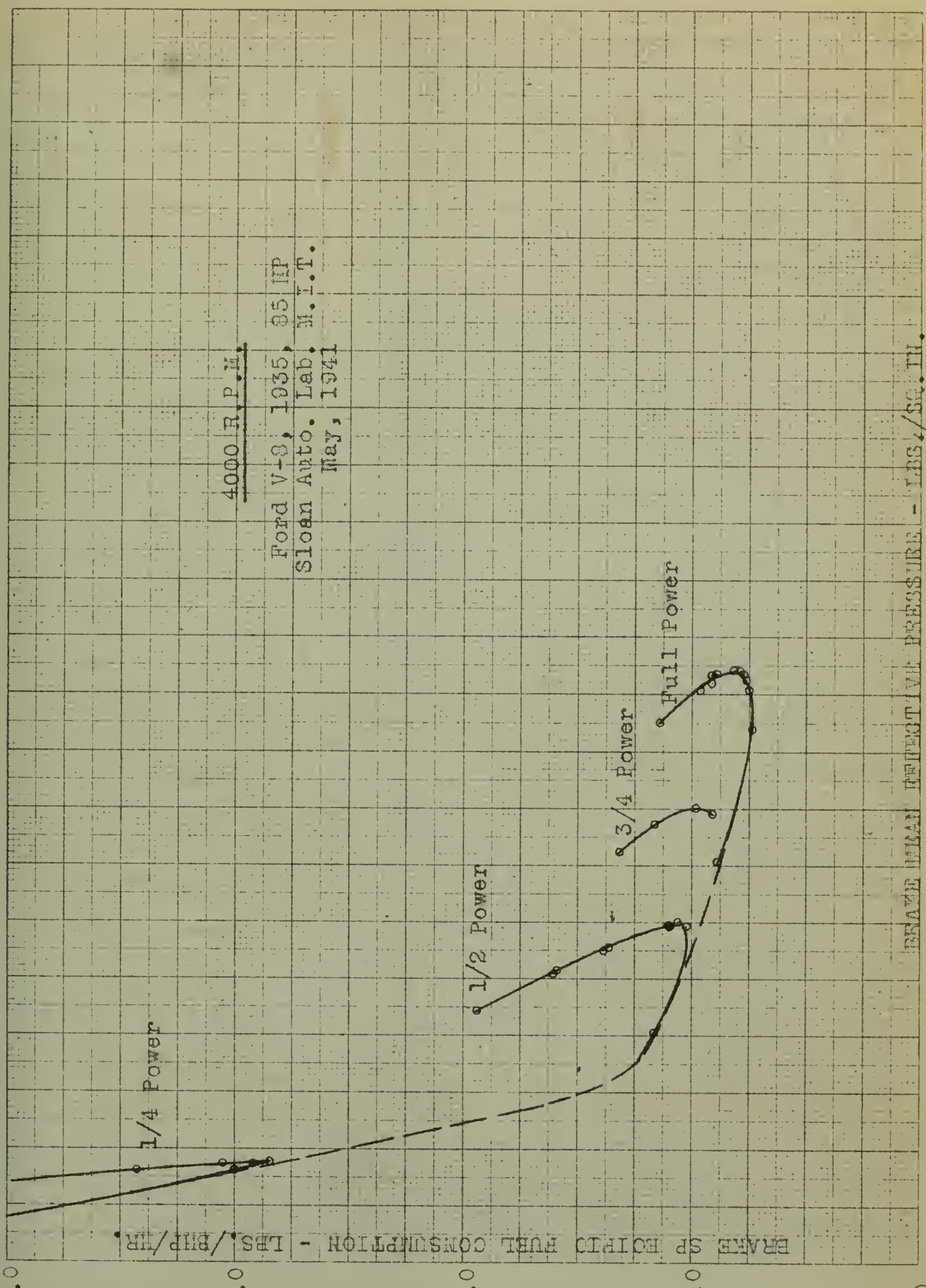


FIGURE 17
Variation of BSFC with BMEP - 4000 R.P.M.

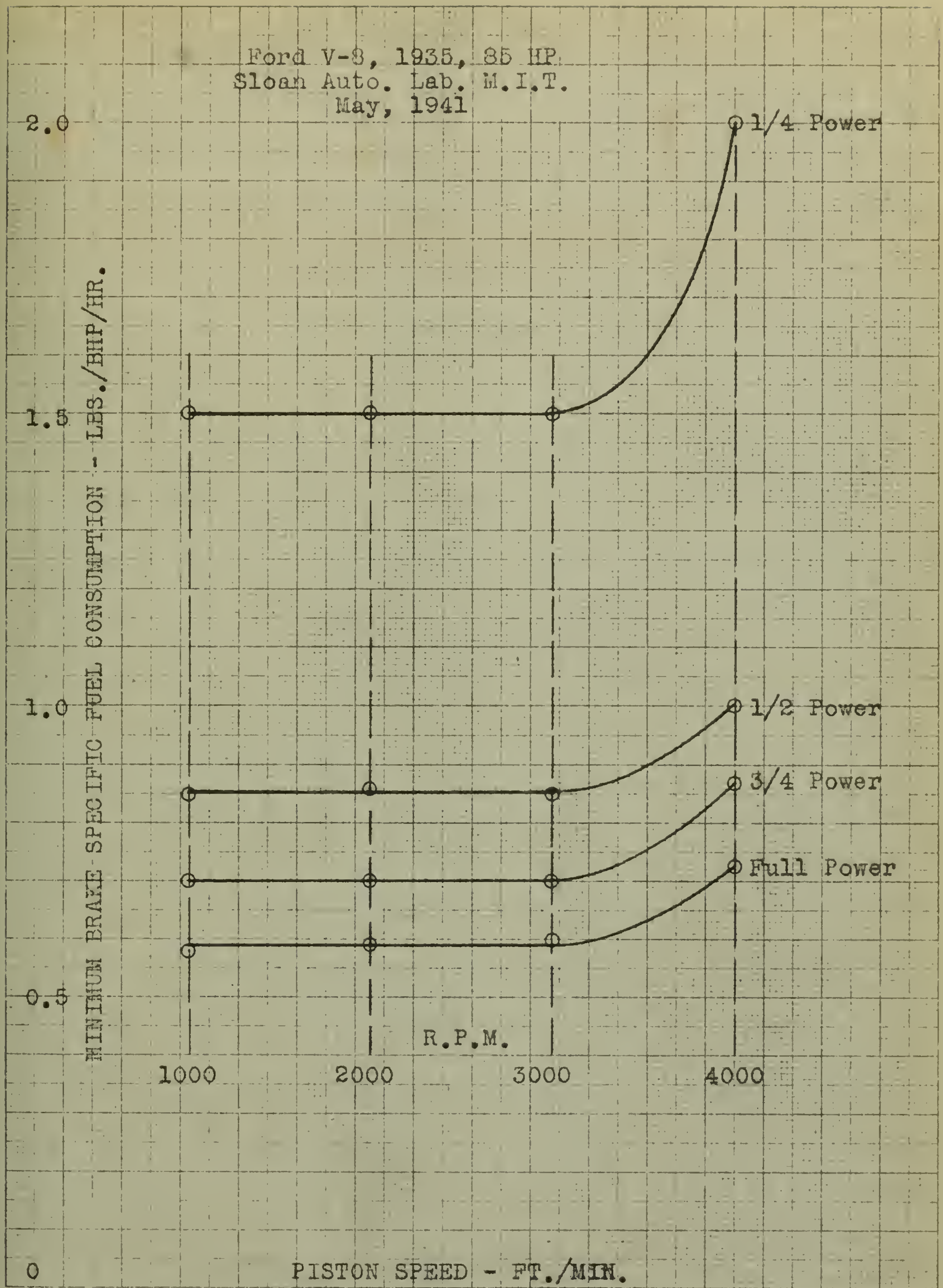


FIGURE 18
Variation of Minimum BSFC with SPEED

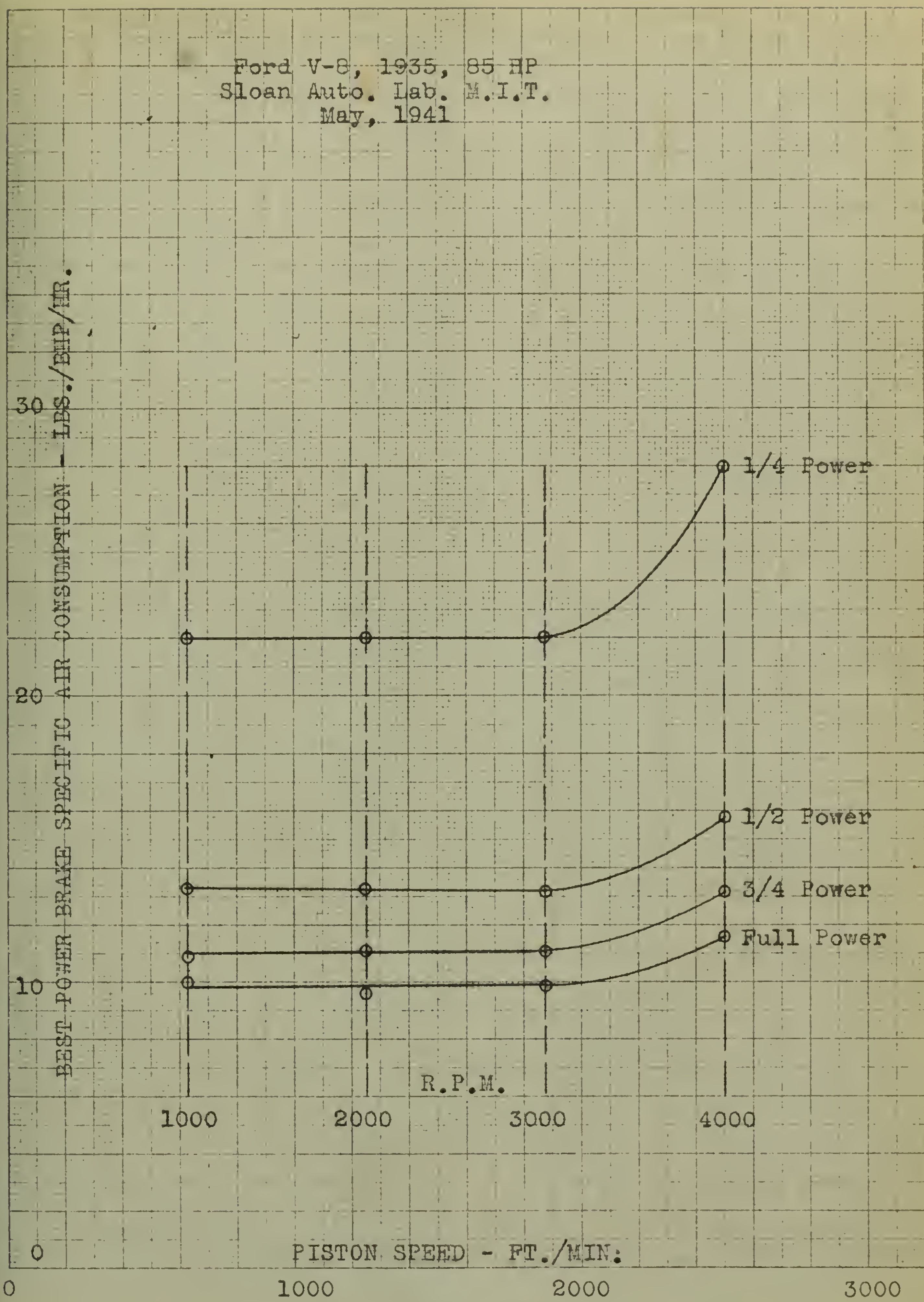


FIGURE 19

Variation of Best Power BSAC with SPEED

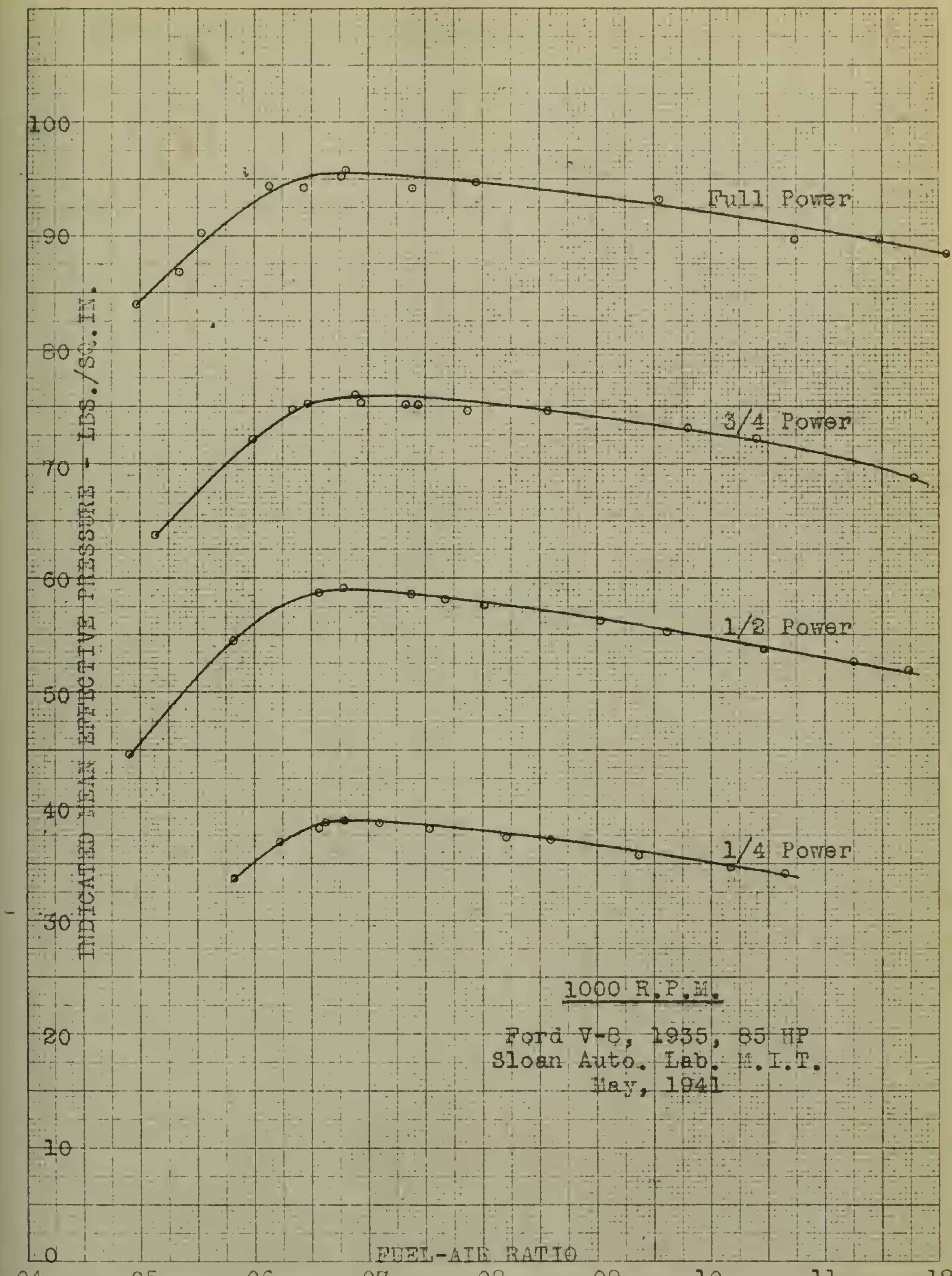


FIGURE 20

Variation of IMEP with F/A - 1000 R.P.M.

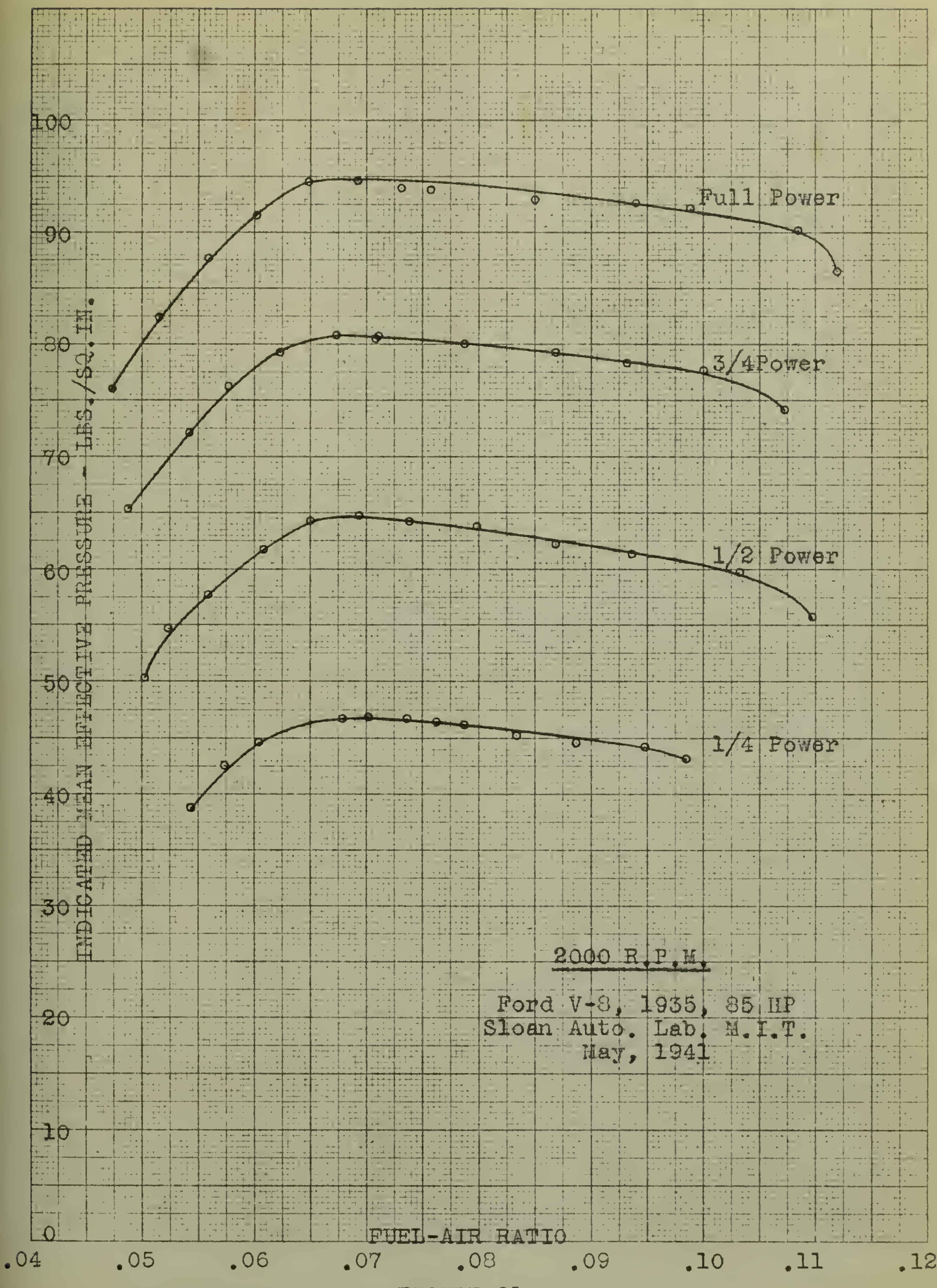


FIGURE 21
Variation of IMEP with F/A - 2000 R.P.M.

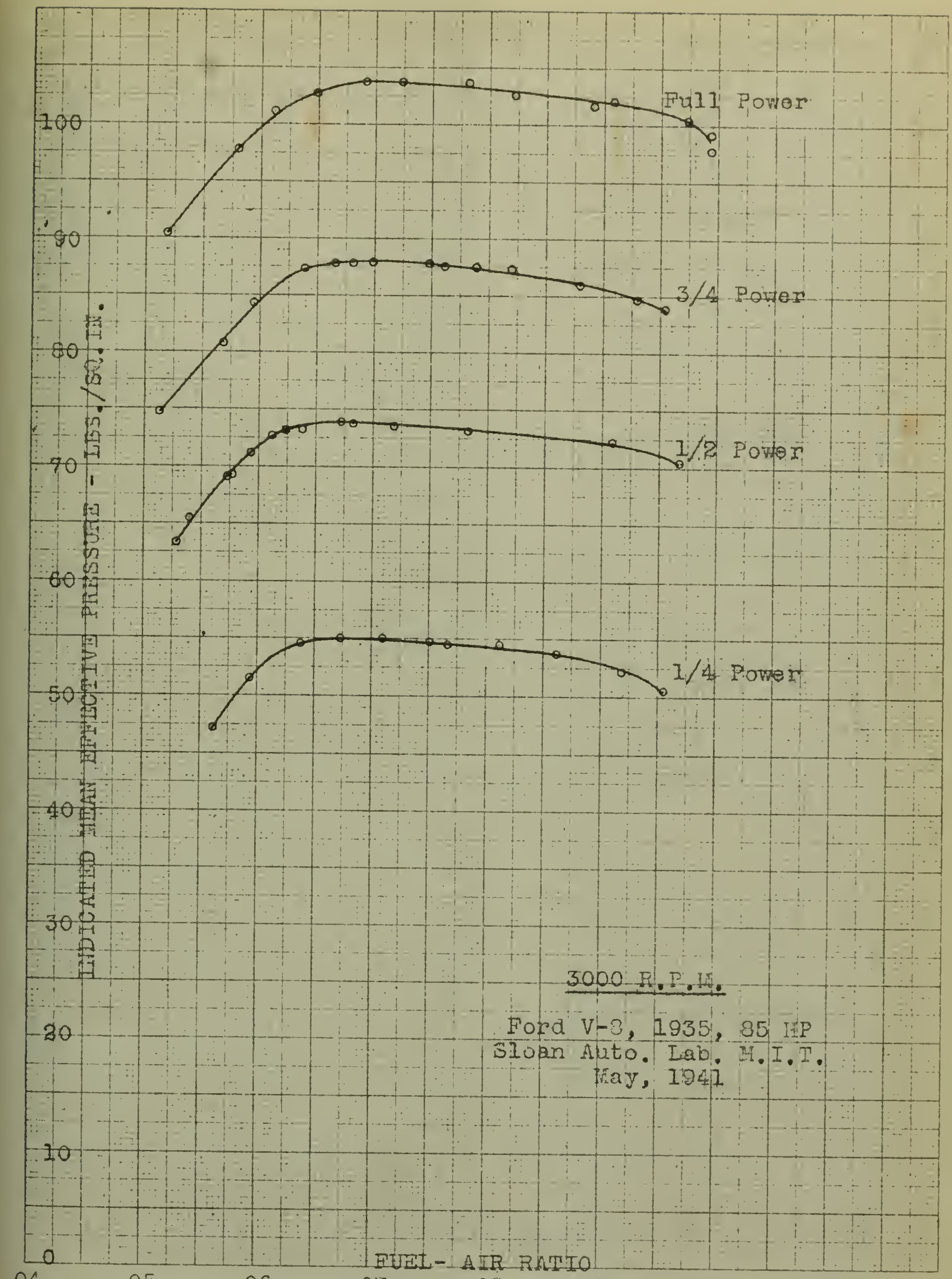


FIGURE 22
Variation of IMEP with F/A - 3000 R.P.M.

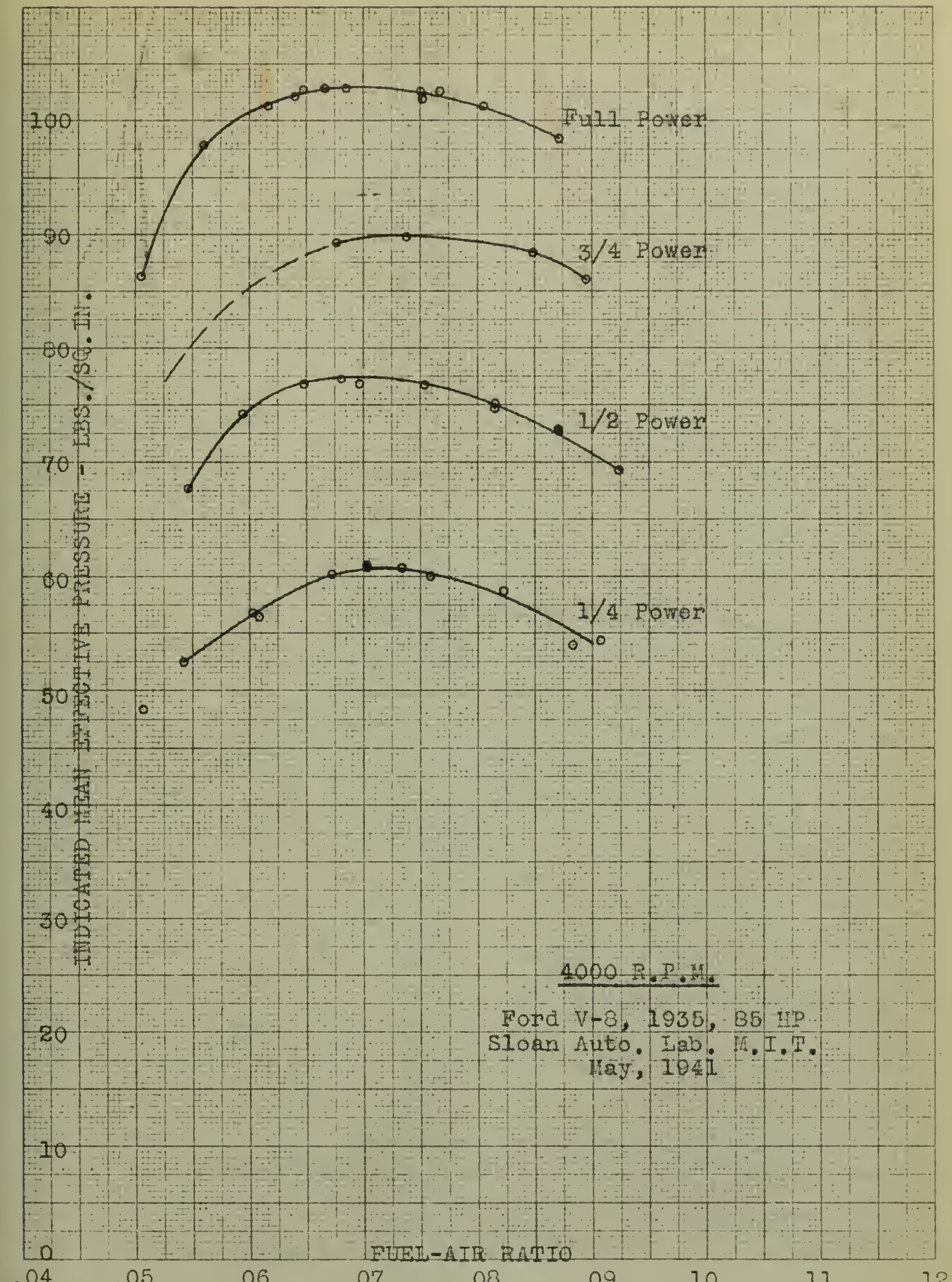


FIGURE 23
Variation of IMEP with F/A - 4000 R.P.M.

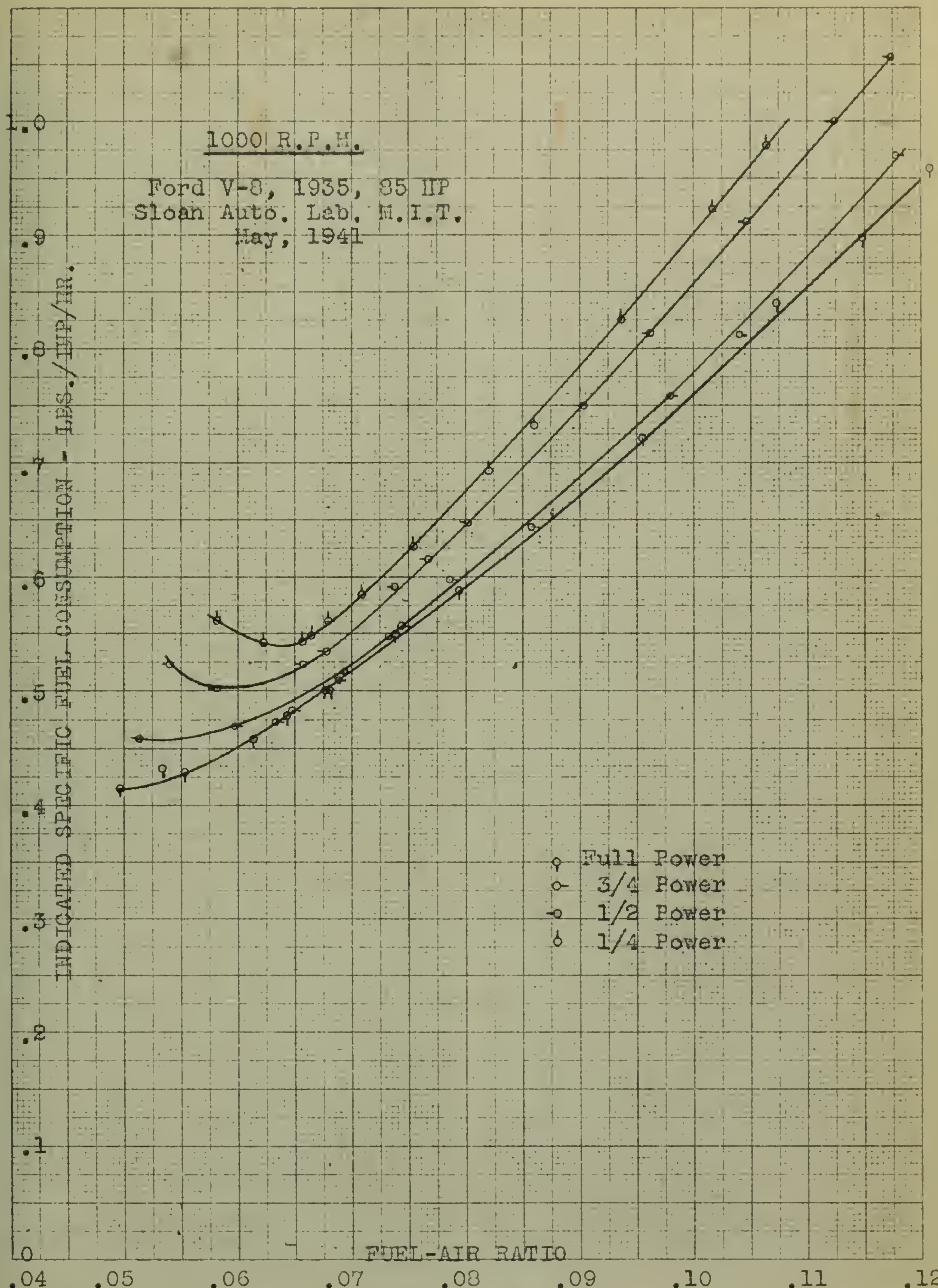


FIGURE 24
Variation of ISFC with F/A - 1000 R.P.M.

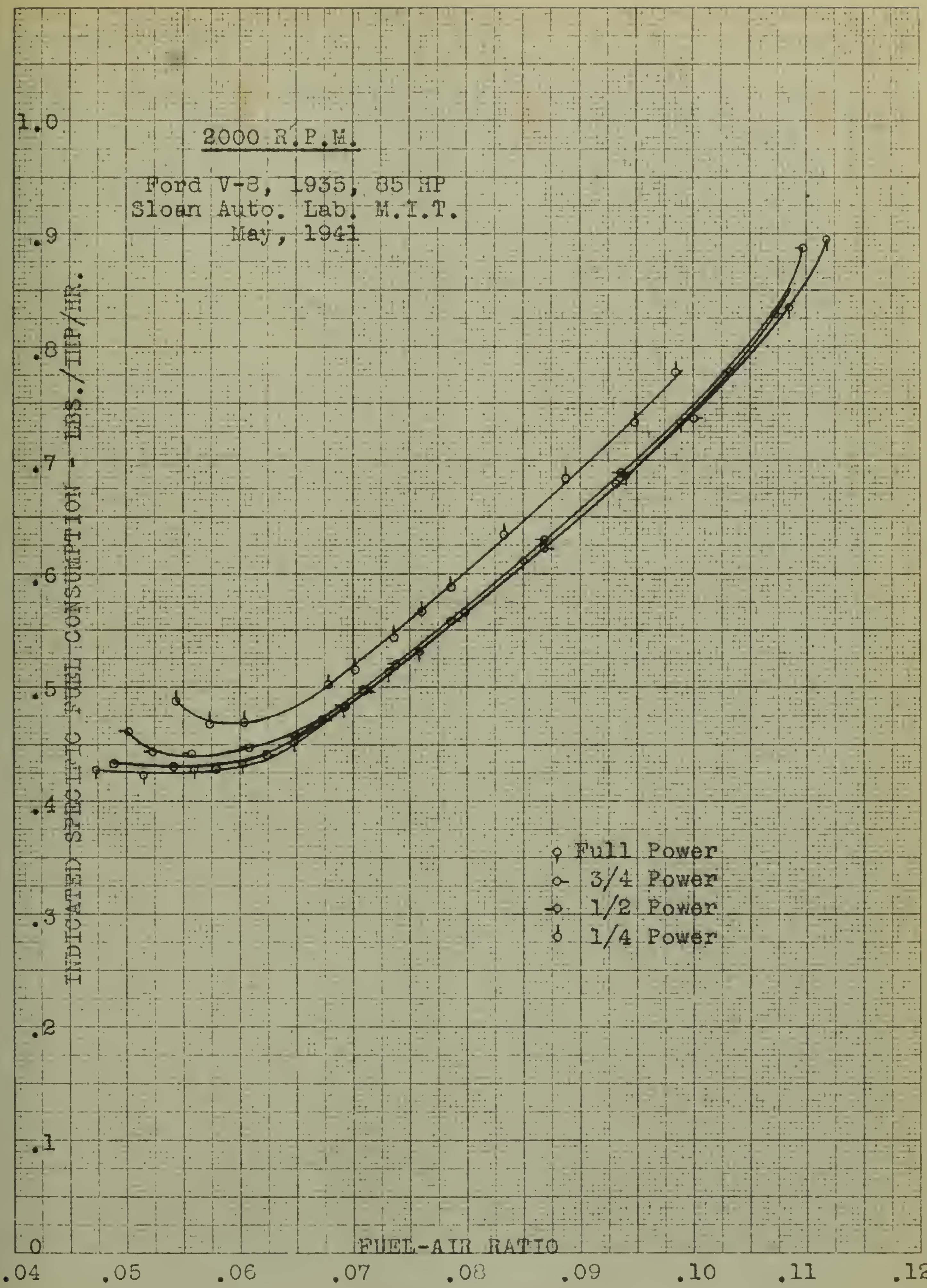


FIGURE 25
Variation of ISFC with F/A - 2000 R.P.M.

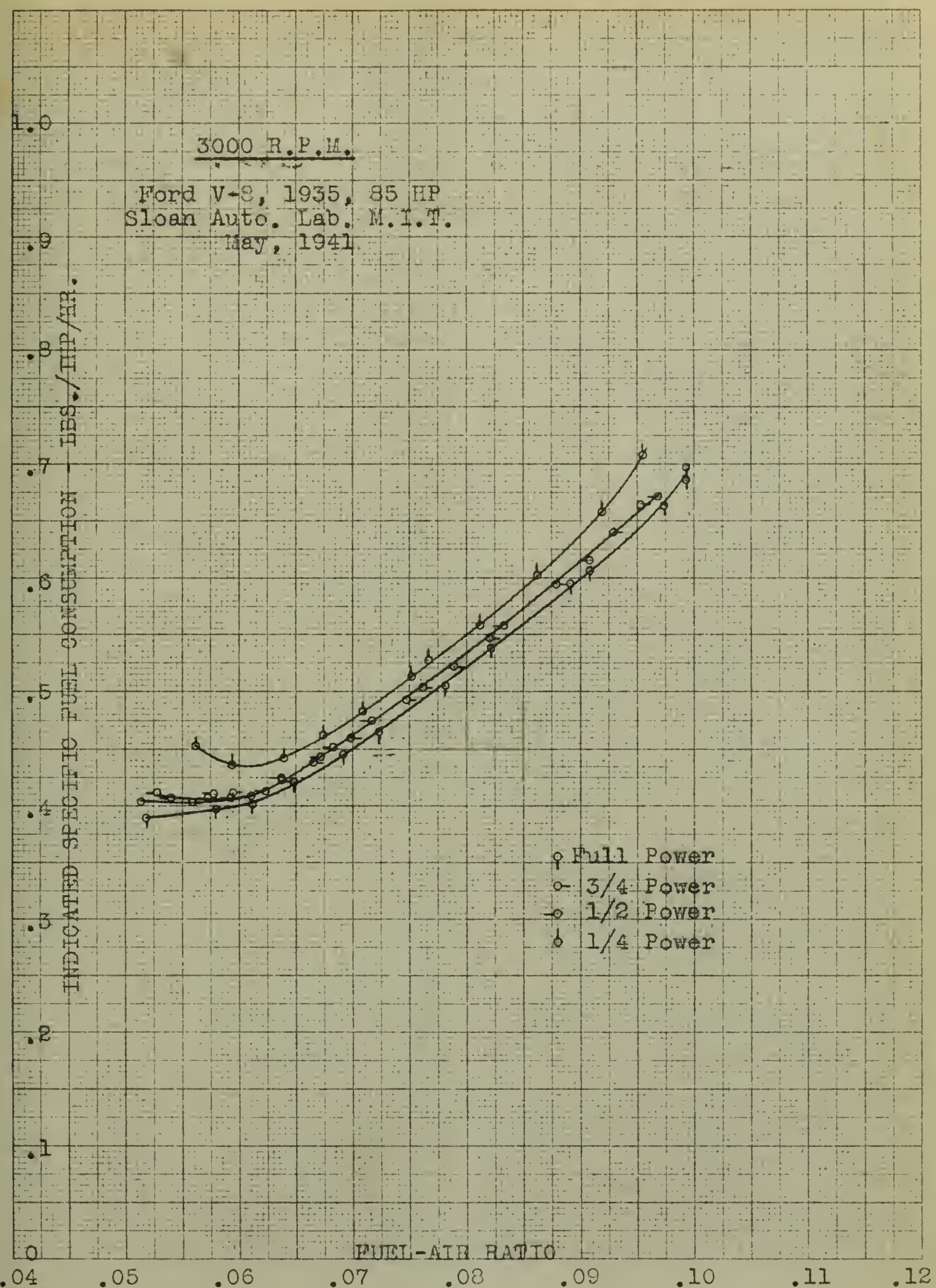


FIGURE 26
Variation of ISFC with F/A - 3000 R.P.M.

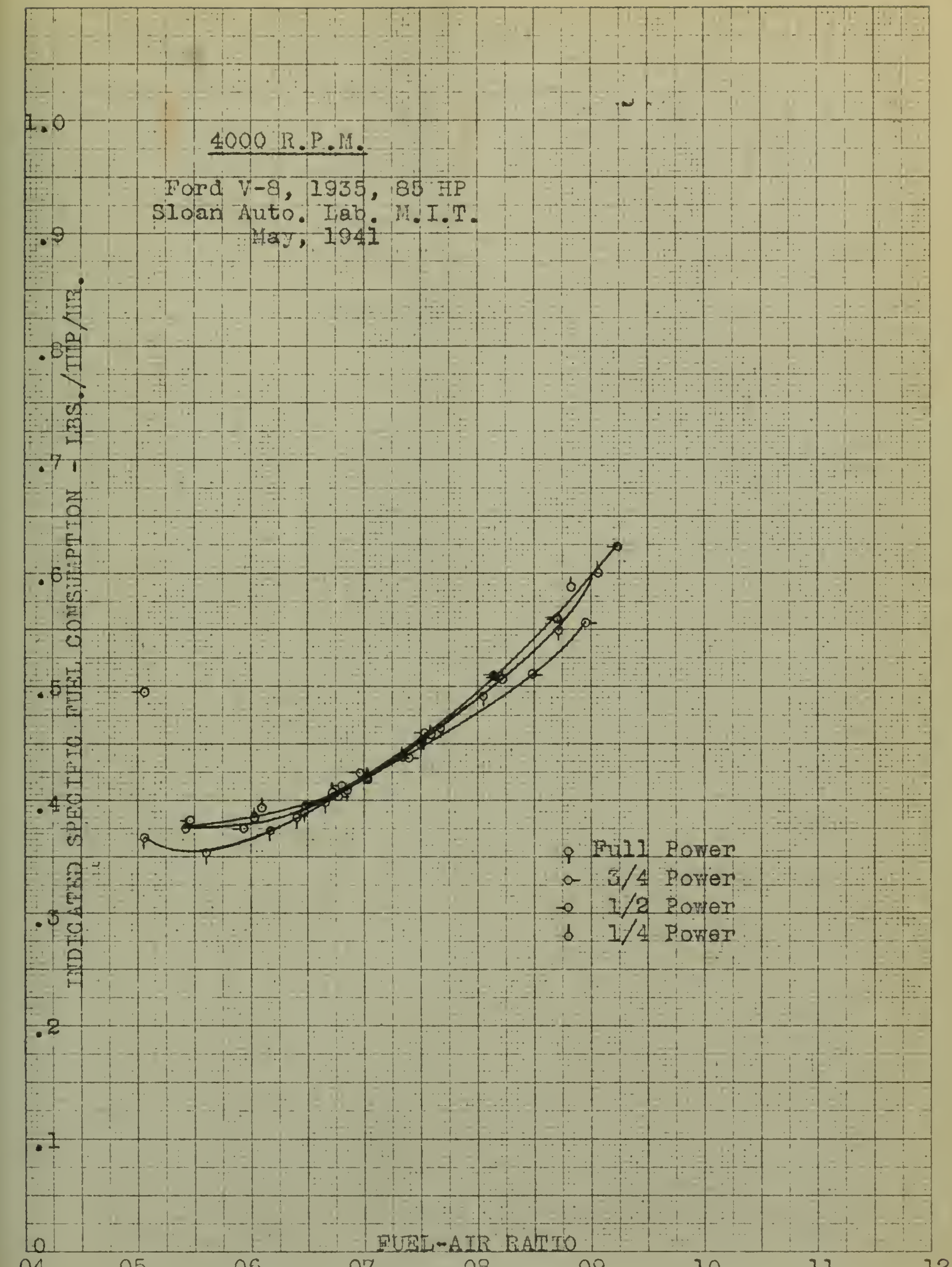


FIGURE 27
Variation of ISFC with F/A - 4000 R.P.M.

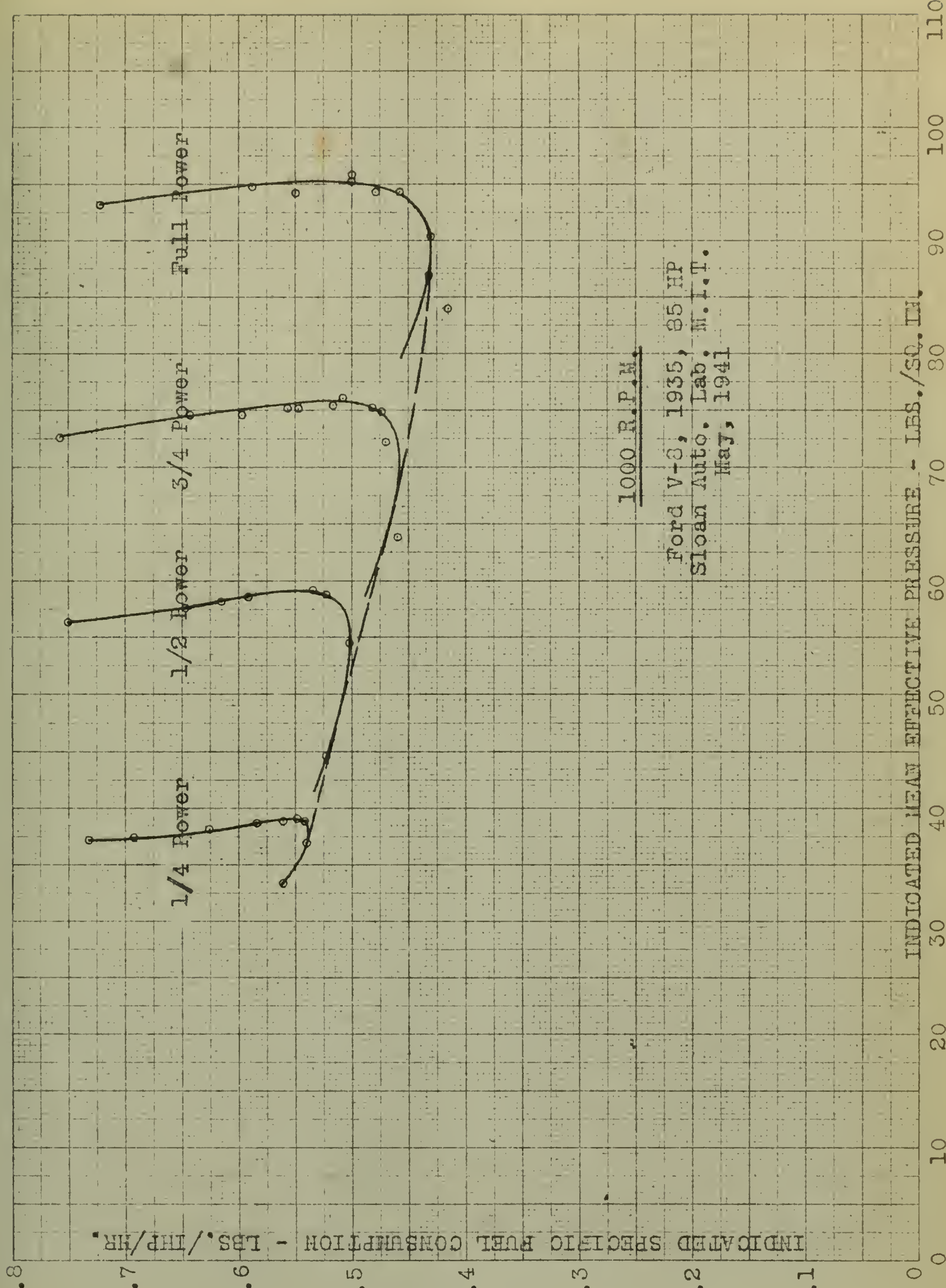


FIGURE 28

Variation of ISFC with IMEP - 1000 R.P.M.

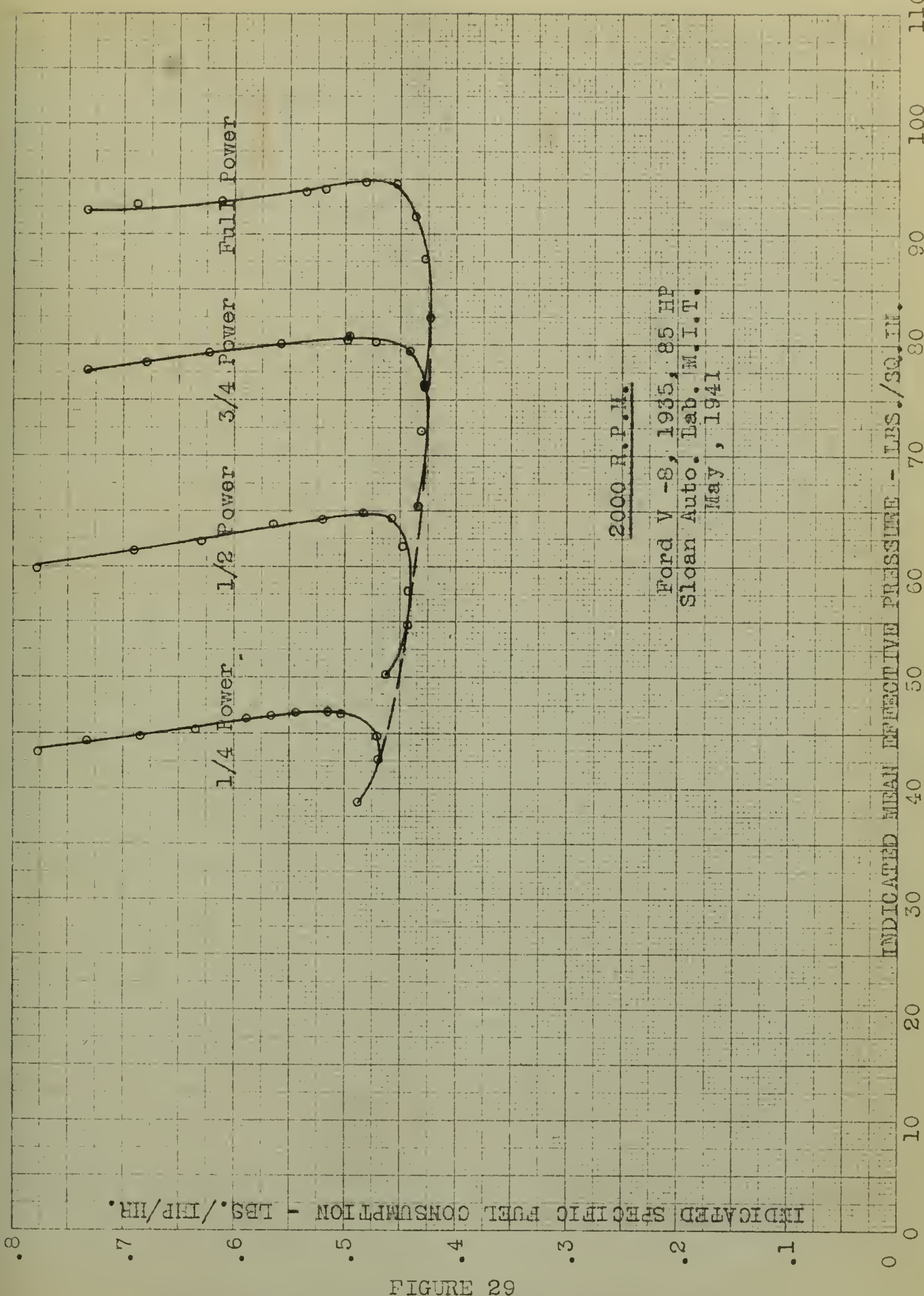


FIGURE 29

Variation of ISFC with IMEP - 2000 R.P.M.

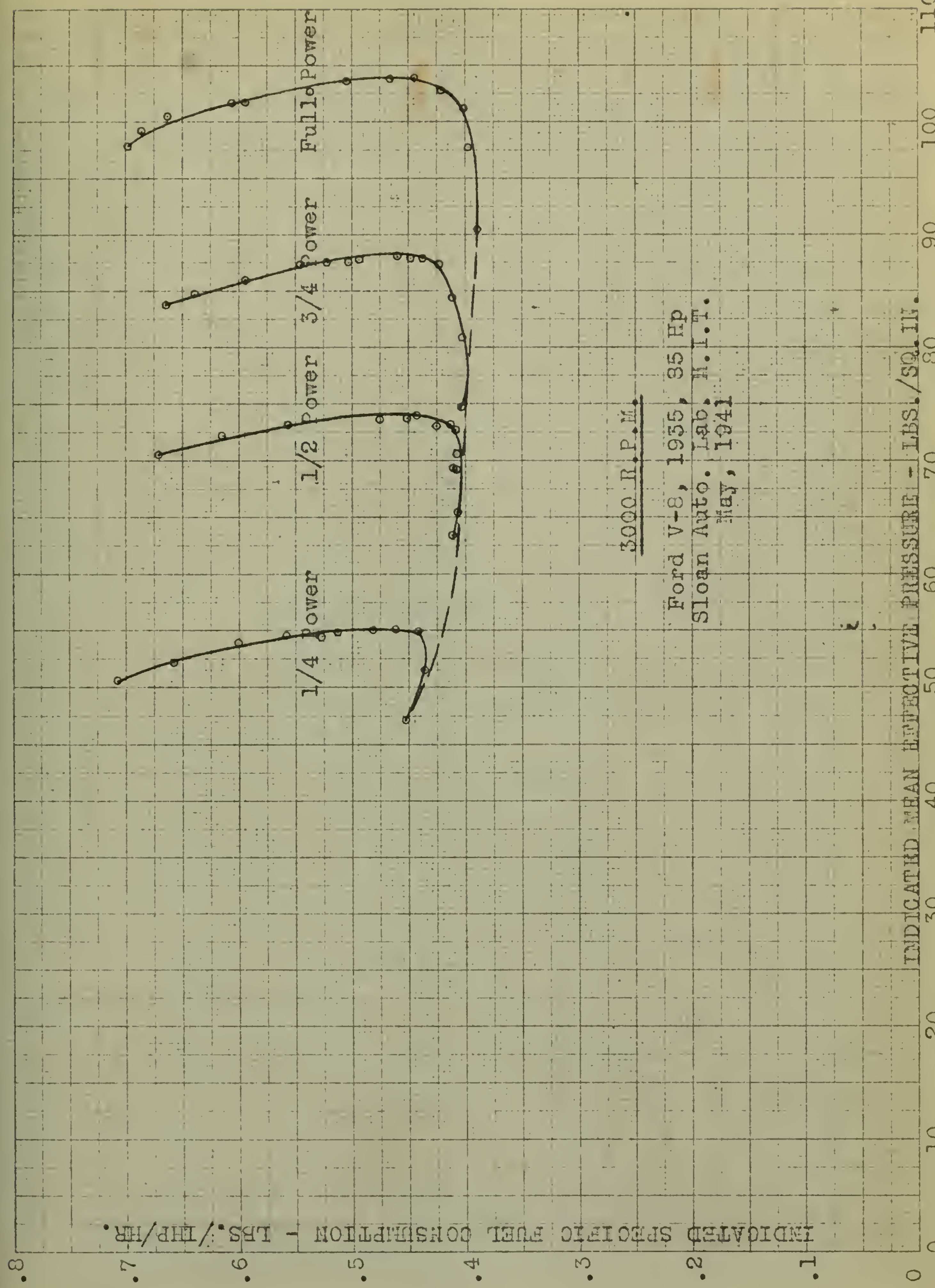


FIGURE 30
<u>Variation of ISFC with IMEP - 3000 R.P.M.</u>

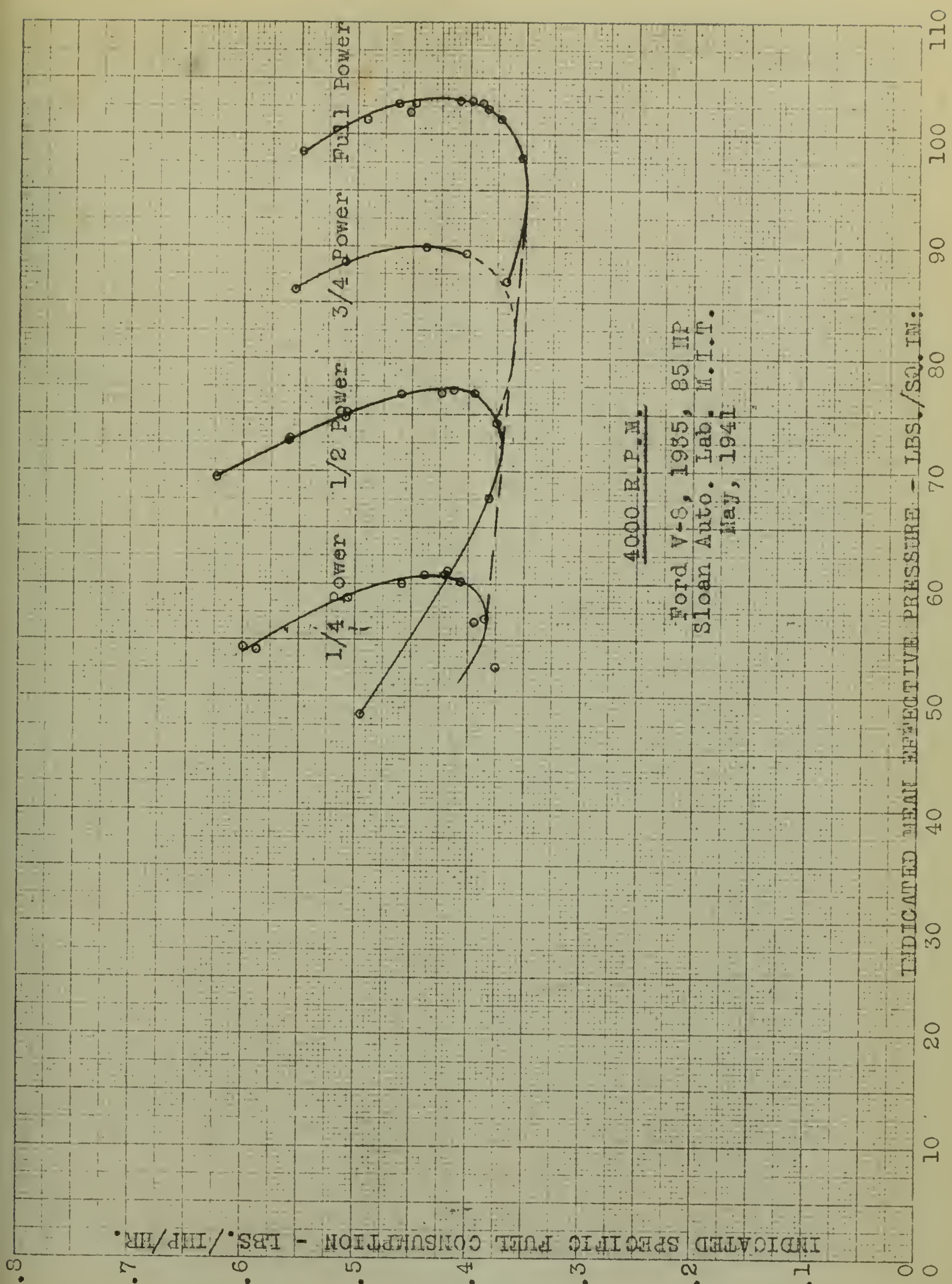


FIGURE 31
Variation of ISFC with IMEP - 4000 R.P.M.

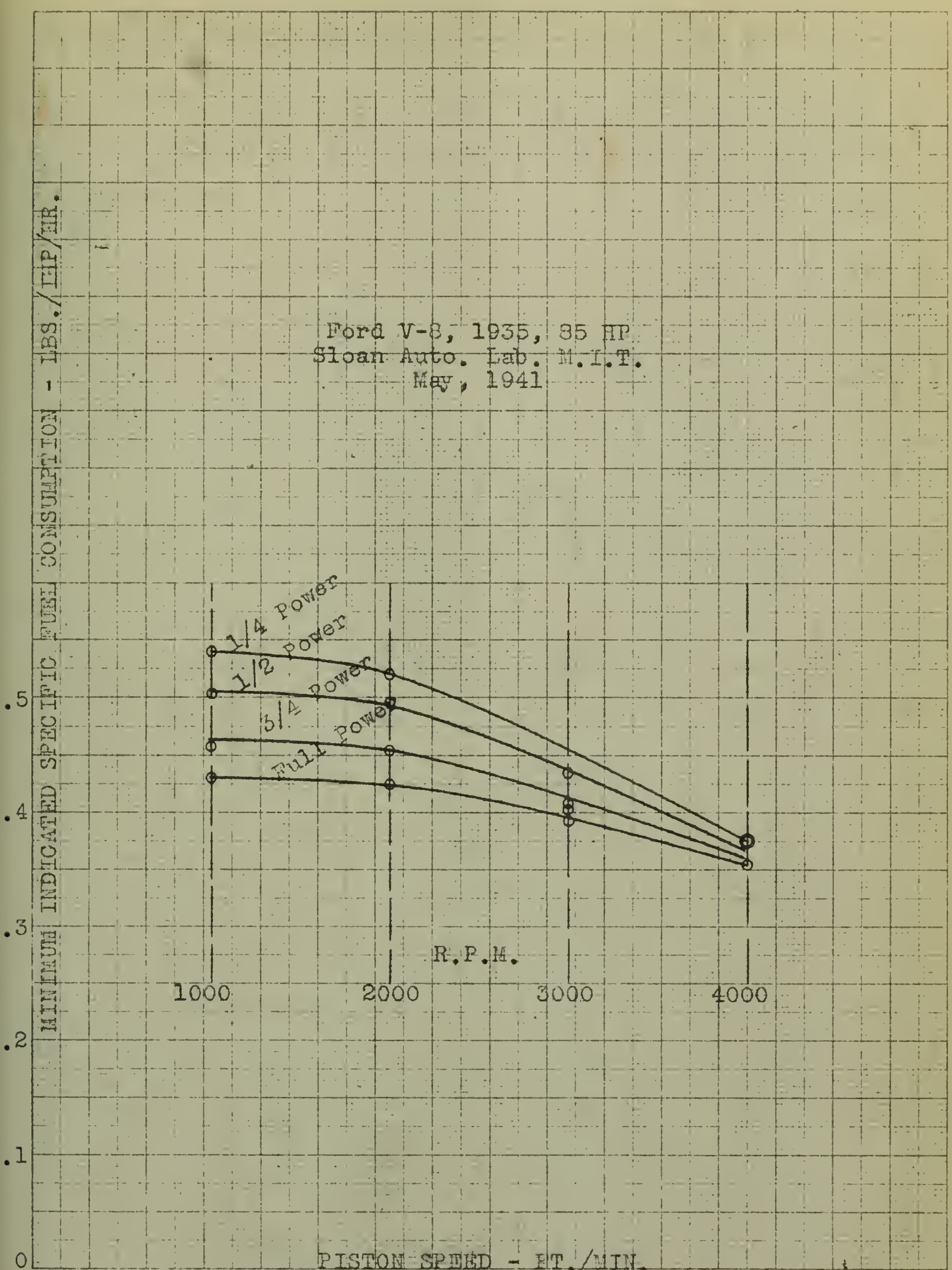


FIGURE 32

Variation of Minimum ISFC with SPEED

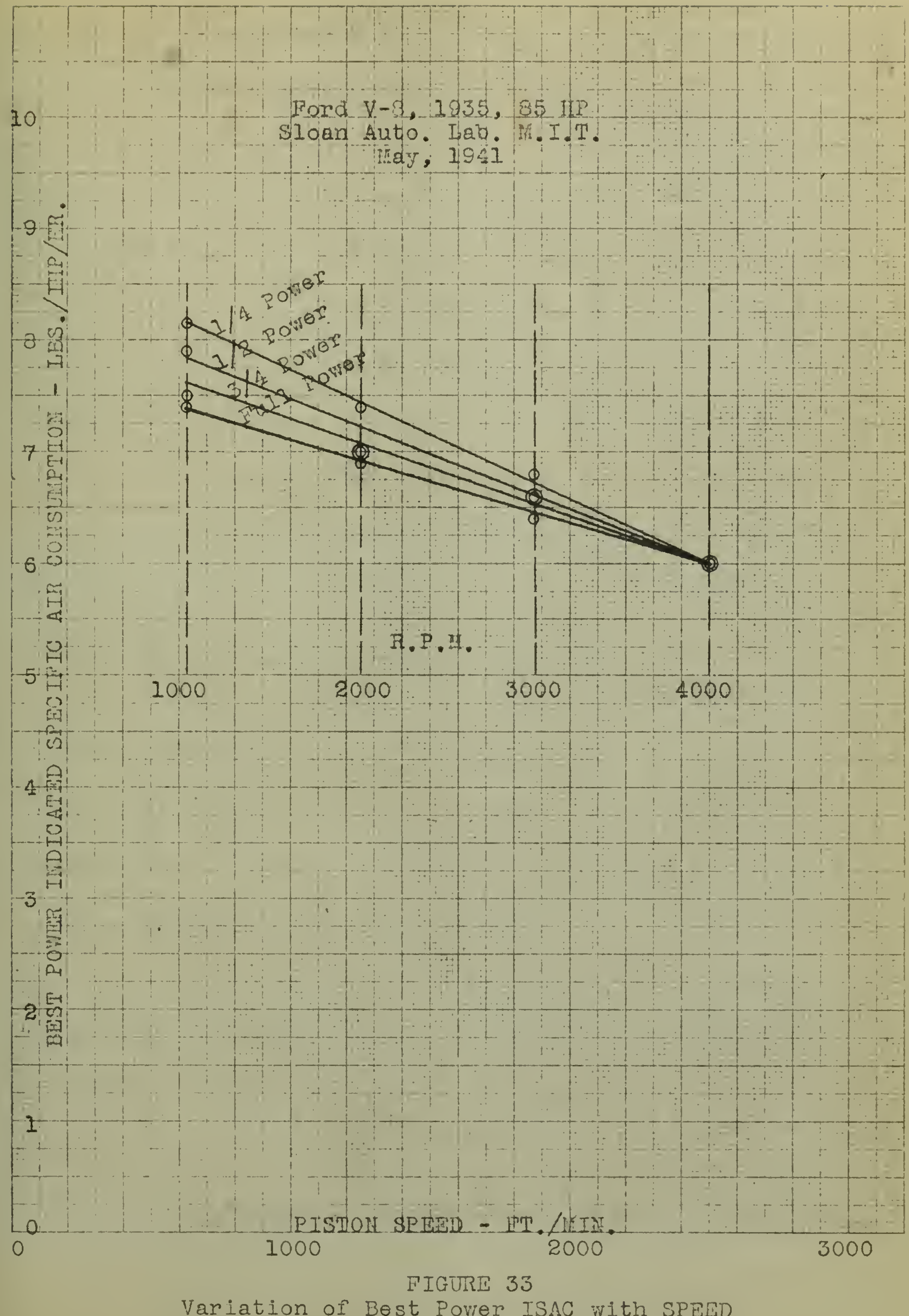


FIGURE 33

Variation of Best Power ISAC with SPEED

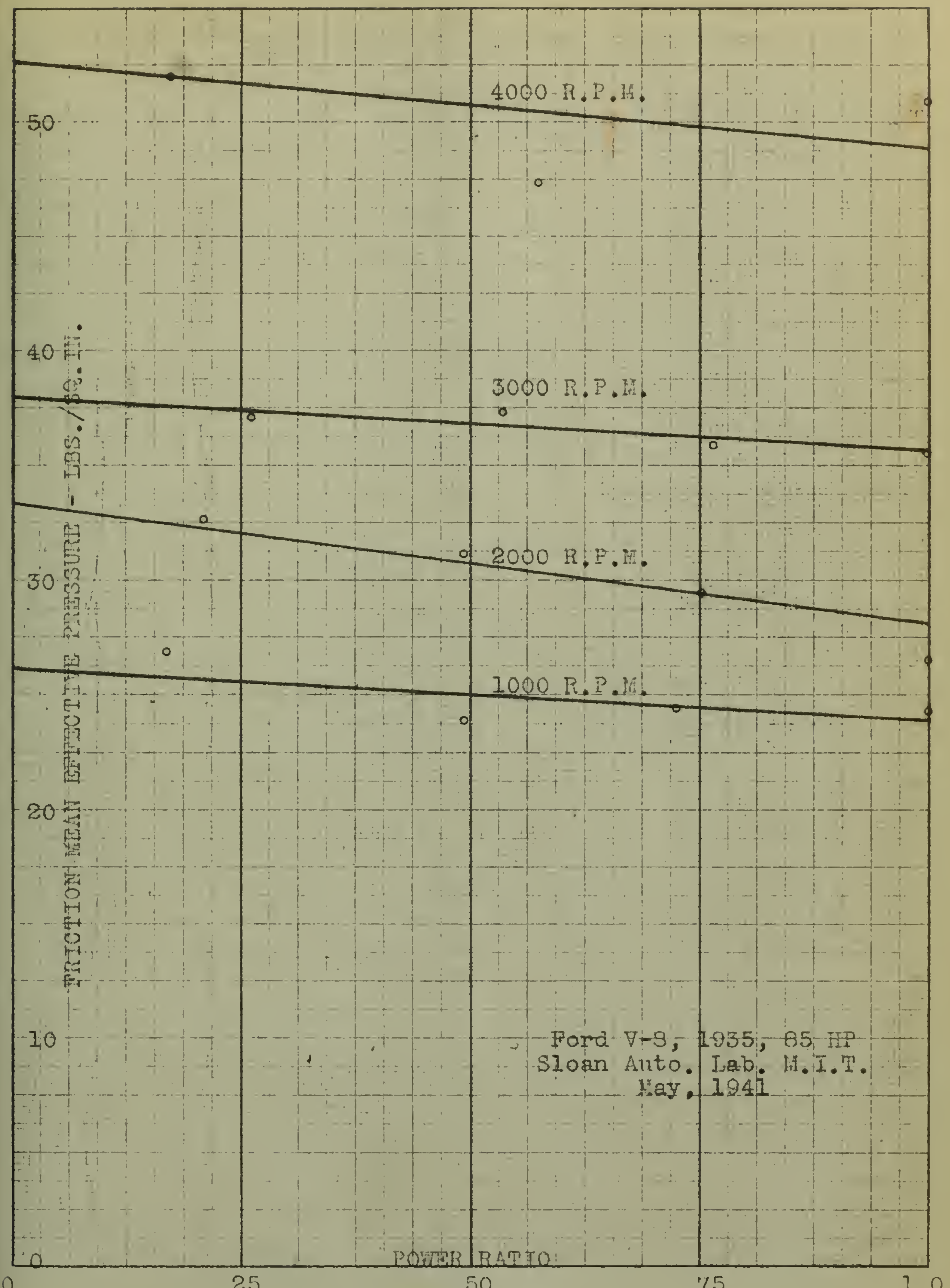


FIGURE 34
Variation of FMEP with SPEED and POWER RATIO

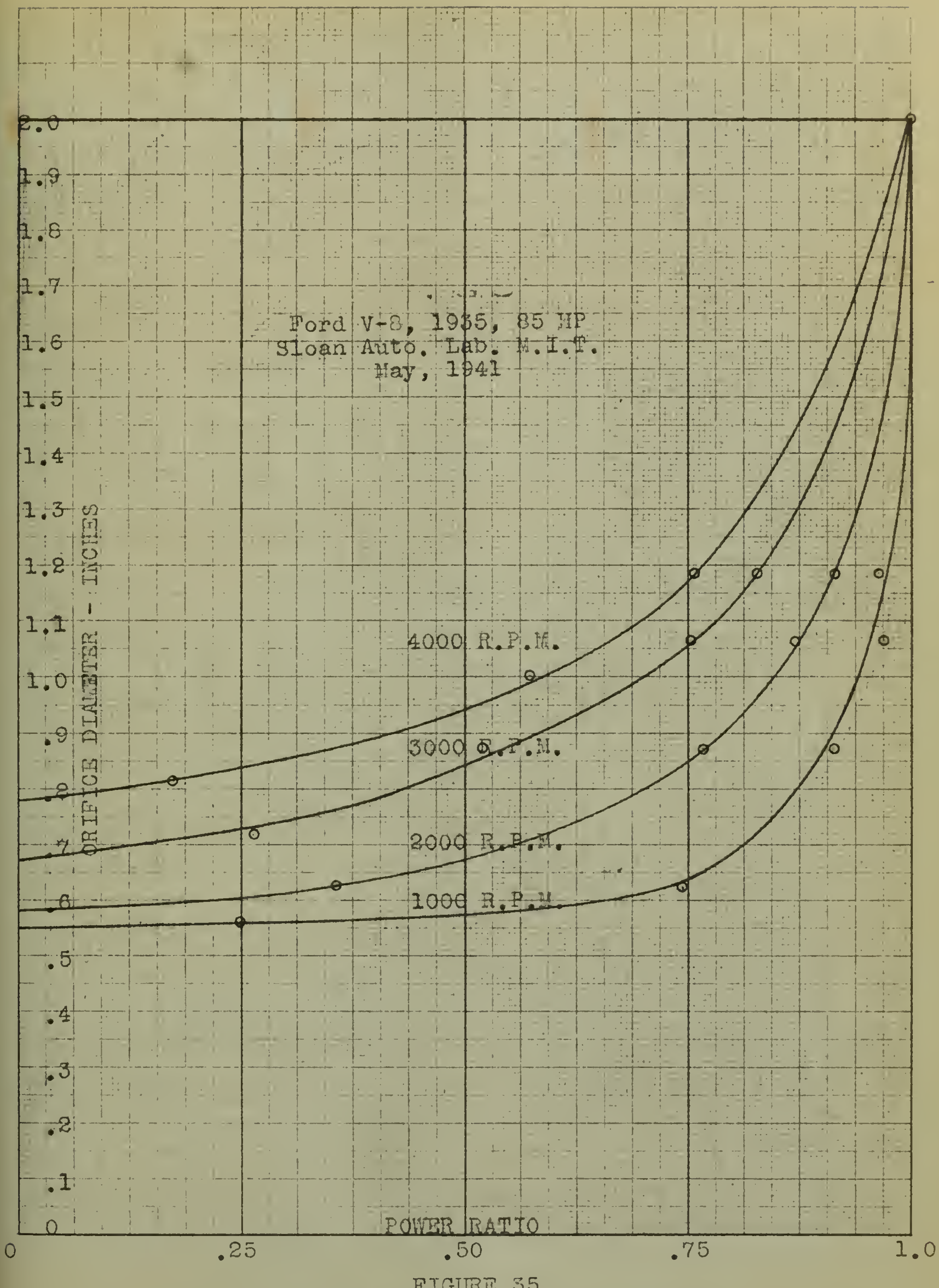


FIGURE 35
Throttling Orifice Calibration Curves.

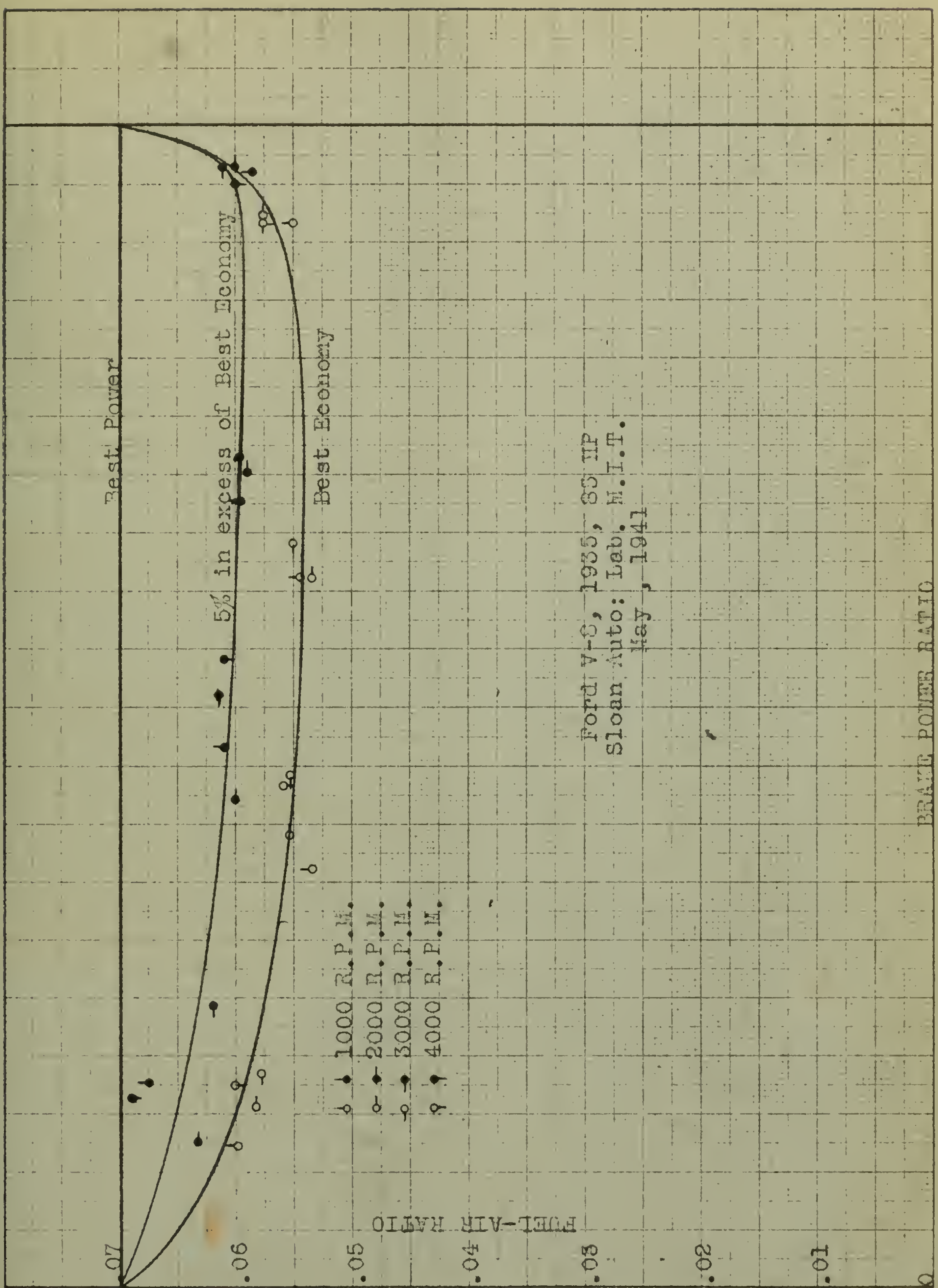


FIGURE 36

Variation of Best Power and Best Economy F/A with Power Ratio

Brake Power Basis

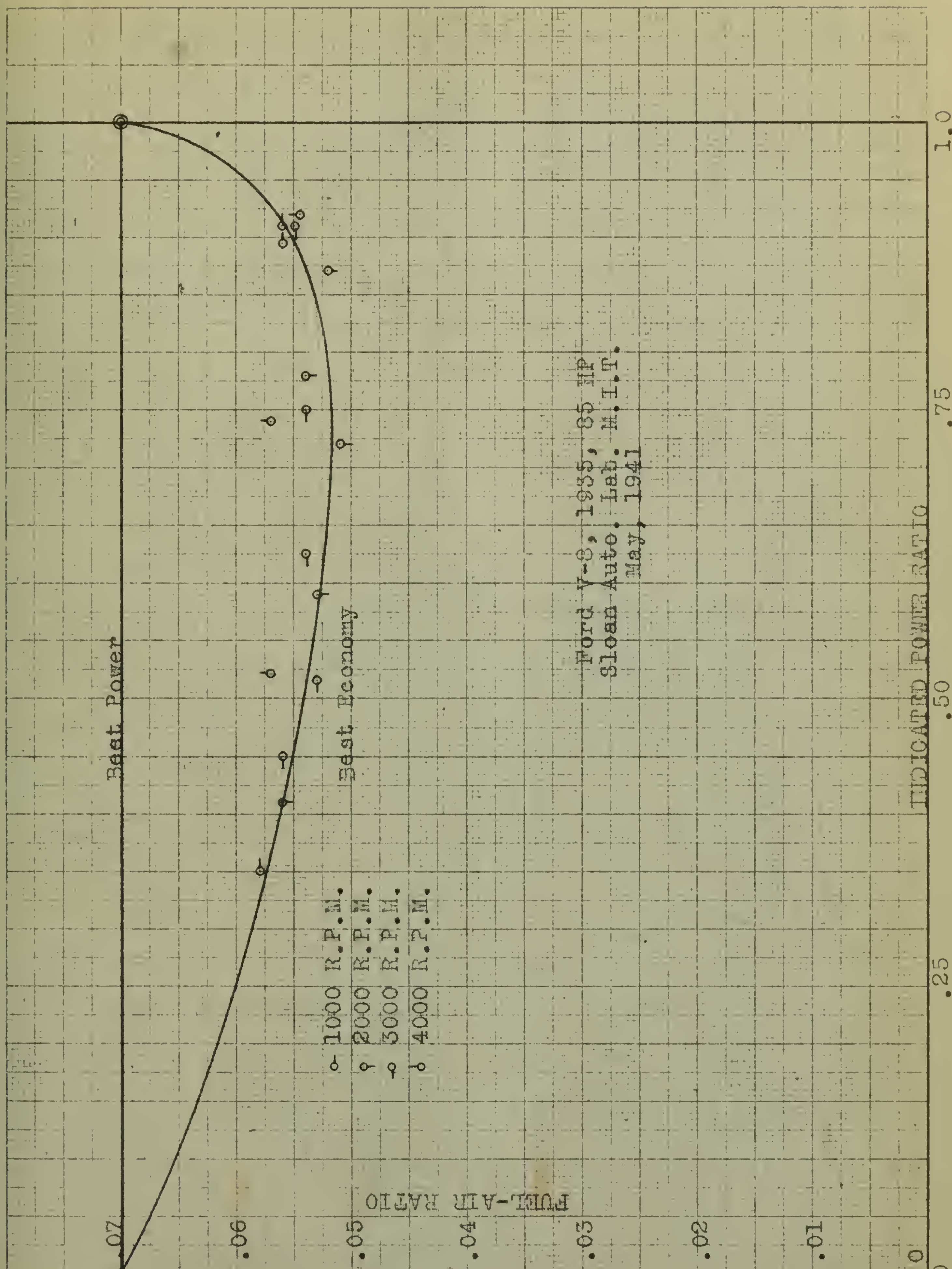


FIGURE 37

Variation of Best Power and Best Economy F/A with Power Ratio
Indicated Power Basis

PART III

APPENDIX

1. Detailed Description of Special Apparatus.

Fuel System. Fuel under pressure was furnished by a Nichols rotary pump capable of pumping more fuel than that consumed at highest powers. The pump, driven by a constant speed electric motor, discharged to an accumulator tank as shown in Fig. 2. From this tank there were direct connections to a three-way valve and to a pressure control valve (Fig. 5), which discharged to the suction side of the fuel pump, and which was set to maintain a constant gage pressure of 10 pounds per square inch. Into the accumulator tank from directly below projected the standpipe of a burette with a series of calibrated bulbs. From the bottom of this burette a line connected to the three-way valve. By this valve fuel to the mixing tank could be drawn from either the accumulator tank or from the burette. Fuel level in the accumulator tank was controlled by varying the volume of air in the tank so that the level of fuel was at all times below the top of the burette standpipe.

Since adjustment of fuel level caused a change in fuel pressure, with consequent change in flow rate and fuel-air ratio, all necessary adjustments of level were made immediately upon completion of a run, and equilibrium obtained before data for a subsequent run was taken.

Only occasional adjustment of this level was required during operation. From the three-way valve fuel to the mixing tank passed through a flow control needle valve, and then through a steam jacketed line approximately one foot long. Continual steam supply to this jacket furnished heat to insure maximum possible vaporization before discharge into the mixing tank below.

In normal operation the three-way valve was turned to interconnect all three lines so that fuel flowed directly from the accumulator tank to the mixing tank, and the fuel level in the burette was the same as that in the accumulator. Excess fuel from the pump flowed through the pressure regulating valve back to the suction side of the pump. To measure fuel flow, the valve was turned to close the direct line to the accumulator, and to take fuel from the burette. The fuel system was most satisfactory in operation. Flow control was accurate, and the pressure remained substantially constant, except as discussed under "Accuracy".

Pressure Control Valve. The sketch of this valve, Fig. 5, is self explanatory. Modification was accomplished by simple machining operations, by substitution of neoprene for rubber throughout, and by use of a control spring which was weaker than the original. The operation of this valve was most satisfactory.

Fuel Flow Control Needle Valve. Hoke and other types of metering valves were found unsatisfactory, so a special valve was designed and made. This valve is shown in Fig. 4. The thread of fifty turns per inch on the valve stem, and the small taper of the needle valve, permitted accurate control of fuel flow for the range desired. The valve seat and guide holder were made separate from the valve body so that different seats might be used. It was originally intended to use various valve stems if necessary, each with a different taper for the needle valve so that different seats might have been required. A steel scale was mounted parallel to the valve stem so that a graduated brass dial on the end of the stem could be used to indicate needle valve settings.

Induction System. The air flow was measured by means of an air barrel, in the end of which was mounted any of a graduated set of orifices whose calibration curves, as furnished by the Bureau of Standards, are appended. Pressure differential between the barrel and the atmosphere was measured by an inclined manometer. In the air line just beyond the barrel was a throttle valve, followed by a check valve to prevent excessive pressures reaching the air barrel, should explosions occur in the mixing tank. The check valve was mounted with the flapper hinge line slightly displaced from the

vertical so that the force of gravity held the valve open against the stop when air was not flowing. The fact that the mixture inlet manometer indicated very steady pressure differences throughout all runs is evidence that the amount of check valve opening did not fluctuate during engine operation.

The mixing tank was surrounded by a water jacket whose temperature was controlled by steam and water so that the fuel-air mixture temperatures could be maintained within very close limits. A manometer to the mixing tank measured inlet pressure. Between the mixing tank and engine manifold were placed two brass wire screens as flame traps to prevent backfires from igniting the contents of the mixing tank. As an additional precaution, in the event of backfires, the mixing tank had a pressure relief valve set to operate at a pressure of four pounds per square inch. The throttle orifices were placed just before the inlet manifold.

Ignition System. The distributor was the standard Ford 1941 model, except for modifications to permit manual control of the spark advance through a range of about 50 degrees. In these modifications the counterweights of the automatic spark control were taped and wired down. No vacuum connection from engine manifold to distributor was made. Slots were cut in the distrib-

utor and counterweight cases so that the disk holding the breaker points could be rotated from the outside, through an angle of about 50 degrees, thus changing the spark advance. A wire connection between the breaker arms and the coil was soldered in place. The disk holding the breaker assembly was moved to and retained at various settings by means of a small locking bolt extending outside the case.

Cooling Water System. The cooling system finally adopted is shown in Fig. 3. Water discharged from each bank passed through an adjustable thermostat and then through a common line to the reservoir tank, where it was cooled by the addition of cold water. The common discharge line was used only to take advantage of equipment available. A line from the bottom of the reservoir tank branched to the water inlets. Excess water, equal in amount to that of the cold water flowing into the reservoir tank, passed to the drain through an overflow line. Connections were also made directly from the cold water lines to the water jacket inlets to permit introduction of cold water to each bank if the thermostats became steam bound or did not operate. In order to relieve any pressure which might build up when the system was not in equilibrium, standpipes were placed between the pumps and the thermostats. The system

was quite satisfactory in operation. When equilibrium was reached the jacket temperatures could be closely controlled. Under these conditions the water circulated much as in the ordinary automobile installations, except that the water was cooled in the reservoir tank instead of by a radiator. By carefully controlling the amount of cold water entering the system, the inlet temperature to the jackets could be held to any desired value so that the thermostats could maintain the jacket temperature at the outlets within very close limits. The cold water supply direct to the jackets, and the standpipe overflows, came into operation only when the system was still warming up or for any other reason was not in equilibrium.

2. Difficulties Encountered.

Fuel Pressure. Control of fuel pressure was the first problem encountered. Several types of pressure relief and control valves were tried and found unsatisfactory before the modified control valve previously described was installed. For the control of fuel flow, Hoke and other available metering valves were tried, but they were found to be too sensitive or of too small capacity for the purpose. This fuel control problem was solved by the construction of the needle valve shown in Fig. 4.

<u>Cooling Water Temperatures</u>. As soon as the engine was operated it became apparent that to maintain constant speed, constant cylinder jacket temperatures were imperative. If these temperatures changed, the speed and power changed accordingly. A number of different cooling systems were tried. The original installation provided cooling by a simple recirculating line with cold water injected directly into the suction side of the pumps, and equivalent overflow from standpipes. This control was not sufficiently accurate so adjustable thermostats were installed in the original system without other change. This arrangement was very unstable and produced large and rapid oscillations of jacket temperatures.

Cooling by use of a reservoir tank was then tried. In the initial installation of this system the discharge lines from the banks led to the bottom of the reservoir tank so that good mixing of the warm discharge and the cold water entering at the top would result. However, convection currents were so great that they overcame the engine water pump pressure and so prevented coolant water circulation. When the discharge lines were changed to enter the top of the tank, satisfactory operation and accurate control of inlet temperature resulted. The discharge lines to the tank and the suction lines from the

tank to the engine pumps were placed as nearly level as possible to closely simulate the actual conditions in an automobile installation. With constant and controllable inlet temperatures, the outlet temperatures remained constant.

Backfires. Despite safety valve, flame traps, and a check valve, the first backfire buckled hose connections and blew fluid from manometers. As a result, flame traps were improved, and the safety valve on the mixing tank was reset to a lifting pressure of four pounds per square inch by use of lighter springs. It was found that careful checking of spark advance prior to starting was most effective in eliminating backfires.

Incompleted Runs at 4000 r.p.m. To save the engine, calibration runs were not made at 4000 r.p.m. as it was originally intended to run at only the lower speeds until all other runs had been completed. When these lower speed runs were completed, a series of runs was made at 4000 r.p.m. full power. Upon their completion, a series at about one-quarter power was made using a throttling orifice of estimated size. From the data of these two series, the orifice calibration curve for 4000 r.p.m. was then faired in to get the orifice sizes necessary for runs at one-half and three-quarter power.

Half power runs were then made. On the next series

of runs, at three-quarter power, four runs had been made when a bearing burned out and the connecting rods on the two rear cylinders broke. This accident damaged the engine beyond repair and terminated the investigation.

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(3) E.E.Fawkes, E.H.Guilbert, J.H.Morse, jr., R.R.Porter, and Harry Sosnoski, THE RELATION BETWEEN OCTANE NUMBER, COMPRESSION RATIO and FUEL-AIR RATIO AS DETERMINED BY INCIPIENT KNOCK, Special Report, Course 2.802, Massachusetts Institute of Technology, Cambridge, Mass., 1941.

4. <u>Data Sheets.</u>

Summary of Some Engine Conditions.

Laboratory Data Sheets.

Air Measuring Orifices - Calibration Curves.

SUMMARY OF SOME ENGINE CONDITIONS

Approx. Power Ratio	Throttle Orifice		Ave. Oil Temp. °F.	Ave. Water Inlet Temp. °F.	Ave. Water Out Temp. °F.	Max. BMEP Lbs. per sq.in.	Best-Power Ratio
	Diam. in.	Area Sq.in.					
Speed - 1000 r.p.m.							
1/4	3/8	0.110	159	150	173	12.08	0.169
1/2	1/2	0.196	162	149	171	35.25	0.493
3/4	5/8	0.307	167	150	176	51.65	0.723
1.0	Full	Full	166	149	180	71.50	1.000
Speed - 2000 r.p.m.							
1/4	9/16	0.249	187	150	169	14.34	0.210
1/2	11/16	0.371	187	150	168	33.70	0.495
3/4	7/8	0.602	191	150	172	51.20	0.751
1.0	Full	Full	196	150	172	68.20	1.000
Speed - 3000 r.p.m.							
1/4	23/32	0.406	199	151	173	17.92	0.263
1/2	7/8	0.602	196	141	170	36.70	0.538
3/4	1-1/16	0.887	197	140	170	52.10	0.765
1.0	Full	Full	198	141	172	68.20	1.000
Speed - 4000 r.p.m.							
1/4	13/16	0.518	210	140	151	8.97	0.173
1/2	1	0.785	216	140	153	30.00	0.576
3/4	1-3/16	1.108	216	140	154	40.00	0.769
1.0	Full	Full	208	140	156	52.00	1.000

Inlet Pressure - 710 mm Hg.
Inlet Temperature - 125 °F.
Exhaust Pressure - 32 in. Hg. at 1000, 2000 amd 3000.
35 in. Hg. at 4000 r.p.m.

DATE 2 Mar. 1941 EXPERIMENT No. Orifice Calibration M. I. T. AERO ENGINE LABORATORY

ENGINE FORD V-8, 1935 FUEL 100 OCTANE S.G. ______ WET BULB ______ DRY BULB ______

BORE 3 1/16" STROKE 3 3/4" COMPRESSION RATIO 6.30 — 1 SPARK ADVANCE VARIABLE BAROMETER (ACT.) ______ (CORR.) ______

CONSTANTS B.M.E.P. = B.L. x B.H.P. = $\frac{\text{B.L. x R.P.M.}}{3000}$ B.S.F.C. = $\frac{}{\text{SECS. x B.H.P.}}$

REMARKS	TIME	RUN	R.P.M.	BRAKE #	B.M.E.P. #/□	B.H.P.	B.S.F.C.	TIME FOR CC	TEMP. WATER	TEMP. OIL	OIL PRES.	F/A	Spark Advance		Throttle Orifice
		1	2000	21.6		14.4							18½°		5/8"
		2	2000	22.0		14.68							18½		5/8"
		3	1000	44.9		14.97							16		5/8"
		4	3000	4.4		4.4							23		5/8"
		5	1000	55.3		18.43							17		7/8"
		6	2000	47.0		31.35							16		7/8"
		7	3000	32.0		32.0							20		7/8"
		8	2000	53.0		35.4							17		1 1/16"
		9	1000	58.4		19.5							19		1 1/16"
		10	3000	46.3		46.3							19		1 1/16"
		11	2000	56.2		37.5							17		1 3/16"
		12	3000	51.3		51.3							19		1 3/16"
		13	1000	58.3		19.4							17		1 3/16"
		14	1000	60.4		20.1							16		Full
		15	2000	61.5		41.0							19		Full
		16	3000	61.5		61.5							18		Full

P_{ex} = 32" Hg.

P_{inlet} = 710 mm Hg.

DATE 10 APRIL '41 EXPERIMENT NO. THESIS — M. I. T. AERO ENGINE LABORATORY

ENGINE FORD V-8, 1935 FUEL 100 OCTANE S.G. ______ WET BULB ______ DRY BULB ______

BORE 3 1/16" STROKE 3 3/4" COMPRESSION RATIO 6.30 – 1 SPARK ADVANCE Variable BAROMETER (ACT.) ______ (CORR.) ______

CONSTANTS B.M.E.P. = B.L. × 1.195 B.H.P. = $\frac{B.L. \times R.P.M.}{3000}$ B.S.F.C. = $\frac{\quad}{SECS. \times B.H.P.}$

REMARKS	TIME	RUN	R.P.M.	BRAKE #	B.M.E.P. #/□	B.H.P.	B.S.F.C.	TIME FOR 100 CC	TEMP. WATER	TEMP. OIL	OIL PRES	F/A	FUEL #/SEC	AIR #/SEC	LAB. TEMP.	AIR BARREL (NAKOM)	FUEL TEMP.	AIR IN TEMP.	WATER IN TEMP.	SPARK ADV.	I.H.P.	I.M.E.P.	I.S.F.C.
	1035	64	1000	5.5	6.57	1.833	2.840	106.65	173/172	155	43	.0582	1.448	2.49	73	2.78	75	126	150	24°	9.300	33.37	.561
	1044	65		8.5	10.15	2.833	1.970	99.8	174/173	156	44	.0621	1.549	2.49	72	2.78	75	125	150	27	10.3	36.96	.541
	1053	66		10.05	12.02	3.35	1.755	94.4	174/173	157	44	.0656	1.637	2.49	72	2.78	75	125	151	25	10.817	38.81	.543
	1100	67		10.1	12.08	3.37	1.806	91.24	173/173	159	43	.0679	1.690	2.49	71½	2.78	75	125	149	21	10.837	38.87	.561
	1108	68		9.9	11.83	3.30	1.910	88.07	175/173	160	43	.0709	1.750	2.47	72	2.75	75	125	151	21	10.767	38.62	.585
	1119	69		9.5	11.35	3.17	2.102	83.27	174/173	160	43	.0754	1.851	2.457	73	2.73	76	126	150	19	10.637	38.16	.626
	1125	70		8.9	10.63	2.97	2.405	76.70	173/173	160	43	.0820	2.010	2.452	73	2.72	76	125	151	18	10.437	37.43	.693
	1134	71		8.6	10.28	2.87	2.642	73.41	172/172	160	43	.0860	2.105	2.446	73	2.70	76	125	150	21	10.337	37.08	.733
	1140	72		7.6	9.08	2.53	3.255	67.45	171/171	160	42	.0936	2.290	2.446	73	2.70	76	125	151	21	9.997	35.88	.825
	1149	73		6.6	7.89	2.20	4.055	62.32	171/171	160	43	.1017	2.480	2.438	73	2.68	76	125	151	26	9.667	34.68	.923
	1159	74		6.1	7.29	2.03	4.577	59.85	171/171	159	44	.1064	2.582	2.424	71½	2.66	76	125	151	26	9.497	34.09	.978
	1215	75		9.9	11.83	3.30	1.788	94.10	175/174	160	43	.0663	1.639	2.47	72	2.75	76	126	151	21	10.767	38.62	.549
FRICTION RUN		76	↓	22.4	26.79	7.47							×10⁻³ ↑	×10⁻² ↑									
17 APRIL, 1941	1029	129	1000	28.2	33.7	9.40	1.105	53.3	181/177	165	40	.0801	2.887	3.605	71	1.17	75	126	150	15°	16.07	57.6	.648
	1035	130		28.7	34.3	9.57	1.045	55.4	175/170	165	40	.0767	2.778	3.620	72	1.18	76	126	149	15	16.24	58.2	.615
	1043	131		29.5	35.25	9.83	.900	62.73	176/176	165	40	.0678	2.453	3.620	72	1.18	76	125	150	15½	16.50	59.2	.535
	1053	132		25.6	30.6	8.53	.895	72.70	172/169	163	40	.0582	2.117	3.640	72	1.19	76	126	148	19	15.20	54.5	.501
	1103	133		17.3	20.7	5.77	1.133	84.87	169/166	160	42	.0490	1.813	3.70	72	1.23	77	126	146	23	12.44	44.6	.523
	1111	134		29.2	34.9	9.73	.886	64.38	175/171	161	42	.0657	2.390	3.64	72	1.19	77	126	150	20	16.40	58.8	.523
	1120	135		29.0	34.7	9.67	1.001	57.30	174/170	162	41	.0738	2.686	3.64	72	1.19	77	124	150	16	16.34	58.6	.591
	1128	136		26.2	31.4	8.73	1.436	44.22	171/168	163	41	.0961	3.480	3.62	72	1.18	77	125	148	19	15.40	55.3	.814
	1135	137		27.1	32.4	9.03	1.302	47.10	172/168	162	41	.0903	3.268	3.62	72	1.18	77	125	149	18	15.70	56.3	.750
	1141	138		24.9	29.8	8.30	1.642	40.66	171/168	162	41	.1046	3.785	3.62	72	1.18	77	125	149	21	14.97	53.7	.911
	1149	139		24.0	28.7	8.00	1.852	37.80	170/167	161	41	.1125	4.071	3.62	72	1.18	77	125	149	27	14.67	52.6	1.000
	1157	140		23.4	28.0	7.80	1.962	36.20	170/167	161	41	.1174	4.251	3.62	72	1.18	77	124	149	27	14.47	51.9	1.060
FRICTION RUN			↓	20.0	23.9	6.67							×10⁻³ ↑	×10⁻² ↑									

Notes (10 April):
BAR. = 764.2 mm Hg.
AIR ORIFICE = 1"
THROTTLE " = 3/8"
FUEL PRESS = 10#/in²

Notes (17 April):
BAR = 760.6 mm Hg.
AIR ORIFICE = 1 1/2"
THROTTLE " = 1/2"
FUEL PRESS. = 10#/in²

DATE 17 APRIL 1941 EXPERIMENT NO. THESIS — M. I. T. AERO ENGINE LABORATORY

ENGINE ______ FUEL ______ S.G. ______ WET BULB ______ DRY BULB ______

BORE ___ STROKE ___ COMPRESSION RATIO ______ SPARK ADVANCE ______ BAROMETER (ACT.) ______ (CORR.) ______

CONSTANTS B.M.E.P. = B.L. × 1.195 B.H.P. = $\frac{B.L. \times R.P.M.}{3000}$ B.S.F.C. = $\frac{}{SECS. \times B.H.P.}$

REMARKS	TIME	RUN	R.P.M.	BRAKE #	B.M.E.P. #/□"	B.H.P.	B.S.F.C.	TIME FOR/ALC	TEMP. WATER	TEMP. OIL	OIL PRES	F/A	FUEL #/SEC	AIR #/SEC	LAB TEMP	AIR BARREL ("Alcohol)	FUEL TEMP	AIR-IN TEMP	WATER IN TEMP	SPARK ADV.		I.H.P.	I.M.E.P.	I.S.F.C.	
	1315	141	1000	43.2	51.65	14.40	.747	51.47	17[illegible]/173	164	41	.0689	2.99	4.35	74	1.73	78	124	150	15		21.2	76.0	.508	
	1327	142	↓	42.0	50.20	14.00	.956	41.35	178/173	166	41	.0857	3.72	4.35	74	1.73	79	126	150	15		20.8	74.6	.643	
	1337	143		40.7	48.7	13.57	1.134	36.0	173/170	167	41	.0980	4.275	4.37	75	1.74	79	125	150	17		20.37	73.1	.757	
	1347	144		40.0	47.85	13.33	1.225	33.9	176/172	167	41	.1040	4.54	4.37	76	1.74	79	124	148	19		20.13	72.2	.812	BAR. = 757.7 m.m. Hg.
	1401	145		37.2	44.6	12.40	1.500	29.78	173/169	166	40	.1178	5.17	4.39	77	1.75	80	124	151	25		19.20	68.9	.970	
	1415	146		42.0	50.2	14.00	.883	44.8	186/173	166	41	.0786	3.435	4.37	77	1.74	81	124	150	15		20.8	74.6	.597	AIR ORIFICE = 1½"
	1425	147		42.4	50.75	14.13	.820	47.7	184/173	169	40	.0744	3.23	4.35	76	1.73	80	124	152	14		20.93	74.15	.556	
	1430	148		42.4	50.75	14.13	.810	48.4	178/173	169	40	.0733	3.18	4.35	76	1.73	80	125	150	15		20.93	76.15	.547	THROTTLE ORIFICE = 5/8"
	1438	149		42.6	51.0	14.20	.763	51.24	179/173	169	40	.0693	3.01	4.35	76	1.73	80	124	150	15		21.0	75.4	.516	
	1446	150		42.2	50.5	14.07	.704	56.1	177/173	169	40	.0634	2.75	4.34	76	1.72	80	125	150	15		20.87	74.9	.474	FUEL PRESSURE = 10 #/in²
	1500	151		40.0	47.85	13.33	.710	58.63	179/173	169	40	.0599	2.63	4.39	76	1.75	80	125	150	16		20.13	72.2	.470	
	1512	152		33.0	39.45	11.00	.743	67.75	174/170	167	40	.0513	2.27	4.43	77	1.79	81	125	149	23		17.80	63.8	.459	
	1524	153	↓	42.5	50.8	14.17	.713	54.58	190/173	167	40	.0647	2.81	4.35	77	1.73	81	126	149	15		20.97	75.2	.482	
FRICTION RUN				20.4	24.4	6.80	.						×10⁻³ ↑	×10⁻² ↑											
8 APRIL, 1941	1427	49	1000	50	59.7	16.67	.585	56.9	177/176	156	42	.0496	2.705	5.45	75	2.68	77	124	148	20		23.44	84.0	.415	
	1435	50	↓	52.3	62.5	17.43	.601	52.9	179/178	164	40	.0534	2.913	5.45	76	2.68	77	127	150	18		24.20	86.8	.433	
	1442	51		59.3	70.9	19.77	.671	41.85	188/180	165	40	.0676	3.682	5.45	76	2.68	77	125	150	15		26.54	95.2	.500	
	1448	52		59.0	70.5	19.67	.792	35.6	181/178	161	40	.0794	4.325	5.45	76	2.68	77	125	150	13		26.44	94.8	.589	BAR = 760.6 m.m. Hg.
	1453	53		57.6	68.9	19.20	.983	29.58	179/178	167	40	.0955	5.21	5.45	75	2.68	77	126	150	13.5		25.87	93.2	.722	
	1459	54		59.8	71.5	19.93	.670	41.5	175/175	166	40	.0680	3.710	5.45	75	2.68	77	125	149	14		26.70	95.8	.500	AIR ORIFICE = 1½"
	1507	55		58.6	70.0	19.53	.644	44.02	183/179	166	40	.0643	3.498	5.44	75	2.67	77	125	150	12		26.30	94.3	.479	
	1515	56		58.7	70.1	19.57	.615	46.08	185/179	167	40	.0614	3.340	5.44	75	2.67	77	125	152	13		26.34	94.4	.457	THROTTLE ORIFICE = FULL
	1521	57		55.2	66.0	18.40	.588	51.2	186/180	167	40	.0553	3.007	5.44	75	2.67	77	125	150	13		25.17	90.3	.430	
	1529	58		58.5	69.9	19.50	.742	38.3	185/178	165	40	.0739	4.020	5.44	75	2.67	77	125	150	13.5		26.27	94.2	.550	FUEL PRESSURE = 10 #/in²
	1535	59		54.7	65.4	18.23	1.152	26.37	178/176	171	40	.1073	5.840	5.44	75	2.67	77	125	150	18		25.00	89.7	.840	
	1545	60		54.7	65.4	18.23	1.231	24.66	173/173	170	40	.1148	6.245	5.44	75	2.67	77	125	146	21		25.00	89.7	.898	
	1552	61	↓	53.7	64.2	17.90	1.321	23.42	174/174	170	40	.1209	6.563	5.44	75	2.67	77	125	149	27		24.67	88.5	.958	
FRICTION RUN				20.3	24.3	6.77							×10⁻³ ↑	×10⁻² ↑											

DATE ______ EXPERIMENT NO. THESIS ______ M. I. T. AERO ENGINE LABORATORY

ENGINE ______ FUEL ______ S.G. ______ WET BULB ______ DRY BULB ______

BORE ___ STROKE ___ COMPRESSION RATIO ______ SPARK ADVANCE ______ BAROMETER (ACT.) ______ (CORR.) ______

CONSTANTS $B.M.E.P. = B.L. \times 1.195$ $B.H.P. = \frac{B.L. \times R.P.M.}{3000}$ $B.S.F.C. = \frac{}{SECS. \times B.H.P.}$

REMARKS	TIME	RUN	R.P.M.	BRAKE #	B.M.E.P. #/□"	B.H.P.	B.S.F.C.	TIME FOR 100 CC	TEMP. WATER	TEMP. OIL	OIL PRES	F/A	FUEL #/SEC	AIR #/SEC	LAB. TEMP.	AIR BORED ("Alcohol)	FUEL TEMP.	AIR IN TEMP.	WATER IN TEMP.	SPARK ADV.		IHP	IMEP	ISFC	
17 APRIL, 1941	1630	154	2000	12.0	14.34	8.00	1.688	41.0	173/168	187	53	.0701	3.75	5.35	78	2.61	79	125	150	22°		26.20	46.96	.515	
	1637	155		11.8	14.10	7.86	1.665	42.3	170/169	188	54	.0678	3.64	5.36	76	2.61	80	126	150	22°		26.06	46.72	.502	
	1641	156		10.1	12.08	6.74	1.742	47.1	170/167	188	53	.0604	3.26	5.38	76	2.63	80	125	148	26°		24.94	44.70	.470	BAR = 755.4 mm Hg.
	1648	157		8.3	9.92	5.53	2.011	49.78	169/167	187	53	.0574	3.09	5.38	76	2.64	80	125	149	26°		23.73	42.54	.469	
	1654	158		5.2	6.21	3.47	3.050	52.4	168/166	186	53	.0544	2.94	5.40	75	2.65	80	125	150	24°		21.67	38.83	.488	AIR ORIFICE = 1½"
	1659	159		11.4	13.62	7.60	2.000	36.45	170/168	187	53	.0787	4.22	5.36	76	2.61	80	125	151	22°		25.80	46.24	.589	
	1705	160		11.6	13.87	7.74	1.898	37.7	169/167	187	52	.0761	4.08	5.36	76	2.61	80	124	149	22°		25.94	46.49	.566	THROTTLE ORIFICE = 9/16"
	1709	161		11.9	14.22	7.94	1.792	38.93	170/169	188	53	.0736	3.95	5.36	76	2.61	80	124	150	22°		26.14	46.84	.544	
	1714	162		10.6	12.67	7.07	2.270	34.5	171/168	189	53	.0834	4.46	5.35	76	2.60	80	126	151	21°		25.27	45.29	.635	FUEL PRESSURE = 10#/in²
	1720	163		10.1	12.07	6.74	2.537	32.4	168/167	187	54	.0888	4.75	5.35	76	2.60	79	125	148	24°		24.94	44.69	.685	
	1727	164		9.8	11.72	6.53	2.778	30.5	168/166	187	53	.0949	5.04	5.31	75	2.56	79	124	149	22°		24.73	44.34	.733	
	1737	165		8.9	10.63	5.93	3.165	29.5	168/166	186	54	.0985	5.21	5.29	75	2.54	79	127	150	24°		24.13	43.25	.777	
FRICTION RUN			↓	27.3	32.62	18.20							×10⁻³ ↑	×10⁻² ↑											
								135.8 cc.																	
18 APRIL, 1941	0913	166	2000	26.0	31.07	17.33	1.261	34.38	166/169	185	52	.0868	6.07	6.99	76	4.43	77	124	150	20		34.66	62.14	.630	
	0920	167		24.0	28.68	16.00	1.621	28.98	165/169	184	52	.1032	7.20	6.98	76	4.42	77	125	150	23		33.33	59.75	.778	
	0930	168		20.6	24.62	13.73	2.009	27.24	[illegible]	183	53	.1097	7.66	6.98	76	4.42	77	125	148	21		31.06	55.69	.888	BAR. = 759.5 mm Hg.
	0938	169		27.4	32.73	18.27	1.103	37.27	167/169	185	52	.0798	5.59	7.00	76	4.44	78	127	149	20		35.60	63.80	.565	
	0944	170		27.8	33.22	18.53	.886	45.68	[illegible]	188	52	.0650	4.56	7.01	76	4.46	79	124	152	21		35.86	64.29	.458	AIR ORIFICE = 1½"
	0950	171		25.6	30.60	17.07	.902	48.85	[illegible]	189	52	.0608	4.28	7.03	76	4.48	79	125	153	21		34.40	61.67	.447	
	0956	172		22.3	26.66	14.88	.957	52.72	[illegible]	188	52	.0557	3.46	7.10	78	4.59	80	124	150	24		32.21	57.73	.442	THROTTLE ORIFICE = 11/16"
	1003	173		16.0	19.12	10.67	1.212	58.03	[illegible]	186	53	.0502	3.59	7.15	78	4.65	80	124	150	24		28.00	50.19	.462	
	1010	174		19.7	23.53	13.13	1.029	55.48	[illegible]	185	53	.0526	3.76	7.16	78	4.66	80	125	149	26		30.46	54.60	.444	FUEL PRESS. = 10#/in²
	1018	175		28.2	33.70	18.81	.931	42.93	171/171	189	52	.0694	4.86	7.00	78	4.46	80	125	152	20		36.14	64.77	.484	
	1024	176		27.7	33.12	18.48	1.006	40.42	168/170	189	52	.0738	5.16	7.00	78	4.46	80	124	149	20		35.81	64.19	.520	
	1035	177		25.4	30.34	16.93	1.394	31.82	[illegible]	189	52	.0936	6.56	7.01	77	4.47	80	125	149	22		34.36	61.41	.690	
FRICTION RUN			↓	26.0	31.07	17.83							×10⁻³ ↑	×10⁻² ↑											

DATE ______ EXPERIMENT NO. Thesis M. I. T. AERO ENGINE LABORATORY

ENGINE ______ FUEL ______ S.G. ______ WET BULB ______ DRY BULB ______

BORE ___ STROKE ___ COMPRESSION RATIO ______ SPARK ADVANCE ______ BAROMETER (ACT.) ______ (CORR.) ______

CONSTANTS B.M.E.P. = B.L. x 1.195 B.H.P. = $\frac{B.L. \times R.P.M.}{3000}$ B.S.F.C. = $\frac{\quad}{SECS. \times B.H.P.}$

REMARKS	TIME	RUN	R.P.M.	BRAKE #	B.M.E.P. #/□"	B.H.P.	B.S.F.C.	TIME FOR ⅓ CC	TEMP. WATER	TEMP. OIL	OIL PRES	F/A	FUEL #/SEC	AIR #/SEC	LAB. TEMP.	AIR BARREL ("Alcohol)	FUEL TEMP.	AIR IN TEMP	WATER IN TEMP	SPARK ADV.		I.H.P.	I.M.E.P.	I.S.F.C.	
18 April, 1941	1055	178	2000	40.2	48.08	26.80	1.192	23.53	169/170	189	52	.1000	8.86	8.86	78	2.23	80	125	151	21		43.28	77.61	.737	
	1101	179	↓	37.4	44.70	24.92	1.378	21.83	166/168	188	53	.1072	9.53	8.89	78	2.24	80	125	146	22		41.40	74.23	.829	
	1108	180		40.9	48.88	27.25	1.092	25.21	171/171	189	52	.0932	8.26	8.86	78	2.23	80	125	152	20		43.73	78.41	.680	
	1113	181		41.6	49.72	27.65	.994	27.28	171/171	190	52	.0868	7.64	8.80	79	2.20	80	125	151	19		44.13	79.25	.622	BAR = 759.5
	1121	182		42.2	50.48	28.15	.884	30.16	173/173	192	51	.0786	6.91	8.80	79	2.20	80	125	152	18		44.63	80.01	.558	AIR ORIFICE = 2"
	1126	183		42.9	51.20	28.60	.782	33.54	171/172	192	52	.0710	6.21	8.75	78	2.17	80	125	150	18		45.08	80.73	.496	THROTTLE ORIFICE = 7/8"
	1132	184		41.7	49.90	27.80	.703	38.41	175/174	193	51	.0624	5.43	8.70	78	2.15	80	125	151	21		44.38	79.43	.441	FUEL PRESS. = 10 #/in.²
	1138	185		42.3	50.55	28.20	.748	35.54	174/174	194	51	.0673	5.86	8.71	79	2.16	81	126	152	17		44.58	80.08	.472	
	1145	186		39.1	46.70	26.07	.701	40.99	171/171	194	51	.0577	5.08	8.80	79	2.20	81	126	150	22		42.55	76.23	.429	
	1150	187		35.6	42.55	23.72	.730	43.42	168/170	192	52	.0540	4.81	8.90	79	2.25	81	125	148	23		40.20	72.08	.431	
	1158	188		30.0	35.85	20.00	.790	47.45	169/169	190	52	.0488	4.39	8.99	80	2.31	81	125	150	25		36.48	65.38	.433	
	1206	189		42.5	50.80	28.32	.788	33.62	171/171	192	51	.0709	6.20	8.75	79	2.15	82	124	149	18		44.80	80.33	.478	
FRICTION RUN			↓	24.7	29.53	16.48							×10⁻³ ↑	×10⁻² ↑											
18 April, 1941	1322	190	2000	56.9	68.0	37.9	.630	31.25	177/175	198	50	.0648	6.64	10.23	81	3.00	83	125	152	20		52.7	94.5	.453	
	1327	191	↓	54.4	65.0	36.2	.615	33.60	175/174	198	50	.0602	6.18	10.27	81	3.02	83	126	151	21		51.0	91.5	.436	
	1332	192		51.1	61.1	34.1	.614	35.68	171/171	197	50	.0559	5.81	10.39	81	3.10	83	125	149	22		48.9	87.6	.428	
	1339	193		46.7	55.9	31.2	.624	37.35	171/171	195	51	.0515	5.41	10.50	80	3.16	83	125	151	26		46.0	82.4	.423	BAR. = 760.0 mm Hg
	1350	194		41.4	49.5	27.6	.656	41.22	171/171	193	52	.0473	5.04	10.63	81	3.25	84	124	153	26		42.4	76.0	.428	AIR ORIFICE = 2"
	1401	195		56.5	67.5	37.7	.720	27.52	175/174	197	50	.0732	7.55	10.31	80	3.04	84	125	152	17		52.5	94.0	.518	THROTTLE ORIFICE = FULL
	1407	196		57.1	68.2	38.1	.676	29.34	175/174	198	50	.0692	7.09	10.23	81	3.00	84	125	151	17		52.9	94.7	.482	FUEL PRESS. = 10 #/in.²
	1412	197		56.4	67.4	37.6	.746	26.62	174/174	199	50	.0757	7.80	10.29	82	3.03	84	125	150	19		52.4	93.9	.536	
	1418	198		55.6	66.5	37.1	.855	23.56	172/172	198	50	.0850	8.81	10.37	82	3.09	84	124	150	20		[illegible]	93.0	.611	
	1424	199		55.4	66.2	36.9	.965	21.03	170/171	196	51	.0940	9.88	10.51	83	3.19	84	127	[illegible]	[illegible]		51.7	92.7	.688	
	1430	200		54.9	65.6	36.6	1.028	19.88	171/171	196	50	.0989	10.45	10.57	84	3.22	84	125	151	21		51.4	92.1	.734	
	1436	201		53.2	63.6	35.5	1.181	17.84	169/170	195	51	.1085	11.63	10.72	82	3.30	84	125	150	23		50.3	90.1	.835	
	1441	202		50.2	60.0	33.5	1.289	17.30	168/170	193	51	.1119	12.0	10.72	82	3.30	84	125	148	22		48.2	86.5	.895	
FRICTION RUN			↓	22.2	26.5	14.8							×10⁻³ ↑	×10⁻² ↑											

DATE ______ EXPERIMENT NO. THESIS ______ M. I. T. AERO ENGINE LABORATORY

ENGINE ______ FUEL ______ S.G. ______ WET BULB ______ DRY BULB ______

BORE ____ STROKE ____ COMPRESSION RATIO ______ SPARK ADVANCE ______ BAROMETER (ACT.) ______ (CORR.) ______

CONSTANTS $B.M.E.P. = B.L. \times 1.195$ $B.H.P. = \frac{B.L. \times R.P.M.}{3000}$ $B.S.F.C. = \frac{}{SECS. \times B.H.P.}$

REMARKS	TIME	RUN	R.P.M.	BRAKE #	BMEP #/□"	B.H.P.	B.S.F.C.	TIME FOR 25CC	TEMP. WATER	TEMP. OIL	OIL PRES	F/A	FUEL #/SEC	AIR #/SEC	LAB TEMP	AIR BARREL ("Alcohol)	FUEL TEMP	AIR IN TEMP.	WATER IN TEMP.	SPARK ADV. °		I.H.P.	IMEP	ISFC	
10 APRIL, 1941	1424	77	3000	12.6	15.06	12.6	2.278	101.88	172/172	197	55	.0918	7.97	8.69	74	2.11	78	125	150	22		43.6	52.1	.658	
	1429	78	↓	14.0	16.73	14.0	1.932	107.26	172/172	197	54	.0862	7.51	8.71	74	2.13	78	125	149	23		45.0	53.8	.601	
	1437	79		14.6	17.45	14.6	1.743	114.88	173/173	198	55	.0811	7.07	8.71	72	2.13	78	124	152	21		45.6	54.5	.558	BAR. = 764.4 mm Hg
	1447	80		14.55	17.40	14.55	1.652	121.35	172/172	198	55	.0766	6.68	8.71	75	2.13	78	127	150	23		45.55	54.45	.527	AIR ORIFICE = 2"
	1510	81		15.0	17.92	15.0	1.483	131.40	174/174	199	57	.0709	6.18	8.71	73	2.13	79	125	153	25		46.0	55.0	.483	THROTTLE ORIFICE = 23/32"
	1518	82		15.0	17.92	15.0	1.416	137.73	172/172	200	57	.0674	5.90	8.75	73	2.14	79	125	148	27		46.0	55.0	.462	FUEL PRESS. = 10#/in²
	1526	83		14.6	17.45	14.6	1.385	144.89	173/173	200	57	.0639	5.61	8.77	73	2.15	79	125	152	23		45.6	54.5	.443	
	1539	84		12.1	14.47	12.1	1.555	155.38	172/172	200	57	.0594	5.225	8.80	73	2.16	79	124	150	26		43.1	51.5	.436	
	1548	85		8.4	10.04	8.4	2.178	42.53	172/172	200	57	.0561	4.96	8.84	73	2.18	80	125	150	22		39.4	47.1	.453	
	1559	86		11.3	13.51	11.3	2.650	25.28	172/172	200	57	.0955	8.31	8.70	73	2.12	80	124	151	22		42.3	50.6	.708	
	1608	87		14.8	17.69	14.8	1.591	32.11	175/175	204	57	.0751	6.54	8.70	73	2.12	80	125	156	23		45.8	54.7	.514	
FRICTION RUN			↓	31.0	37.06	31.0							×10⁻³ ↑	×10⁻² ↑											
								135.8cc																	
11 APRIL, 1941	1008	88	3000	26.8	32.0	26.8	.889	31.63	169/169	193	57	.0577	6.61	11.45	72	3.65	75	125	140	23		58.0	69.3	.411	
	1015	89	↓	23.5	28.1	23.5	.947	33.93	169/169	195	56	.0540	6.18	11.47	70	3.67	75	125	140	25		54.7	65.4	.407	
	1023	90		21.8	26.1	21.8	1.001	34.56	170/170	195	55	.0578	6.06	11.48	70	3.68	75	125	144	23		53.0	63.4	.412	BAR. = 767.3
	1031	91		26.7	31.9	26.7	.885	31.98	171/170	195	57	.0573	6.55	11.42	69	3.64	75	125	143	23		57.9	69.2	.407	AIR ORIFICE = 2"
	1042	92		28.3	33.8	28.3	.856	31.11	171/171	198	57	.0593	6.73	11.35	69	3.60	75	126	143	27		59.5	71.1	.407	THROTTLE ORIFICE = 7/8"
	1048	93		29.6	35.4	29.6	.840	30.30	170/170	196	57	.0611	6.91	11.33	68	3.58	75	125	141	24		60.8	72.7	.409	FUEL PRESS = 10#/in²
	1052	94		30.0	35.8	30.0	.846	29.69	170/170	195	57	.0623	7.05	11.31	70	3.57	75	125	139	22		61.2	73.15	.414	
	1058	95		30.1	36.0	30.1	.865	28.93	171/171	196	57	.0639	7.24	11.31	70	3.57	76	126	140	22		61.3	73.3	.425	
	1106	96		30.7	36.7	30.7	.894	27.52	172/171	198	57	.0672	7.61	11.31	70	3.57	76	125	143	21		61.9	74.0	.443	
	1114	97		30.5	36.4	30.5	.913	27.07	171/170	197	57	.0683	7.74	11.31	70	3.57	76	126	141	23		61.7	73.75	.451	
	1121	98		30.4	36.3	30.4	.963	25.77	170/170	197	57	.0717	8.13	11.33	70	3.58	76	124	140	24		61.6	73.6	.475	
	1128	99		30.0	35.8	30.0	1.135	22.12	170/169	196	56	.0834	9.46	11.33	69	3.58	76	125	141	24		61.2	73.15	.557	
	1134	100		29.2	34.9	29.2	1.272	20.27	169/168	195	55	.0909	10.31	11.34	69	3.59	76	125	139	23		60.4	72.2	.615	
	1148	101		27.8	33.2	27.8	1.425	19.02	168/168	193	55	.0969	11.00	11.35	71	3.60	76	125	139	21		59.0	70.5	.671	
FRICTION RUN			↓	31.2	37.3	31.2							×10⁻³ ↑	×10⁻² ↑											

DATE ________ EXPERIMENT NO. THESIS ________ **M. I. T. AERO ENGINE LABORATORY**

ENGINE ________ FUEL ________ S.G. ________ WET BULB ________ DRY BULB ________

BORE ____ STROKE ____ COMPRESSION RATIO ________ SPARK ADVANCE ________ BAROMETER (ACT.) ________ (CORR.) ________

CONSTANTS B.M.E.P. = B.L. x 1.195 B.H.P. = $\frac{B.L. \times R.P.M.}{3000}$ B.S.F.C. = $\frac{\quad}{SECS. \times B.H.P.}$

REMARKS	TIME	RUN	R.P.M.	BRAKE #	B.M.E.P. #/□"	B.H.P.	B.S.F.C.	TIME FOR [illegible] cc	TEMP. WATER	TEMP. OIL	OIL PRES	F/A	FUEL #/SEC	AIR #/SEC	LAB. TEMP.	AIR BARREL ("Alcohol)	FUEL TEMP	AIR IN TEMP	WATER IN TEMP	SPARK ADV.		IHP	IMEP	ISFC	
11 APRIL, 1941	1340	102	3000	43.2	51.6	43.2	.883	19.68	172/172	200	56	.0789	10.61	13.48	72	2.05	76	125	140	24		73.3	87.5	.521	
	1450	103	↓	43.2	51.6	43.2	.854	20.38	172/169	199	56	.0761	10.25	13.48	70	2.05	76	125	140	24		73.3	87.5	.503	
	1400	104		43.6	52.1	43.6	.778	22.19	171/170	196	56	.0699	9.41	13.48	70	2.05	76	126	144	20		73.7	88.0	.460	
	1410	105		43.5	52.0	43.5	.761	22.71	171/171	198	56	.0681	9.19	13.50	70	2.06	77	124	143	20		73.6	87.9	.449	
	1418	106		43.5	52.0	43.5	.742	23.26	172/172	198	56	.0665	8.96	13.48	71	2.05	77	126	143	19		73.6	87.9	.438	BAR = 768.7 m.m. Hg.
	1424	107		43.0	51.4	43.0	.719	24.28	171/171	197	56	.0637	8.59	13.50	70	2.06	77	123	141	20		73.1	87.3	.423	AIR ORIFICE = 2½"
	1432	108		40.5	48.4	40.5	.716	25.82	172/170	198	56	.0595	8.07	13.55	70	2.07	77	125	139	23		70.6	84.3	.411	THROTTLE ORIFICE = 1⅙"
	1439	109		37.5	44.9	37.5	.728	27.54	169/169	197	57	.0558	7.59	13.61	70	2.10	77	126	140	25		67.6	80.8	.403	FUEL PRESS. = 10 #/in²
	1450	110		32.5	38.9	32.5	.780	29.65	169/169	195	54	.0513	7.04	13.71	69	2.14	77	125	143	24		62.6	74.7	.404	
	1456	111		43.4	51.9	43.4	.835	20.72	172/171	195	56	.0748	10.08	13.48	71	2.05	77	125	141	20		73.5	87.8	.493	
	1503	112		43.0	51.4	43.0	.930	18.78	172/170	197	56	.0820	11.11	13.55	72	2.07	77	125	140	21		73.1	87.3	.546	
	1509	113		41.9	50.1	41.9	1.025	17.52	172/169	197	56	.0880	11.91	13.55	70	2.07	77	125	141	21		72.0	86.0	.595	
	1517	114		40.8	48.8	40.8	1.113	16.54	170/169	195	54	.0930	12.61	13.58	69	2.09	77	126	139	23		70.9	84.7	.640	
	1522	115		40.1	48.0	40.1	1.163	16.10	169/167	195	53	.0953	12.98	13.61	69	2.10	77	125	139	23		70.2	83.9	.665	
FRICTION RUN			↓	30.1	35.9	30.1							x10⁻³ ↑	x10⁻² ↑											
12 APRIL, 1941	0955	116	3000	56.1	67.0	56.1	.823	16.35	171/171	198	55	.0824	12.82	15.6	72	2.77	75	125	140	20		85.9	102.6	.538	
	1003	117	↓	56.9	68.0	56.9	.769	17.28	172/172	198	55	.0781	12.15	15.56	71	2.75	75	125	140	18		86.7	103.6	.505	
	1009	118		57.1	68.2	57.1	.676	19.55	174/172	199	55	.0692	10.73	15.5	70	2.72	76	124	142	18		86.9	103.8	.445	
	1016	119		54.8	65.5	54.8	.620	22.15	174/172	198	55	.0611	9.43	15.43	72	2.70	77	126	145	19		84.6	101.1	.401	BAR = 772.1 m.m. Hg
	1025	120		45.8	54.8	45.8	.644	25.50	170/169	198	56	.0618	8.19	15.8	72	2.83	78	125	139	26		75.6	90.4	.390	AIR ORIFICE = 2½"
	1037	121		52.0	62.1	52.0	.626	23.03	173/172	198	55	.0580	9.05	15.6	72	2.77	79	125	144	20		81.8	97.7	.398	THROTTLE ORIFICE = FULL
	1044	122		56.2	67.2	56.2	.644	20.72	174/173	199	55	.0649	10.07	15.5	72	2.72	79	126	143	19		86.0	102.8	.421	FUEL PRESS. = 10 #/in²
	1056	123		57.0	68.1	57.0	.709	18.58	172/172	200	55	.0723	11.22	15.52	74	2.73	80	125	143	18		86.8	103.7	.465	
	1104	124		55.4	66.1	55.4	.915	14.80	172/171	199	55	.0891	14.08	15.8	72	2.83	79	126	141	21		85.2	101.7	.595	
	1111	125		54.3	64.9	54.3	1.029	13.43	169/168	197	55	.0974	15.51	15.92	72	2.88	79	125	137	22		84.1	100.5	.664	
	1117	126		52.0	62.1	52.0	1.097	13.16	170/169	196	56	.0994	15.85	15.97	73	2.90	78	126	140	20		81.8	97.7	.698	
	1121	127		53.2	63.6	53.2	1.071	13.19	170/169	196	56	.0994	15.82	15.92	73	2.88	78	126	140	20		83.0	99.2	.686	
	1126	128		55.6	66.5	55.6	.930	14.50	174/174	199	56	.0908	14.38	15.83	72	2.84	78	124	144	21		85.4	102.1	.606	
FRICTION RUN			↓	29.8	35.6	29.8							x10⁻³ ↑	x10⁻² ↑											

DATE ______ EXPERIMENT NO. THESIS ______ M. I. T. AERO ENGINE LABORATORY

ENGINE ______ FUEL ______ S.G. ______ WET BULB ______ DRY BULB ______

BORE ___ STROKE ___ COMPRESSION RATIO ______ SPARK ADVANCE ______ BAROMETER (ACT.) ______ (CORR.) ______

CONSTANTS B.M.E.P. = B.L. × 1.195 B.H.P. = $\frac{B.L. \times R.P.M.}{3000}$ B.S.F.C. = $\frac{}{SECS. \times B.H.P.}$

REMARKS	TIME	RUN	R.P.M.	BRAKE #	B.M.E.P. #/□	B.H.P.	B.S.F.C.	TIME FOR 3 5CC	TEMP. WATER	TEMP. OIL	OIL PRES	F/A	FUEL #/SEC	AIR #/SEC	LAB. TEMP.	AIR BARREL ("Alcohol)	FUEL TEMP.	AIR IN TEMP	WATER IN TEMP.	SPARK ADV. °		I.H.P.	IMEP	ISFC	
21 APRIL, 1941	1358	215	4000	3.7	4.42	4.93	5.02	30.11	155/150	210	52	.0608	6.88	11.32	85	3.70	87	125	142	20		62.9	56.4	.394	
	1405	216	↓	6.7	8.01	8.93	3.43	24.35	155/150	207	52	.0758	8.51	11.22	83	3.64	88	125	143	23		66.9	60.0	.458	
	1419	217		7.5	8.97	10.00	2.84	26.30	152/148	210	52	.0702	7.88	11.22	84	3.64	89	127	139	19		68.0	61.0	.418	BAR. = 756.4 mm Hg
	1424	218		6.8	8.13	9.06	3.00	27.4	154/149	210	52	.0672	7.55	11.22	84	3.64	90	125	140	20		67.1	60.1	.406	AIR ORIFICE = 2"
	1435	219		4.0	4.78	5.34	4.56	30.60	153/149	210	52	.0602	6.76	11.25	84	3.65	90	126	140	22		63.3	56.8	.385	THROTTLE ORIFICE = 13/16"
	1449	220		0.4	.478	.534	41.1	34.00	151/150	201	52	.0541	6.08	11.27	85	3.67	91	126	140	18		58.5	52.5	.375	FUEL PRESS. = 10 #/in²
	1455	221		5.6	6.69	7.47	4.43	22.50	148/153	209	52	.0821	9.20	11.20	85	3.62	90	126	139	21		65.5	58.7	.506	
	1503	222		1.7	2.03	2.27	15.65	21.00	158/150	210	52	.0883	9.85	11.17	85	3.60	90	125	140	17		60.3	54.0	.588	
	1513	223		7.3	8.72	9.73	2.92	26.20	153/148	211	52	.0702	7.90	11.25	84	3.65	90	125	140	19		67.7	60.7	.420	
	1516	224		7.3	8.72	9.73	3.05	25.10	150/150	211	52	.0734	8.25	11.22	84	3.64	90	125	140	19		67.7	60.7	.439	
	1525	225		2.0	2.39	2.67	13.6	20.50	153/148	210	52	.0906	10.10	11.17	84	3.60	90	125	140	17		60.7	54.4	.600	
FRICTION RUN			↓	43.5	52.0	58.0							×10⁻³ ↑	×10⁻² ↑											

REMARKS	TIME	RUN	R.P.M.	BRAKE #	B.M.E.P. #/□	B.H.P.	B.S.F.C.	TIME FOR 3 5CC	TEMP. WATER	TEMP. OIL	OIL PRES	F/A	FUEL #/SEC	AIR #/SEC	LAB. TEMP.	AIR BARREL ("Alcohol)	FUEL TEMP.	AIR IN TEMP	WATER IN TEMP.	SPARK ADV. °		I.H.P.	IMEP	ISFC	
22 APRIL, 1941	1015	226	4000	23.3	27.85	31.07	1.369	17.65	165/153	215	48	.0815	11.82	14.52	76	2.44	80	125	140	27		83.74	75.05	.508	
	1025	227	↓	21.3	25.45	28.40	1.600	16.52	155/148	218	45	.0870	12.62	14.52	76	2.44	80	128	139	25		81.07	72.65	.560	
	1033	228		25.1	30.00	33.47	1.062	21.08	157/151	216	45	.0680	9.88	14.52	76	2.44	80	127	140	22		86.14	77.20	.413	BAR. = 765.3 mm Hg
	1042	229		24.8	29.65	33.07	1.023	22.13	158/153	215	45	.0648	9.41	14.52	76	2.44	81	123	142	24		85.74	76.85	.395	AIR ORIFICE = 2 1/2"
	1047	230		22.6	27.02	30.13	1.033	24.15	158/153	217	45	.0594	8.63	14.52	76	2.44	81	125	141	24		82.80	74.22	.375	THROTTLE ORIFICE = 1"
	1052	231		17.0	20.32	22.67	1.269	26.05	157/151	217	47	.0546	8.00	14.44	76	2.49	81	125	140	24		75.34	67.52	.382	FUEL PRESS = 10 #/in²
	1058	232		1.00	1.195	1.33	20.10	28.08	151/145	213	50	.0506	7.42	14.64	76	2.49	82	123	140	24		54.00	48.40	.495	
	1106	233		24.8	29.65	33.07	1.100	20.61	151/146	216	43	.0696	10.11	14.52	76	2.44	81	125	140	25		85.74	76.85	.424	
	1112	234		24.7	29.53	32.95	1.190	19.07	156/150	214	44	.0754	10.93	14.52	76	2.44	82	122	140	25		85.62	76.73	.460	
	1120	235		23.0	27.48	30.67	1.386	17.63	156/150	219	43	.0814	11.81	14.52	77	2.44	82	125	142	24		83.34	74.68	.510	
	1128	236		21.5	25.70	28.67	1.583	16.51	173/150	216	43	.0870	12.62	14.52	76	2.44	82	125	142	22		81.34	72.90	.559	
	1139	237		18.6	22.22	24.80	1.943	15.55	161/153	219	41	.0923	13.40	14.52	77	2.44	83	125	142	17		77.47	69.42	.623	
FRICTION RUN			↓	39.5	47.20	52.67							×10⁻³ ↑	×10⁻² ↑											

DATE ____ EXPERIMENT NO. ____ M. I. T. AERO ENGINE LABORATORY

ENGINE ____ FUEL ____ S.G. ____ WET BULB ____ DRY BULB ____

BORE ____ STROKE ____ COMPRESSION RATIO ____ SPARK ADVANCE ____ BAROMETER (ACT.) ____ (CORR.) ____

CONSTANTS B.M.E.P. = B.L. x ____ B.H.P. = $\frac{B.L. \times R.P.M.}{\quad}$ B.S.F.C. = $\frac{\quad}{SECS. \times B.H.P.}$

REMARKS	TIME	RUN	R.P.M.	BRAKE #	B.M.E.P. #/□	B.H.P.	B.S.F.C.	TIME FOR ? CC	TEMP WATER	TEMP OIL	OIL PRES	F/A	FUEL #/SEC	AIR #/SEC	LAB TEMP	AIR BARREL ("Alcohol)	FUEL TEMP	AIR IN TEMP	WATER IN TEMP	SPARK ADV.		IHP	IMEP	ISFC
[illegible] April, '41	1335	238	4000	30.3	36.2	40.4	1.318	14.87	158/153	216	43	.0895	14.79	16.51	79	3.27	83	125	140	20		95.9	86.0	.555
	1340	239	↓	32.4	38.7	43.2	1.162	14.88	155/150	216	44	.0848	13.98	16.48	78	3.24	84	124	140	21		98.7	88.5	.510
	1349	240		33.5	40.0	44.7	.981	17.03	156/151	216	41	.0738	12.18	16.50	80	3.27	85	124	138	21		100.2	79.8	.438
	1356	241		33.0	39.5	44.0	.911	18.65	157/152	217	43	.0676	11.13	16.46	79	3.25	85	125	140	21		99.5	89.3	.403
Estimated Friction			↓	41.6	49.8	55.5							x10⁻³ ↑	x10⁻² ↑										

Bar = 765.5 m.m. Hg

Air Orifice = 2½"

Throttle Orifice = 1 3/16"

Fuel Press. = 10 #/in.²

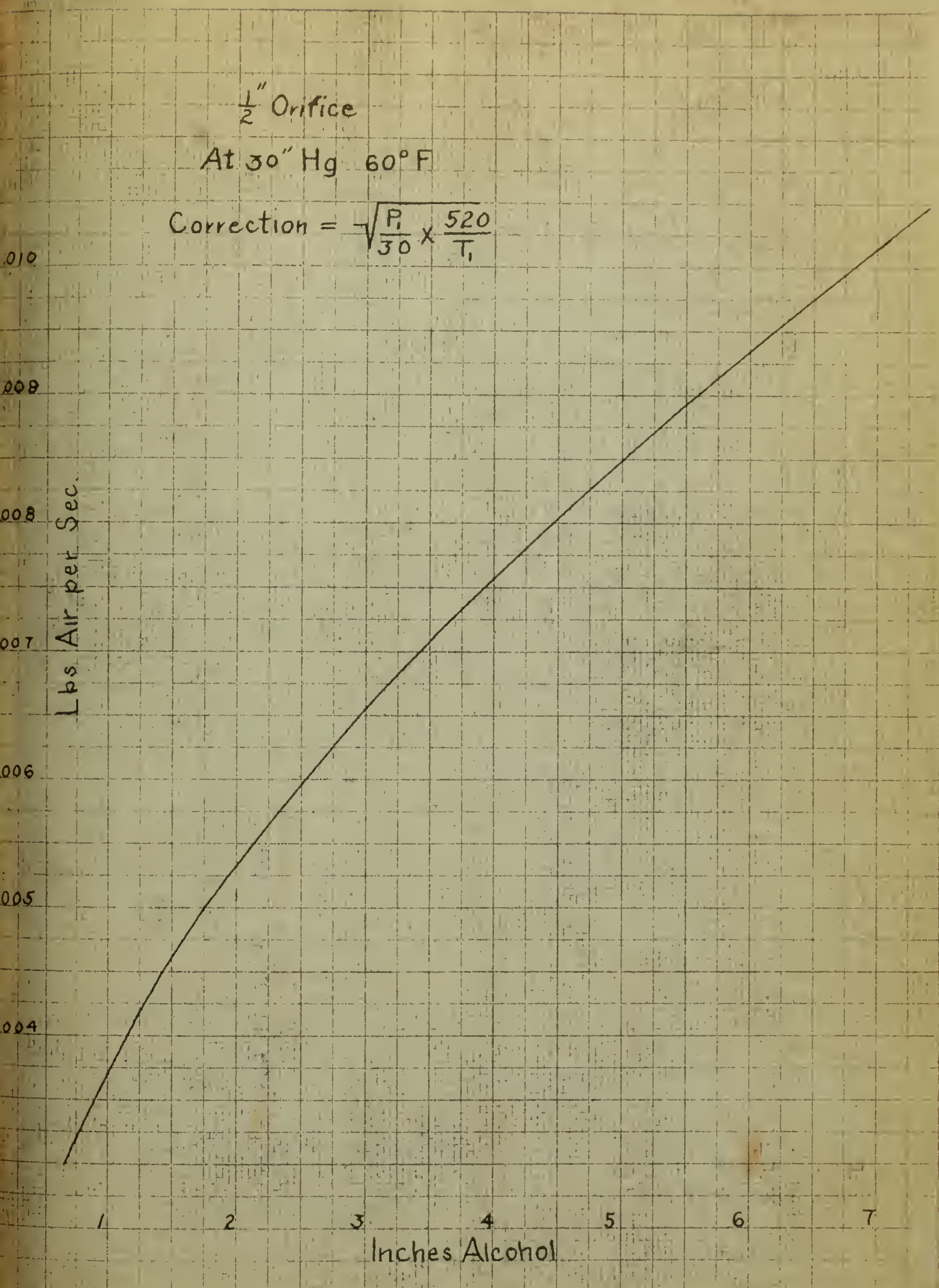
REMARKS	TIME	RUN	R.P.M.	BRAKE #	B.M.E.P. #/□	B.H.P.	B.S.F.C.	TIME FOR ? CC	TEMP WATER	TEMP OIL	OIL PRES	F/A	FUEL #/SEC	AIR #/SEC	LAB TEMP	AIR BARREL ("Alcohol)	FUEL TEMP	AIR IN TEMP	WATER IN TEMP	SPARK ADV.		IHP	IMEP	ISFC
19 April 1941	1040	203	4000	43.2	51.7	57.6	.890	14.58	161/153	208	51	.0750	14.26	19.00	81	2.02	84	125	142	26		114.3	102.6	.449
	1047	204	↓	43.5	52.0	58.1	.809	16.00	159/153	205	52	.0685	13.02	19.00	81	2.02	83	125	140	25		114.8	102.9	.409
	1051	205		43.5	52.0	58.1	.787	16.40	160/153	204	51	.0665	12.70	19.10	81	2.05	83	124	140	25		114.8	102.9	.398
	1059	206		39.3	47.0	52.5	.735	19.43	157/151	208	52	.0560	10.70	19.10	82	2.05	84	125	140	26		109.2	97.9	.353
	1108	207		29.5	35.3	39.4	.892	21.32	157/152	207	56	.0505	9.75	19.3	82	2.10	85	126	140	21		96.1	86.2	.366
	1115	208		43.3	51.8	57.8	.770	16.79	158/152	212	53	.0647	12.37	19.1	82	2.04	86	125	140	24		114.5	102.7	.389
	1121	209		42.1	50.4	56.2	.751	17.76	160/155	211	52	.0616	11.70	19.0	81	2.02	87	125	141	25		112.9	101.3	.373
	1129	210		426	50.9	56.7	.907	14.52	156/15?	211	53	.0751	14.28	19.0	81	2.03	88	127	139	21		113.4	101.8	.453
	1135	211		42.0	50.3	56.1	.985	13.49	158/152	212	53	.0805	15.37	19.1	82	2.05	88	125	140	23		112.8	101.2	.491
	1226	212		42.8	51.2	57.1	.767	17.02	158/151	200	50	.0640	12.18	19.0	82	2.03	89	126	140	24		113.8	102.1	.385
	1241	213		43.2	51.7	57.7	.919	14.09	160/155	198	51	.0767	14.72	19.2	83	2.07	89	125	143	23		114.4	102.6	.464
	1246	214		39.7	47.5	53.0	1.138	12.42	159/151	214	50	.0871	16.71	19.2	83	2.07	77	126	140	22		109.7	98.4	.549
Friction Run			↓	42.5	50.9	56.7							x10⁻³ ↑	x10⁻² ↑										

Bar = 763.7 m.m. Hg

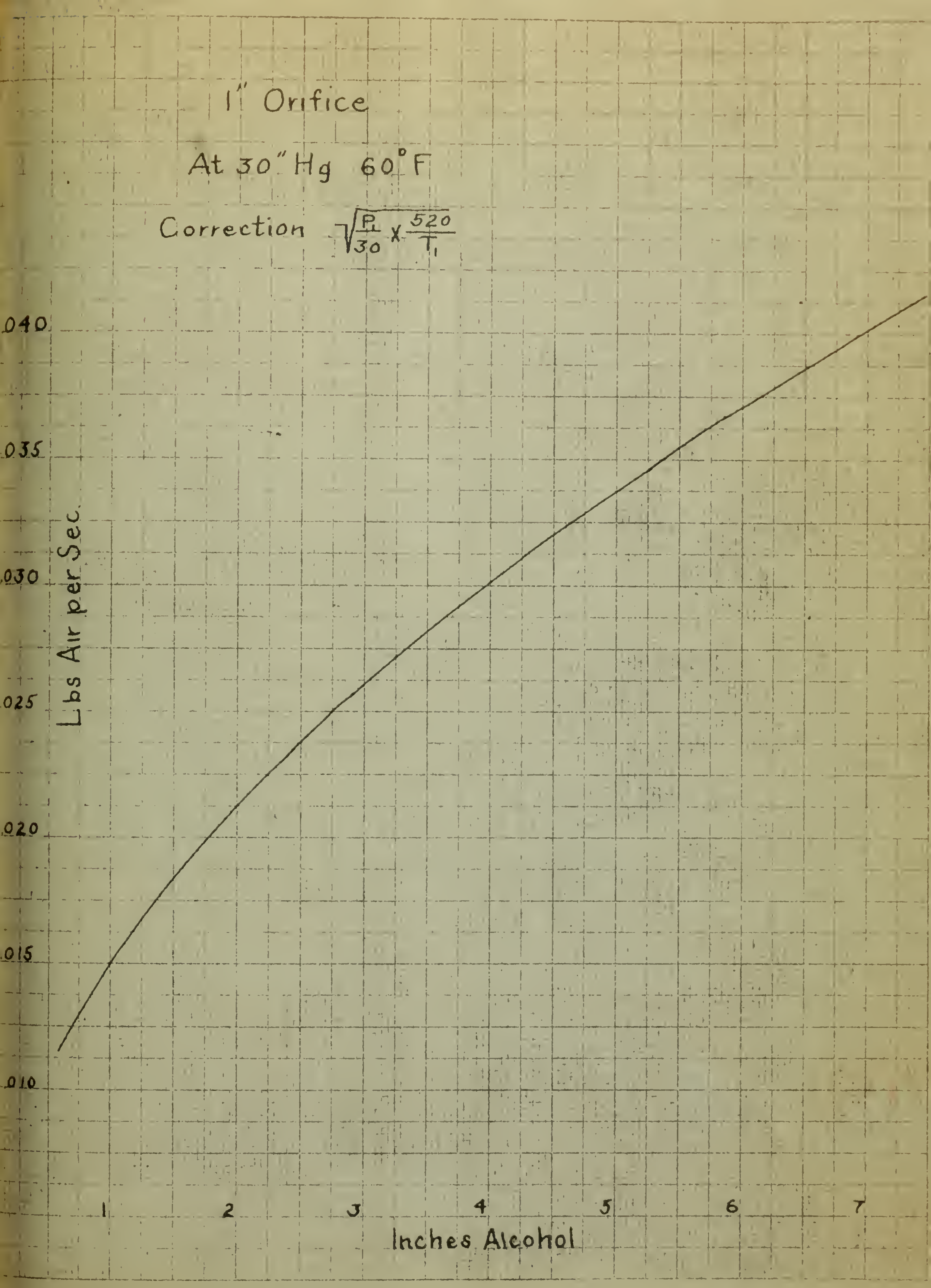
Air Orifice = 3"

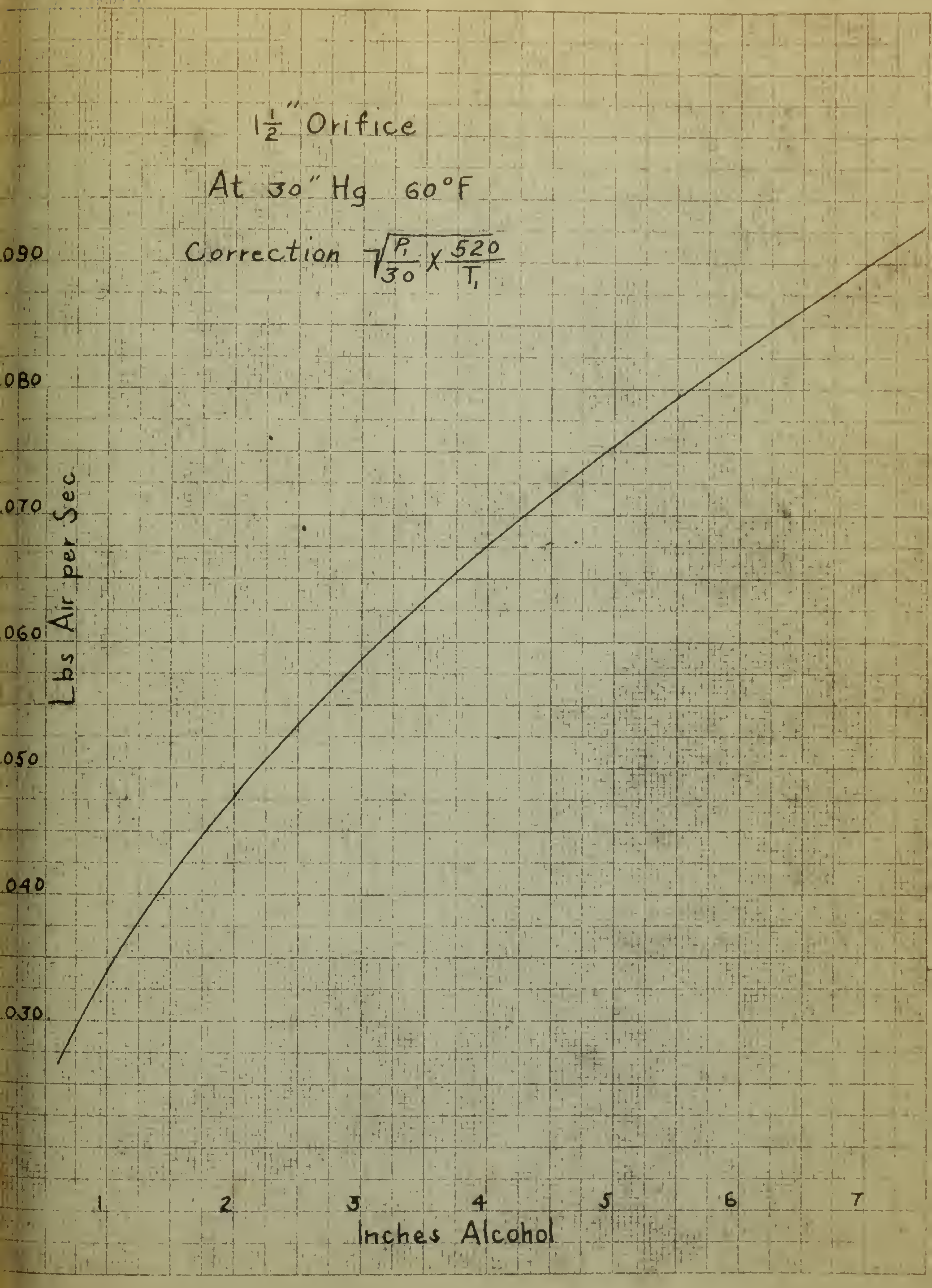
Throttle Orifice = Full

Fuel Press. = 10 #/in.²

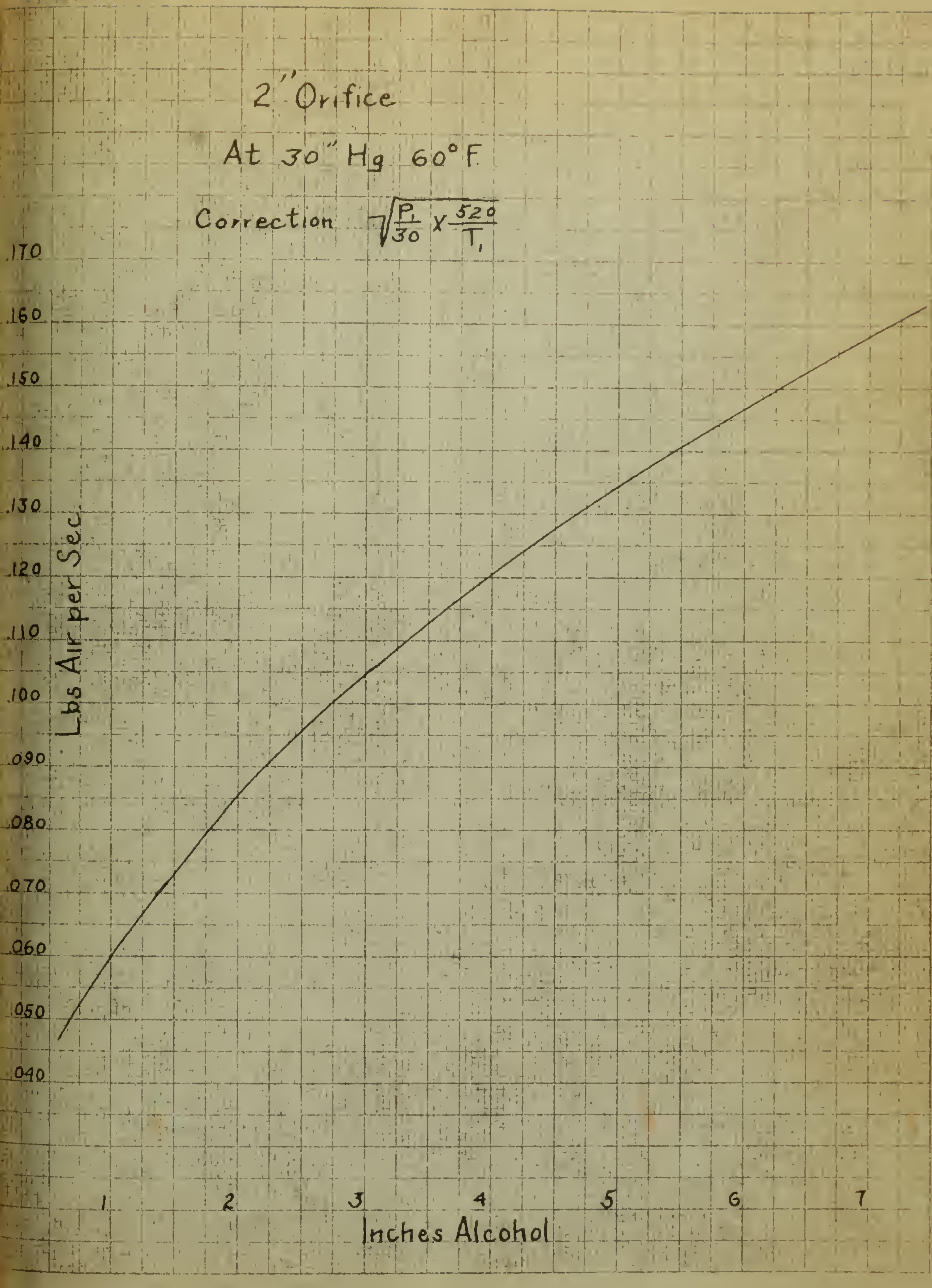


$\frac{1}{2}''$ Orifice
At 30" Hg 60° F
Correction = $\sqrt{\frac{P_1}{30} \times \frac{520}{T_1}}$
.010
008
.008
.007
.006
.005
.004
Lbs Air per Sec.
1
2
3
4
5
6
7
Inches Alcohol

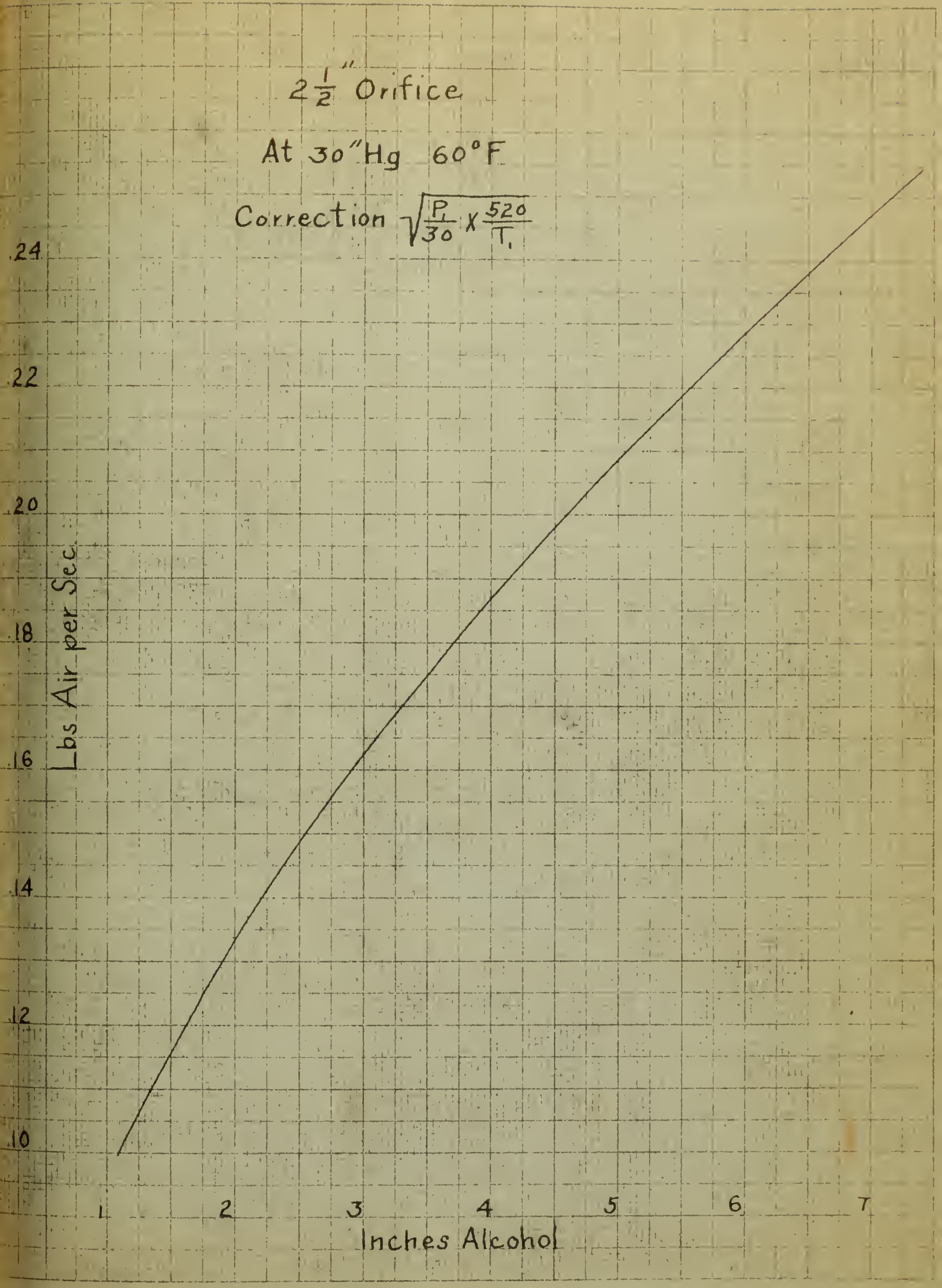

1" Orifice
At 30" Hg 60°F
Correction $\sqrt{\frac{P_1}{30} \times \frac{520}{T_1}}$
Lbs Air per Sec.
.040
.035
.030
.025
.020
.015
.010
1
2
3
4
5
6
7
Inches Alcohol



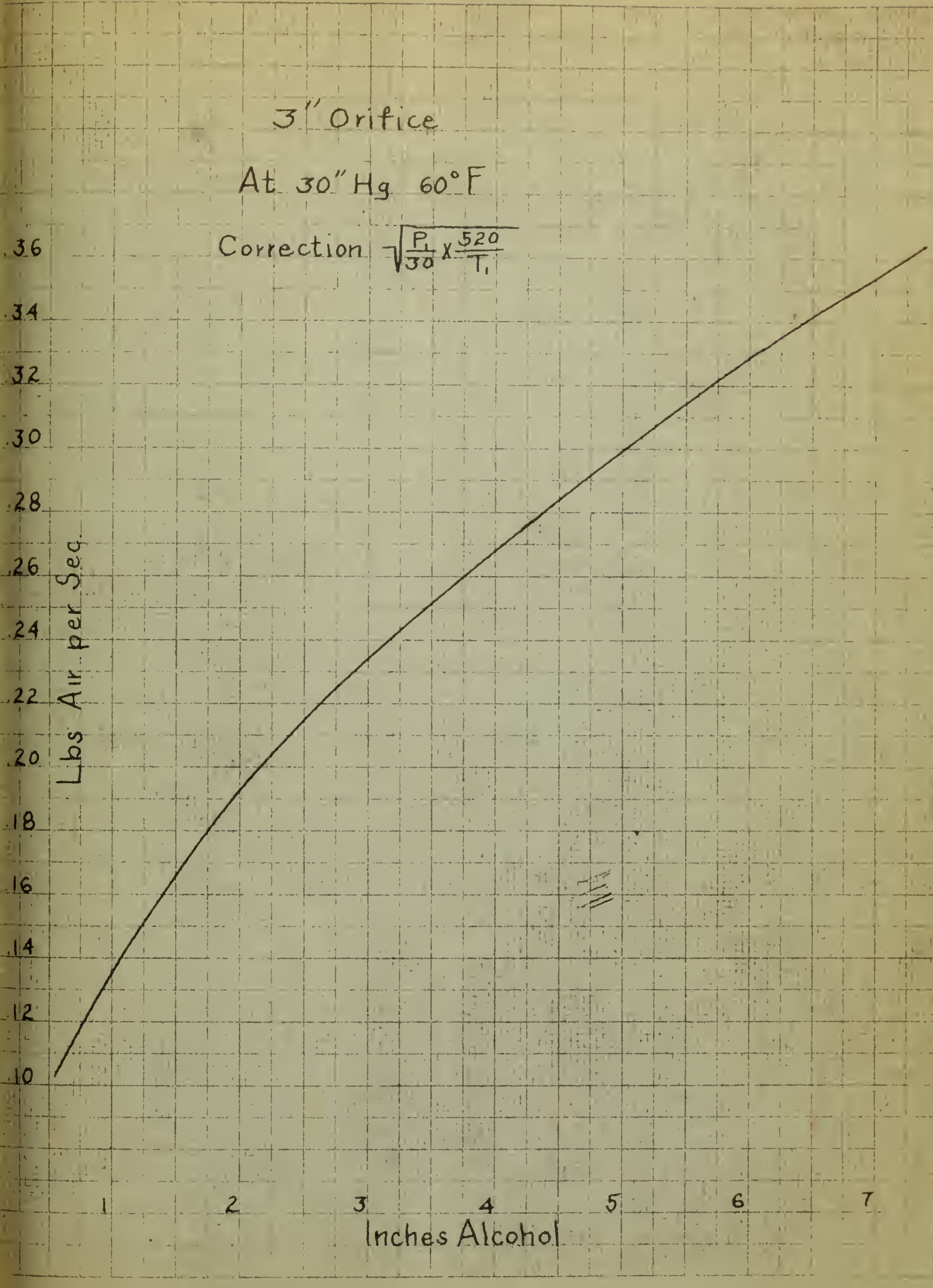
1½" Orifice
At 30" Hg 60°F
Correction $\sqrt{\frac{P_1}{30} \times \frac{520}{T_1}}$
Lbs Air per Sec.
.090
.080
.070
.060
.050
.040
.030
1
2
3
4
5
6
7
Inches Alcohol



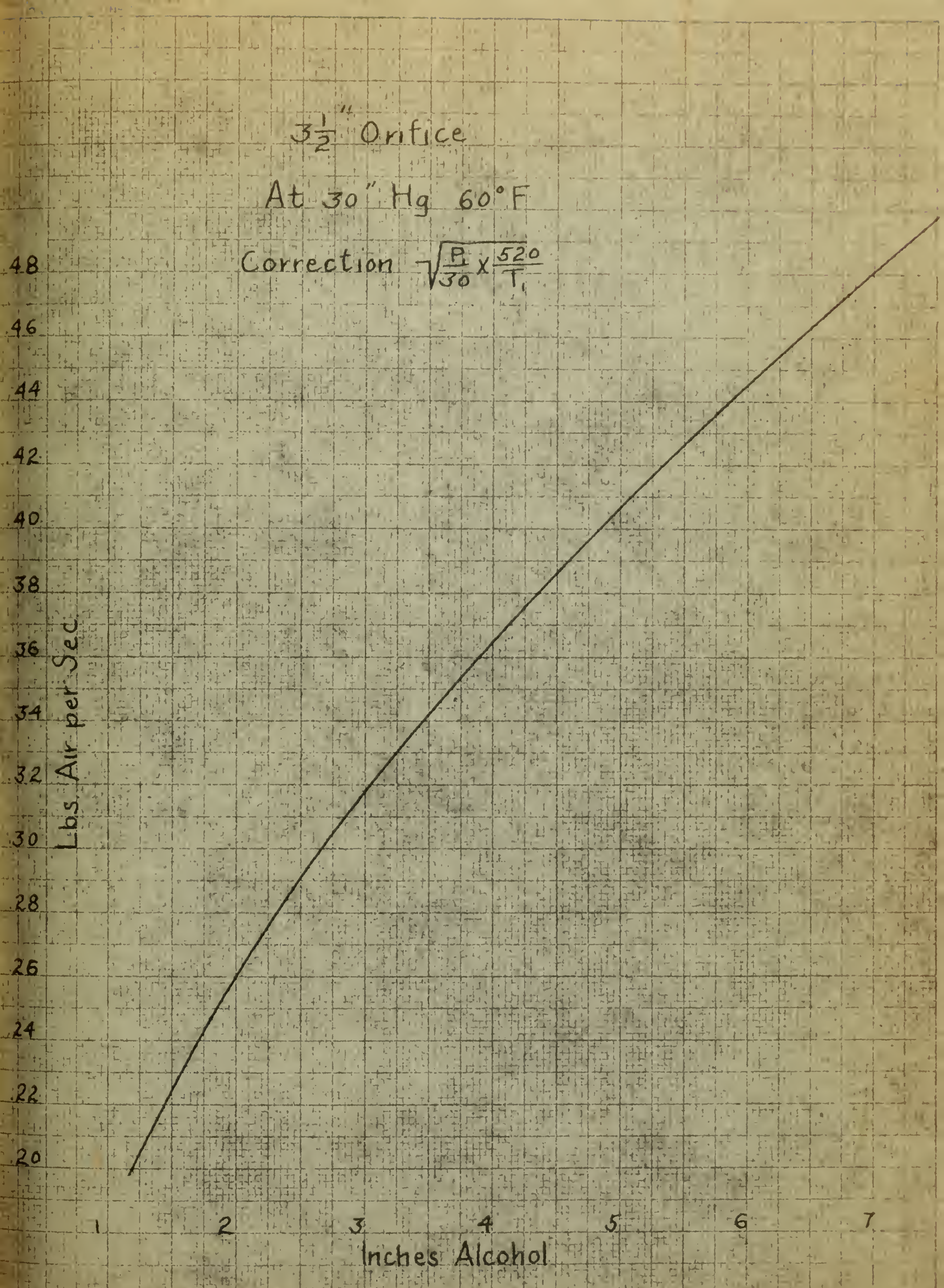
2" Orifice
At 30" Hg 60° F
Correction $\sqrt{\frac{P_1}{30} \times \frac{520}{T_1}}$
Lbs Air per Sec.
.170
.160
.150
.140
.130
.120
.110
.100
.090
.080
.070
.060
.050
.040
1
2
3
4
5
6
7
Inches Alcohol



2 1/2" Orifice
At 30"Hg 60°F
Correction $\sqrt{\frac{P_1}{30} \times \frac{520}{T_1}}$
Lbs Air per Sec.
.24
.22
.20
.18
.16
.14
.12
.10
1
2
3
4
5
6
7
Inches Alcohol



3" Orifice
At 30" Hg 60°F
Correction $\sqrt{\frac{P_1}{30} \times \frac{520}{T_1}}$
Lbs Air per Sec.
.36
.34
.32
.30
.28
.26
.24
.22
.20
.18
.16
.14
.12
.10
1
2
3
4
5
6
7
Inches Alcohol



3½" Orifice
At 30" Hg 60°F
Correction $\sqrt{\frac{P_1}{30} \times \frac{520}{T_1}}$
Lbs. Air per Sec.
.48
.46
.44
.42
.40
.38
.36
.34
.32
.30
.28
.26
.24
.22
.20
1
2
3
4
5
6
7
Inches Alcohol

 Fawkes 45638
The mixture requirements of an internal combustion engine at various speeds and loads.

Thesis
F245 Fawkes 45638
The mixture requirements of an internal combustion engine at various speeds and loads.