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PART-THROTTLE OPERATION AND CONTROL OF A PISTON-PORTED TWO-STROKE CYLINDER

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SUMMARY

The single-cylinder piston-ported two-stroke engine described in Technical Note No. 674 was used to investigate means of obtaining reliable operation under idling conditions. The fuel-consumption characteristics of the engine were also studied over the useful range of speeds and loads, using several different fuels. Fuel was supplied to the engine as vapor mixed with the inlet air and also by direct injection into the cylinder.

With fuel injection into the cylinder, it was found possible to idle satisfactorily; the only adjustments necessary were scavenging pressure and fuel rate. Fuel consumption with injected fuel was comparable with that of small four-stroke aircraft engines. A specific output of 2.73 net brake horsepower per square inch of piston area was obtained at an engine speed of 1800 rpm per minute and a scavenging ratio of 1.4.

INTRODUCTION

Previous work with the piston-ported engine described herein was for the purpose of determining the practical limits of scavenging efficiency for this form of cylinder in order that its optimum performance might be compared with the performance of other engine types. Tests were made first with various inlet- and exhaust-port timings and with many inlet-port shapes and arrangements (reference 1). With the optimum port timing and arrangement determined by the first series of tests, a second series was made to determine the optimum piston-head and cylinder-head shapes (reference 2).

Nearly all of the runs referred to were made at relatively high rotative speeds (1200 to 1800 rpm) and at a scavenging ratio of 1.4. Little information was therefore obtained relative to the performance of this type of engine when operating at light loads, that is, at low scavenging ratios, low rotative speeds, or a combination of both. Such combinations would be encountered at part throttle with propeller load and therefore would be of importance in connection with the cruising and idling operation of an airplane engine.

At low scavenging ratios, the percentage of residual gases present in the combustion chamber of a two-stroke cylinder may be several times as great as in a four-stroke cylinder operating at the same fraction of full load. In the cylinder of a four-stroke engine, the residual gases occupy only the clearance space. In the cylinder of a two-stroke engine the residual gases occupy the entire cylinder volume before the fresh charge enters, and at low scavenging ratios only a small fraction of these gases is pushed out by the incoming charge. The presence of this large fraction of residual gas is often the cause of the irregular combustion, misfiring, or preignition of the entering charge, which has been characteristic of many two-stroke spark-ignition engines operating at light loads.

The present investigation was undertaken principally to determine means of obtaining reliable and regular operation at light loads, particularly when combined with low speed, in this type of engine. A second object of the work was to determine the fuel-consumption characteristics of the engine with particular reference to loads and speeds corresponding to cruising operation. Several different fuels and systems of fuel supply were used.

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GENERAL PLAN OF TESTS

Since the primary object of previous tests with this engine had been the study of scavenging efficiency, most of the previous work had been carried out with a mixture of fuel and air for scavenging. This procedure naturally involved an appreciable loss of fuel from the exhaust ports with consequent high specific fuel consumption. In the present series of tests, runs were made not only with carbureted mixtures of this kind but also with fuel injection directly into the cylinder.

Runs were made over a range of speed and load sufficient to cover the probable useful range of an engine of this kind with the following fuel and fuel-system combinations:

1. Illuminating gas introduced into the air supply near the inlet ports.

2. Vaporized gasoline introduced into the air supply near the inlet ports.

3. Gasoline injected directly into the cylinder after completion of the scavenging process.

4. Safety fuel injected directly into the cylinder after completion of the scavenging process.

DEFINITIONS

In order to avoid confusion, several terms used in this report are defined:

Scavenging ratio, as defined in references 1 and 2, is the ratio of the volume of charge at inlet density passing through the cylinder per stroke to the piston displacement.

Scavenging efficiency is the ratio of the weight of fresh charge retained in the cylinder per stroke to the product of total cylinder volume and inlet density. Inlet density was taken as 75° F and 29.92 inches of mercury in every case.

Gross brake mean effective pressure is defined as the brake mean effective pressure calculated from the dynamometer brake load when the scavenging blower is driven by a separate source of power.

Net brake mean effective pressure is calculated by assuming 70 percent adiabatic blower efficiency and subtracting the resulting blower mean effective pressure from the engine gross brake mean effective pressure.

Scavenging pressure difference is the difference between the scavenging pressure and the pressure in the exhaust surge tank.

Mixture volumes or air quantities are the volumes supplied to the engine per minute and are measured in cubic feet at 75° F and 29.92 inches of mercury. These conditions existed at the inlet to the scavenging pumps.

Fuel rate is the rate of fuel flow to the engine in cubic centimeters per minute.

Specific fuel consumption is the number of pounds of fuel consumed by the engine per horsepower hour of output.

Propeller load is a curve of horsepower required which is assumed to vary as the cube of the engine speed.

DESCRIPTION OF APPARATUS

Engine.- The cylinder used for this work is shown in figure 1 and is the one described in detail in reference 1. Briefly, it is a piston-ported, two-stroke cylinder of $4\frac{1}{2}$-inch bore and 6-inch stroke. It consists of an outer barrel into which a longitudinally split cylinder sleeve fits. One half of this sleeve contains the inlet ports; the other half contains the exhaust ports. The port timing may be varied by sliding one or the other half up or down in the outer barrel and locking it in the desired position. The shape of any of the ports may be changed by means of removable inserts. The cylinder head is similar to an inverted piston the skirt of which extends out of the top of the cylinder bore and includes a flange for bolting it in position. The compression ratio may be changed by sliding the head up or down in the cylinder. The inlet and exhaust passages are partly contained in the outer barrel.

The piston is made of aluminum alloy and has edges rounded to a 5/8-inch radius. The shape is shown as No. 4 in figure 7 of reference 2. The cylinder head is the standard cast-iron shallow spherical design shown as B in figure 8 of reference 2. The port arrangement used is the one designated E in reference 1. The combination of port timing, port arrangement, piston and cylinder-head shape used in these tests was the combination that had been found to give optimum full-load performance in the previous investigations.

A heavy cast-iron crankcase was used with counter-weighted crankshaft. Water and oil pumps were separately driven by electric motors.

Auxiliary apparatus.- Figure 2(a) is a schematic layout of the engine and its accessories set up for spark-ignition operation on illuminating gas. The setup was similar to that described in references 1 and 2 except for the additional gas-measuring equipment required because of the wide range of gas quantities to be measured. Gas and air were separately measured, and supplied by separate blowers. Large surge tanks were used in both inlet and exhaust systems to prevent dynamic effects that might interfere with the scavenging process. The air measurements were made with a standard orifice box attached to the air-blower inlet. High rates of flow of illuminating gas were measured with a pitot tube in the gas line; the pitot tube was calibrated against the large industrial gas meter. Intermediate flows were indicated by means of appropriate calibrated orifices inserted in the gas line, and very small quantities were read directly from the small gas meter. Carbon dioxide under pressure was used to quickly extinguish backfires in the inlet system. Backfiring was often encountered when investigating performance with lean mixtures, retarded spark, or very low scavenging ratios. Normal firing could be immediately reestablished without stopping the engine or gas blowers by introducing a small quantity of CO_2 by means of a quick-acting hand valve.

The setup for vaporized gasoline (fig. 2(b)) was similar to that used with illuminating gas except that provision was made for measuring the fuel rate both volumetrically and by means of a Rotameter. The gasoline was completely vaporized by passing it through a steam-jacketed tube into a baffled vaporizing tank surrounded by a hotwater jacket. For a given set of running conditions, the volume of auxiliary air supplied to the vaporizing tank was the same as the volume of illuminating gas passing through this part of the system in the earlier tests. The main air-supply system was the same as before, and the total volume of gas passing through the engine was also the same at the same running conditions (neglecting the volume of the vaporized gasoline, which would amount to about 2 percent of the total volume). The performance of the engine under these conditions should represent that obtainable with gasoline under conditions of very good carburetion and distribution.

Figure 2(c) is a diagram of the setup used with injected fuel. The total air supply was the same as before, except that all the air passed through the main air system; the rate of air flow was measured with the orifice box at the inlet to the blower. The fuel was supplied under a pressure of about 20 pounds per square inch and was measured in a fuel burette, which had an air dome to balance the fuel pressure. Instantaneous readings of rate of fuel flow were taken with a Rotameter. The fuel pump was a standard two-cylinder Bosch pump with cams set at 180°. The pump was operated at half engine speed, and fuel was pumped from alternate pump cylinders once per crankshaft revolution. This system was chosen because pump operation is more reliable at low pump speeds, and it was further desired to try the effect of injecting fuel on alternate engine power strokes by cutting out one of the pump cylinders. The spray nozzle used for most of the tests was a 12° Bosch pintle type, which was chosen for low penetration and good atomization. The nozzle was located on the inlet side of the cylinder head with its axis parallel to the cylinder bore, so that it would discharge downward into the upward blast of scavenging air. With the fuel pump operating at half engine speed, the injection period occupied a relatively large number of crank degrees, which probably caused most of the air in the cylinder to pass through the spray during the injection period. It was thought that the homogeneity of the charge would be improved in this way over that obtained by injecting the fuel in one short burst early in the compression stroke. Figure 3 is a photograph of the engine setup for cylinder injection. The injection nozzle may be seen in the cylinder head, and the fuel pump is in the left foreground.

TEST PROCEDURE

With illuminating gas and vaporized gasoline, changes in the fuel-air ratio and spark advance were found necessary to reduce the mean effective pressure sufficiently to reach the idling range on a propeller-load curve and at the same time to preserve satisfactory engine operation without misfiring or backfiring. With fuel injection, backfiring was not experienced, and it was found that idling could be best obtained by leaving the spark advance at 14° B.T.C. and reducing the quantity of air supplied by reducing the scavenging-pressure difference.

Engine speeds between 150 and 1800 rpm were investigated. Scavenging ratios greater than 2.5, or engine conditions requiring scavenging pressures exceeding 18 inches of mercury, were not considered of interest. Except for idling conditions, the spark advance was maintained constant at 13° to 14° B.T.C. For all runs, the port timing was 52° to 55° B.B.C. for the inlet and 65° B.B.C. for the exhaust. These values had previously been found optimum for most conditions of operation with illuminating gas. Pressures in the exhaust surge tank were virtually atmospheric. Compression ratio, based on the full stroke, was 7.0 for all runs.

Illuminating gas.- Runs with illuminating gas were made at a fuel-air ratio of 0.227 by volume, which was found to be the best-power mixture. The mixture volume supplied to the engine was held constant during each run, and the engine output was determined over a series of increasing engine speeds with consequent decreasing scavenging ratios. With large mixture volumes, each run started at an engine speed resulting in a scavenging ratio of 2.5, and continued until an engine speed of 1800 rpm or a required scavenging pressure of 18 inches of mercury was exceeded. With low mixture volumes a low speed of 150 rpm could be used without reaching a scavenging ratio of 2.5. The high speed then was limited by erratic engine operation, which usually occurred when the scavenging ratio fell below 0.25.

Vaporized gasoline.- Test procedure with the vaporized gasoline was similar to that with illuminating gas. The fuel-air ratio was held constant at 0.075, a value that was found by means of preliminary runs to correspond to best power under conditions of medium to full output. Fuel flow was controlled by a needle valve in the gasoline line to the vaporizing tank and was measured directly by means of a carefully calibrated Rotameter. The fuel used for the idling runs was an unleaded domestic aviation gasoline of about 74 octane number. With this fuel, detonation consistently occurred above scavenging ratios of 1.4 to 1.5 and seemed to be a function of scavenging ratio alone. In order to eliminate this effect, the fuel for the runs at high scavenging ratio was changed to a leaded aviation fuel of 100 octane number (Army method specification 92.2). Check runs under conditions where detonation was absent showed no difference in engine performance with these fuels.

During the illuminating-gas runs enough leakage developed between the inlet and exhaust port sleeves to affect somewhat the engine performance at very low speeds. For the vaporized gasoline, and all subsequent runs, the two halves of the sleeve were clamped tightly together by set screws inserted in the outer cylinder barrel, and oil was pumped into the space between cylinder sleeves and outer barrel to serve as a seal at this joint. Indicator cards showed that this procedure was effective in eliminating the leakage.

Injected gasoline.- When fuel is injected directly into a two-stroke cylinder during the compression stroke, in an attempt to prevent fuel loss during scavenging, homogeneity of the charge must be established within less than 180° of crank travel. This requirement usually necessitates a very careful choice of injection timing, nozzle position, and nozzle-spray characteristics.

Preliminary runs to determine best spark timing were made at 1800 rpm and a scavenging ratio of 1.4, which might be considered rated speed and output; with the start of fuel injection taking place 5° before the exhaust ports closed; that is, at 240° A.T.C. Spark advance was not critical and was optimum at about 14° B.T.C. With spark set at 14° B.T.C., the injection timing was then varied, keeping the fuel rate constant. Optimum injection timing was found to be 200° A.T.C. (See figs. 14 and 15). Engine performance was not affected by wide variations in injection pressure or by a change to a nozzle with a 30° spray angle.

With a given air quantity, runs were made at each speed and scavenging ratio at several fuel rates to obtain best power and minimum specific fuel consumption at each scavenging ratio, engine speed, and air quantity. This procedure was necessary because the fraction of air retained in the cylinder in each instance, as well as the homogeneity of the mixture, was unknown. The fuel rate was held constant during a run by watching the rotameter and adjusting the pump control. (See fig. 2(c)). Since the rotameter calibration was somewhat affected by the frequency of the pump inlet surge (in spite of the use of a small surge chamber with neoprene diaphragm) the actual fuel rate was measured each time with a burette and a stop watch. In all other respects the runs were similar to those made with vaporized gasoline.

Injected safety fuel.- The procedure followed with the injected safety fuel was similar to that for the injected gasoline. Preliminary runs were made to find optimum injection timing. A range of fuel rates was investigated at each speed and scavenging ratio for each air quantity.

The safety fuel used was 95 octane number, with a flash point of 105° F, initial boiling point of 307° F, and end point of 380° F. The fuel had a tendency to form carbon in the combustion space. This deposit had to be periodically removed to prevent detonation and preignition, at low speeds and high scavenging ratios.

RESULTS AND DISCUSSION

Illuminating Gas

Preliminary runs.- Figure 4 is a plot of the variation of gross brake mean effective pressure with engine speed for various mixture volumes and represents the data as taken. As the engine speed is reduced, with a constant mixture volume, there is an increase in gross brake mean effective pressure due to increasing scavenging ratio, which results in increasing scavenging efficiency. This effect is not so noticeable with large mixture volumes because scavenging is fairly complete at all speeds. The brake mean effective pressure is seen to fall off at low engine speeds because the spark advance and port timing are no longer optimum, and the loss in thermal and scavenging efficiency from these causes is greater than the improvement due to increased scavenging ratio.

From the elementary theory of fluid flow:

$$M = K_1 A \sqrt{\rho \Delta p}$$

where

M weight of gas per unit time

K_1 constant

A port area

ρ density of gas

Δp required pressure difference across ports

With constant inlet temperature, ρ would vary with p_s, the scavenging pressure; therefore under steady-state flow conditions

$$\Delta p \approx \frac{M^2}{K_2 A^2 p_s}$$

Since for a constant scavenging ratio, the weight M is proportional to the engine speed, it would be expected that the scavenging-pressure difference required to force a constant weight of charge through the ports during the scavenging period (that is, to give a constant scavenging ratio) would vary with the square of the engine speed. This effect is seen to be nearly true from figure 5, in which the scavenging-pressure difference required (with atmospheric exhaust pressure) is approximately proportional to $(\text{rpm})^{2.3}$ at higher speeds. Two operating factors present affect the required pressure difference.

1. The increase in p_s associated with high Δp tends to reduce the required pressure difference.

2. The inertia of the gases may tend to increase the required pressure. Since the flow in the inlet system is started and stopped once each engine revolution, at high engine speeds the average gas velocity for the entire scavenging period may be only a fraction of the equilibrium static-flow velocity corresponding to the Δp available. This effect is a function of the engine speed and the length of the inlet system and is predominant in this engine; it causes the required scavenging-pressure difference to increase more rapidly than the square of the speed. This effect is clearly shown in figure 6, where it may be seen that the pressure difference required to force a constant weight of gas through the engine per unit of time, increases with speed. As the inlet pipe from the surge tank to the engine is quite short, it is felt that a similar effect would be found in most two-stroke engines. The absence of resonance peaks in the air flow is to be noted in the straightness of the lines in figure 6.

From the required scavenging-pressure difference and mixture volumes, the scavenging mean effective pressures were calculated (see reference 1), assuming an adiabatic efficiency for the blower of 0.70. The net brake mean effective pressure obtained by subtracting this value from the gross brake mean effective pressure is plotted against engine speed in figure 7 for various scavenging ratios. The lines of constant scavenging ratio may be considered roughly as lines of constant throttle position, since a geared centrifugal blower would supply scavenging air at pressures approximately proportional to $(\text{rpm})^2$, or almost enough to maintain constant scavenging ratio. The corresponding lines of constant mixture volume in cubic feet per minute, measured at 75° F and 29.92 inches of mercury are also indicated. An airplane propeller-load curve drawn through 80 net brake mean effective pressure at 1800 rpm is superimposed on the plot.

It may be observed from this plot that little additional output is obtained in going from a scavenging ratio of 1.4 to a scavenging ratio of 2.5.

The curves of net brake mean effective pressure against engine speed at constant scavenging ratio are roughly horizontal at the higher engine speeds, indicating that scavenging efficiency improves sufficiently with engine speed to offset the increase in friction and blower horsepower.

As seen in figure 7, at a scavenging ratio of 0.4 (and a scavenging pressure of about 0.5 in. Hg), the engine speed on a propeller-load curve was still about 1200 rpm. With lower mixture volumes, the engine would not run at speeds high enough (scavenging ratios low enough) to cause the engine-power-curve to cross the propeller-load curve. In order to operate under the idling conditions it was therefore necessary to reduce either the power developed at a given scavenging ratio or the scavenging ratio at which engine operation became unsatisfactory.

Idling runs.- With normal 14° B.T.C. spark advance and 0.227 fuel-air ratio, backfiring into the inlet system would always occur when the scavenging ratio dropped to about 0.25. The lowest mixture volume tried under these conditions (fig. 9, dashed lines) was 4.91 cubic feet per minute, at which backfiring occurred at speeds above 220 rpm. The brake mean effective pressure developed was 16 pounds per square inch, which was far above the propeller-load curve at this point. The scavenging-pressure difference required was 1.7 inches of water.

With the mixture volumes from 4.91 to 18.4 cubic feet per minute, firing was fairly regular until backfiring occurred at a scavenging ratio of 0.25. There was little misfiring, but some preignition of the fresh charge occurred after the inlet ports closed. This preignition could be detected by rough running, and the engine would usually fire regularly for a few seconds after cutting off the ignition. The preignition was evidently caused by afterburning in the cylinder because, with ignition turned off, firing would not resume after a single nonfiring stroke.

Four-stroke runs.- In order to reduce the power output at low scavenging ratios, there was devised a special ignition circuit that would fire on alternate revolutions if desired.

This method proved to be of little value for two reasons. At the lowest mixture volumes, only a small reduction in brake mean effective pressure was realized because a considerable fraction of the small scavenging charge entering the cylinder for the nonfiring stroke remained there and was supplemented by the fresh charge blown in for the firing stroke. Thus, the effective scavenging ratio was raised and nearly as much energy was liberated per revolution as before. At the higher engine speeds the fresh charge of the nonfiring stroke was ignited by the residual gases of the firing stroke; two-stroke operation could therefore not be prevented, and little or no reduction in power could be obtained. These effects are shown in figure 8(a). The curves for four-stroke operation are discontinued at the point where automatic two-stroke operation started. Four-stroke runs, made with advanced spark in the hope that afterburning and preignition might thereby be prevented, produced no improvement.

With normal two-stroke ignition, preignition seemed to occur at speeds slightly higher than those at which preignition occurred with four-stroke ignition.

Effect of fuel-air ratio.- Runs were made with mixture volumes of 8.6 and 18.4 cubic feet per minute with normal spark advance and abnormally rich mixtures. The power was not greatly affected, as is shown in figure 8(b). At the higher speeds and lower scavenging ratios, the normal fuel-air ratio resulted in smooth running until backfiring occurred at scavenging ratios of about 0.25. The rich mixtures did not backfire, probably owing to their lower flame temperatures, but misfiring would occur at scavenging ratios below 0.6 and sometimes at ratios considerably higher. Abnormally lean mixtures caused backfiring into the inlet system at such high scavenging ratios that formal runs were not attempted.

Effect of spark advance.- With normal fuel-air ratio, a retarded spark reduced the power but caused backfiring even at very low engine speeds. An advanced spark reduced engine power at low engine speeds and removed the backfiring limitation at the higher speeds. Power at the higher speeds tended to be independent of spark advance, probably owing to preignition. (See fig. 8(c).)

It was felt that the misfiring associated with rich mixtures was objectionable and that perhaps lean mixtures would give the required reduction in power without backfiring at the higher speeds, if used in conjunction with an advanced spark. This combination appeared to be successful, as shown in figure 8(d). A series of idling runs was therefore made at a spark advance of 45° B.T.C. and a fuel-air ratio of 0.143. Mixture volumes were varied from 18.4 to 4.3 cubic feet per minute. At the lowest mixture volume, the propeller-load curve was crossed at 340 rpm. No backfiring was experienced throughout the runs. The results are shown in figure 9, with the values for normal spark and normal fuel-air ratio also given for comparison. Misfiring occurred at the lowest scavenging ratios, and the curves are discontinued at the points where good brake readings became difficult to obtain. The angle at which the curves of engine mean effective pressure cross the propeller-load curve indicates a high degree of speed stability.

Vaporized Gasoline

Preliminary runs.- Preliminary fuel-air runs were made with a spark advance of 13.5° and a scavenging ratio of 1.4 at 1270 rpm. This curve is shown in figure 10(a). The best power fuel-air ratio was found to be approximately 0.075, which corresponds to the value generally found for four-stroke engines with good carburetion. Runs then were made with a fuel-air ratio of 0.075 and various spark positions to determine best power spark adjustment. Spark advance was not critical, as may be seen in figure 10(b), and the best power spark advance appeared to be about 12° B.T.C. For subsequent tests, a spark advance of 14° B.T.C. was used as being conducive to good operation at the higher engine speeds. The change in power due to using a spark advance of 14° instead of 12° B.T.C. is within experimental error.

The gross brake mean effective pressure developed with gasoline is shown in figure 11. The curves are of the same general shape as those obtained with illuminating gas but are about 16 percent higher owing to the greater heating value per cubic foot of charge. With 74 octane fuel, detonation was encountered at scavenging ratios above 1.4 or 1.5. The engine could therefore be operated on this fuel over most of the propeller-load curve. Occasional detonation was experienced at high engine speeds and relatively low scavenging ratios. This detonation was probably due to preignition. Since the study of detonation in this engine was beyond the scope of this investigation, detonation was prevented at the higher scavenging ratios by changing to a leaded aviation fuel of 100 octane number.

The scavenging pressures required with the gasoline-air mixtures were slightly higher than those with illuminating gas at the same scavenging ratio and speed. This effect may be due to higher pressure in the cylinder at the time the inlet port opens. It is, therefore, possible that a somewhat later inlet or earlier exhaust opening would be optimum for operation on gasoline-air mixtures.

Net brake mean effective pressure at constant scavenging ratios and constant air quantities is shown plotted against engine speed in figure 12 with a propeller-load curve superimposed. With a scavenging ratio of 1.4, a net brake mean effective pressure of 100 pounds per square inch was developed at 1800 rpm. The corresponding scavenging pressure was 14.7 inches of mercury. The general shape of these curves is very similar to those for illuminating gas shown in figure 7.

Idling runs.- As with illuminating gas, special means had to be found to keep the engine running steadily at outputs and speeds corresponding to the lower portion of the propeller-load curve. This result may be seen in figure 13(a), which shows runs made with the normal spark advance of 14° B.T.C. and a fuel-air ratio of 0.075.

In figure 13 the curves are plotted for small air quantities. Figure 13(a) shows that, with normal spark timing and fuel-air ratio, as the load is reduced and the speed allowed to rise, operation becomes unsatisfactory before the mean effective pressure is sufficiently reduced. The dashed portion of the curves indicates the region of fairly steady running with four-stroke and six-stroke operation. The curves end where good brake load readings could not be obtained.

With an actual propeller load, figure 13(a) shows that satisfactory operation could be obtained down to 950 rpm by throttling to an air quantity of 18.4 cubic feet per minute, with a scavenging pressure of 0.7 inch of mercury required. With more throttling, the engine might stall unless some means were provided for increased loading. Mean effective pressures less than 15 or 20 pounds per square inch were not possible, regardless of engine speed or air quantity used.

<u>Effect of spark advance</u>.- The effect of spark advance in reducing the output is shown in figure 13(c). A retarded spark resulted in violent backfiring and was therefore eliminated from consideration after a few trials. Both 30° and 45° B.T.C. spark advances suppressed the backfiring and permitted operation at high enough speeds and low enough scavenging ratios to cross the propeller-load curve. The steadiest operation was obtained with a spark advance of 45° B.T.C.

<u>Advanced spark and variable fuel-air ratio</u>.- Runs were made with 45° B.T.C. spark advance and various fuel-air ratios (fig. 13(b)) in an effort to reduce still further the engine power at low air quantities. It was found that a reduction in power could be obtained with either a rich or a lean mixture, but the lean mixture (0.065 fuel-air ratio) caused rough operation before the propeller-load curve was reached. The fuel-air ratio of 0.12 gave the greatest reduction, but firing was somewhat steadier at 0.10. The runs of figures 13(b) and 13(c) were made with an air quantity of 8.59 cubic feet per minute. As a check, a limited number of idling runs were also made at air quantities of 4.9 and 3.7 cubic feet per minute with similar results.

Figure 13(d) shows the final idling runs made with a spark advance of 45° B.T.C. and a fuel-air ratio of 0.10. It was found possible to idle to about 600 rpm. These idling curves are replotted in figure 12 and give a complete picture of net performance on vaporized gasoline. Idling at 600 rpm is seen to require a scavenging ratio of 0.15.

Cutting out half of the cylinders in a multicylinder engine would permit somewhat lower idling speeds or more steady operation of the remaining cylinders at the same engine speeds. Adjustment of spark advance and fuel-air ratio still would be advisable.

<u>Specific fuel consumption</u>.- The fuel consumption obtained with vaporized and injected fuel will be discussed later.

Gasoline Injection

Preliminary runs.— Preliminary runs were made at 1800 rpm and a scavenging ratio of 1.4 to determine a satisfactory injection nozzle and nozzle position. It was found that a nozzle position on the inlet side of the cylinder, as shown in figure 2(c), gave slightly better power and that peak power occurred at leaner mixtures than when other locations were used. For these runs the spark advance was set at 14° B.T.C. and fuel injection was timed to start just before the exhaust ports closed (240° A.T.C.). Tests with 8°, 12°, and 30° Bosch pintle nozzles showed little difference in performance. The 12° nozzle was used for all subsequent runs.

The effect of spark advance at best-power fuel rate is shown in figure 14(a). A spark advance of 14° B.T.C. was chosen for the final performance runs. The effect of variations in fuel rate at 1800 rpm and a scavenging ratio of 1.4 were then made with a spark advance of 14° B.T.C. These curves are shown in figures 14(b) and 14(c).

The effect of variations in the injection timing is shown in figure 15 at 1800 rpm, scavenging ratio of 1.4, spark advance of 14° B.T.C. and best-power fuel rate, as previously determined. In order to make it possible to adjust for any desired injection angle, the position of the pump drive coupling corresponding to start of injection at 240° A.T.C. was determined by observing the fuel spray with a stroboscope, while motoring the engine at about 1500 rpm, with pump delivery adjusted to correspond approximately to best-power fuel rate at this speed and a scavenging ratio of 1.4. The pump drive coupling was marked for this timing, and other injections adjustments were made by changing the coupling by the desired angle. Small changes in actual injection timing are to be expected with variations in pump speed and fuel rate This method was considered sufficiently accurate, however, because the output is not very sensitive to small deviations from the optimum injection timing. The effect of injection timing at three widely different speeds and air quantities is shown in figure 15. For these runs, the maximum power, or the best economy fuel rate, was first determined at an injection timing of 240° A.T.C. Since the exhaust ports closed at 245°, little or no fuel would be lost from the exhaust at this injection timing. The fuel rate thus determined was thereafter held constant for all injection-timing runs.

It was found that for all the speeds and loads tried, an injection timing of about 200° A.T.C. resulted in maximum output. This value was therefore chosen as a good compromise between homogeneity of charge and loss of fuel from the exhaust. Figure 16 shows that, if the fuel loss caused by earlier injection timing is made up by increasing the fuel rate, the increase in power caused by the improved homogeneity is sufficient to compensate for the fuel loss and the brake specific fuel consumption is actually lower. It would appear that further study of these effects would be worth while. With the extremely small fuel quantities required for idling, it was anticipated that fuel pump delivery would be more consistent if lower injection pressures could be employed. It was found that, as long as there was enough spring pressure to hold the injection valve against its seat, wide variations in injection pressure had no noticeable effect on output.

Power runs.- Power runs were all made with 100 octane gasoline; injection timing was held constant at 200° A.T.C.; and spark advance was held constant at 14° B.T.C. In other respects the runs were similar to those with vaporized gasoline except that a series of fuel rates was run at each speed and each scavenging ratio. These additional runs were necessary because the quantity of air remaining in the cylinder, and therefore the best fuel rate for a given condition, could not be determined in advance. Both the maximum brake mean effective pressure and the minimum brake specific fuel consumption could be obtained by operating over a range of fuel rates at each point. Typical fishhook curves of brake specific fuel consumption against brake mean effective pressure are shown in figure 17.

It was found that, with fuel injection, backfiring and preignition of the fresh charge were absent. The principal reason for this absence is undoubtedly the fact that no fuel was introduced until roughly 70° of crank travel after the start of scavenging. The limits of operation were marked only by irregular misfiring. At high scavenging ratios, the fuel rate could be varied from richer than best power to leaner than best economy before encountering irregular operation. At low scavenging ratios, only that portion of the fishhook curve corresponding to the richer mixtures could be obtained.

Idling runs.- Preliminary runs had indicated that an injection timing of 200° A.T.C. was also optimum for low air quantities. (See fig. 15.) Therefore all idling runs were made with this injection timing. At each speed and scavenging ratio, as with the power runs, a wide range of fuel rates was investigated. An advanced spark was found to result in markedly irregular operation. This effect was believed to be caused by the shorter time available for proper mixing of the fuel and air. A similar effect often has been noted in four-stroke gasoline engines operating with direct cylinder injection. For example, a spark advance of 40° B.T.C. would permit only 120° of crank travel between the start of injection and the ignition of the charge. A retarded spark (T.C.) resulted in slightly smoother operation at scavenging ratios above 0.5. Below this value, the standard 14° B.T.C. advance was equally good, however, and it was therefore decided to operate at this advance for all idling runs.

Figure 18 shows performance using injected gasoline with scavenging ratios down to 0.2. Satisfactory idling was obtained as low as 480 rpm with a net brake mean effective pressure of 6 pounds per square inch. At 1800 rpm and a scavenging ratio of 1.4, a net brake mean effective pressure of 100 pounds per square inch was obtained. This result is the same as that obtained with vaporized fuel and indicates comparable combustion efficiency. In other respects the curves are similar to those of figures 7 and 12. As previously mentioned, the fuel rate was much more critical at the lower scavenging ratios.

Owing to the success of the idling runs with normal injection, only a few experiments were made with fuel injection on alternate strokes. It was established, however, that little or no reduction power resulted because additional fresh air was supplied during the second scavenging period. It is possible that lower scavenging ratios could have been employed under these conditions.

Injected Safety Fuel

The setup, the auxiliary apparatus, and the general procedure used with safety fuel was the same as that used with injected gasoline.

Injection timing runs were made at 1520 rpm, an air quantity of 73.6 cubic feet per minute, and constant fuel rate, to determine whether the reduced volatility of the safety fuel would require a change in timing. Spark advance was held at 14° B.T.C. An injection timin of 210° A.T.C. was found to be optimum and was used throughout.

At each air quantity, fishhook curves were obtained at various speeds, exactly as with the vaporized gasoline. Power and fuel consumption obtained with the safety fuel were almost the same as with gasoline. Idling runs were made without changing the spark advance. The results of the power runs are shown in figure 19.

One serious trouble encountered with the safety fuel was carbon formation. Apparently some of the heavy ends did not take part in the combustion, and deposited on all parts of the combustion space, including the fuel nozzle. Although the fuel was 95 octane, preignition and detonation resulted, even at fairly low air quantities. In one instance carbon on the fuel nozzle interfered sufficiently with the spray to affect seriously the engine operation. Carbon formation with gasoline was negligible, although lead deposits on spark plugs required attention at long intervals. Idling with safety fuel appeared to be only slightly more difficult than with gasoline.

Brake Specific Fuel Consumption

Vaporized gasoline.- When the fuel-air mixture is used for scavenging, specific fuel consumption depends upon the relationship between power and scavenging ratio. As the scavenging ratio is increased, engine power increases but more fuel is lost out of the exhaust ports during the scavenging process. Fuel used will be equal to air supplied multiplied by the fuel-air ratio:

$$\frac{\text{Pounds of fuel}}{\text{hour}} = \text{S.R.} \times \rho_i \times V_d \times \frac{\text{rpm} \times 60}{1728} \times F \qquad (1)$$

where

S.R. scavenging ratio

ρ_i inlet density

V_d displacement volume

F fuel-air ratio

$$\text{Horsepower output} = \frac{\text{net bmep} \times V_d \times \frac{1}{12} \times \text{rpm}}{33000} \qquad (2)$$

Dividing (1) by (2):

$$\text{bsfc} = \frac{\text{S.R.} \times F \times \rho_i}{\text{net bmep}} \; \frac{60 \times 33000 \times 12}{1728}$$

If $\rho_i = 0.0742$ (75° F)

and $F = 0.075$

$$\text{bsfc} = \frac{\text{S.R.}}{\text{net bmep}} \; 76.6 \qquad (3)$$

The net brake mean effective pressure required for a propeller load is shown in figure 12. This curve is proportional to $(\text{rpm})^2$ and is drawn through an arbitrarily chosen "full throttle" point at an engine speed of 1800 rpm and a scavenging ratio of 1.4. Brake specific fuel consumption may be calculated directly from figure 12 by using equation (3). Curves of specific fuel consumption against engine speed were calculated for propeller load, full-throttle, and cruising conditions and are plotted in figure 20. The throttle position for cruising was chosen to give propeller-load power at 1550 rpm. As previously mentioned, a geared centrifugal supercharger having an outlet pressure proportional to $(\text{rpm})^2$ should supply, at fixed throttle, a nearly constant scavenging ratio over a range of speeds. The observed scavenging pressures for various constant scavenging ratios were plotted against engine speed, as was done for illuminating gas in figure 5. If a line of pressure proportional to $(rpm)^2$ is drawn through the point corresponding to full throttle or cruising, it is possible to obtain the actual change in scavenging ratio as the engine speed is reduced under these conditions. The corresponding air quantities are shown in figure 21. This figure also includes the air quantities required on the propeller-load curve. Straight dashed lines of constant scavenging ratio are shown for comparison with the constant-throttle lines.

In reference to figure 20, the high brake specific fuel consumption near full throttle is due to the relatively large scavenging ratio used, with the attendant loss of fuel from the exhaust in the excess scavenging air. As the engine is throttled, the brake mean effective pressure falls off more slowly than the scavenging ratio, and consequently, less air (and fuel) is wasted and the brake specific fuel consumption improves. At low speeds and low scavenging ratios, a small reduction in scavenging ratio produces a larger reduction in scavenging efficiency and power output. It is also probable that the port shapes and the timing are not correct for low speeds and low scavenging air velocities. The high percentage of residual gases undoubtedly increases combustion-time losses. The foregoing factors coupled with a decreasing mechanical efficiency lead to an increase in brake specific fuel consumption at low outputs. The minimum brake specific fuel consumption, which occurs at about 1280 rpm, is 0.825. All values of brake specific fuel consumption are high compared with an equivalent four-stroke engine. Although few runs were made at other than a fuel-air ratio of 0.075 with vaporized gasoline, the data from preliminary runs indicates that brake specific fuel consumption may be reduced roughly 12 percent by running at best economy instead of best power fuel-air ratio. The curves of brake specific fuel consumption at the fixed throttle settings reflect the drop in mean effective pressure at reduced speeds and constant scavenging ratio, as shown in figure 12.

<u>Injected gasoline</u>.- From the standpoint of fuel-pump control it is of interest to know the fuel requirements of the engine in weight of fuel per working stroke. The weight of fuel required per stroke for best power, plotted against engine speed for various constant air quantities, is obtained directly from the test data and is shown in figure 22. The fuel required on the propeller-load curve

is obtainable by noting the engine speeds at which the propeller-load curve intersects the lines of constant air quantity in figure 18. From figure 21 the air quantities that would be supplied over the speed range by the scavenging blower at full throttle, or at cruising throttle, may be obtained. These values of air quantity and engine speed may be plotted on figure 22 to give the fuel-pump-delivery requirements for these two throttle settings.

At full throttle with the scavenging ratio somewhat above 1.4 for most of the speed range, the fuel required per stroke remains nearly constant for speeds above 600 rpm. At cruising condition, the scavenging ratio is also nearly constant over a range of speeds. It will be seen that somewhat more fuel is required per stroke at the lower speeds. Since net brake mean effective pressure (and indicated mean effective pressure) is less at these speeds (fig. 18), the quantity of fresh air remaining in the cylinder per stroke (scavenging efficiency) probably also is less. Thus the extra fuel supplied is not consumed as efficiently, if at all. It is probable that the scavenging flow changes at low engine speeds and low scavenging pressures. This result may interfere with establishment of charge homogeneity, thus requiring a greater quantity of fuel to utilize the air present, or it may be that under these conditions more fuel is lost through the exhaust ports. Figure 15 shows that optimum injection timing remains at 200° A.T.C. This result would seem to indicate that the relative importance of homogeneity and exhaust-port loss remained the same. Both factors probably become worse at low speeds and scavenging pressures. The increased quantity of residual gases present with poor scavenging may require rich mixtures for best power, but this possibility is thought to be unlikely because the occurrence of increased residual gases does not affect the fuel-air ratio for best power in four-stroke engines (reference 3, p. 144).

The fuel quantities required per stroke at fixed throttle are constant enough so that a simple linkage connecting air throttle and fuel pump could be used and would give a predetermined constant fuel quantity at each throttle setting. The fuel required per stroke falls off on the propeller-load curve as the air quantity supplied becomes less.

Fuel-air ratio supplied.- The fuel-air ratio supplied is plotted against engine speed in figure 23. The ineffective use of the fuel supplied, at the lower engine speeds, is reflected here as an increase in required fuel-air ratios. The higher scavenging ratios used at full throttle mean that much of the air supplied does not remain in the cylinder; thus the best power fuel-air ratio for full throttle, based on total air supplied, appears as the lowest curve in this figure.

Figure 18 shows that the mean effective pressure (and therefore the quantity of fresh air remaining in the cylinder per stroke) is fairly constant above 600 rpm if the throttle position or scavenging ratio is held constant. Since the fuel-air ratio supplied for best power is also nearly constant above this speed, it would appear that the factors influencing the utilization of the fuel supplied are constant at the higher engine speeds.

The net brake specific fuel consumption for the injected-fuel runs is obtained most easily from the pounds of fuel per stroke of figure 22 and the corresponding net brake mean effective pressures of figure 18.

$$\text{Net bsfc (lb/hp-hr)} = \frac{\text{lb fuel}}{\text{stroke}} \times \frac{\text{rpm} \times 60 \times 33000}{\text{net bmep} \times \frac{V_d}{12}\ \text{rpm}}$$

$$= \frac{\frac{\text{lb fuel}}{\text{stroke}}}{\text{net bmep}} \times 249 \times 10^3$$

Figure 24 presents the net brake specific fuel consumption plotted against engine speed for various constant air quantities. These values were obtained at the best-power fuel rate. An inspection of figure 17 indicates that the use of a best economy fuel rate would improve these values about 10 percent at the higher outputs. With very low air quantities satisfactory operation was not obtainable at minimum brake specific fuel consumption fuel rate.

For each air quantity shown in figure 24, there is an engine speed at which net brake specific fuel consumption is a minimum. With constant air quantity, the low scavenging ratios associated with high engine speeds result in poor scavenging and wasted fuel caused by four-stroke and six-stroke operation. This effect causes the brake specific fuel consumption curve to rise at high speed. It is interesting to note that, at higher speeds and air quantities, the brake specific fuel consumption is not greatly affected by large changes in scavenging ratio. This result may indicate that the percentage of the fuel lost from the exhaust was more or less independent of scavenging ratio. The losses associated with low output, as already discussed, are evident in the curve for an air quantity of 24.54 cubic feet per minute. The values of brake specific fuel consumption for the very low air quantities are not plotted. They are higher and more scattered, owing to the rich mixtures required. and the four-stroke and six-stroke operation encountered. Fuel consumption was not considered of particular interest at these extremely low outputs. Figure 25 shows net brake specific fuel consumption on the propeller-load curve and the two chosen constant-throttle settings. The brake specific fuel consumption is seen to be quite reasonable for an engine of this size in the usual operating range.

Figure 26 shows the motoring friction mean effective pressures measured on the M.I.T. engine at various engine speeds and scavenging pressures. These curves show the usual rise in friction mean effective pressure with engine speed. The reason for the increase in motoring friction mean effective pressure with scavenging pressure is not apparent. Cylinder pressure acting on the piston rings may be responsible. The relationship between the actual friction mean effective pressure under firing conditions (imep - bmep) and the motoring friction mean effective pressure is not known. The points shown in figure 26 are the average of several determinations; consistent values were difficult to obtain, even though great care was exercised in keeping the temperature of the oil and the water jacket constant.

CONCLUSIONS

From tests made with a piston-ported two-stroke cylinder of $4\frac{1}{2}$-inch bore and 6-inch stroke, it has been concluded that;

1. Fuel injection directly into the cylinder of the two-stroke engine is to be preferred over the use of a pre-mixed charge for the following reasons:

(a) Specific fuel consumption could be reduced to figures comparable with small four-stroke aircraft engines.

(b) Idling was possible without backfiring or preignition and without change in engine adjustment other than reducing the scavenging pressure and fuel rate.

(c) At engine speeds above 600 rpm, the fuel quantity required per stroke varied principally with throttle setting. It would therefore appear practicable to control the fuel pump by means of a simple linkage to the air throttle.

(d) Net brake mean effective pressure with injected fuel was just as high as with vaporized fuel; that is, about 100 pounds per square inch at an engine speed of 1800 rpm and a scavenging ratio of 1.4.

2. Early injection timing (200° A.T.C.) was necessary under all conditions, probably owing to the effect on charge homogeneity. An advanced spark resulted in irregular operation with fuel injection, probably for the same reason.

3. Since optimum fuel-injection timing remained the same at low speeds, although there was evidence of increased fuel loss, it would appear that both charge homogeneity and fuel loss from the exhaust became worse at low speeds.

4. With a given scavenging ratio, reduced mean effective pressure and increased fuel losses occurred at low speeds with all fuels and injection systems.

5. The use of injected safety fuel is not recommended on account of its tendency to deposit carbon on combustion-space and nozzle surfaces.

6. Operation with fuel mixed with the scavenging air resulted in cruising fuel consumption about 70 percent

higher and in full-throttle fuel consumption about double the values obtained with injected fuel.

7. With normal spark timing and fuel-air ratio, backfiring with vaporized fuel seemed to be a function of scavenging ratio only and occurred in the neighborhood of 0.25, regardless of engine speed.

8. Idling could be accomplished with vaporized fuel by advancing the spark and using a very rich or very lean mixture. The use of a retarded spark resulted in violent backfiring.

9. The use of four-stroke ignition was not a satisfactory method of reducing power for idling purposes,

10. A high specific output was attained with the engine, namely; 2.73 net brake horsepower per square inch piston area, at the rather moderate piston speed of 1800 feet per minute. This result, together with its great mechanical simplicity, would seem to make the type attractive for development as a low-cost airplane power plant.

Massachusetts Institute of Technology,
Cambridge, Mass., January 1943.

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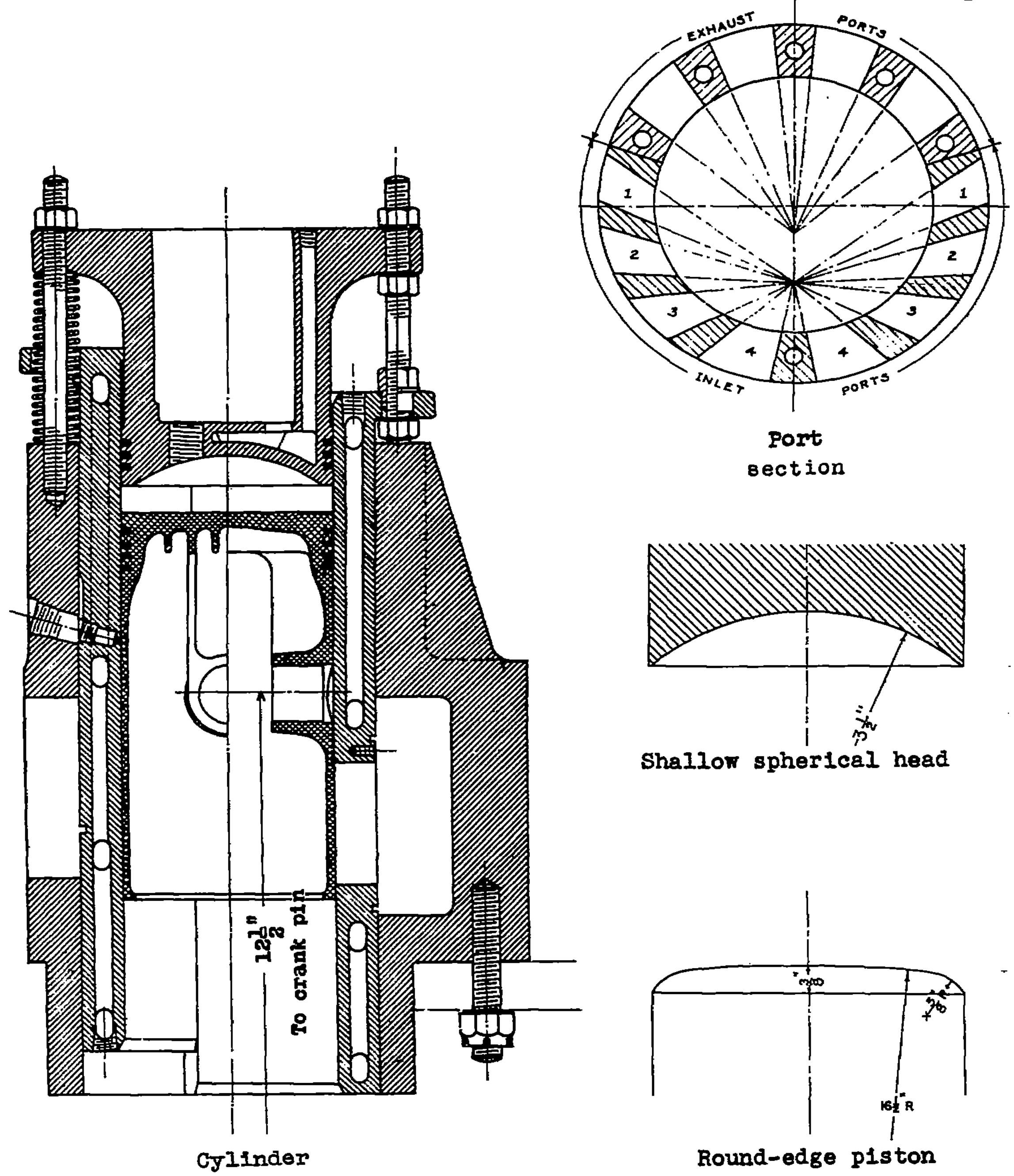
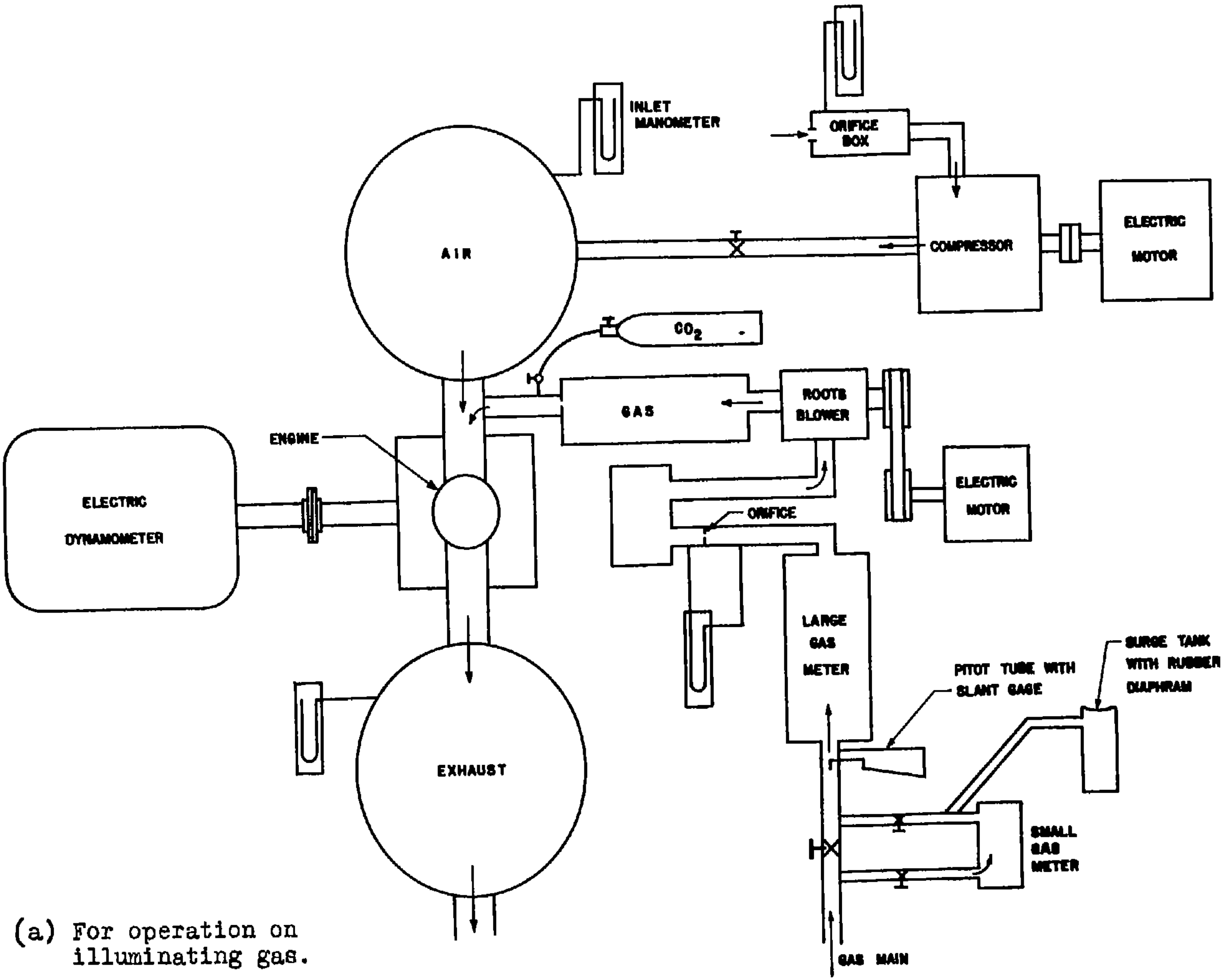
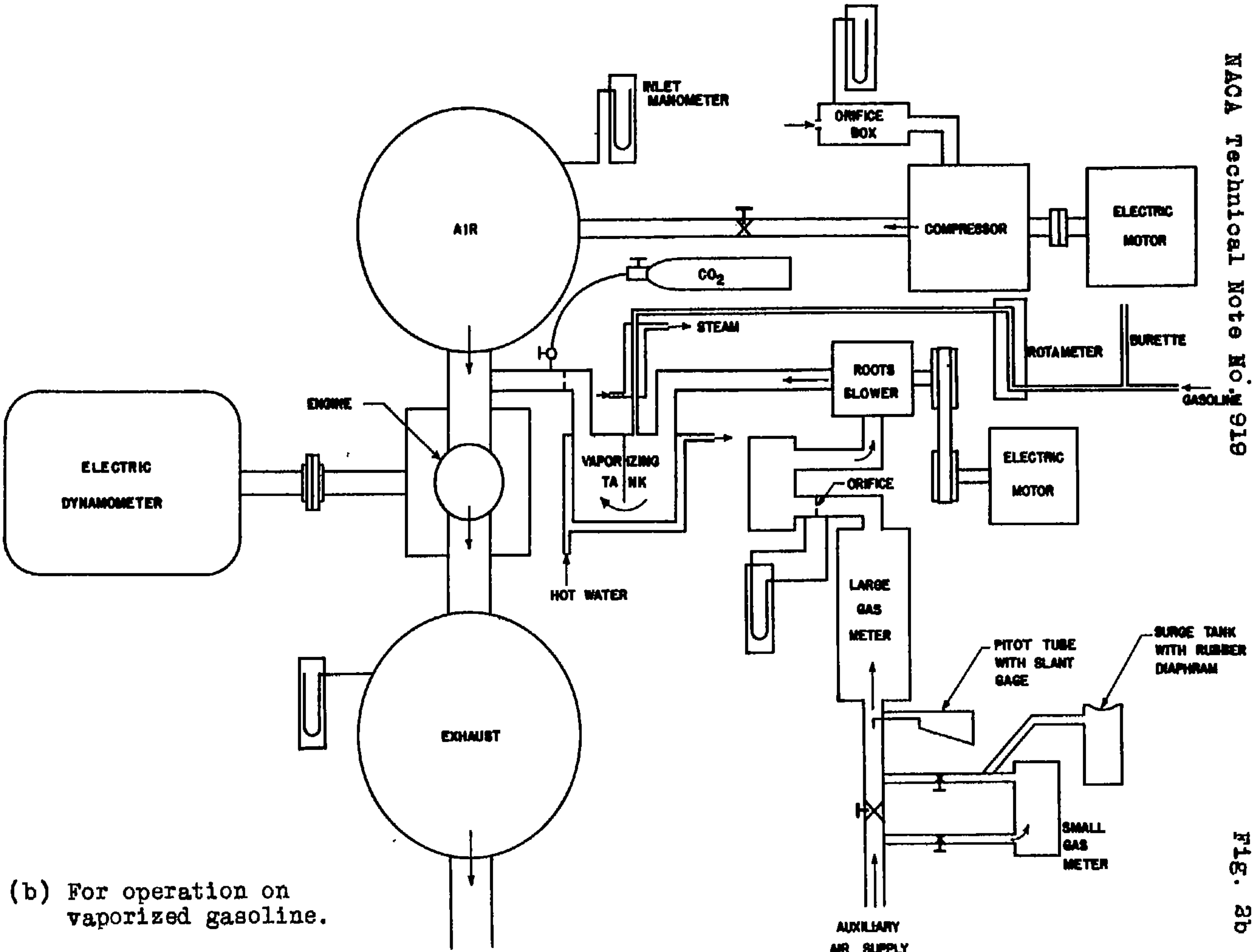


Figure 1.- Engine details.



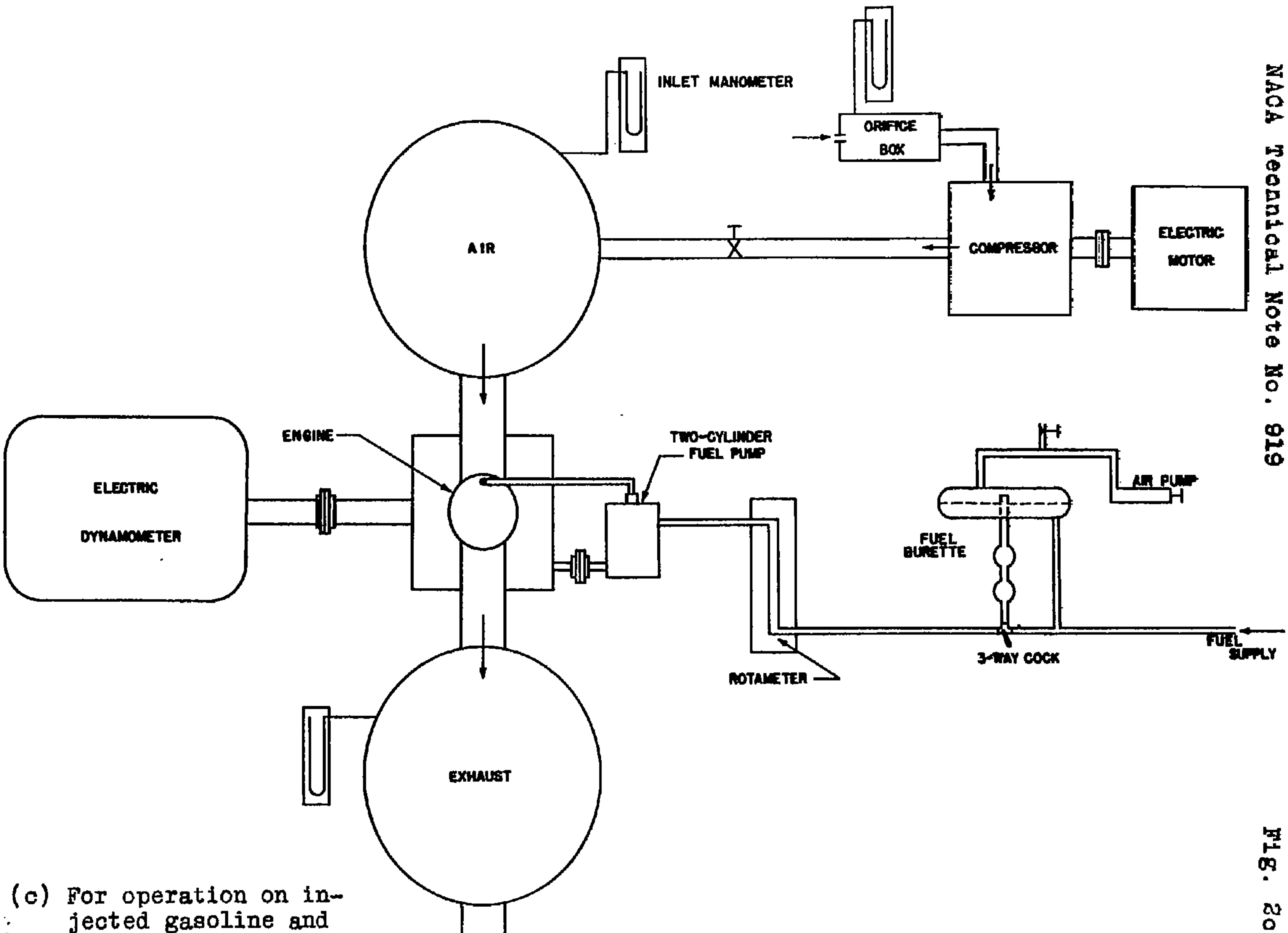
(a) For operation on illuminating gas.

Figure 2.- Schematic layout of test engine and accessories.



(b) For operation on
vaporized gasoline.

Figure 2.- Continued.



(c) For operation on injected gasoline and safety fuel.

Figure 2.- Concluded.

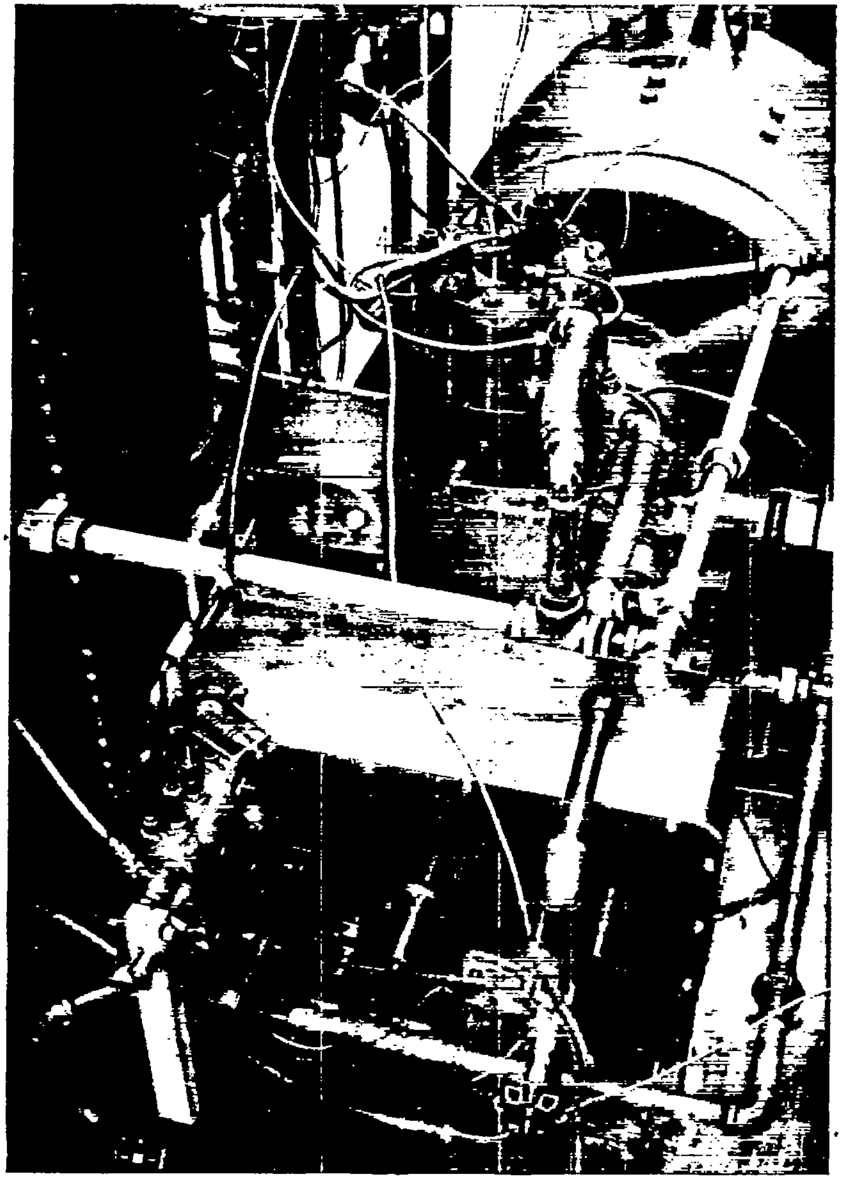


Figure 3 .- Engine set-up for cylinder injection.

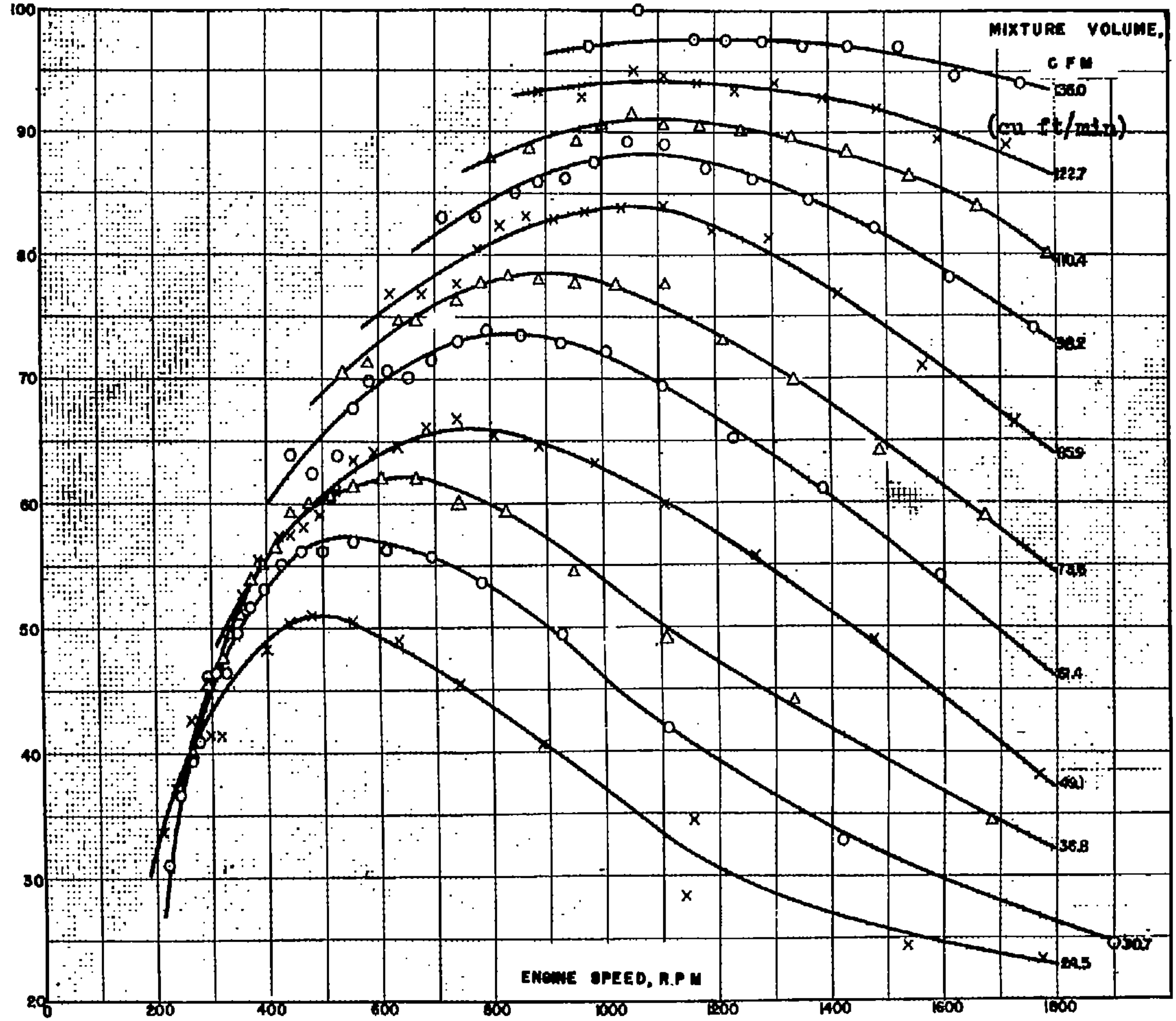


Figure 4.- The effect of engine speed on gross brake mean effective pressure for various mixture volumes; spark advance, 14° B.T.C.; fuel-air ratio, 0.227; illuminating gas. (Volume measured at 75°F, 29.9 in.Hg).

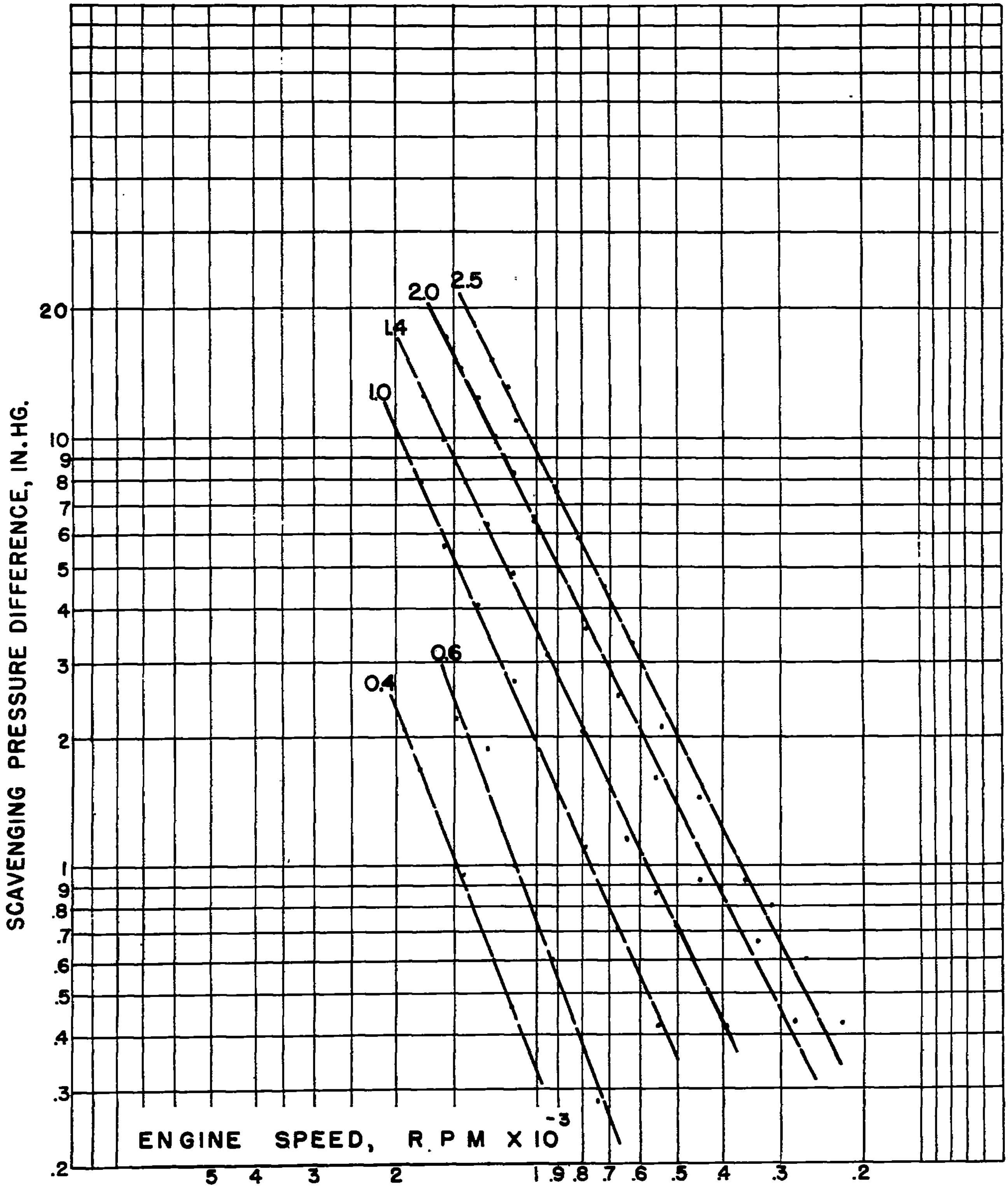


Figure 5.- The effect of engine speed on scavenging pressure difference at various scavenging ratios; illuminating gas.

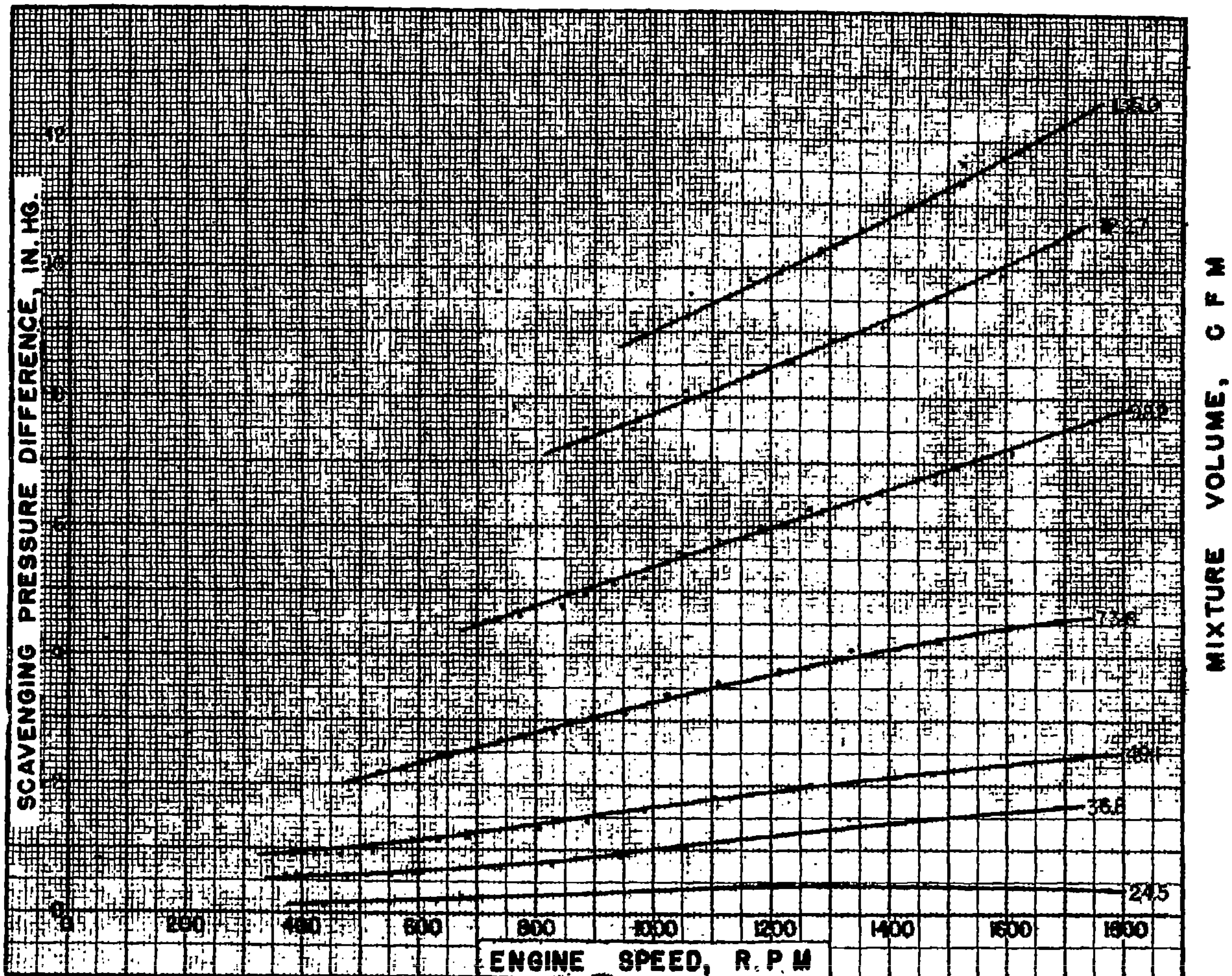


Figure 6.- The effect of engine speed on scavenging pressure difference at various mixture volumes, measured at 75°F and 29.9 in.Hg; illuminating gas.

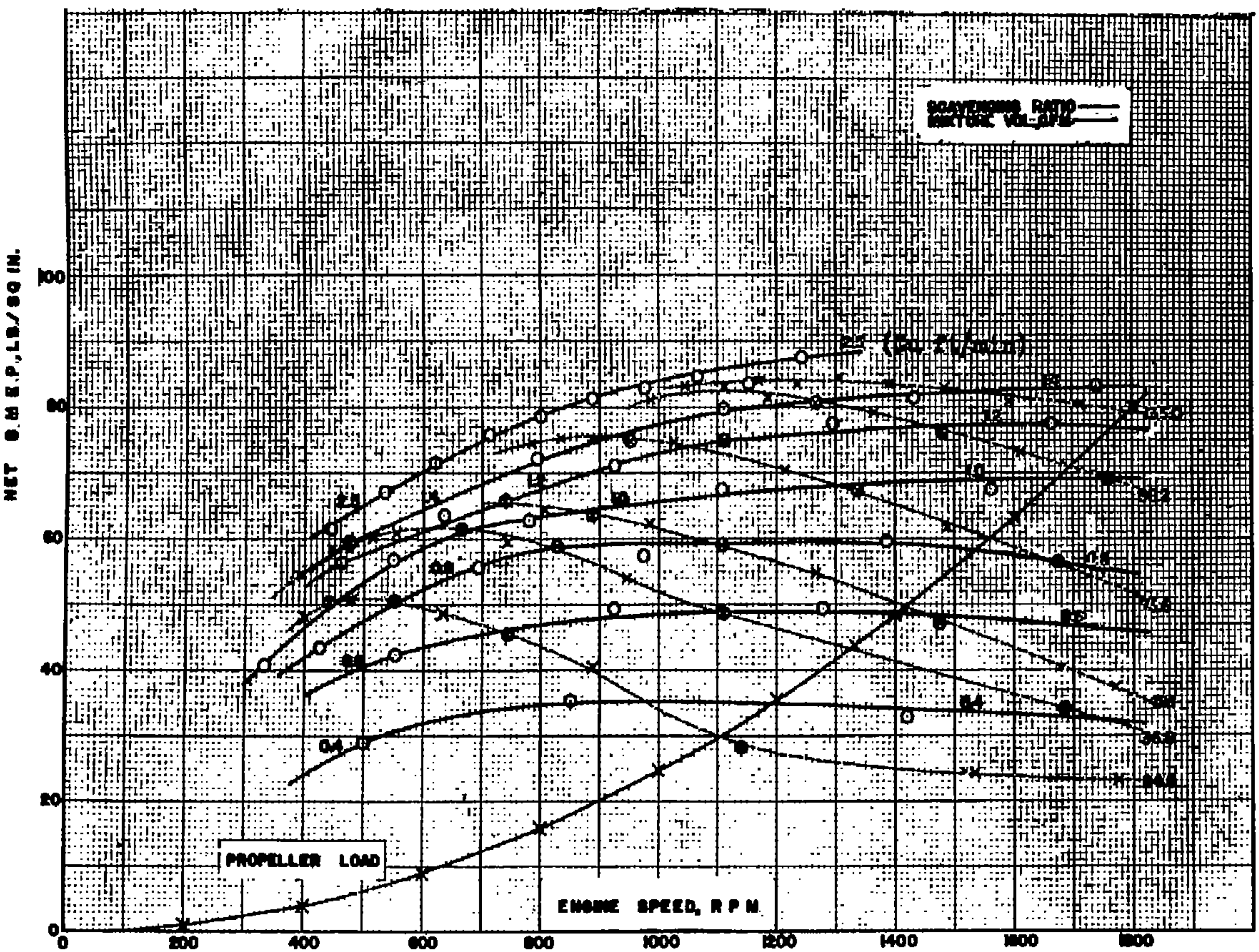
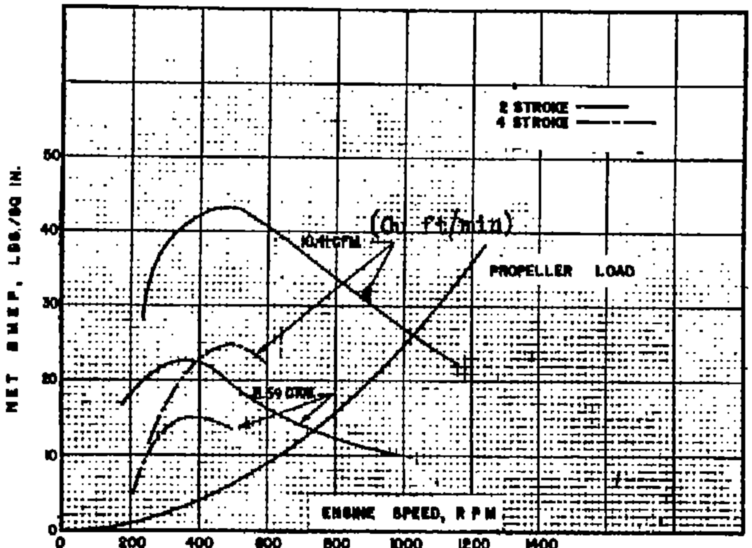
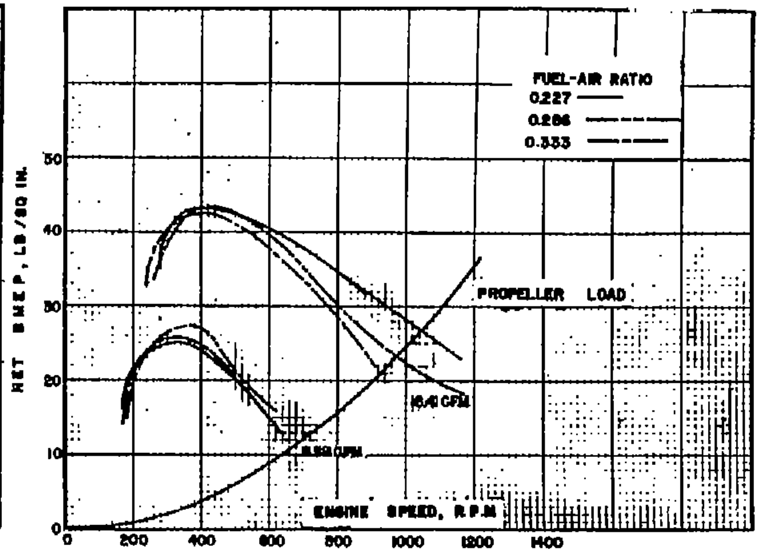


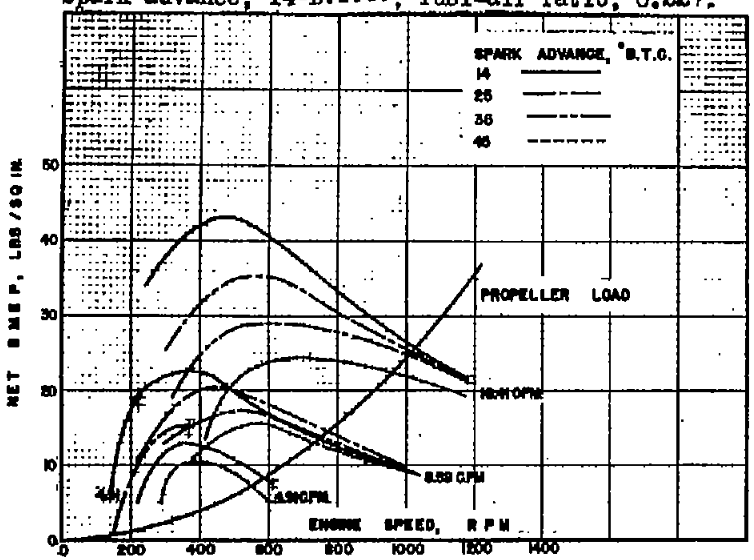
Figure 7.- The effect of engine speed on net brake mean effective pressure at various scavenging ratios and mixture volumes; spark advance, 14° B.T.C.; fuel-air ratio, 0.227; illuminating gas.



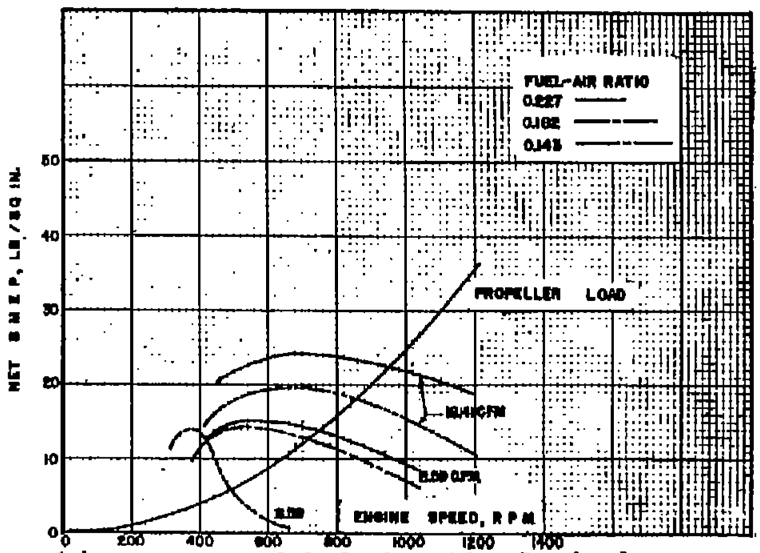
(a) Comparison of two- and four-stroke operation. Spark advance, 14°B.T.C.; fuel-air ratio, 0.227.



(b) The effect of fuel-air ratio. Spark advance, 14°B.T.C.



(c) The effect of spark advance. Fuel-air ratio, 0.227.



(d) The effect of fuel-air ratio. Spark advance, 45°B.T.C.

Figure 8. - Idling runs showing the effect of fuel-air ratio and spark advance on brake mean effective pressure at various mixture volumes; illuminating gas.

Fig. 8

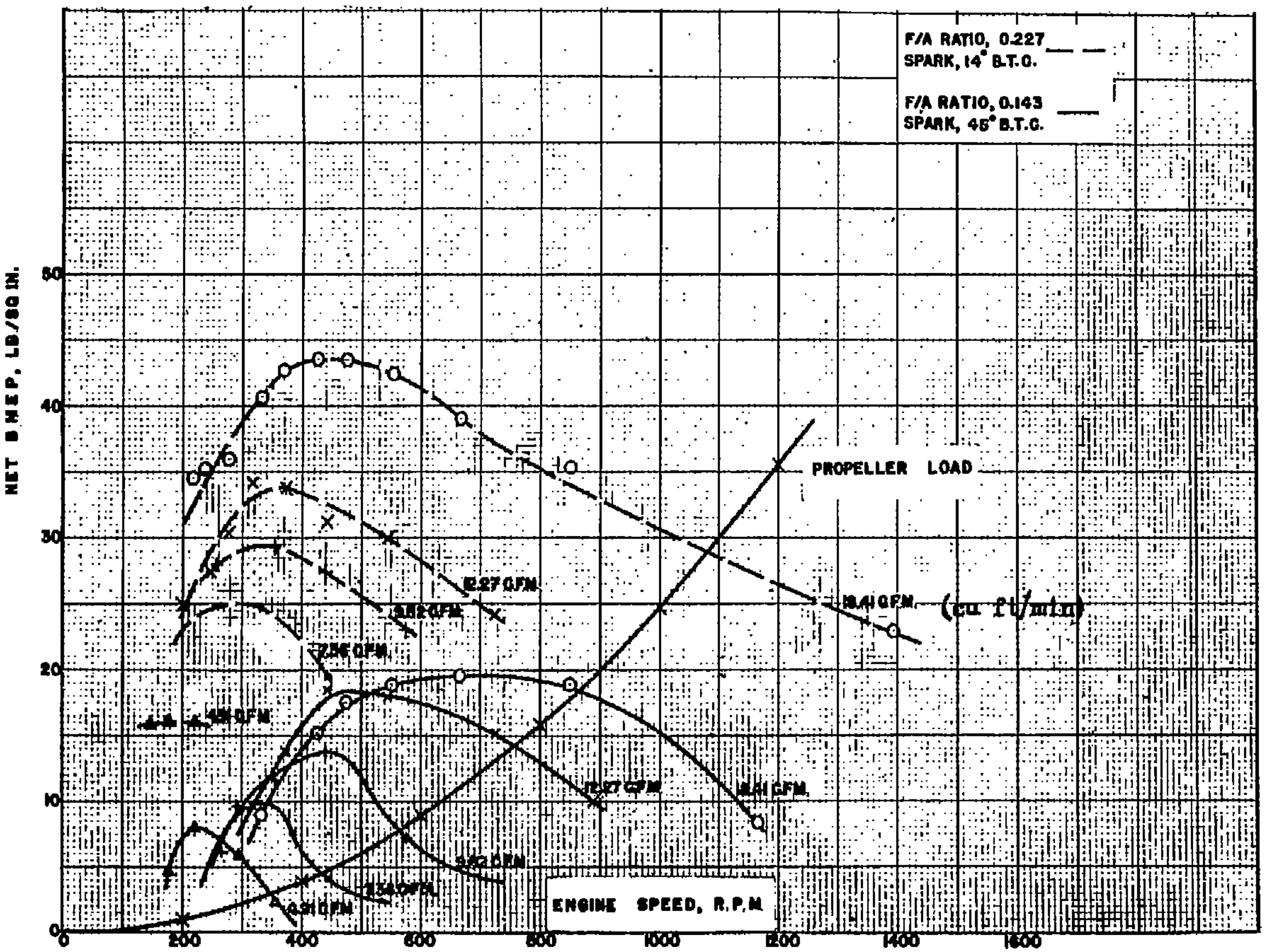
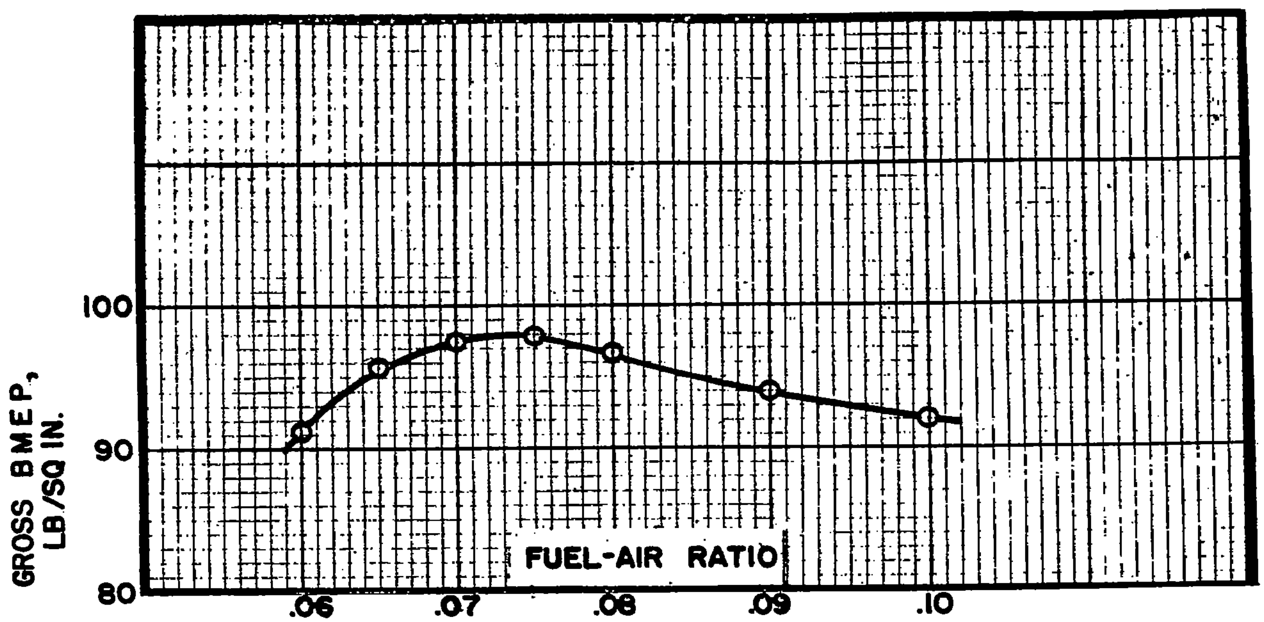
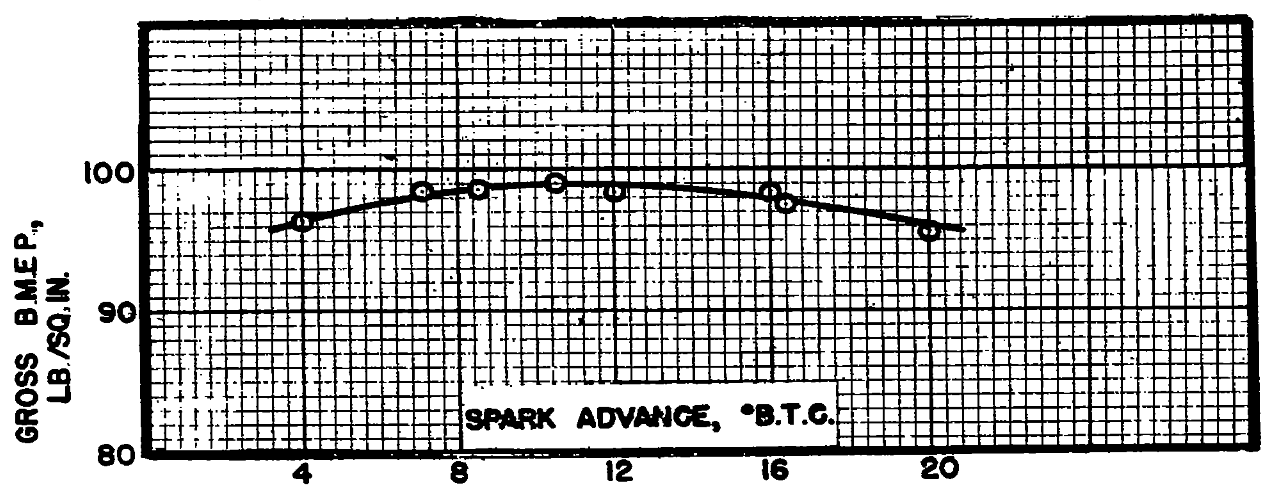


Figure 9.- The effect of engine speed on brake mean effective pressure, for small mixture volumes; illuminating gas. (Final idling runs)



(a) - The effect of fuel-air ratio on brake mean effective pressure.



(b) - The effect of spark advance on brake mean effective pressure.

Figure 10.- Determination of best fuel-air ratio and best spark advance; vaporized gasoline; mixture volume, 98.2 cu ft/min; scavenging ratio, 1.4; engine speed, 1268 rpm.

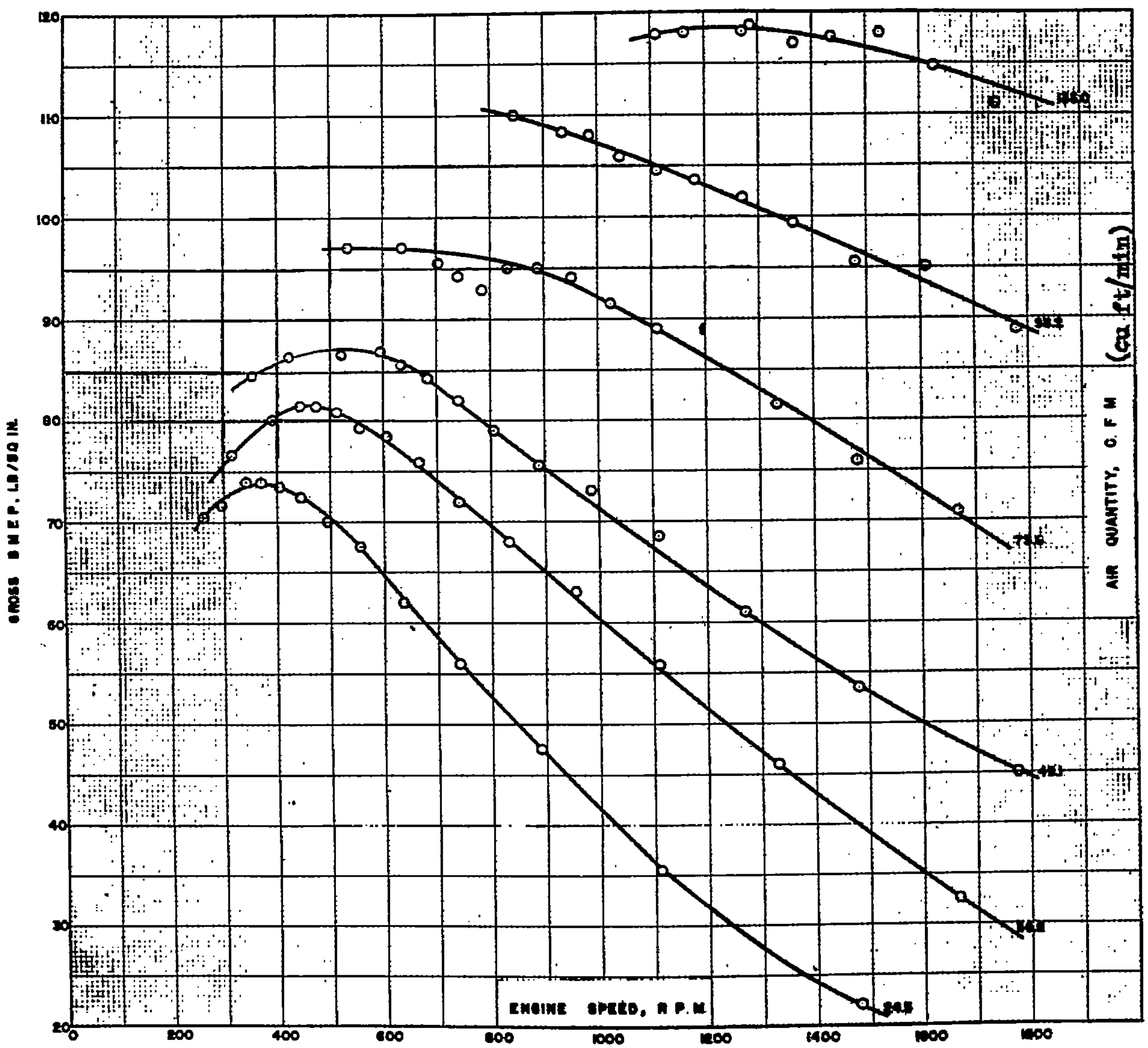


Figure 11.- The effect of engine speed on gross brake mean effective pressure for various air quantities; vaporized gasoline; spark advance, 14° B.T.C.; fuel-air ratio, 0.075.

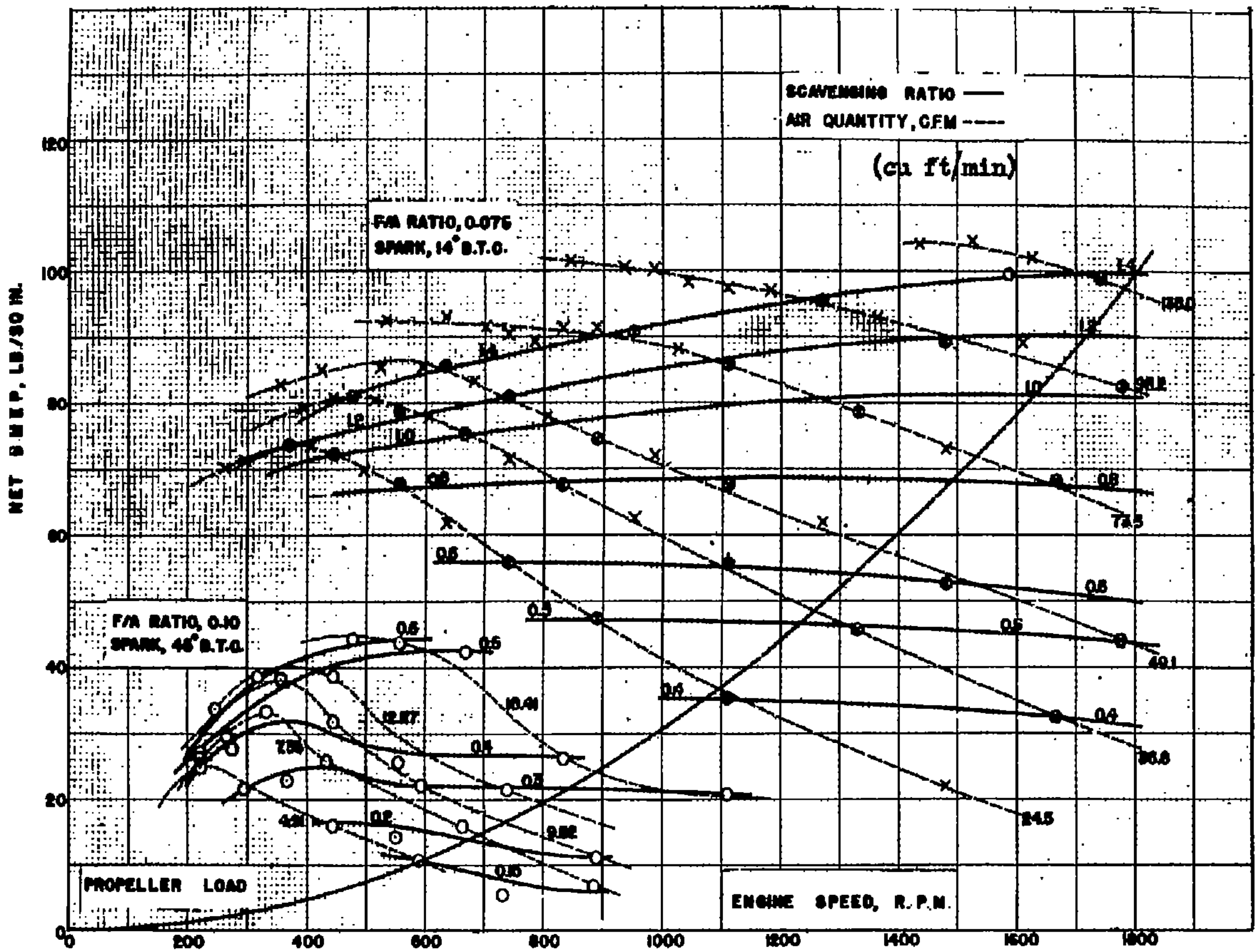
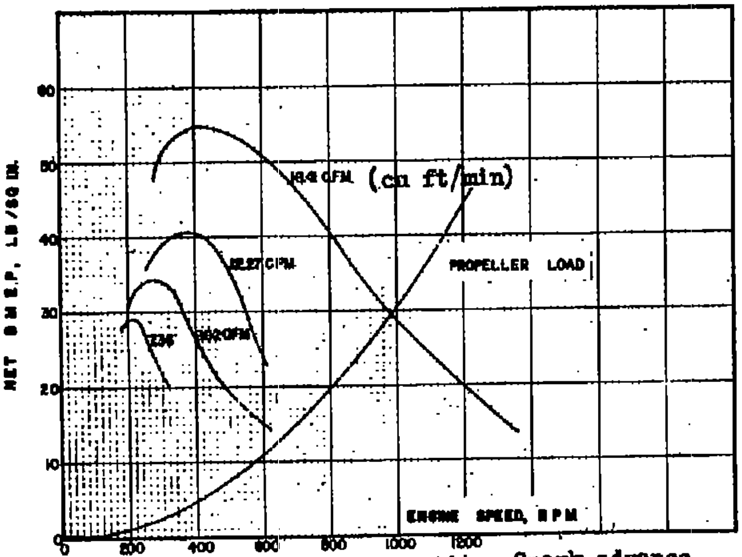
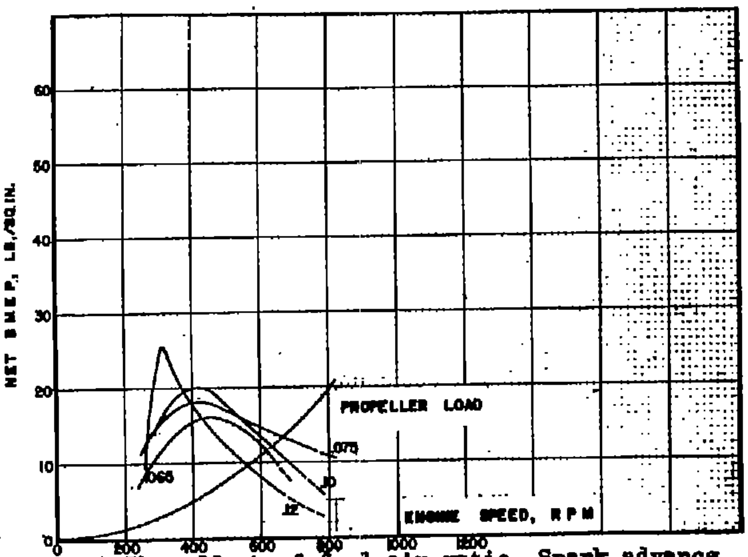


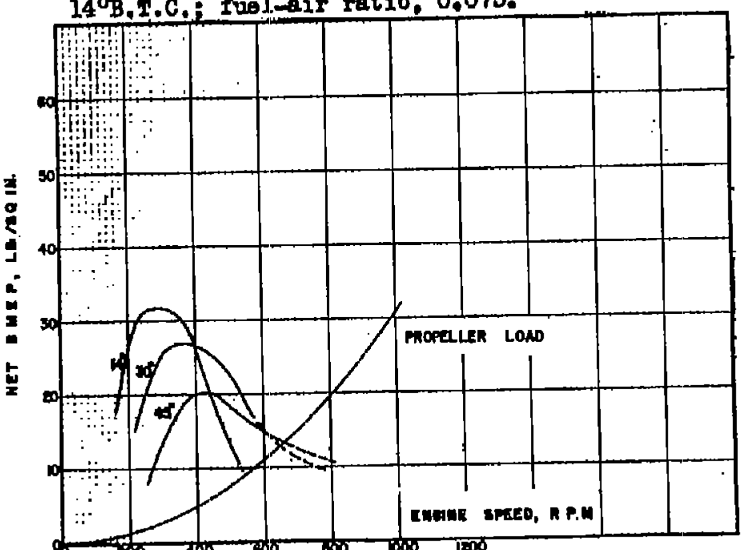
Figure 12.- The effect of engine speed on net brake mean effective pressure at various scavenging ratios and air quantities; vaporized gasoline.



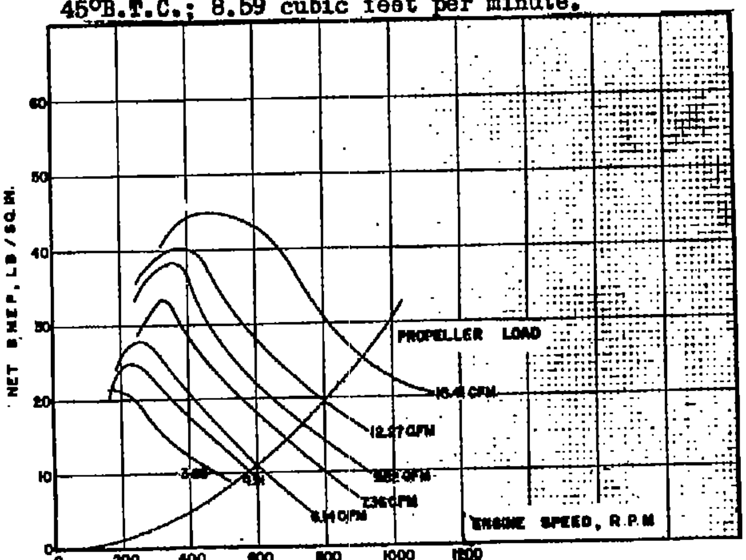
(a) The effect of air quantity. Spark advance, 14°B.T.C.; fuel-air ratio, 0.075.



(b) The effect of fuel-air ratio. Spark advance, 45°B.T.C.; 8.59 cubic feet per minute.

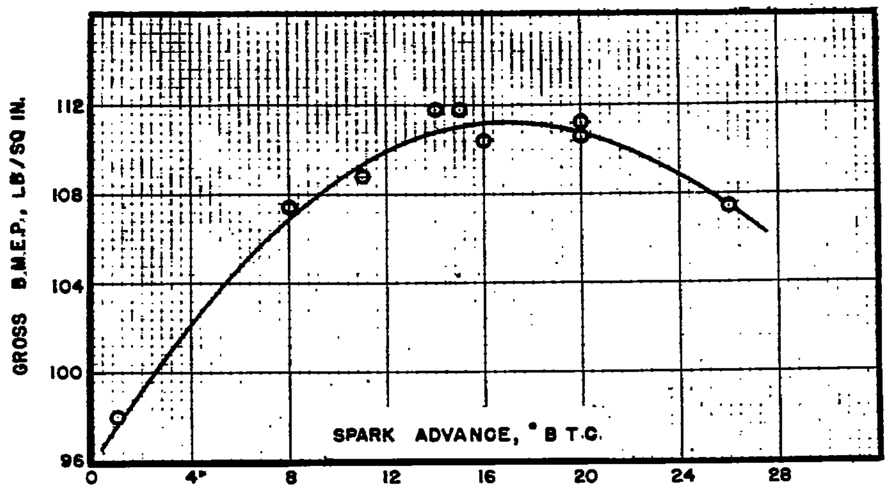


(c) The effect of spark advance. Fuel-air ratio, 0.075; air quantity, 8.59 cubic feet per minute.

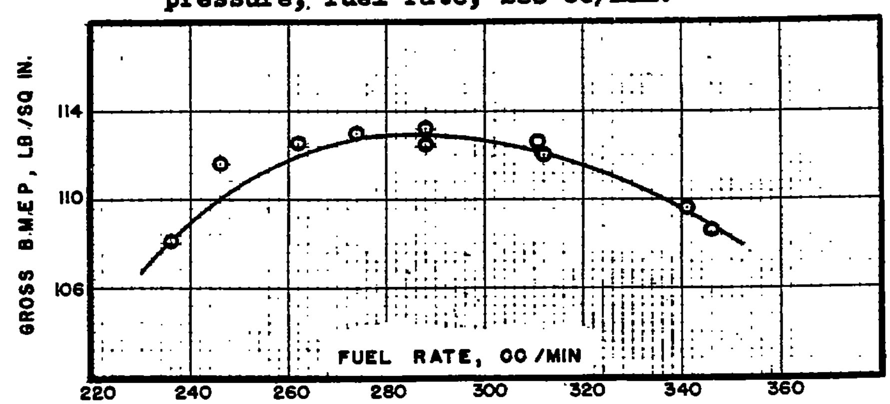


(d) The effect of air quantity. Fuel-air ratio, 0.10; spark advance, 45°B.T.C.

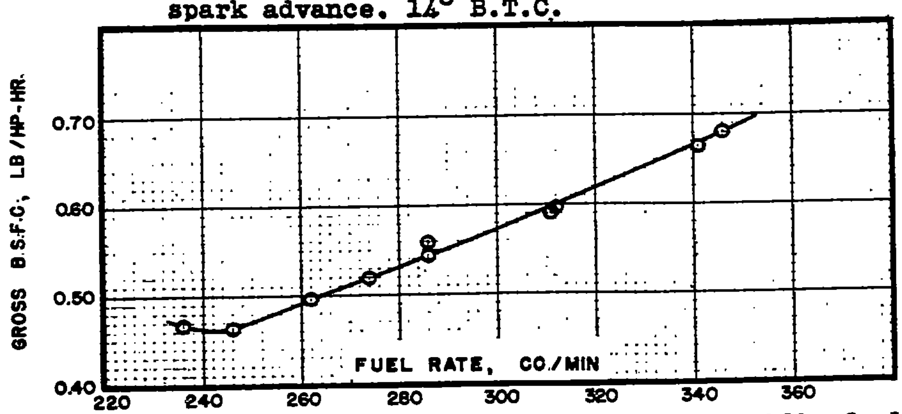
Figure 13.- Idling runs showing the effect of engine speed on brake mean effective pressure for various fuel-air ratios, air quantities, and spark positions; vaporized gasoline.



(a) The effect of spark advance on brake mean effective pressure, fuel rate, 288 cc/min.



(b) The effect of fuel rate on brake mean effective pressure; spark advance, 14° B.T.C.



(c) The effect of fuel rate on brake specific fuel consumption; spark advance, 14° B.T.C.

Figure 14.- Preliminary runs with injected gasoline; injection angle, 240° A.T.C.; engine speed, 1800 rpm; scavenging ratio, 1.4; 30° pintle nozzle.

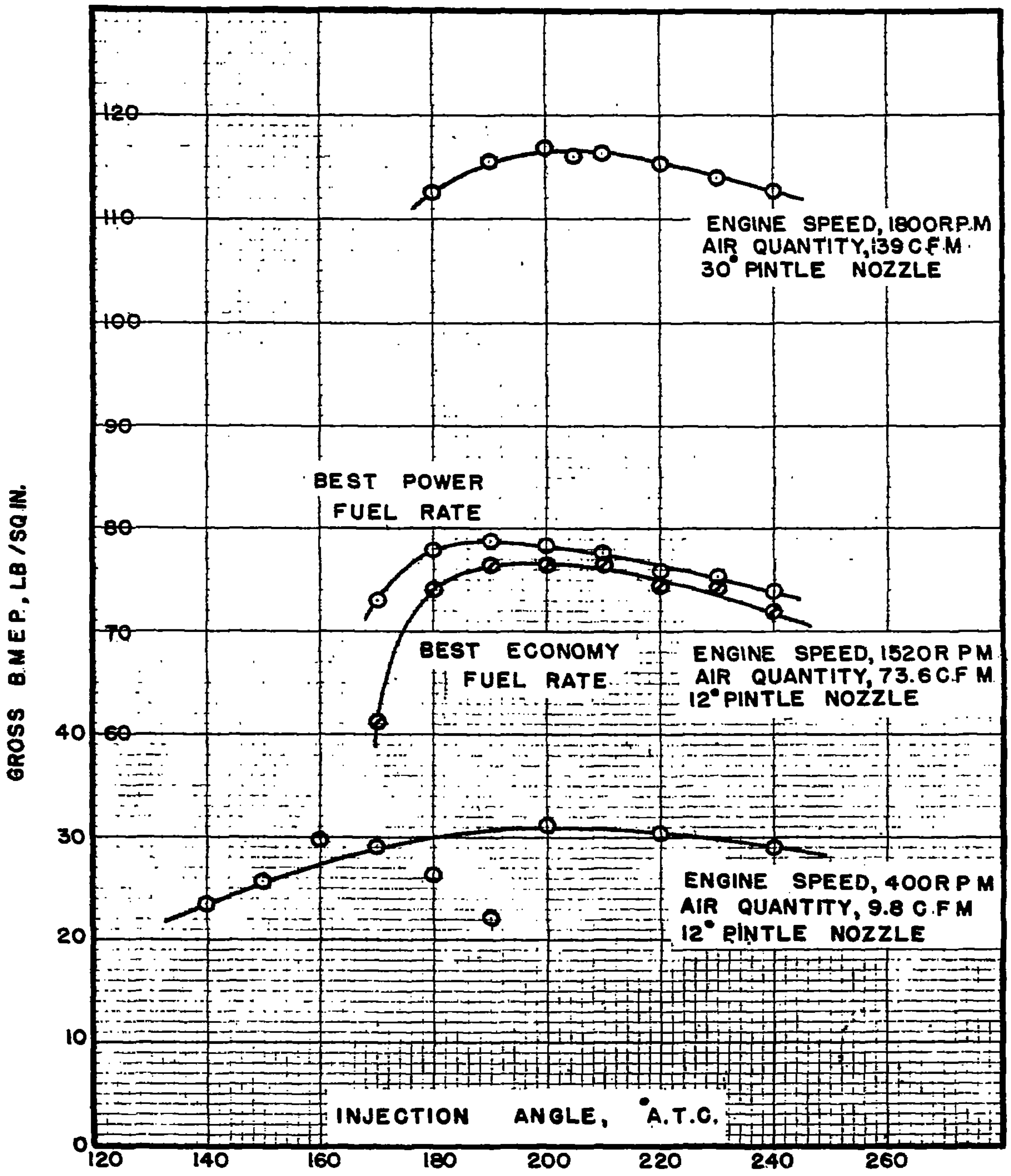


Figure 15.- The effect of injection angle on brake mean effective pressure for various speeds and air quantities; spark advance, 14° B.T.C.; injected gasoline. Best power fuel rate except as noted.

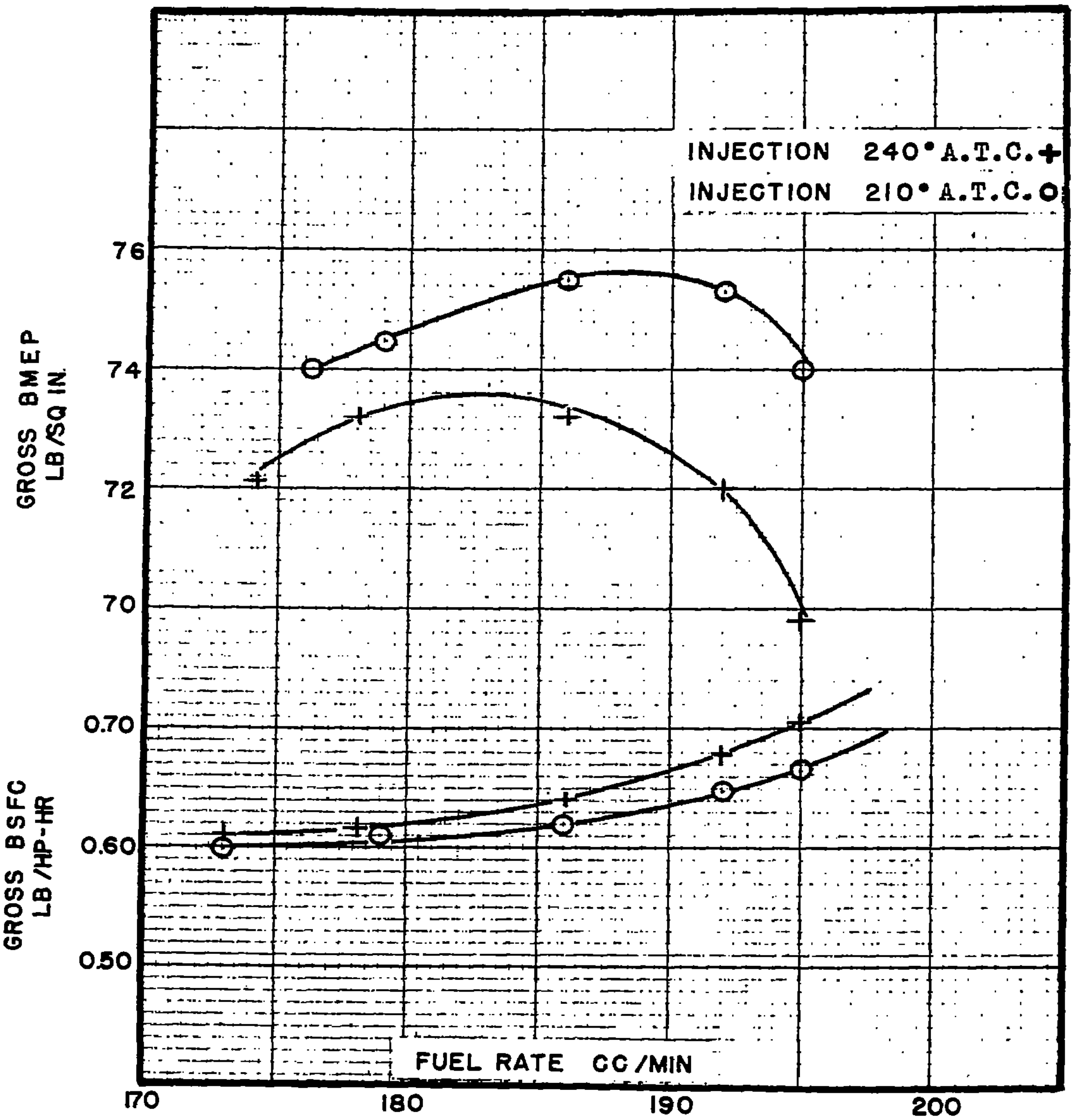


Figure 16 .- The effect of fuel rate on gross brake mean effective pressure and gross brake specific fuel consumption at two injection timings; engine speed, 1520 rpm; air quantity, 73.6 cu ft/min; spark advance, 15° B.T.C.

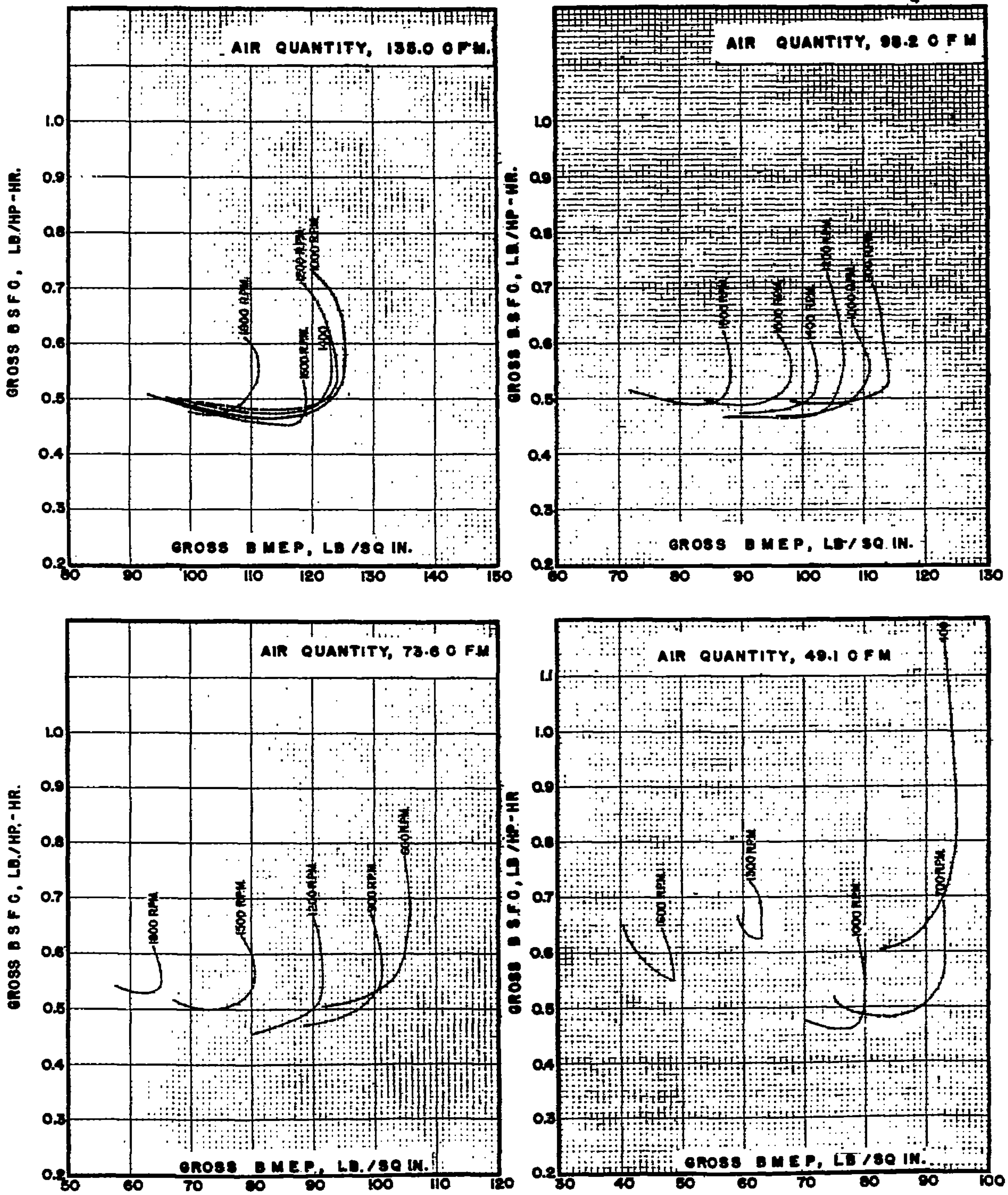


Figure 17. - Typical performance curves for injected gasoline; spark advance, 14° B.T.C.; injection angle, 200° A.T.C.

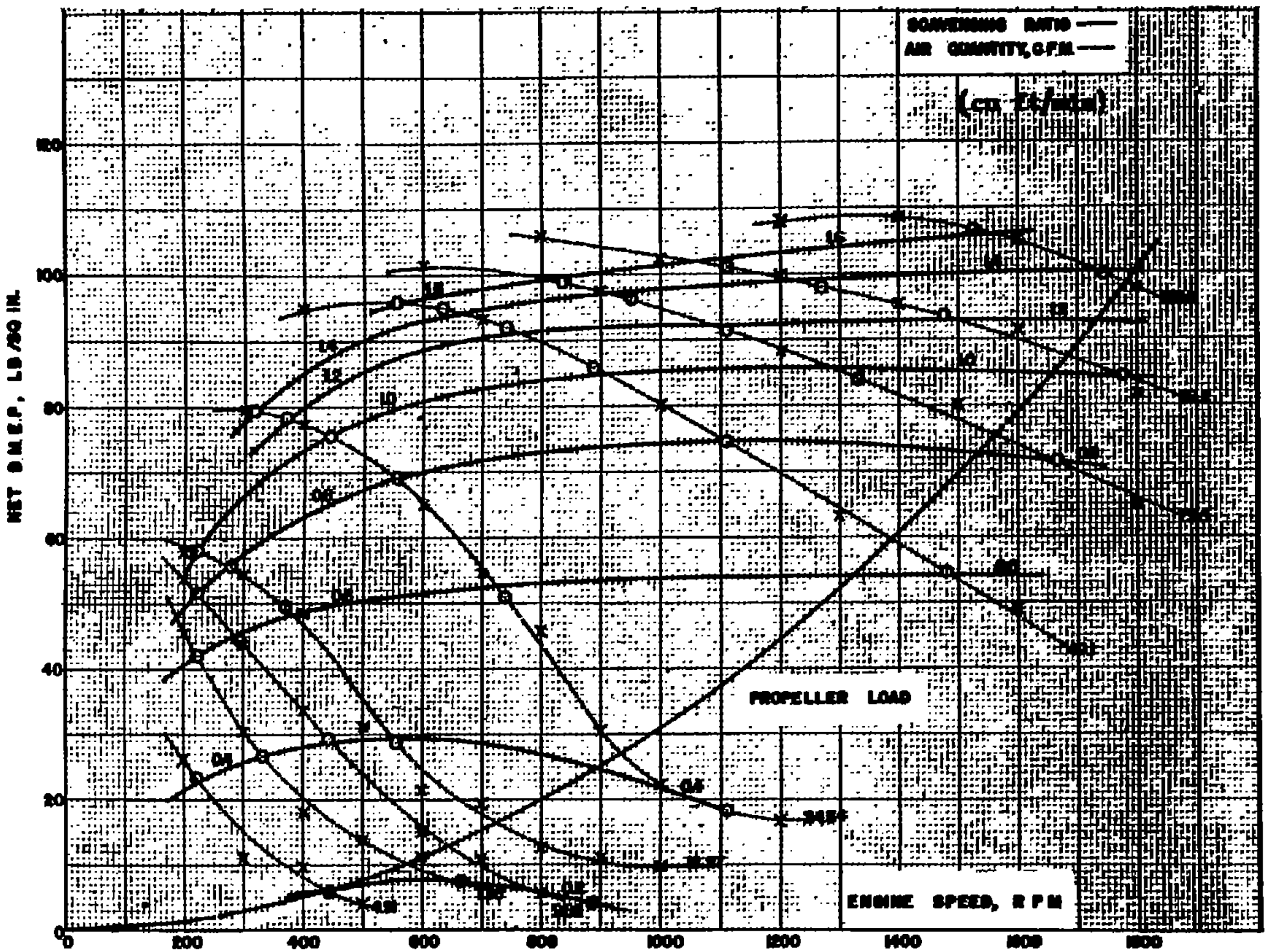


Figure 18. - The effect of engine speed on brake mean effective pressure at various scavenging ratios and air quantities; injected gasoline; spark advance, 14° B.T.C.; best power fuel rate; injection angle, 200° A.T.C.

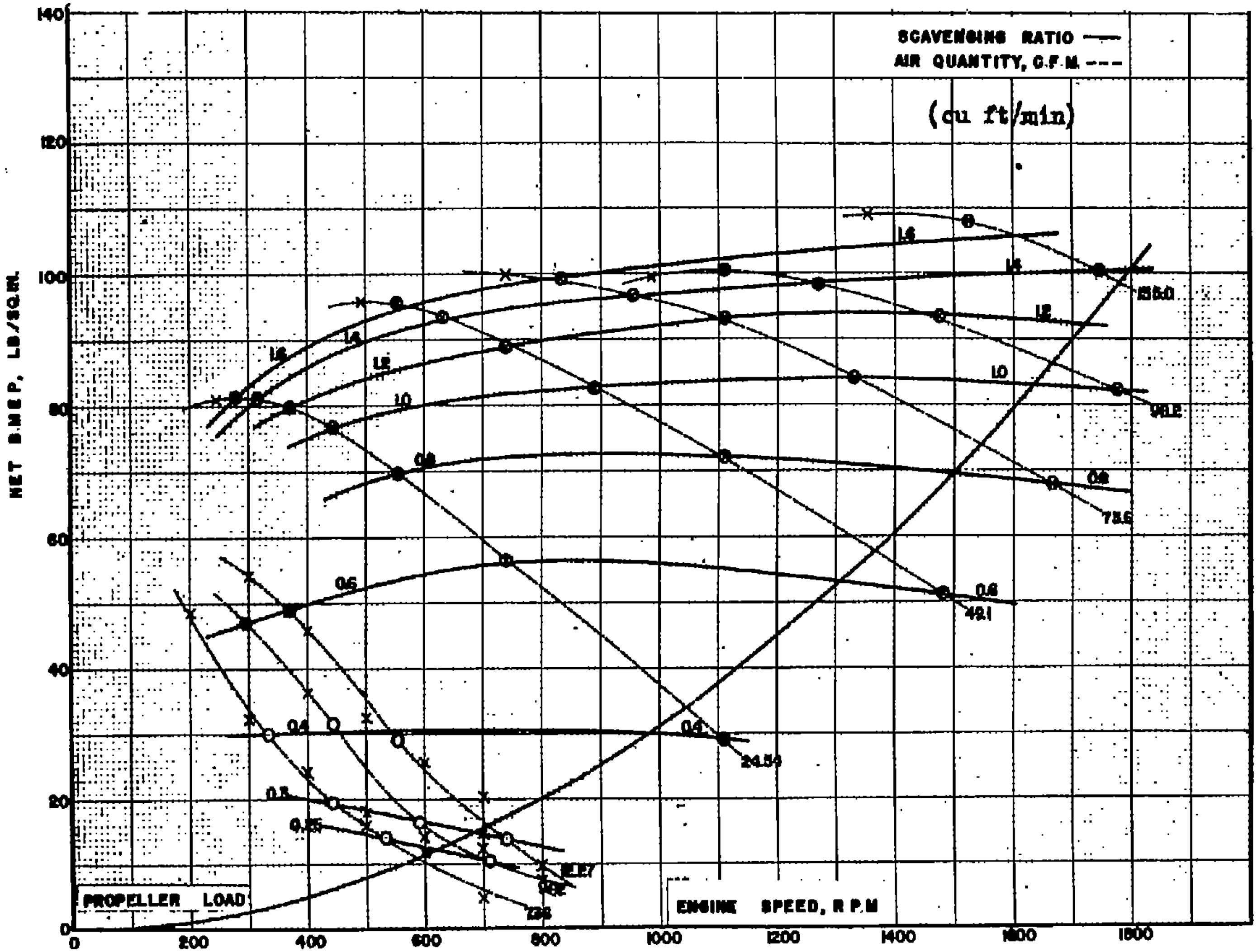


Figure 19. - The effect of engine speed on brake mean effective pressure at various scavenging ratios and air quantities; spark advance, 14° B.T.C.; safety fuel; best power fuel rate; injection angle, 210° A.T.C.

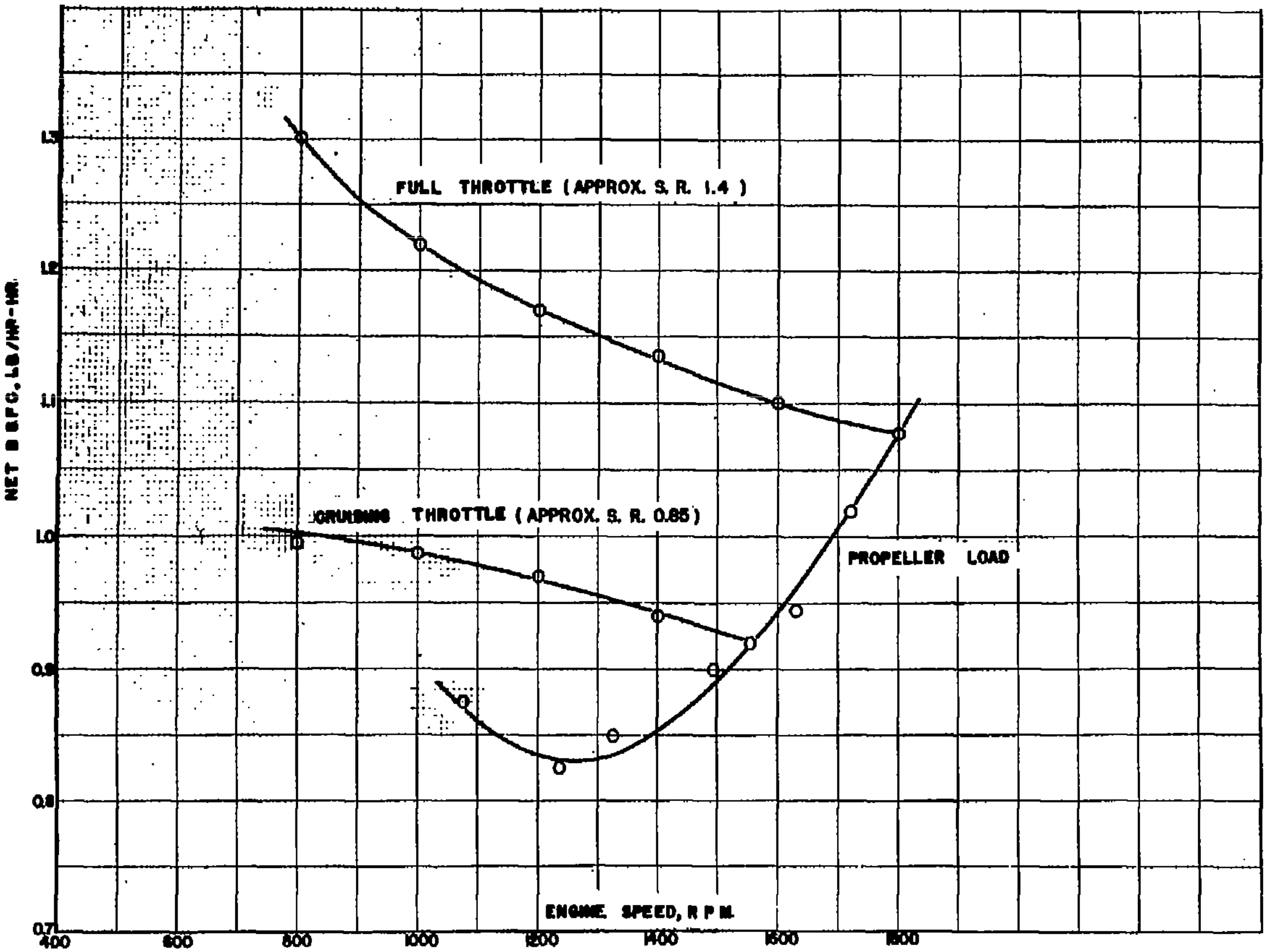


Figure 20. - Net brake specific fuel consumption at constant throttle and on a propeller load curve; vaporized gasoline; spark advance, 14° B.T.C.; fuel-air ratio, .075.

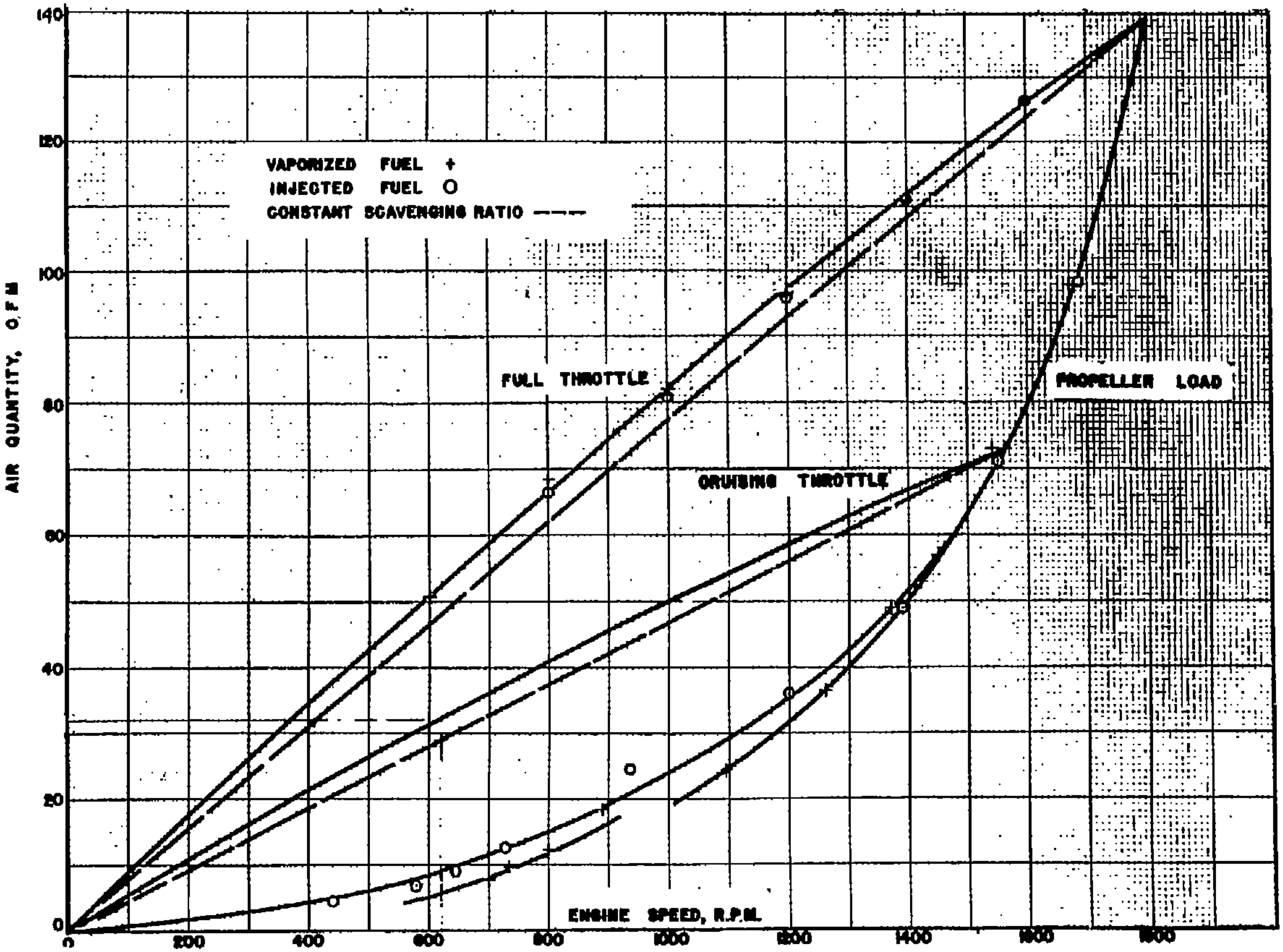


Figure 21. - Air quantity required at constant throttle and on a propeller load curve; injected gasoline; spark advance, 14° B.T.C.; best power fuel rate; injection timing, 200° A.T.C.

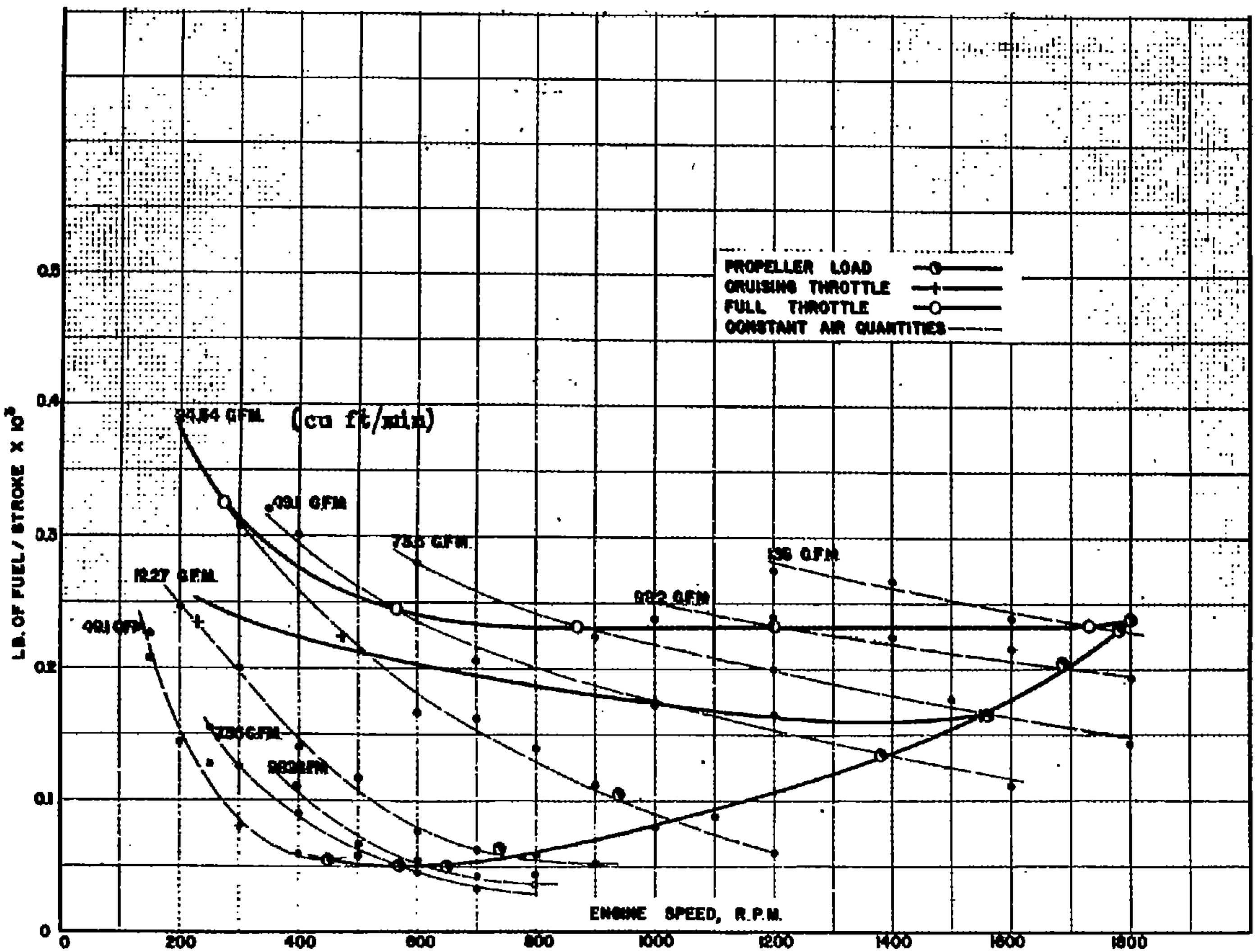


Figure 22.- Fuel quantity required per stroke for various air quantities and throttle settings; injected gasoline; best power fuel rate; spark advance, 14° B.T.C.; injection timing, 200° A.T.C.

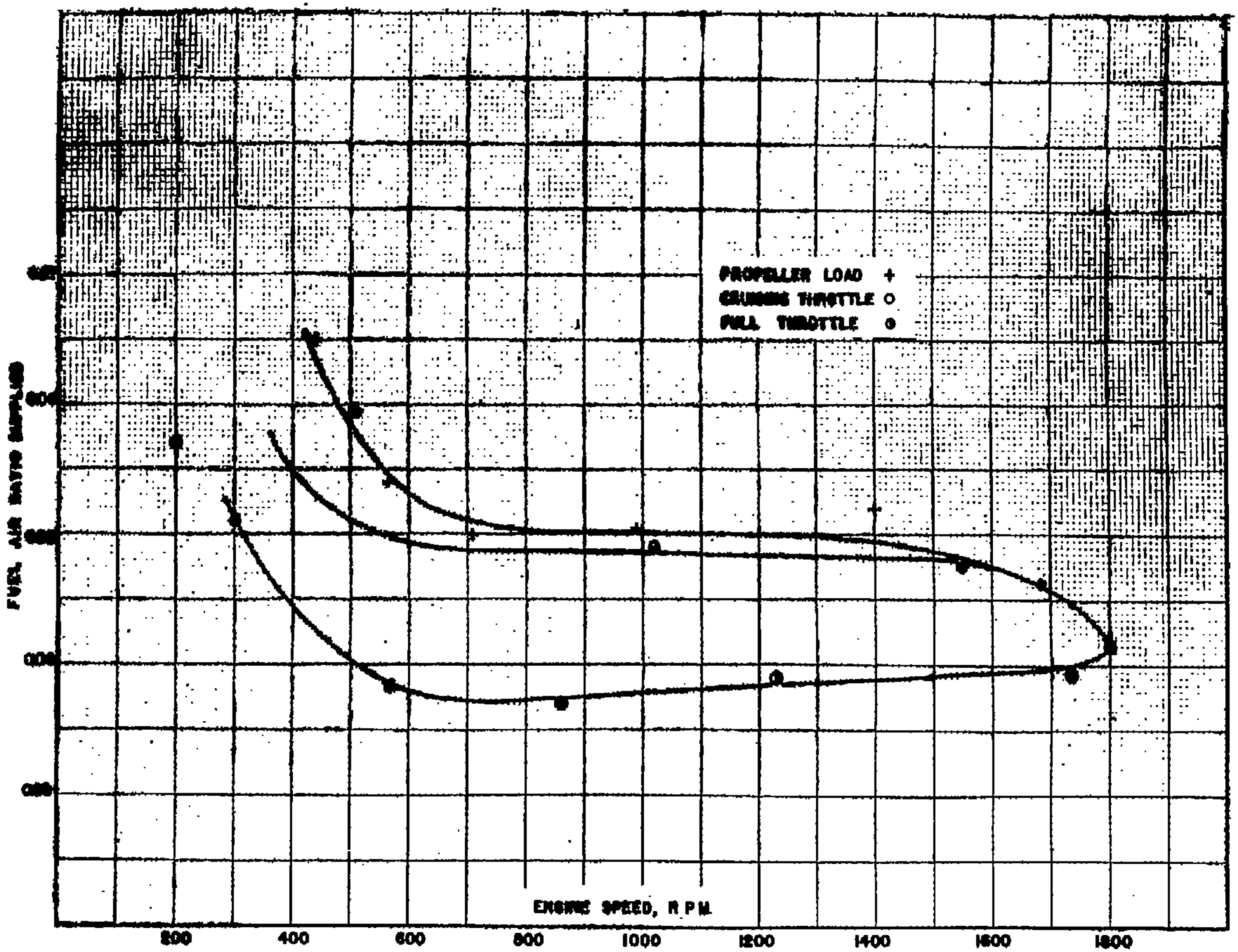


Figure 23.- Fuel-air ratio supplied to engine at constant throttle and on a propeller load curve; injected gasoline; best power fuel rate; spark advance, 14° B.T.C.; injection timing, 200° A.T.C.

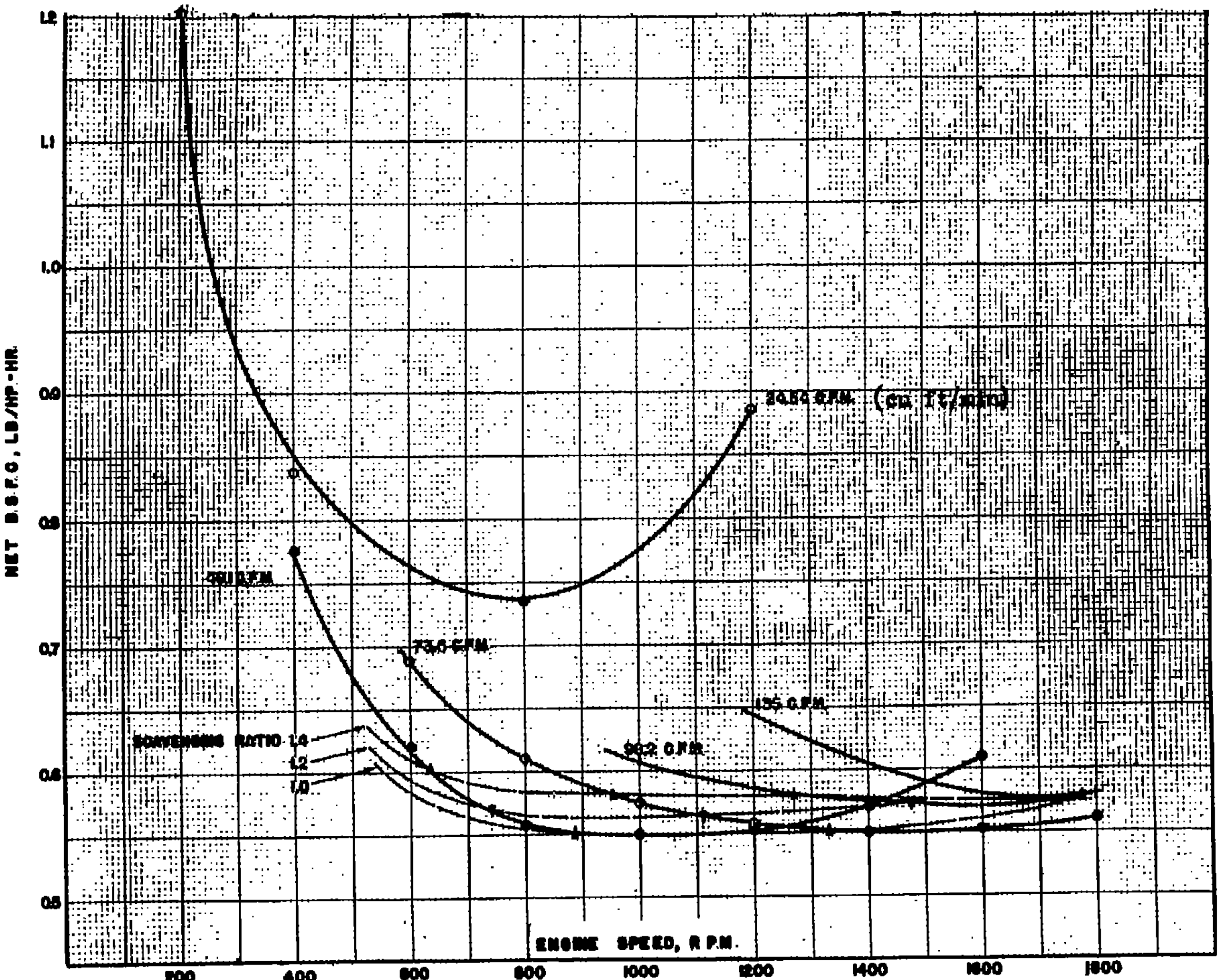


Figure 24.- The effect of engine speed on net brake specific fuel consumption at various constant air quantities and scavenging ratios; injected gasoline; best power fuel rate; injection timing, 200° A.T.C.

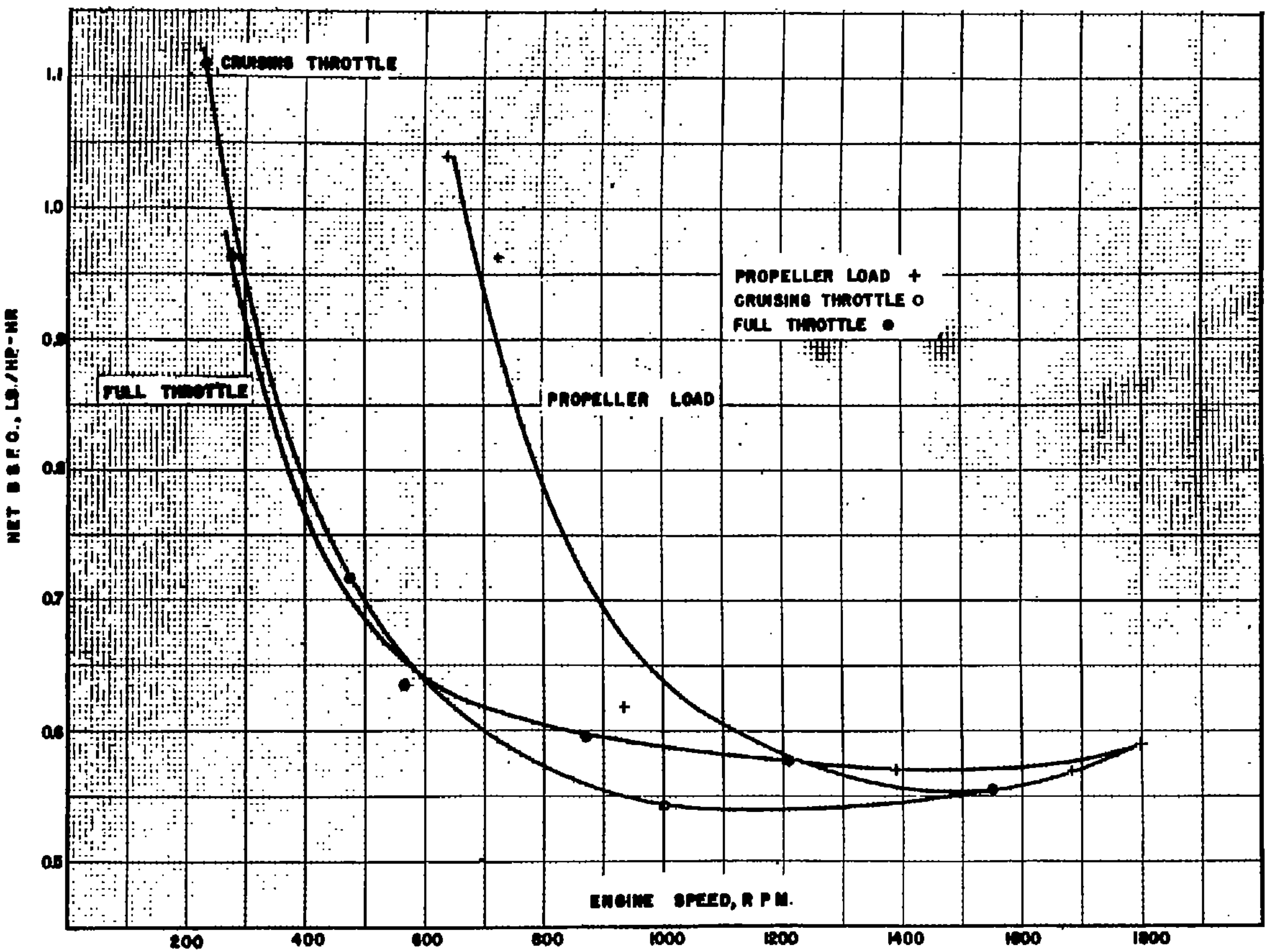


Figure 25.- Net brake specific fuel consumption at constant throttle and on a propeller load curve; injected gasoline, best power fuel rate; injection timing, 200° A.T.C.

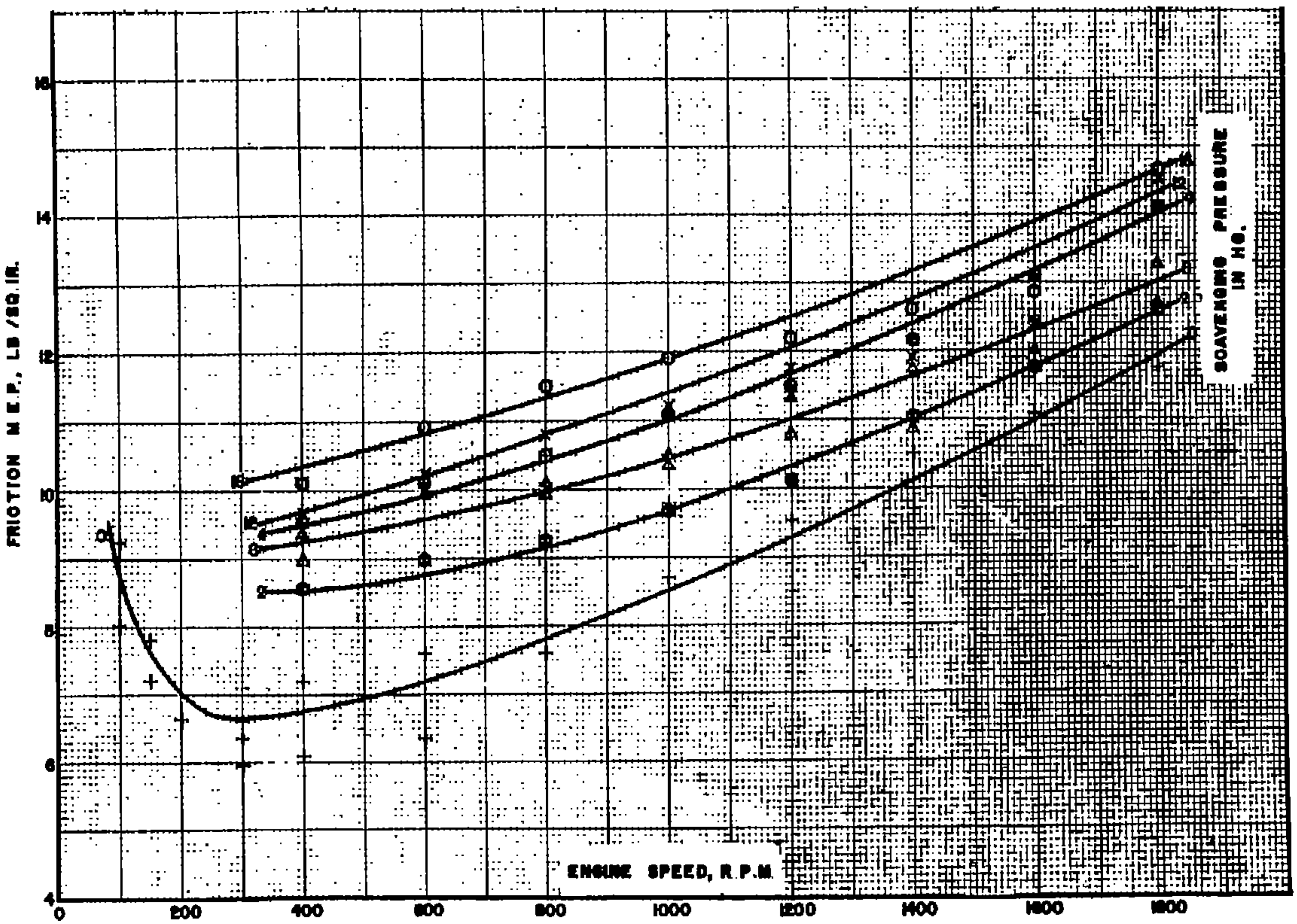


Figure 26. - The effect of engine speed on friction mean effective pressure at various scavenging pressures; motoring runs.