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SUMMARY

An investigation was made with a specially designed engine to determine the scavenging characteristics of a large number of inlet-port shapes and arrangements and the optimum port arrangement and timing for this particular type of engine. A special cylinder construction permitted wide variations in timing as well as in shape and arrangement of both the inlet and exhaust ports.

The study of the effect of port shape combinations and timings on engine performance was made using illuminating gas as a fuel. Through variations in inlet-port arrangement and port timings, the value of the scavenging efficiency was increased from an original 44 percent to approximately 67 percent with a corresponding increase in power. With the optimum port arrangement and timing determined, a large number of performance runs were made under both spark-ignition and compression-ignition operation.

INTRODUCTION

With a view toward increasing the specific output and simplifying the engine construction, much attention and work is being devoted to the development of suitable two-stroke engines for aircraft use. Since the output of a two-stroke engine is primarily dependent upon the scavenging efficiency of its cylinders, the problem of successfully developing an engine of this type is mainly one of obtaining the most complete scavenging possible without the use of a complicated mechanism or of excessively high scavenging pressures.

There are two general methods of scavenging the cylinder of the two-stroke engine, namely, the "through" scavenging in which inlet and exhaust ports are at opposite ends of the cylinder and the "loop" scavenging where inlet and exhaust ports are at the same end of the cylinder. The scavenging patterns of any two-stroke engine may be classified under one of these catagories.

The through-scavenged engine necessitates the use of valve gear or double pistons. This method, however, permits unidirectional flow of the combustion products and of the scavenging air through the cylinder. The loop-scavenged engine, on the contrary, requires neither valve gear nor double pistons. It is necessary to scavenge this type of cylinder by circulating the air in the form of a loop and, consequently, the scavenging air is much more likely to mix with the burned gases, making it more difficult to secure high scavenging efficiency.

Although the specific output of the loop-scavenged engine may never equal that of the through-scavenged engine, its inherent simplicity may outweigh this consideration in certain applications, provided that its scavenging efficiency can be made sufficiently high. The piston-ported engine lends itself particularly well to diesel operation because its lack of valve gear permits extreme flexibility in cylinder-head and combustion-chamber design; important factors in successful diesel operation.

The purpose of this investigation was to determine the practical limits of scavenging efficiency of a loop-scavenged cylinder, in order that its optimum performance might be compared and its advantages properly weighed and balanced against the performance and advantages of other engine types.

It is also the purpose of this investigation to furnish design data as to the arrangement and timing of the ports that give best performance in the loop-scavenged cylinder.

DESCRIPTION OF APPARATUS

Engine

The engine used for this investigation was a specially designed single-cylinder, water-cooled, piston-ported, two-stroke engine having a $4\frac{1}{2}$-inch bore and a 6-inch stroke. It was mounted on a universal crankcase and was directly connected to an electric cradle dynamometer. The engine incorporates three basic ideas, namely, variable inlet-port and exhaust-port timings, variable port shapes, and variable compression ratio. The inlet-port timing may be varied during operation from 45° to 65° from bottom center and the exhaust from 55° to 80° from bottom center. The port shape is varied by fastening inserts into the ports. These inserts may be made to deflect the entering air or charge at various angles toward the cylinder head. The compression ratio may be changed by displacing the head axially in the cylinder sleeve.

Figure 1 is an assembly drawing of the engine without its crankcase. Figure 2 shows photographs of the actual parts assembled. The cylinder sleeve that contains the ports is split longitudinally into two sections. One of the sections contains eight inlet ports and the other, four exhaust ports. Figure 3 shows an inside and outside view of the sleeves. One pair of 30° inserts may be seen in position in the inlet sleeve. Sleeves were independently water-cooled.

Figure 4 shows various sections through the sleeves and ports. Section D-D is taken through the inlet and exhaust ports. It will be noted that the eight inlet ports are arranged so that their center lines intersect slightly off the cylinder axis. This port arrangement appeared, from a study of the literature, to be the most promising design.

Figure 5(a) shows sections through the 0°, 30°, 45°, 60°, and 70° inserts. Figure 5(b) is a photograph of the actual inserts. A pair of inserts for one port consists of an upper and a lower part. Each part is held in place by a machine screw.

Two types of aluminum pistons were used, one having a flat head and the other a slightly rounded head. Figure 6 is a drawing and figure 7 is a photograph of both pistons. The flat-top piston was used a large part of the time while running on illuminating gas. The round-top piston was also used with illuminating gas to investigate the effect of a change in piston shape on scavenging.

A cast-iron head was used for spark-ignition operation. This head was slightly concave, as may be seen from the drawing in figure 8. An aluminum-alloy head, together with the round-top piston, was used for compression-ignition operation. (See fig. 9.) Figure 10(a) is a photograph of the cylinder head with its pear-shaped combustion space. A combustion chamber of this shape was chosen in an attempt to obtain good spray distribution with a minimum of obstruction to the flow of scavenging air.

Auxiliary Apparatus

Figure 11 is a schematic lay-out of the engine and its accessories as set up for spark-ignition operation on illuminating gas. Figure 12 shows two photographs of the engine set up for diesel operation. The fuel pump with its controls and the injection-nozzle holder are plainly visible.

Air was supplied to the engine by a separately driven compressor. The quantity of air was measured by a calibrated orifice box on the suction side of the pump. Illuminating gas from the city mains was compressed to intake pressure by a separately driven Roots blower. Gas flow was measured both by a large industrial gas meter and a pitot-static flowmeter. The pitot-static flowmeter was used to obtain instantaneous readings of the gas flow and the industrial gas meter was used to calibrate the pitot tube.

For compression-ignition operation a standard Bosch high-speed fuel-injection pump was used. The spray nozzle, which was located at the apex of the heart-shaped combustion chamber (see fig. 10(a)), was a special Bosch nozzle having three orifices, each 0.4 millimeter in diameter, located in a horizontal plane, and spaced 30° apart.

Resonance was eliminated in the inlet and exhaust systems by connecting the surge tanks to the engine with short lengths of large diameter pipe.

An M.I.T. balanced-pressure type indicator was used to obtain pressure-time diagrams.

Leakage of the entering mixture down the piston to the crankcase during that portion of the cycle in which the piston covered the inlet ports, was eliminated by keeping the crankcase at scavenging pressure. Leakage from the sealed crankcase was negligible.

Test Procedure

A summary of the various steps taken in the test procedure for the spark-ignition engine using illuminating gas is listed below:

1. With several inlet-port shapes, runs were made covering the entire range of inlet-port and exhaust-port timings. It was concluded from the results of these runs that inlet-port shape had little effect on the optimum inlet timing and even less effect on the optimum exhaust timing.

2. The results having shown that the optimum exhaust-port timing was not appreciably affected by changes in the inlet-port arrangements, the scavenging characteristics of a large number of inlet-port arrangements were investigated at a fixed exhaust-port timing of 65° B.B.C. and at three inlet-port timings.

3. The results from (2) confirmed the observations of (1): that the optimum inlet-port timing was unaffected by variations in inlet-port arrangements. Consequently, an additional number of inlet-port arrangements were tested under fixed inlet-port and exhaust-port timings of 47° and 65°, respectively.

4. A complete set of inlet-port and exhaust-port timing runs was made using the optimum inlet-port arrangement as determined from (1), (2), and (3).

5. With the conditions of optimum port arrangement and timing established, runs were made to determine brake mean effective pressure and scavenge pressure against engine speed at a constant scavenge ratio, constant scavenge pressure, and with the scavenge pressure varying as the square of the engine speed. Runs were also made over a range of scavenge ratios. Since most of these runs were made previous to those of (4), an inlet-port timing of 47°, based on the data of (1), (2), and (3) was used instead of 52° which appears to be the optimum timing for port arrangement, D.

6. With the optimum inlet-port arrangement as previously determined, runs were made covering the inlet-port and exhaust-port timing ranges to determine the optimum timings under compression-ignition operation.

7. With the optimum inlet-port arrangement and timing, runs were made under compression-ignition operation to determine brake mean effective pressure and specific fuel consumption against fuel rate over a range of scavenge ratios at constant engine speed.

8. An investigation similar to that under (7) was made over a range of engine speeds at a constant scavenge ratio.

The engine conditions that were held constant for most of the work, are as follows:

Engine speed 1,800 r.p.m.

Compression ratio ... 7 (spark ignition) and 15 (compression ignition)

Air-fuel ratio .... 4.4 (illuminating gas, best power)

Ignition timing ... 16° B.T.C.

Scavenge ratio 1.4

Fuel .... illuminating gas (spark ignition)
fuel oil (compression ignition)

The blower mean effective pressure was calculated as follows, assuming adiabatic compression and a blower efficiency of 70 percent (see fig. 13):

Blower horsepower

$$= \frac{m\ J\ C_p\ T_i \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]}{0.70 \times 33000}$$

Blower mean effective pressure

$$= \frac{m\ J\ C_p\ T_i \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]}{0.7\ L\ A\ N}$$

in which:

m, pounds of air per minute.

J, 778 foot pounds per B.t.u.

C_p, specific heat of air at constant pressure, 0.235 B.t.u. per pound.

T_i, blower inlet air temperature, °R.

p_2, blower delivery pressure.

p_1, blower inlet pressure.

n, 1.41.

L, stroke, feet, of engine piston.

A, engine piston area, square inches.

N, r.p.m. of engine.

DISCUSSION AND RESULTS

Spark Ignition - Illuminating Gas

The term "scavenge ratio" as used in this work is the ratio of the volume of charge (mixture of air and illuminating gas under spark-ignition conditions) or of air (under compression-ignition conditions) that passes through the engine cylinder per stroke, to the displaced volume of the cylinder. The volume of the entering charge is measured under atmospheric (laboratory) conditions of temperature and pressure.

The term "scavenging efficiency" is defined in this report as the ratio of the weight of fresh charge in the cylinder at the close of the exhaust ports, to the product of the cylinder volume and the inlet density.

Illuminating gas was chosen as a fuel because it was possible to mix it thoroughly with the accompanying air. Thorough mixing of the fuel and air is absolutely necessary when the relative scavenging efficiency is to be determined from changes in brake mean effective pressure due to corresponding changes in inlet-port arrangement. Poor mixing of the fuel and air, as would be likely with a liquid fuel, would affect the combustion process and hence the thermal efficiency and power. Under these circumstances, changes in the mean effective pressure with port arrangement could not be taken as an indication of relative scavenging efficiency.

All runs under spark-ignition operation were made using illuminating gas as a fuel. Since the power obtained from a best power mixture of illuminating gas and air was found from actual tests on a four-stroke C.F.R. engine to be 85 percent of that of a best power mixture of gasoline and air, it is reasonable to believe that an increase of 15 percent in power would be possible in the two-stroke engine by substituting gasoline for illuminating gas.

Although the method of determining the relative scavenging abilities of various inlet-port arrangements by means of changes in the brake mean effective pressure is straightforward and simple, it is a much more difficult problem to determine the absolute scavenging efficiency under any one inlet-port arrangement. The value of an absolute measure of scavenging efficiency resides in the fact that a theoretical limit of the scavenging efficiency may be determined. If it is found that the existing optimum port arrangement yields a scavenging efficiency almost equal to this limit, further research may be unnecessary. The method employed in determining the efficiency of the scavenging process under the optimum port arrangement and timing conditions will be briefly outlined.

As previously defined, the scavenging efficiency is the ratio of the weight of fresh charge in the cylinder at the time the ports close to the product of the total cylinder volume and inlet density. The inlet density conditions were arbitrarily taken as 75° F. and 29.92 inches of mercury. Determination of the denominator of this ratio is quite simple inasmuch as the total cylinder volume and inlet density conditions are known. The numerator of the ratio, the actual weight of fresh charge in the cylinder at the close of the ports, may be determined if the indicated output and the specific air consumption (pounds of air per indicated horsepower per hour) are known.

In order to determine the value of the specific air consumption when running on illuminating gas and air, and at a compression ratio of 7, runs were made at best power fuel-air ratio and ignition timing on a four-stroke C.F.R. engine at several engine speeds. The foregoing conditions were made to reproduce as closely as possible those of the optimum two-stroke operation, in order that the indicated thermal efficiency for the two cases would be as nearly the same as possible. The calculations relative to the foregoing discussion are included.

Actual indicated horsepower of the two-stroke engine: 48.8.

The specific air consumption as determined from the C.F.R. engine operating on illuminating gas: 5.85 pounds of air per indicated horsepower per hour.

The air actually consumed in the two-stroke engine per stroke: $\frac{48.8 \times 5.85}{1800 \times 60} = 0.00264$ pound.

Engine speed: 1,800 r.p.m.

The number of pounds of air in a full cylinder volume containing the best power mixture of illuminating gas and air at 75° F. and 29.92 inches of mercury:

$$\frac{111.3}{1728} \times 0.0742 \times \frac{4.8}{5.8} = \underline{0.00396} \text{ pound.}$$

Total cylinder volume in cubic inches: 111.3.

Inlet density (75° F. and 29.92 in. Hg): 0.0742.

Ratio of $\left(\frac{\text{air}}{\text{air + illuminating gas}}\right)$ by volume: $\frac{4.8}{5.8}$.

Scavenging efficiency $= \frac{0.00264}{0.00396} \times 100 = \underline{66.7}$ percent.

On the basis of this definition of scavenging efficiency, it is evident that even though the cylinder were completely purged of its burned products and it contained only fresh charge under the previously specified conditions of inlet density, the charging efficiency would not be equal to 100 percent. The portion of the piston displacement included between the bottom-center and the port-closing positions is lost.

It is possible to make an estimate of the maximum practical scavenging efficiency attainable in this particular engine. This maximum value of the scavenging efficiency will be reached when the exhaust gases are completely expelled and when the cylinder, at the time of port closing, contains only fresh charge at atmospheric density.

$$\text{Maximum attainable scavenging efficiency} = \frac{89.4}{111.3} \times 100 = 80.3 \text{ percent.}$$

Total cylinder volume in cubic inches less the lost cylinder volume (i.e., portion of cylinder volume contained between the bottom center and port closing position of the piston) = 89.4.

Total cylinder volume in cubic inches: 111.3.

$$\frac{\text{Highest actual scavenging efficiency obtained}}{\text{Maximum attainable scavenging efficiency}} = \frac{0.667}{0.803} \times 100 = 83 \text{ percent.}$$

An independent check on this ratio of the actual optimum to the maximum attainable scavenging efficiency was made. This check was based upon the assumption that the ratio of actual to possible indicated mean effective pressures is equal to the ratio of actual to possible scavenging efficiencies. The indicated mean effective pressure obtained under the actual optimum scavenging efficiency conditions is known. The denominator of the ratio, the indicated mean effective pressure that would be obtained if all the burned gas were expelled and only fresh charge remained in the cylinder, is the unknown quantity. This quantity was determined from runs made on a four-stroke, single-cylinder engine and under the same test conditions as those of the two-stroke engine in order to keep the indicated thermal efficiencies of both engines as nearly equal as possible.

The problem of determining from a four-stroke engine the maximum indicated mean effective pressure that could be obtained from the two-stroke engine is mainly one of charging the displacement volume of the four-stroke engine with a fresh mixture of the same density as in the two-stroke engine. In order to make the density conditions equal, it is necessary to know the temperature and the pressure of the charge in each engine at, for example, the start of compression. As a first approximation, it was assumed that the temperature of charge in both engines at the start of compression was equal. The compression ratio of the two engines was the same, and the compression pressures of both engines were made equal by slightly throttling the four-stroke engine. Under these conditions the charge density and indicated thermal efficiency of the two engines would be equal and the mean effective pressure that the compressed charge was capable of delivering would be constant whether in the combustion space of the two-stroke or the four-stroke engine. Since the clearance volume of the four-stroke engine was not scavenged, it is necessary to multiply its indicated mean effective pressure by the ratio of total cylinder volume to displaced volume. The value thus obtained will be the two-stroke maximum attainable indicated mean effective pressure. The ratio of the best actual indicated mean effective pressure obtained from the two-stroke engine to the maximum attainable mean effective pressure will be the same as the ratio of best actual to the maximum attainable scavenging efficiency.

This ratio of the best actual to the maximum attainable scavenging efficiency as calculated on the foregoing basis is equal to 87 percent as compared with a similar ratio of 85 percent on the <u>four-stroke</u> engine. Since the ratios are almost equal for both engines, the proportion of burned products remaining in the cylinder of each engine must be nearly the same. This being the case, the increase in temperature of the fresh charge due to mixing with the burned products must be approximately equal in both engines and therefore the validity of the original assumption of equal charge temperatures in both engines at the start of compression is established.

The calculation of the probable maximum attainable indicated mean effective pressure of the two-stroke engine as determined from the four-stroke engine is as follows:

The indicated mean effective pressure obtained from the four-stroke engine running on illuminating gas with the charge density and compression ratio equal to that of the two-stroke engine: 110 pounds per square inch.

The probable maximum two-stroke indicated mean effective pressure: $110 \times \frac{7}{6} = \underline{128.4}$ pounds per square inch.

Ratio of total cylinder volume to the displaced volume: $\frac{7}{6}$ in both engines.

Probable maximum two-stroke brake mean effective pressure: 128 - 14 = 114.

Two-stroke friction mean effective pressure (motoring): 14.

The highest actual two-stroke indicated mean effective pressure obtained when running on illuminating gas and with the optimum (E) inlet-port arrangement: 112.1 pounds per square inch.

$$\frac{\text{Highest actual scavenging efficiency}}{\text{Maximum attainable scavenging efficiency}} : \frac{112.1}{128.4} \times 100 = 87.3 \text{ percent.}$$

A comparison of the values of the ratio of the actual to the maximum scavenging efficiencies as determined from both methods shows that they agree reasonably well, the first method giving a value of 83 percent and the second 87 percent. It may therefore be briefly restated that the best actual scavenging efficiency obtained under spark-ignition operation at 1,800 r.p.m. was about 66.7 percent or approximately 85 percent of perfect scavenging for this engine.

Determination of the optimum inlet-port arrangement.- With all the engine conditions held constant, the effects of more than 30 inlet-port arrangements on the relative scavenging efficiency were investigated. Figure 14 is a plot of brake mean effective pressure and scavenging pressure against inlet-port timing for five typical port arrangements. The positions of the curves with respect to one another may be taken as an indication of the relative scavenging abilities of each port arrangement.

A numbering system was found to be the most convenient method of designating the inlet-port arrangements. In figure 15 showing a section through the ports, the numbers are located in each of the eight inlet ports. Each one of the symmetrically opposite pairs of inlet ports bears the same number. As an example, assume that the number 4 ports are fitted with 60° inserts, the number 3 ports with 45° inserts, and the number 2 and 1 sets of ports with 0° inserts. This port arrangement would then be described by the following group of numbers: 0-0-45-60-60-45-0-0. The numbers represent the insert angles and their positions correspond to port positions.

In figure 14 the capital letters opposite each of the five curves represent the following inlet-port arrangements.

INLET PORT NUMBERS

INLET PORT ARRANGEMENTS	1	2	3	4	4	3	2	1
A	45	45	45	45	45	45	45	45
B	0	0	0	0	0	0	0	0
C	0	30	45	60	60	45	30	0
D[1]	0	0	0	60	60	0	0	0
E[2]	0	0	0	60	60	0	0	0

[1] The top sections were left out of the number 1 and 2 pairs of inserts.

[2] The top sections were left out of the number 1, 2, and 3 pairs of inserts, and the lower sections of the same inserts were cut down 1/8 inch to increase the air-flow area. This condition brought the lower port surface flush with the top of the piston at bottom dead center and best inlet timing.

The A inlet-port arrangement of figure 14 represents the lowest brake mean effective pressure. This low value results from the poor scavenging obtained on account of the direction of the entire entering charge which is at an angle of 45° toward the cylinder head. Curve B indicates that the 0° inserts are much more effective in scavenging the cylinder than are the 45° inserts.

In order further to increase the scavenging efficiency, and hence the output, it was found necessary to use mixed port arrangements. The most successful of the many mixed ports investigated were the C, D, and E arrangements.

The C and D arrangements show up very well on the basis of net and gross brake mean effective pressure. In the D arrangement the tops were removed from the number 1 and 2 pairs of inserts, which allowed these two pairs of ports a 3° opening lead and so gave the charge entering these ports a start over the other ports.

The E port arrangement was tried at a late period during the investigation; in fact, only after the various speed and scavenging ratio runs were completed with the D port arrangement. The E arrangement was the optimum, on the basis of scavenging efficiency and power output of all the 30 odd arrangements investigated. Many modifications of the E arrangement were tried and several of these resulted in mean effective pressures very nearly as high as those obtained with the E arrangement.

Although the limit of improvement in scavenging efficiency with inlet-port arrangement in this engine appears to have been reached, it is believed that a study of the flow conditions as viewed through the transparent cylinder of a flow model exactly similar to the actual engine cylinder in size and shape may readily reveal the possibility of a further increase in scavenging efficiency over the best values obtained thus far.

Inlet-port and exhaust-port timings.- It is usually assumed that the exhaust port should open sufficiently ahead of the inlet port so that the cylinder pressure will drop to the scavenge pressure or lower before the inlet opens. Considering the inlet timing to be fixed, it is not surprising to find that an exhaust opening which gives scavenge pressure in the cylinder at the time of inlet opening, gives less power than one with a slightly later exhaust opening. The later exhaust opening results in both a higher expansion ratio and a smaller loss of fresh charge into the exhaust. If a constant scavenge air quantity is supplied to the cylinder, in the case of late exhaust opening, the slight additional losses due to higher scavenge pressure and to slightly greater mixing of the exhaust gases and the fresh charge will not equal the gain from the decreased blow-down loss until the cylinder pressure at the time of inlet opening is somewhat larger than the scavenge pressure.

It is surprising to note (fig. 16) how great an excess of cylinder pressure over scavenge pressure exists with the best power timing. Under these conditions, considerable exhaust must enter the inlet system and later be blown back into the cylinder with the fresh charge. Tests showed that diesel operation required an earlier exhaust opening for best power. Figure 17 is an indicator diagram for diesel operation at the best power timing.

The effect of inlet-port and exhaust-port timing on brake mean effective pressure and scavenge pressure may be seen from figures 18 and 19, respectively. Although the gross brake mean effective pressure increases uniformly with a late exhaust-port opening, the scavenge pressure also increases in order to maintain a constant scavenge ratio of 1.4. Consequently, the net brake mean effective pressure shows considerably less change with exhaust timing.

The curves of brake mean effective pressure against inlet-port timing (fig. 18) peak at an inlet opening of approximately 52° before bottom center, which is the optimum inlet-port timing. In the determination of the optimum exhaust-port timing, consideration must be given to the scavenge pressure as well as to the brake mean effective pressure.

It being desirable to keep the scavenge pressure as low as possible, an exhaust opening of 65° B.B.C. was chosen as optimum.

Brake mean effective pressure against engine speed.- A first glance at the curves in figure 20 would lead one to suspect that resonance in the inlet or exhaust systems was the cause of the peaks in the mean-effective-pressure curves. The scavenge ratio was held constant throughout the range of speed investigated, however, and the presence of peaks in the mean-effective-pressure curves therefore cannot be attributed to resonance.

It is quite possible that at the engine speeds corresponding to peaks in the curves of mean effective pressure, the time-opening characteristic of the inlet ports may be such as to promote a more efficient scavenging process.

Figure 21 shows the effect of engine speed on the gross and net brake mean effective pressures with the scavenge pressure held constant at 11.8 inches of mercury. The scavenge ratio decreases steadily with increasing engine speed, but the curve is quite smooth and shows no indication whatsoever of resonance.

Figure 22 is another set of speed runs in which the increase in scavenge pressure was made proportional to the square of the engine speed. It is notable that although the scavenging ratio decreases from 1,200 to 1,900 r.p.m., the gross mean effective pressure increases, showing an improvement in scavenging. This result is undoubtedly due to the fact that the port timing and arrangement were selected for 1,800 r.p.m. A different port timing and perhaps a different arrangement would be necessary to give the best mean effective pressure at, say, 1,200 r.p.m. The lack of wide fluctuations in the mean effective pressure and the scavenge-ratio curves confirms the foregoing conclusion that the set-up is so made that resonance effects are small.

<u>Brake mean effective pressure and scavenge pressure against scavenge ratio</u>.- Figure 23 shows the effect of an increase in scavenge ratio on the brake mean effective and scavenge pressures. The increase in the mean effective pressure with scavenge ratio is due principally to an improvement in scavenging. At relatively high values of scavenge ratio, the effect of supercharging, that is, of increased pressure at the time of exhaust closing, becomes significant. The previously determined limit to the brake mean effective pressure obtained with maximum attainable scavenging efficiency was 128 - 14 = 114 pounds per square inch. The brake mean effective pressure of an actual engine might be expected to approach this value as the scavenge ratio is indefinitely increased. Reference to figure 23 shows that 114 pounds per square inch appears to be a reasonable value for an asymptote of the gross mean-effective-pressure curve.

<u>The effect of combustion-chamber and piston-head shape on scavenging</u>.- In order to investigate the effect of combustion-chamber shape on the scavenging process, several runs were made with illuminating gas using the compression-ignition cylinder head (fig. 10). Under constant operating conditions, power readings were taken with the cylinder head located in its normal position and then turned 90° and 180° from its normal position. Rotation of the head in this manner placed the pear-shaped combustion space at various angles to the flow of scavenging charge. The mean effective pressure, and hence the scavenging, was unaffected by these changes.

The effect on scavenging of a piston having a slightly rounded head (figs. 6 and 7) was also investigated.

The round-top piston showed a slight improvement in scavenging and mean effective pressure over the flat-top piston. The increase in brake mean effective pressure amounted to approximately 3 or 4 percent.

Compression Ignition

The E inlet-port arrangement was used throughout the investigation of compression-ignition performance on the assumption that the inlet-port arrangement giving the most satisfactory scavenging under spark-ignition conditions would still be the optimum arrangement under compression-ignition operation.

Figure 24 is a set of curves of brake mean effective pressure against fuel rate at various scavenge ratios. It is interesting to note that the brake mean-effective-pressure curves lie closely to one another in the region of low power, where there is an excess of air in the cylinder at all scavenge ratios. As the fuel rate is increased, however, the curves diverge as is to be expected since, at high percentages of air, scavenging is more complete and hence more oxygen is available for combustion.

The fact that the minimum points of the fuel-consumption curves occur at light loads indicates that there is opportunity for considerable improvement in combustion-chamber design. Here again the values of minimum fuel consumption differ only very slightly from one another for the various scavenging ratios. At increasing loads the low scavenging ratios show a higher specific fuel consumption owing to less complete combustion as a result of poorer cylinder scavenging or fuel-air mixing.

Figure 25 is a similar set of curves in which fuel-rate runs were made at various speeds. In this case the scavenge ratio was held constant at 1.4. These curves indicate that the maximum gross mean effective pressure is constant with respect to engine speed, being approximately 72 pounds per square inch. This fact leads to the conclusion that the maximum quantity of fresh air available per stroke for combustion must be constant with respect to engine speed and therefore that the scavenging efficiency must also be constant. It is evident from the curves that, at high speeds, a large quantity of fuel per stroke is necessary to consume all the air. This condition results, possibly, from the shorter period of time available for mixing of the fuel and the air at high speeds.

Estimate of possible maximum output on compression-ignition operation.- Figure 26 is a light-spring pressure-crank-angle indicator diagram taken under firing conditions for the purpose of determining the pressure in the cylinder at the instant the exhaust ports close. A knowledge of the magnitude of the pressure and the temperature at this point allows a calculation to be made of the actual quantity of charge (burned products plus fresh charge) contained in the cylinder. Multiplication of this total quantity by scavenging and combustion efficiency factors gives a net quantity of air available for combustion. The fuel rate necessary to consume this available air at the chemically correct fuel-air ratio is then easily calculated; and a value of the estimated power may be obtained from this fuel rate, the heating value of the fuel, and the cyclic thermal efficiency. A calculation of this estimated power follows.

From the card of figure 26 compression of the air begins at atmospheric pressure and approximately 3° before the closing of the exhaust ports. The cylinder volume at 3° before exhaust closing is:

$$95.5 \times \frac{4.15}{6.00} + 5.5 = 71.5 \text{ cu. in.}$$

Piston displacement in cubic inches: 95.5.

Total stroke less that portion of the stroke included between the bottom center position of the piston and that position of the piston at the beginning of compression from atmospheric pressure: 4.15 inches.

Clearance volume in cubic inches: 5.5.

Actual weight of fresh air in the cylinder at standard conditions (760 mm Hg and 32° F.):

$$\frac{71.5}{1728} \times \frac{492}{650} \times 0.0809 \times 0.85 = 0.00215 \text{ pound.}$$

32° F. = 492° Rankine.

Assumed charge temperature at close of exhaust ports degrees Rankine: 650.

Specific weight of air at standard conditions: 0.0809.

Assumed ratio of actual to maximum scavenging efficiency: 0.85.

Weight of fuel per stroke necessary to consume all the fresh air: 0.067 x 0.00215 = 0.000144 pound.

Ratio of fuel to air by weight for complete combustion: 0.067.

Probable maximum horsepower = 0.000144 x 1,800 x 60 x 0.85 x 0.40 x 18,500 x $\frac{1}{2545}$ = 38.5 horsepower.

Engine speed: 1,800 r.p.m.

Ratio of maximum useful to the chemically correct fuel-air ratio: 0.85.

Indicated thermal efficiency: 0.40.

B.t.u. per pound of fuel: 18,500.

B.t.u. per horsepower per hour: 2,545.

The maximum horsepower obtained under diesel operation was 31. It is believed that with improvement in combustion-chamber design the output may be increased to approximately 40 horsepower.

Figure 27 is a light-spring indicator diagram taken under exactly similar engine conditions as the card of figure 26 except for exhaust port timing. The card of figure 26, corresponding to an exhaust-port closing at 76° A.B.C., shows no appreciable supercharging at the close of the ports. With the exhaust-port closing occurring somewhat earlier, as was the case with the card of figure 27 (exhaust closed at 70° A.B.C.), the effect of slight supercharging is evident, amounting to approximately 2 pounds per square inch. As was previously pointed out, although this exhaust-port timing resulted in a little supercharging, the gross power output was less than while running at 76° A.B.C. exhaust closing.

CONCLUSIONS

Spark-Ignition Operation

1. Inlet-Port Arrangements:

In general, it was found that the most successful inlet-port arrangements from considerations of high scavenging efficiency and power output were those having low air-entry angles in the number 1, 2, and 3 ports and high air-entry angles in the number 4 ports. (See fig. 15.) The best port arrangement tried was one having 0° inserts with the top sections removed in the number 1, 2, and 3 ports and 60° inserts in the number 4 ports (E arrangement).

In view of the fact that modifications of the E inlet-port arrangement showed no improvement in scavenging, it was felt that further improvement would be possible only through a visual study of the flow conditions in a special flow chamber.

2. Inlet and Exhaust Port Timings:

Although there was only a slight change in either the gross or the net output with variations in the inlet port timing between 45° and 65° from bottom center, the optimum inlet-port timing for port arrangement D, was found to be approximately 52° from bottom center.

Varying the exhaust-port timing from 74° to 60° from bottom center resulted in an increase in both the gross brake mean effective and the scavenge pressures, causing the net brake mean effective pressure to increase slightly.

3. The Effect of Combustion-Chamber and of Piston-Head-Shape on the Scavenging Efficiency:

Within the limits investigated, the combustion chamber shape had no effect on the scavenging efficiency.

An improvement in the scavenging efficiency of approximately 3 percent was realized by substituting the round-top for the flat-top piston.

4. Engine Speed:

With the scavenge ratio held constant, the mean effective pressure, and hence the scavenging efficiency, varied irregularly with engine speed. This result is probably due to changes in the scavenging flow pattern with engine speed.

5. Performance:

The results show that when operating on illuminating gas as a fuel, a net brake mean effective pressure of at least 80 pounds per square inch at 12 inches of Hg and at 1,800 r.p.m. can be expected with this type of engine. This value is equivalent to 95 pounds per square inch mean effective pressure when operating on gasoline.

Compression-Ignition Operation

1. The gross brake mean effective pressures at high loads increases with the quantity of scavenging air. How much of this increase in output is due to more efficient scavenging and how much to better mixing of the fuel and air is not certain. The fact that the minimum points of the specific fuel-consumption curves occur at very low fuel-air ratios indicates, however, that there is relatively poor mixing of the fuel and the air. There is, therefore, opportunity for considerable improvement in spray distribution or combustion-chamber shape. In spite of this fact, however, it was usually possible to obtain a brake specific fuel consumption of less than 0.5 at 60 percent of maximum power.

2. The maximum gross brake mean effective pressure remained constant over a range of speeds from 1,200 to 2,000 r.p.m. at a constant scavenge ratio of 1.4. The fuel-air ratio necessary to obtain the maximum mean effective pressure increased with engine speed. These two facts would seem to indicate a nearly constant quantity of fresh air per stroke but poorer mixing of fuel and air at the higher speeds.

3. The exhaust opening for best power was earlier (11°) than for spark-ignition operation.

Massachusetts Institute of Technology,
Cambridge, Mass., May 1938.

BIBLIOGRAPHY

Jalbert, J.: Two-Stroke-Cycle Engines for Airplanes. T.M. No. 347, N.A.C.A., 1926.

Holm, Otto: Crank Case Scavenging of a Two-Stroke-Cycle Engine. T.M. No. 446, N.A.C.A., 1928.

List, Hans: Crank Case Scavenging of Two-Stroke-Cycle Engines. T.M. No. 516, N.A.C.A., 1929.

Geisse, J. H.: Low Cost is Principal Advantage of Two-Cycle Engines for Small Planes. Auto. Indus., vol. 64, no. 12, April 25, 1931, pp. 651-654.

Gasterstädt, Dr.: Development of the Junkers-Diesel Aircraft Engine. T.M. No. 565, N.A.C.A., 1930.

Caunter, C. F.: The Two-Cycle Engine. Sir Isaac Pitman & Sons, Ltd., (London), 1932.

Taylor, E. S., and Williams, G. L.: Further Investigation of Fuel Injection in an Engine Having Spark Ignition. S.A.E. Jour., vol. 30, no. 1, Jan. 1932, pp. 24-29.

Curtis, Chas. G.: Improvement in Scavenging and Supercharging Two-Cycle Diesel Engines. Diesel Power, vol. 11, no. 9, Sept. 1933, pp. 545-549.

Leonhardt, Rene: Scavenging Air Flow in Two-Cycle Diesels. Diesel Power, vol. 11, no. 7, July 1933, pp. 442-444.

Anon.: The Scavenging of Two-Cycle Engines. Sulzer Tech. Review, no. 4, 1933.

Venediger, Herbert J.: Evaluation of Scavenging in Two-Stroke-Cycle Engines. T.M. No. 732, N.A.C.A., 1934.

Venediger, Herbert J.: A Discussion of the Several Types of Two-Stroke-Cycle Engines. T.M. No. 776, N.A.C.A., 1935.

De Juhasz, K. J.: Measuring the Scavenging Efficiency of Two-Stroke Diesel Engines. Auto. Indus., vol. 73, no. 26, Dec. 28, 1935, pp. 858-861.

Earle, Sherod L., and Dutee, Francis J.: Effect of Air-Entry Angle on Performance of a 2-Stroke-Cycle Compression-Ignition Engine. T.N. No. 610, N.A.C.A., 1937.

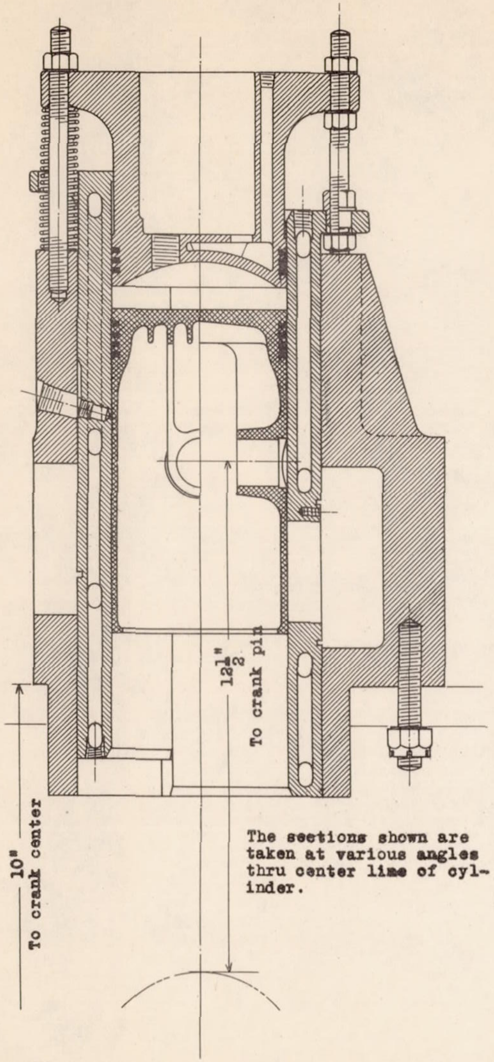


Figure 1.- Engine assembly details.

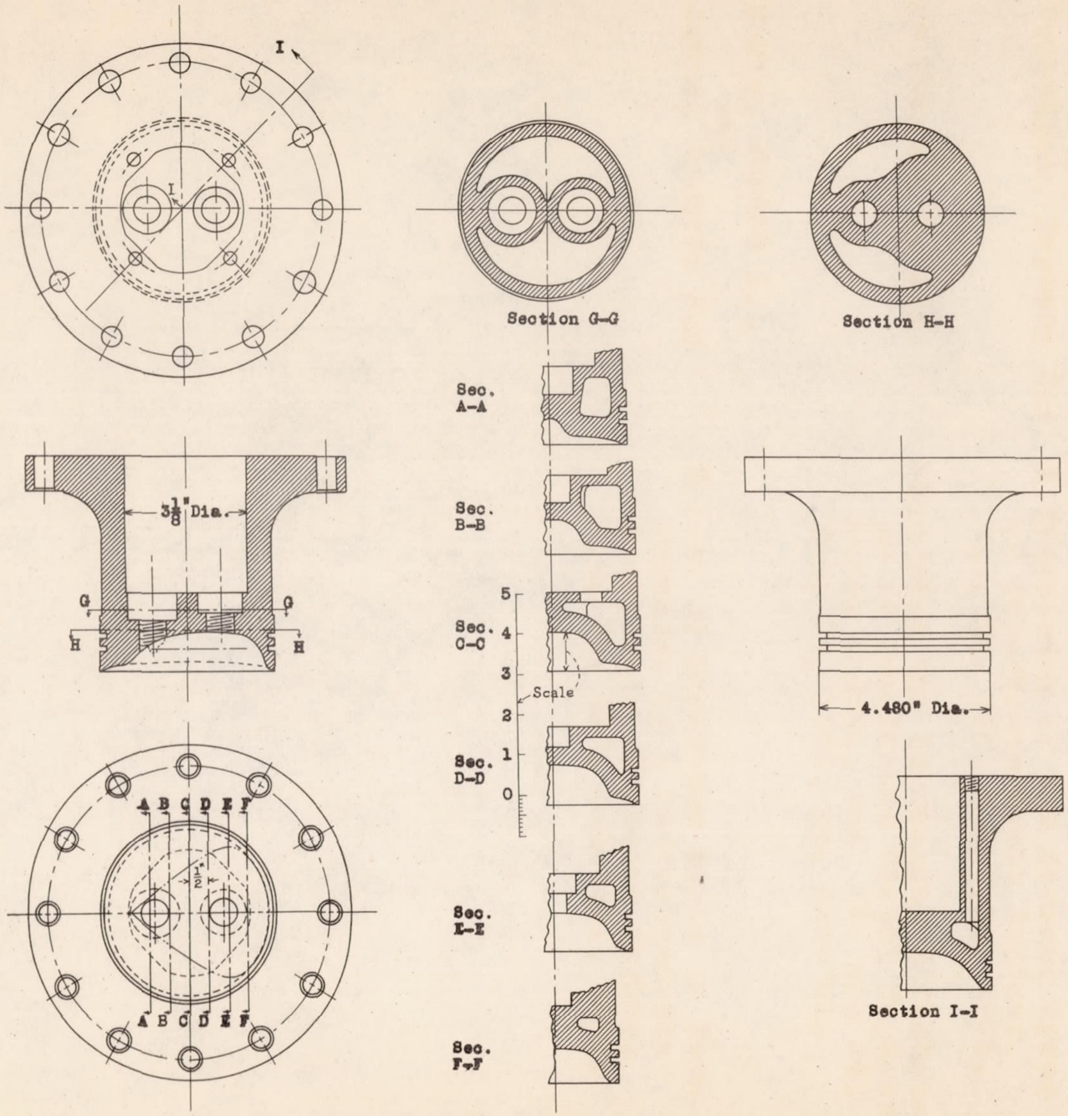


Figure 9.- Cylinder-head details (Compression-ignition).

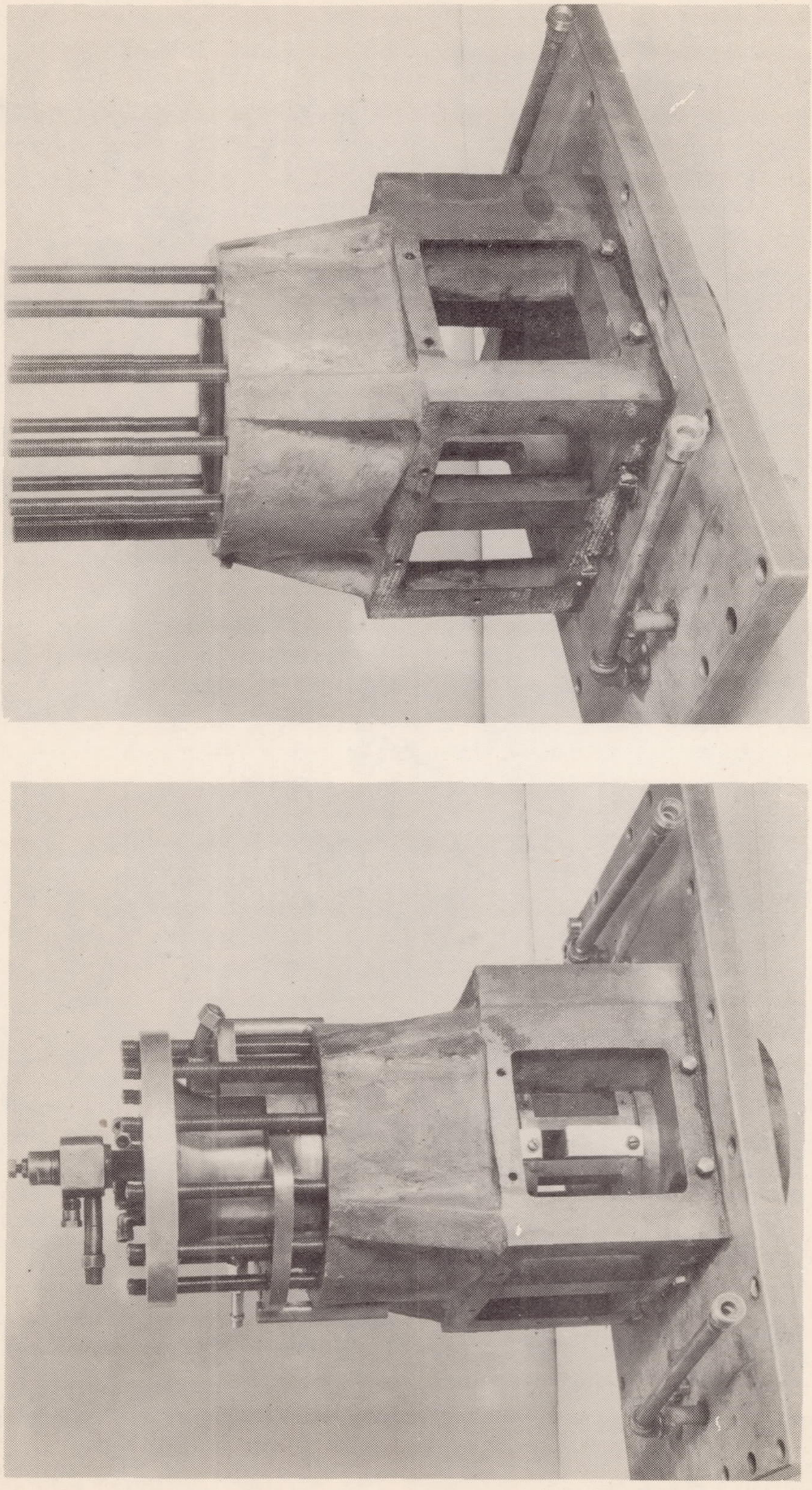


Figure 2. - Engine assembly.

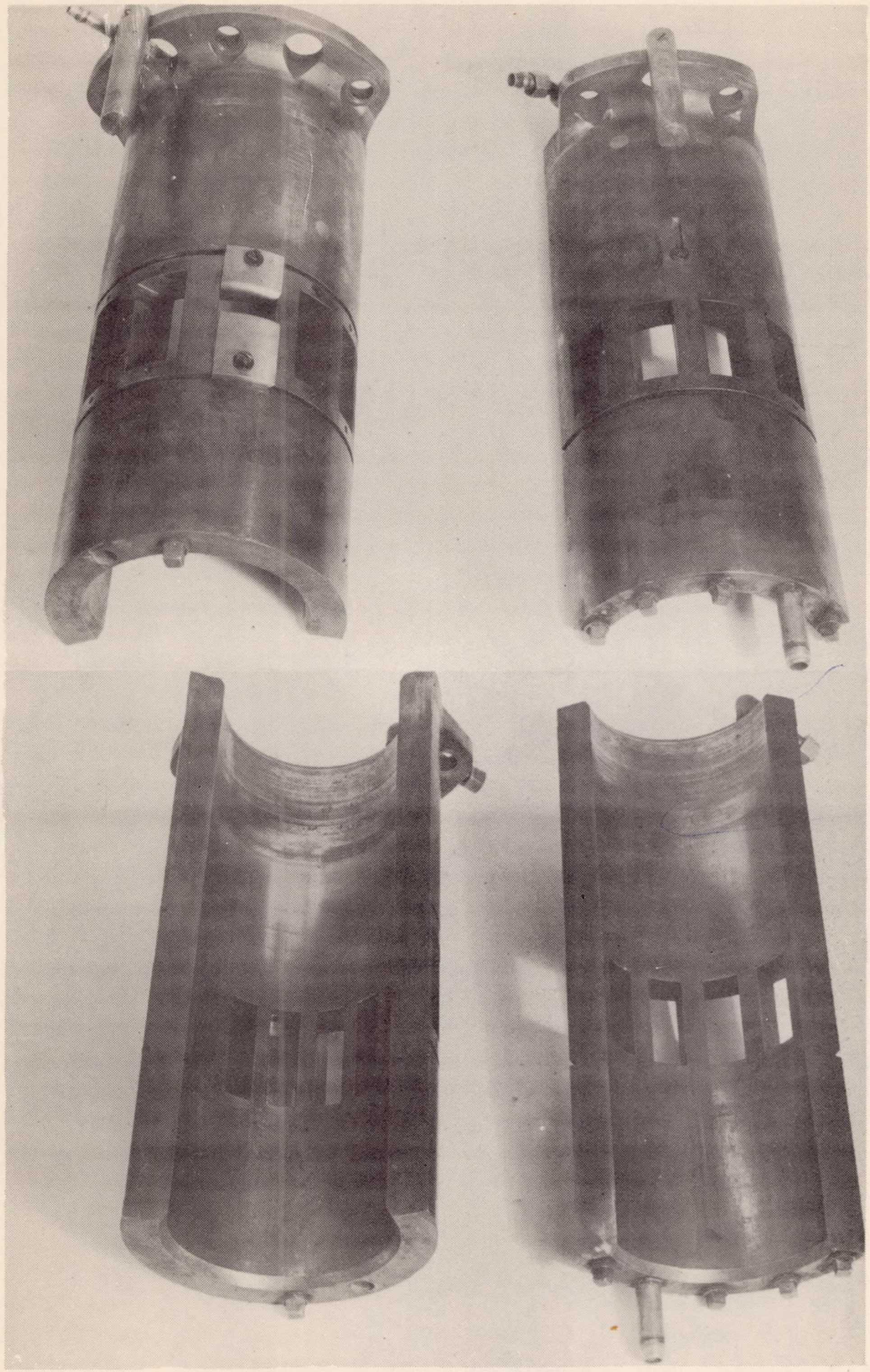


Figure 3.- Cylinder sleeves.

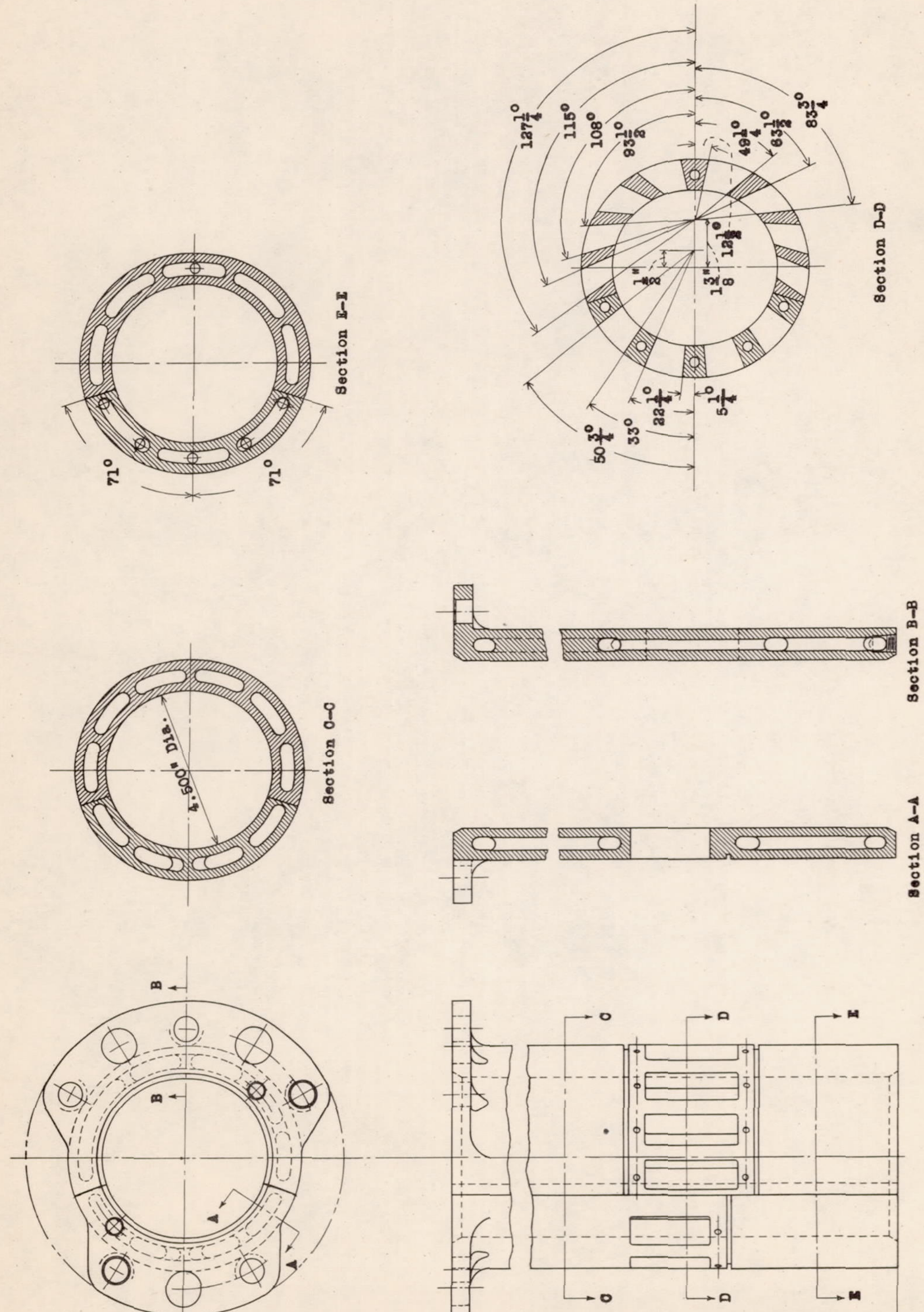


Figure 4.- Sleeve details.

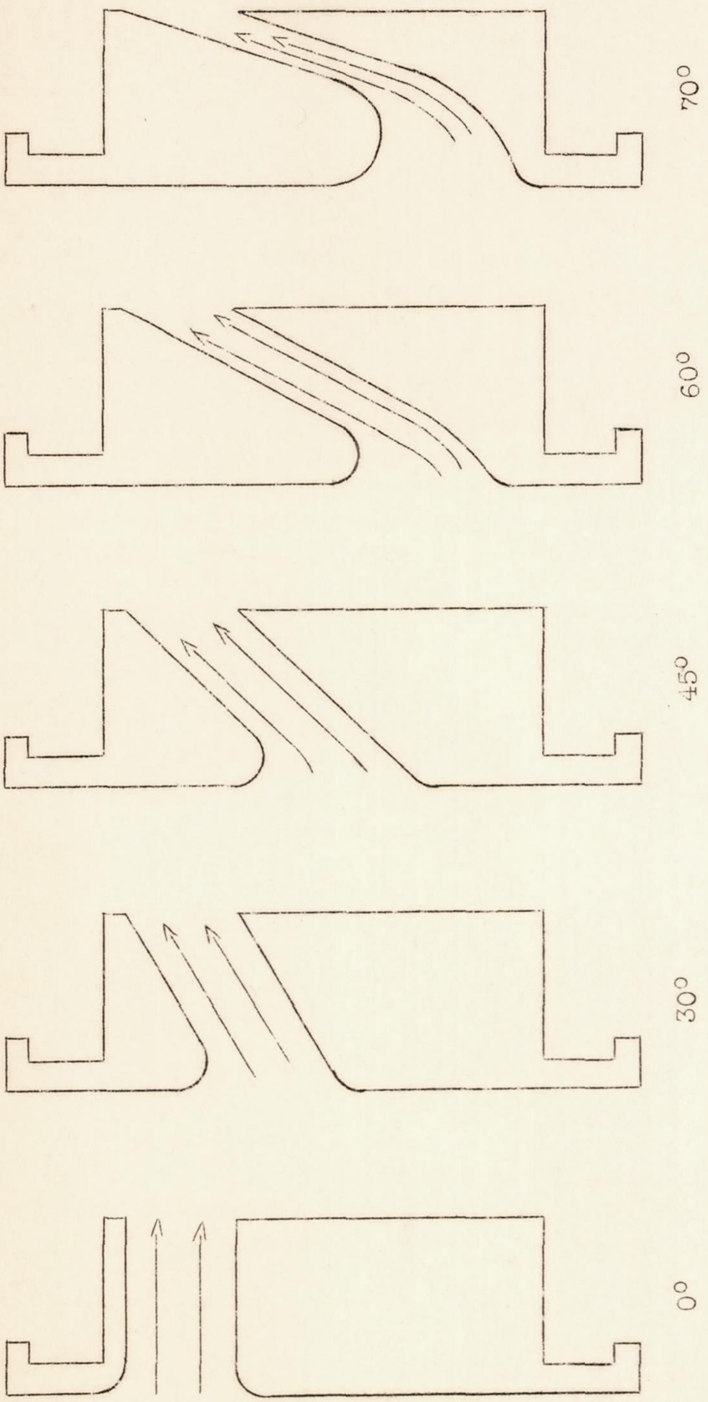


Figure 5(a) - Inlet ports.

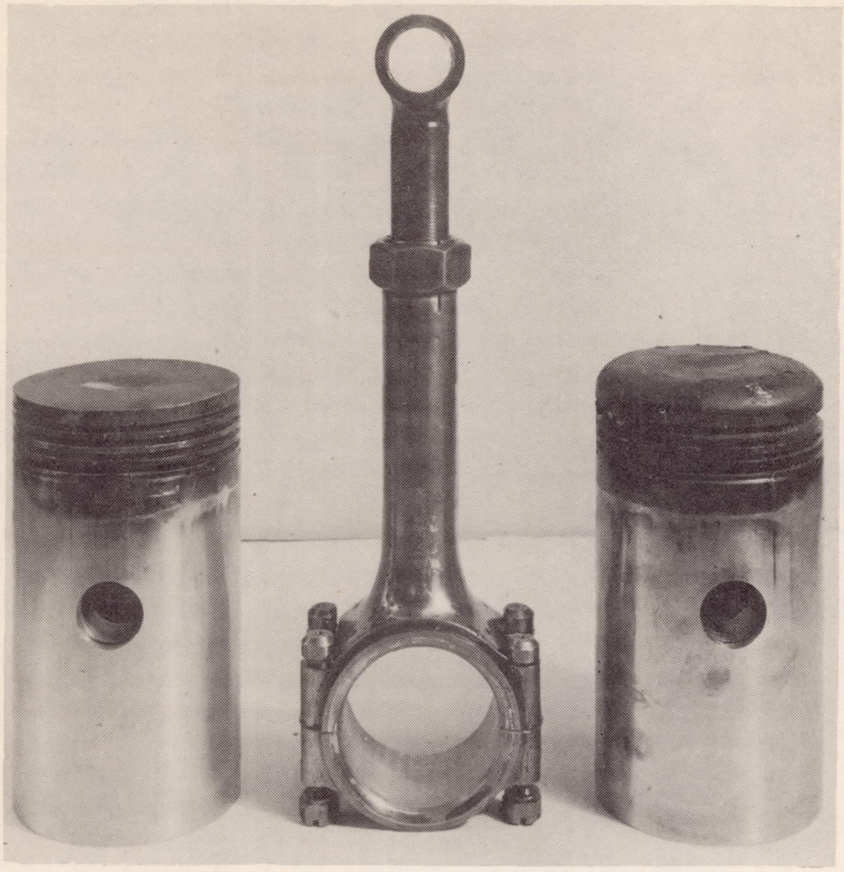
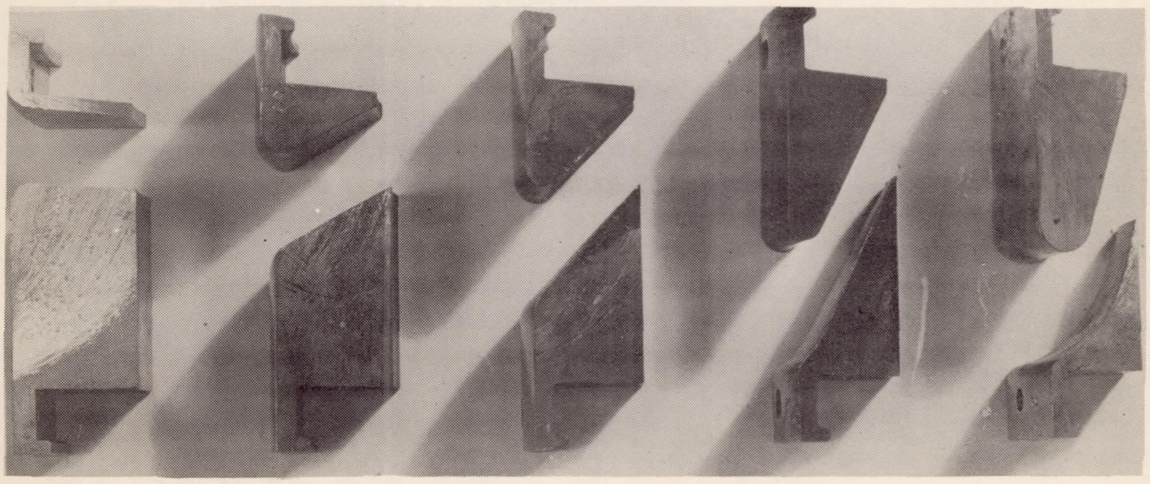


Figure 7.- Connecting rod and pistons.



0° 30° 45° 60° 70°

Figure 5(b).- Inlet ports.

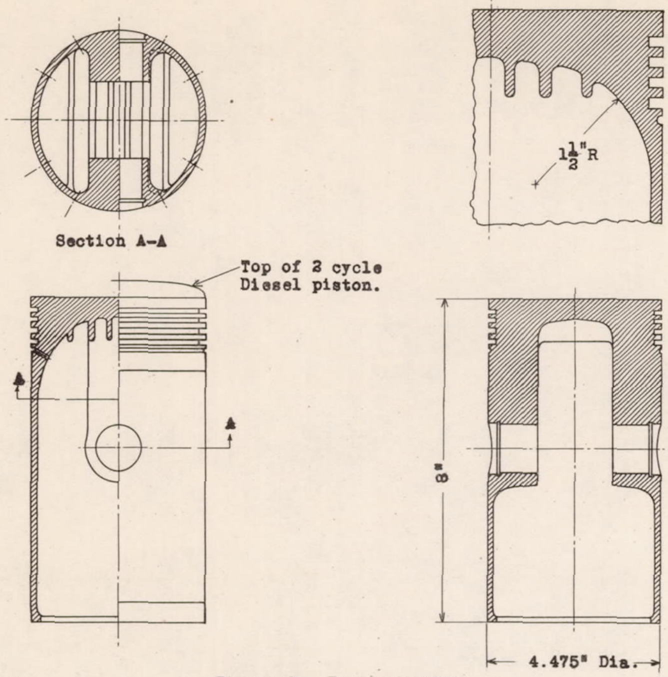


Figure 6.- Piston details.

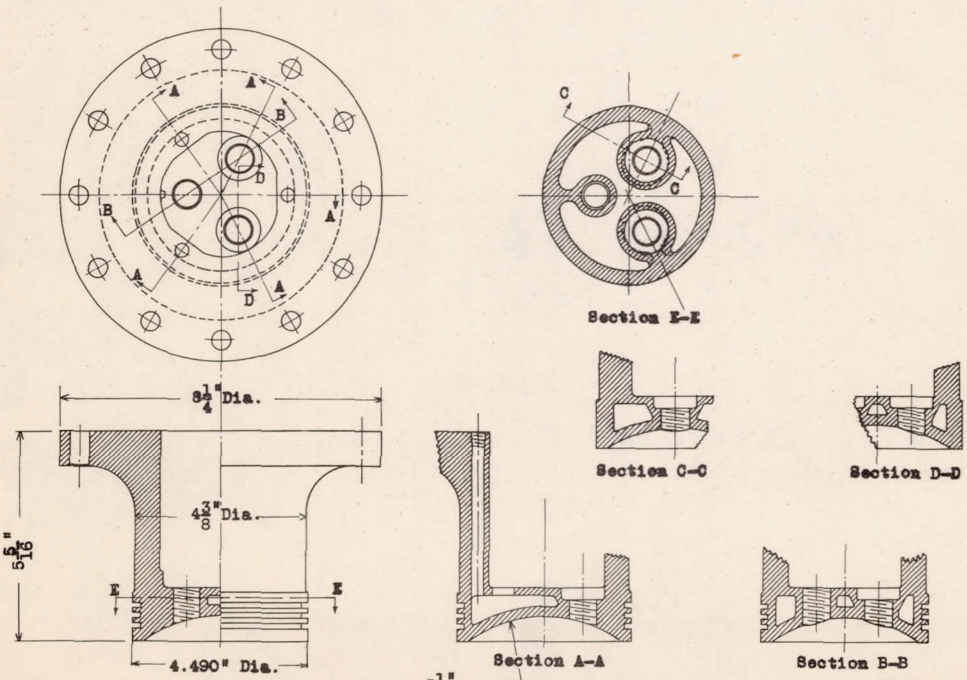


Figure 8.- Cylinder-head details (Spark-ignition).

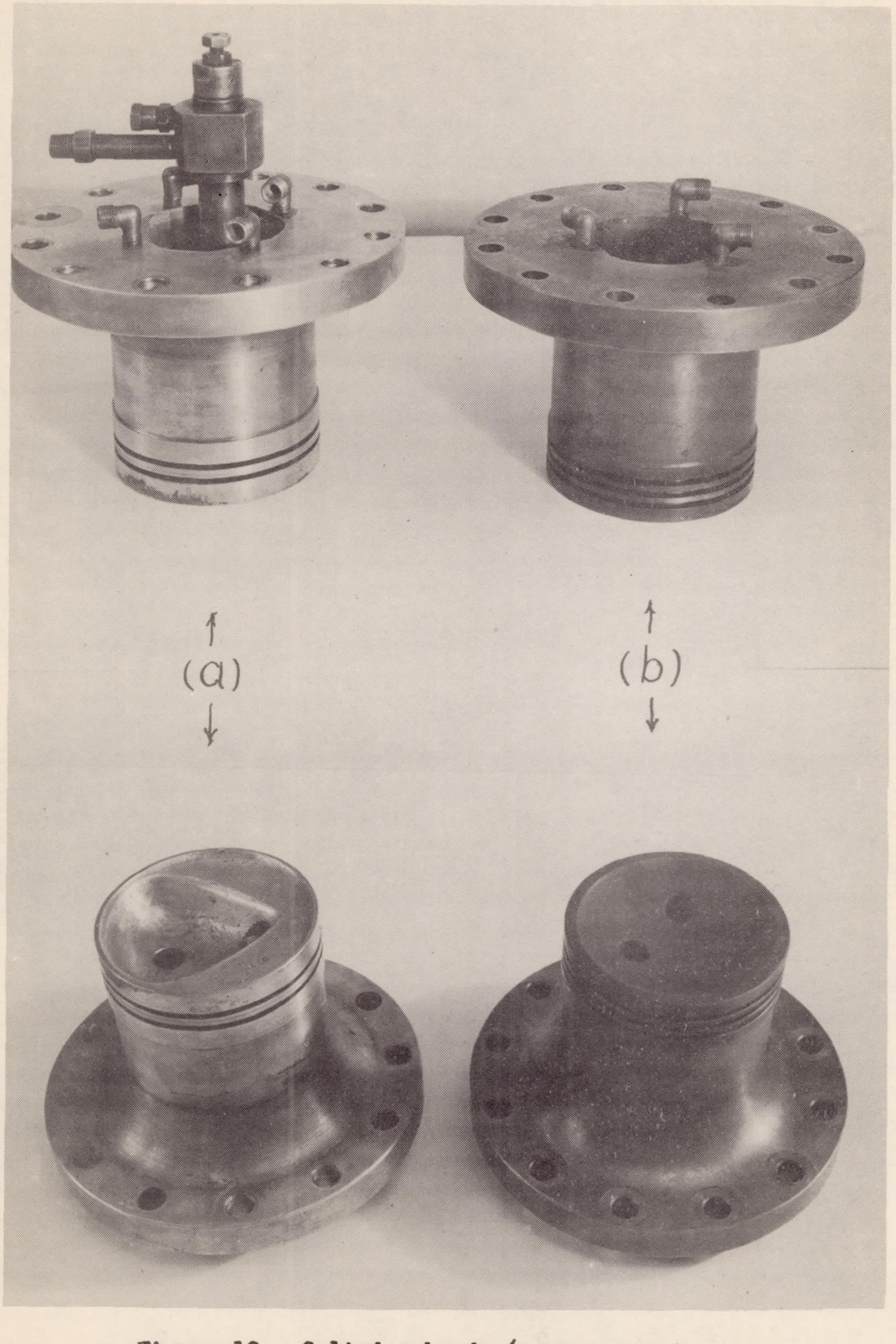


Figure 10.- Cylinder-heads (a, compression ignition, b, spark ignition).

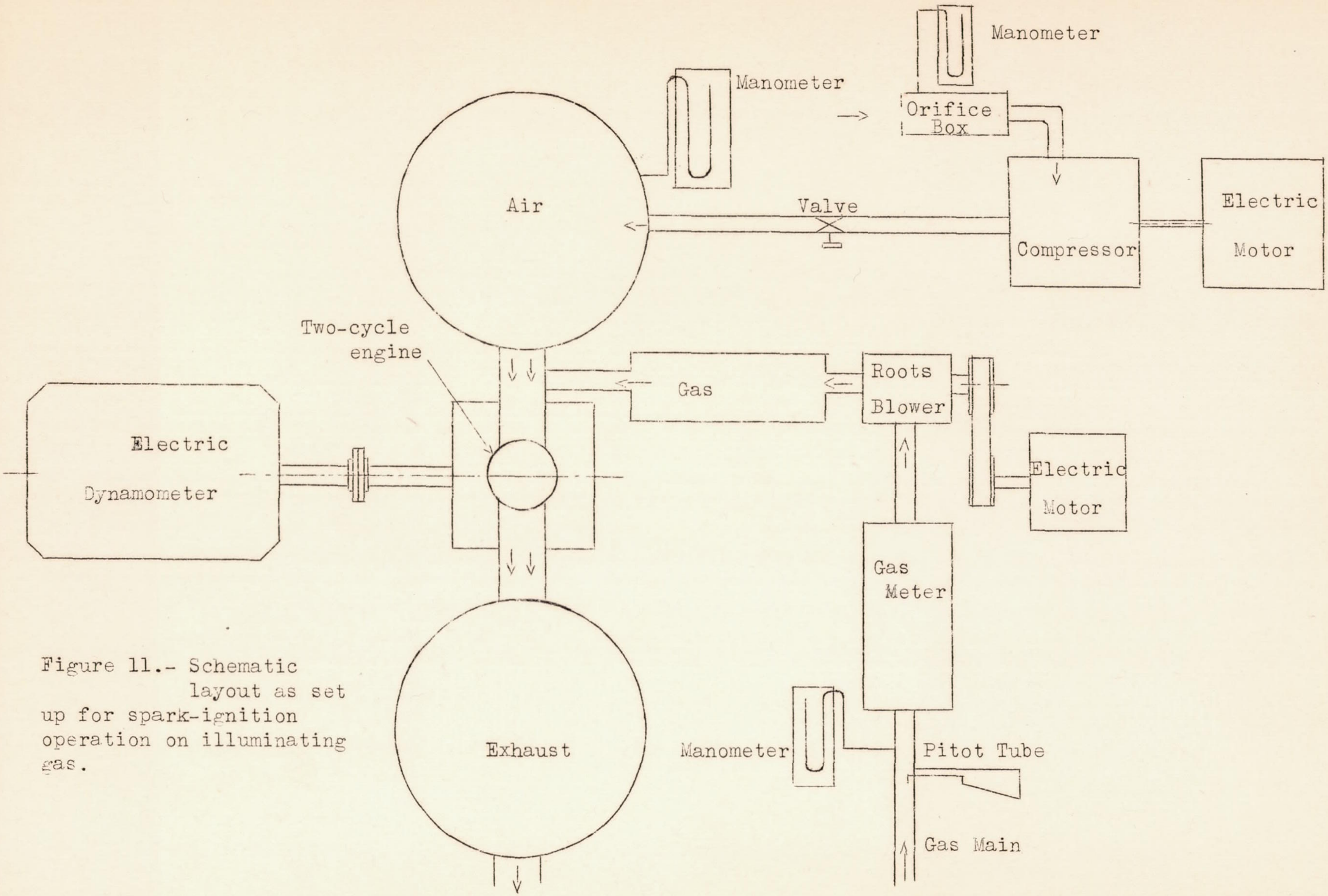


Figure 11.- Schematic layout as set up for spark-ignition operation on illuminating gas.

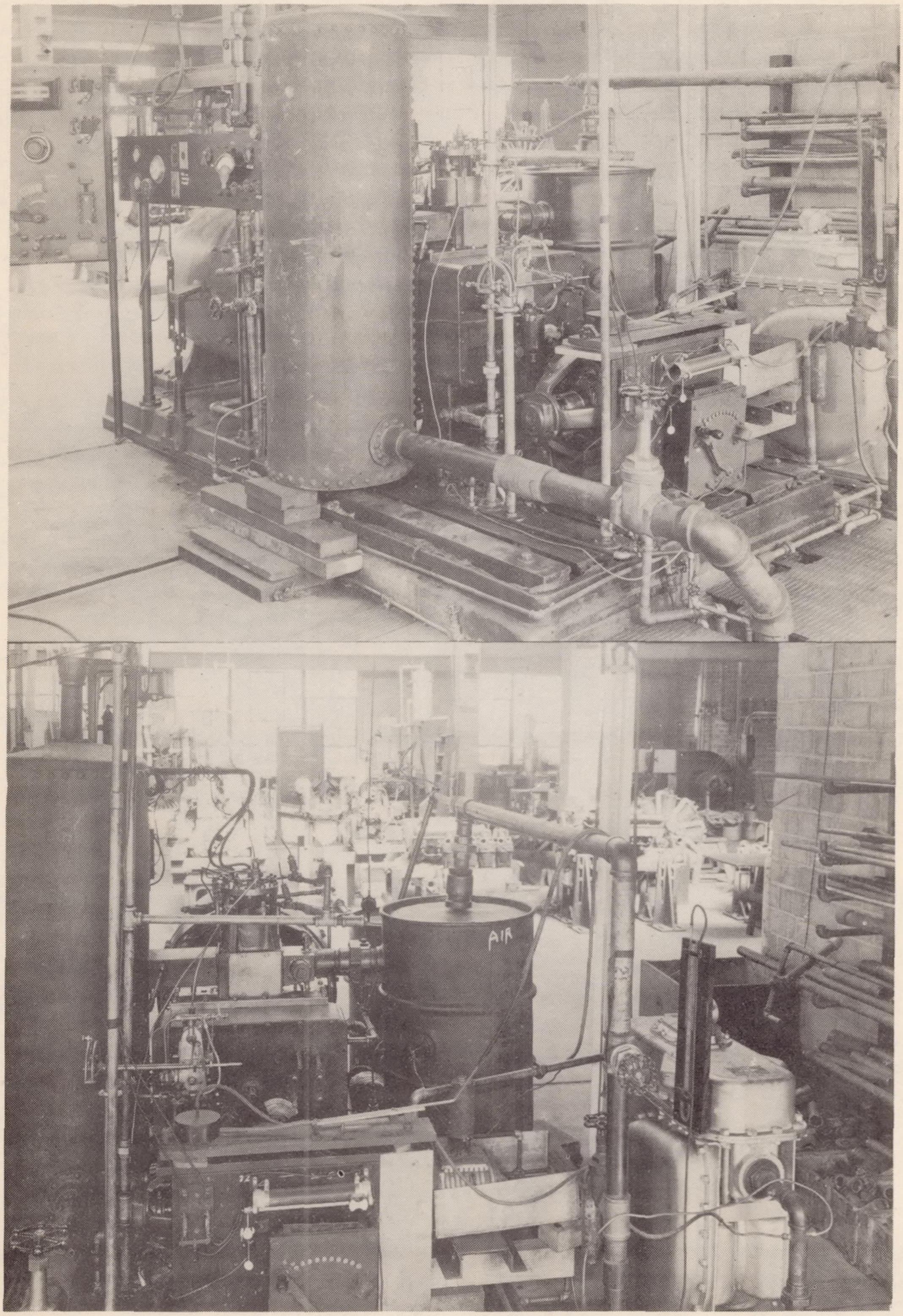


Figure 12.- Engine set-up for Diesel operation.

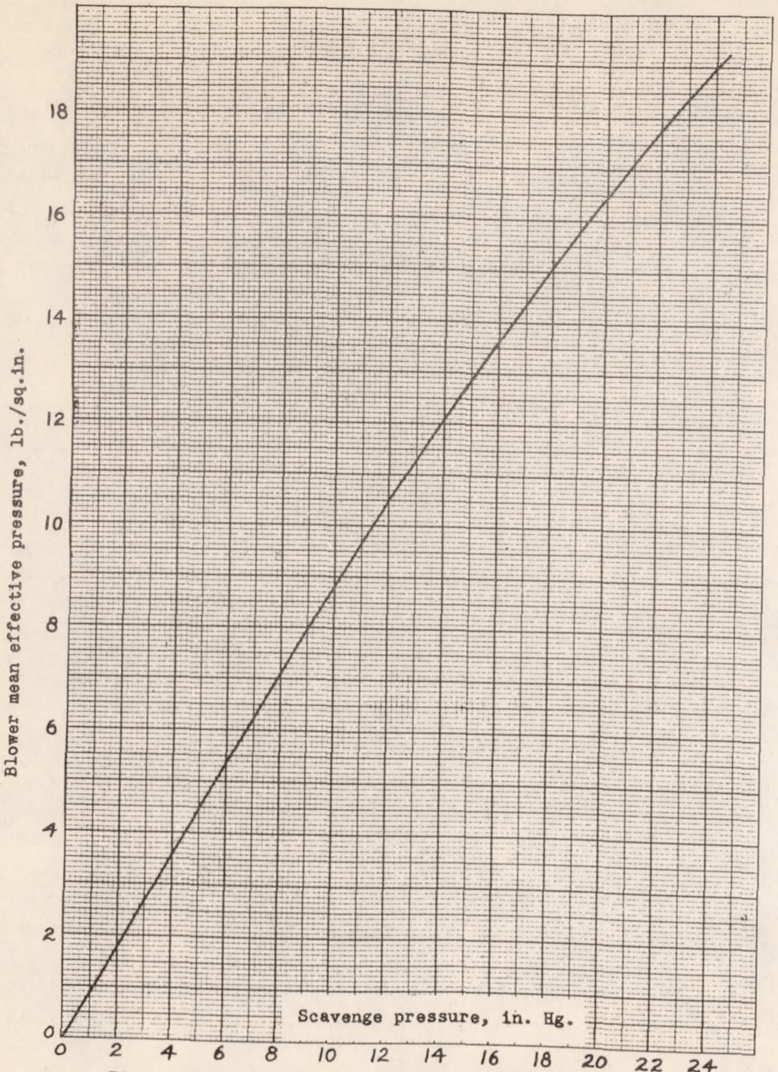


Figure 13. - Blower mean effective pressure against scavenge pressure, scavenge ratio, 1.4.

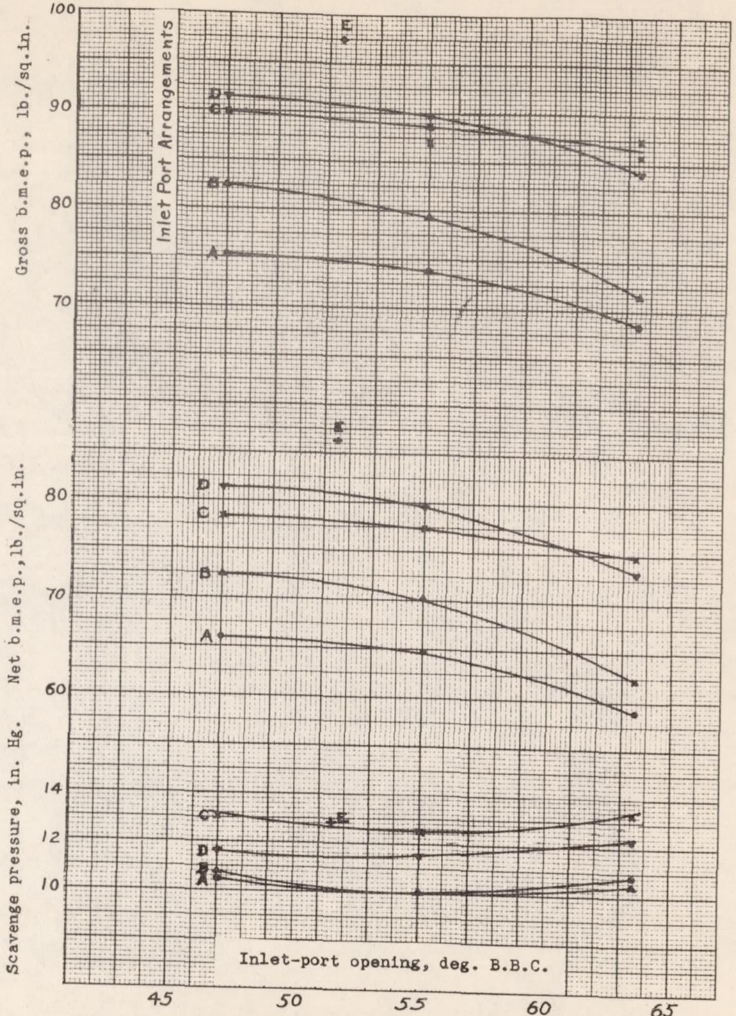


Figure 14. - Effect of inlet-port timing on scavenge pressure and brake mean effective pressure. Exhaust opens 65° B.B.C., engine speed, 1,800 r.p.m.; scavenge ratio, 1.4; spark-ignition; illuminating gas.

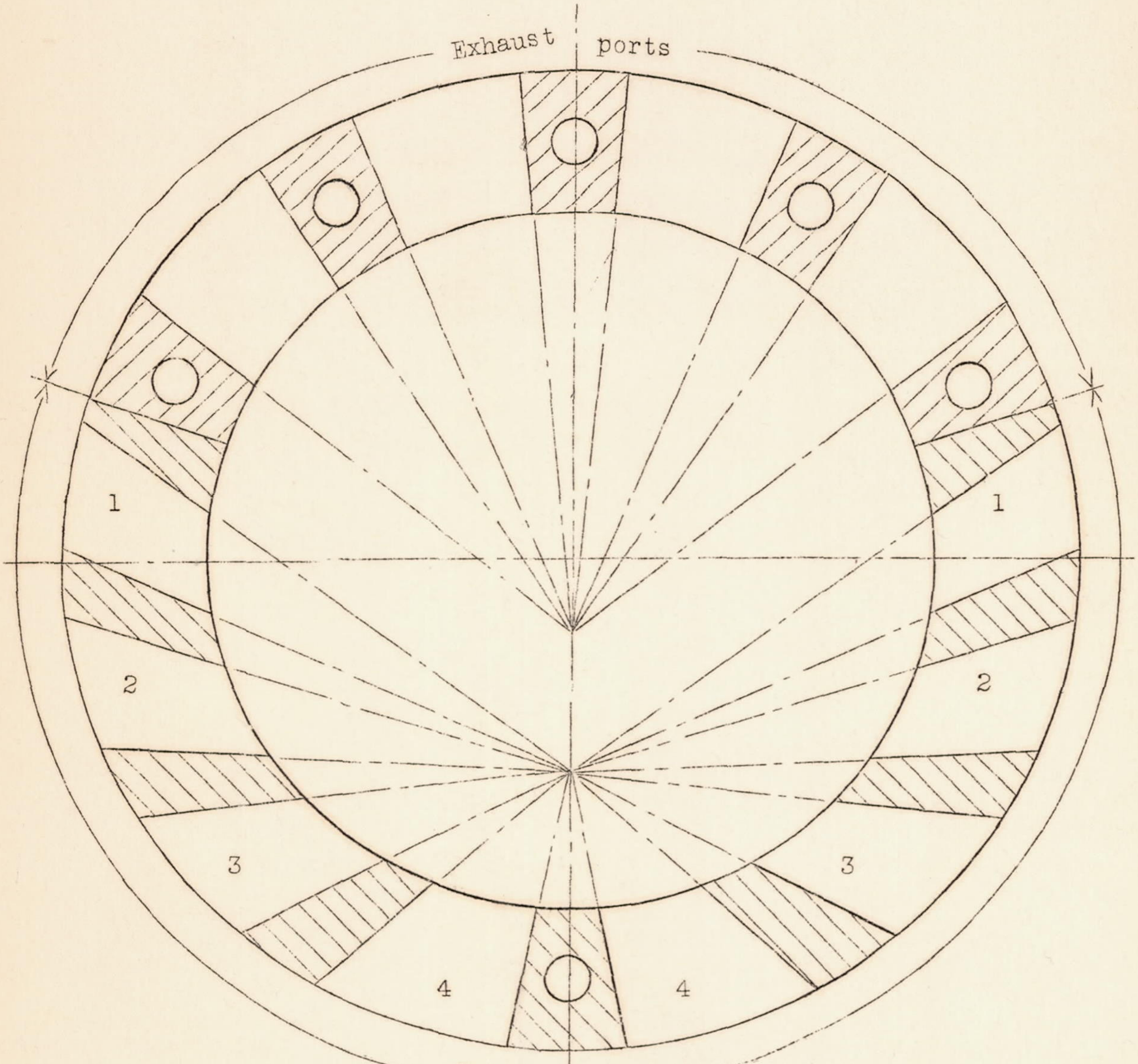


Figure 15.- Inlet-port numbering scheme.

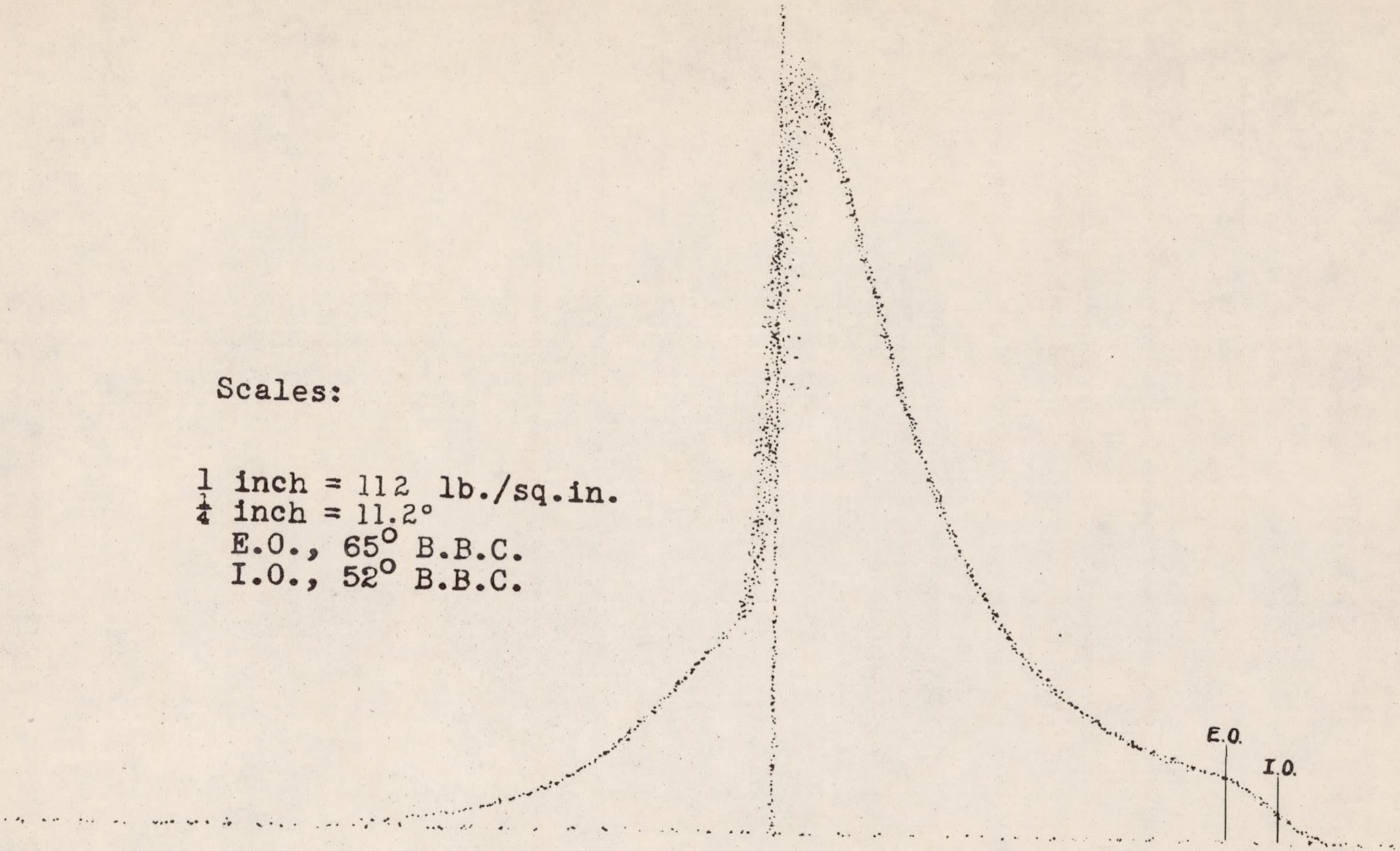


Figure 16. - Indicator diagram for best power under spark-ignition conditions.

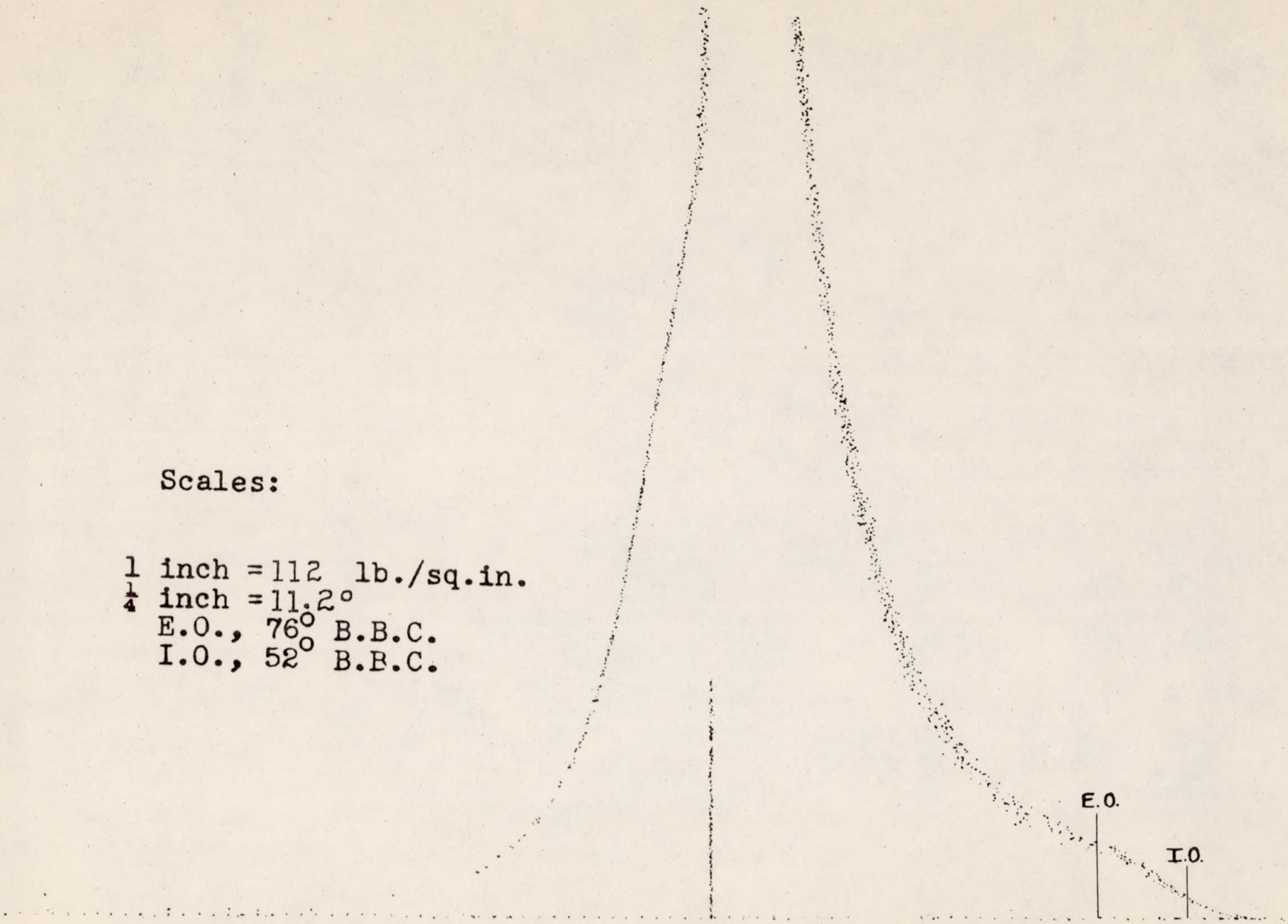


Figure 17.- Indicator diagram for best power under compression-ignition conditions.

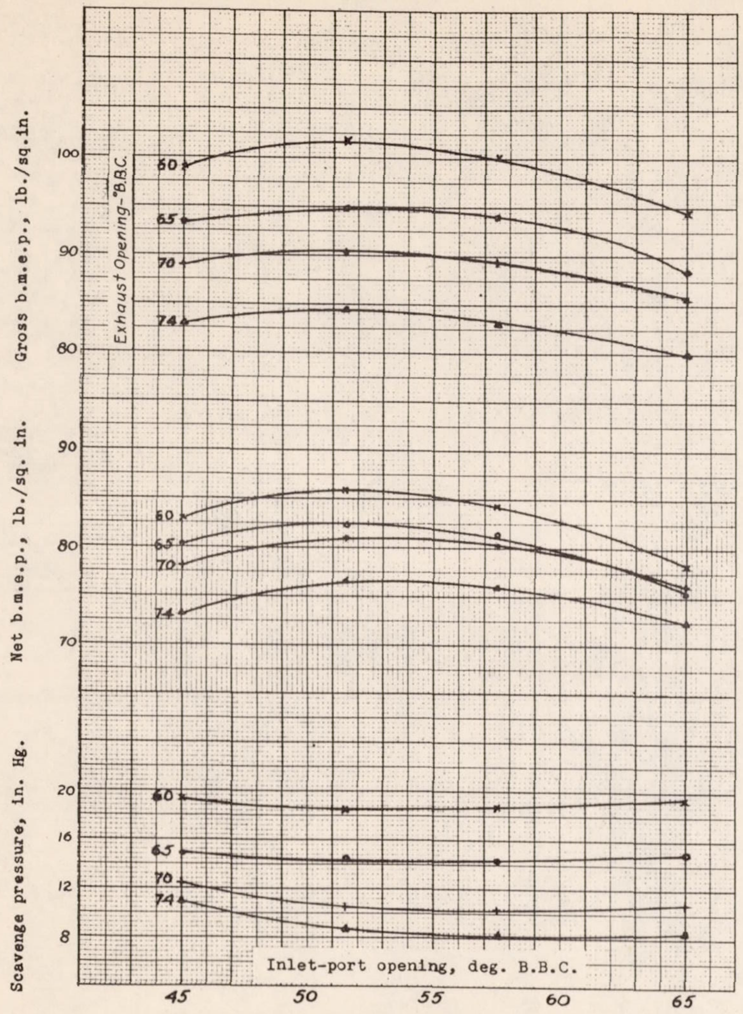


Figure 18. - The effect of inlet-port opening on scavenge pressure and brake mean effective pressure. Port arrangement, D; engine speed, 1,800 r.p.m.; scavenge ratio, 1.4; spark-ignition; illuminating gas.

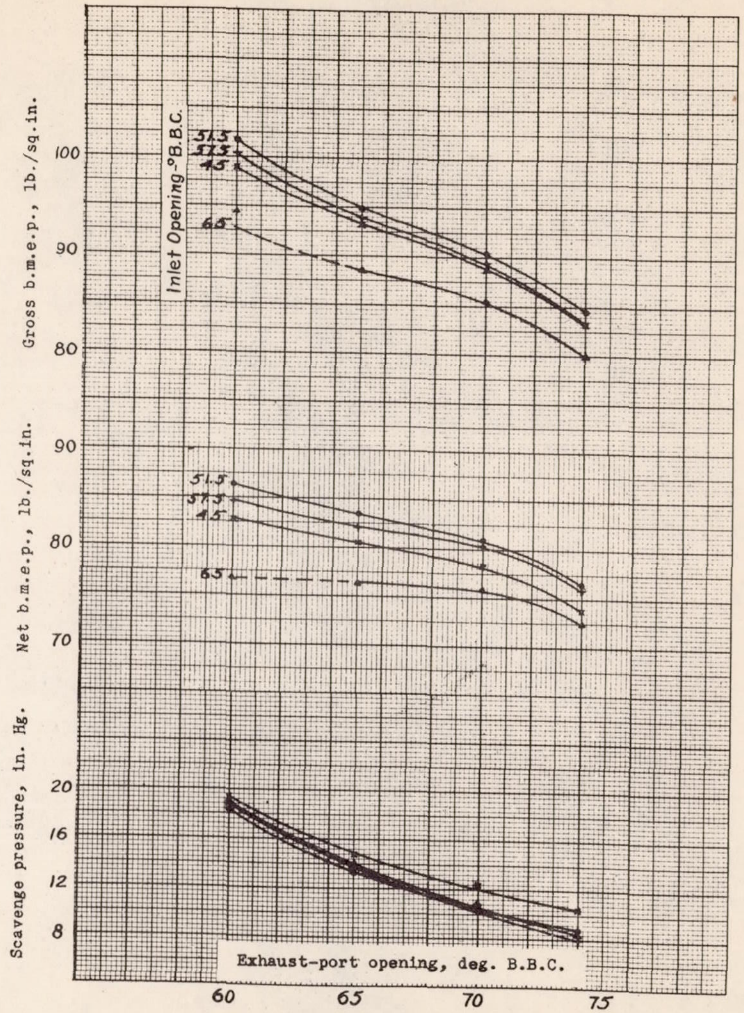


Figure 19. - The effect of exhaust-port opening on scavenge pressure and brake mean effective pressure. Port arrangement, D; engine speed, 1,800 r.p.m.; scavenge ratio, 1.4; spark ignition; illuminating gas.

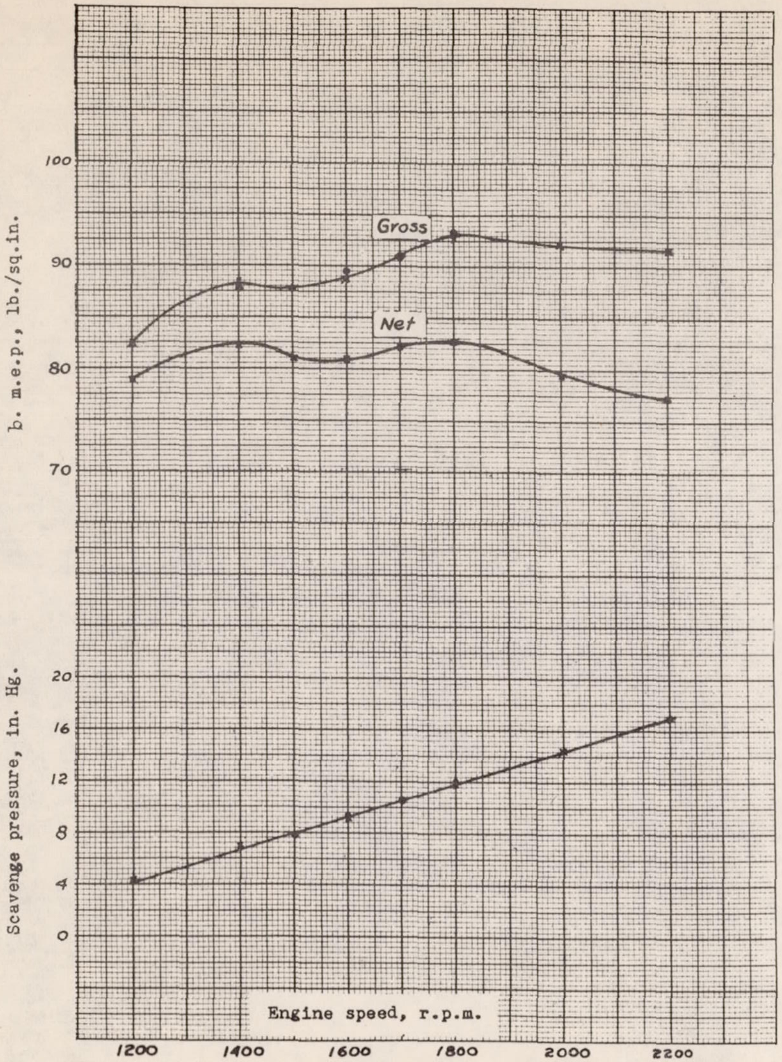


Figure 20.- The effect of engine speed on scavenge pressure and brake mean effective pressure. Port arrangement, D; inlet-port opening, 47° B.B.C.; exhaust-port opening, 65° B.B.C.; scavenge ratio, 1.4; spark-ignition; illuminating gas.

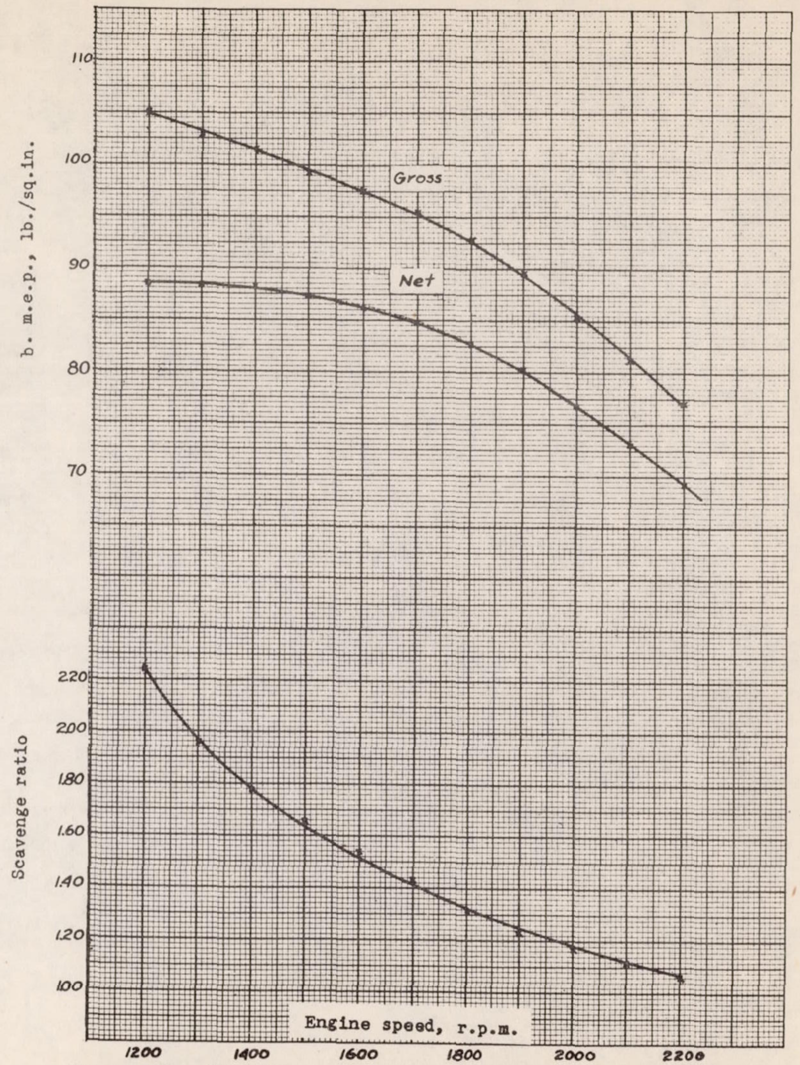


Figure 21.- The effect of engine speed on scavenge ratio and brake mean effective pressure. Port arrangement, D; inlet-port opening, 47° B.B.C.; exhaust-port opening, 65° B.B.C.; scavenge pressure, 11.8 inches Hg.; spark-ignition; illuminating gas.

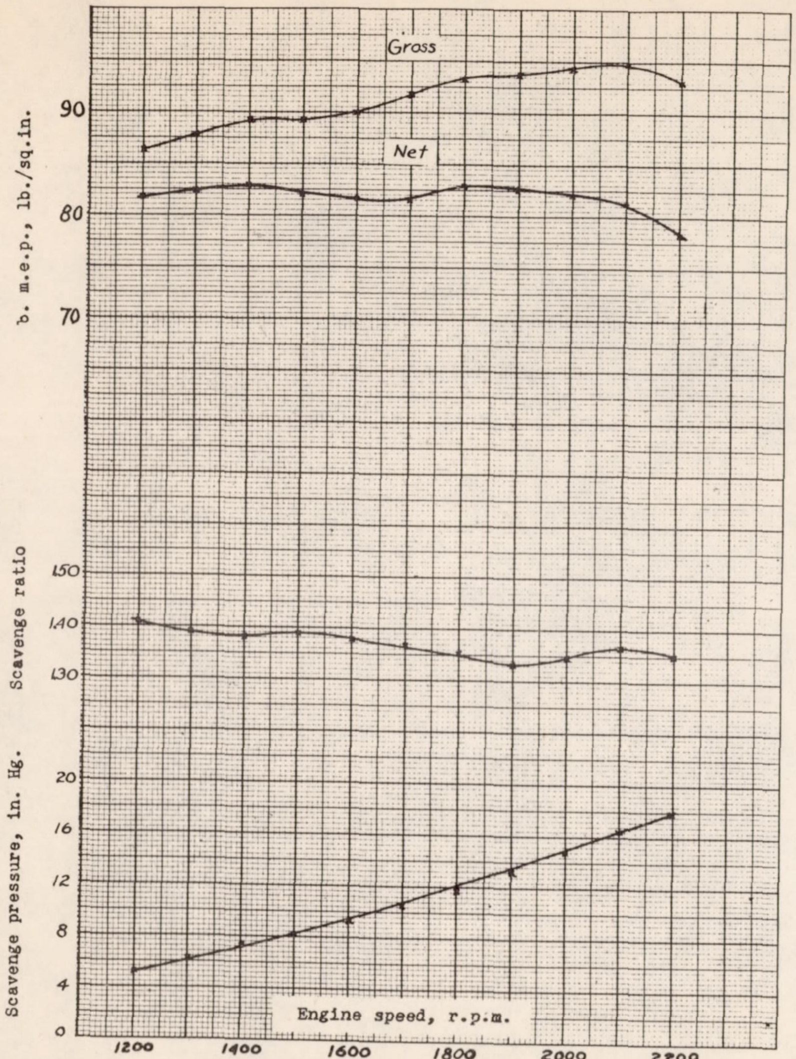


Figure 22.- The effect of engine speed on scavenge pressure, brake mean effective pressure and scavenge ratio. Port arrangement, D; inlet-port opening, 47° B.B.C.; exhaust-port opening, 65° B.B.C., scavenge pressure proportional to (r.p.m.)^2; spark-ignition; illuminating gas.

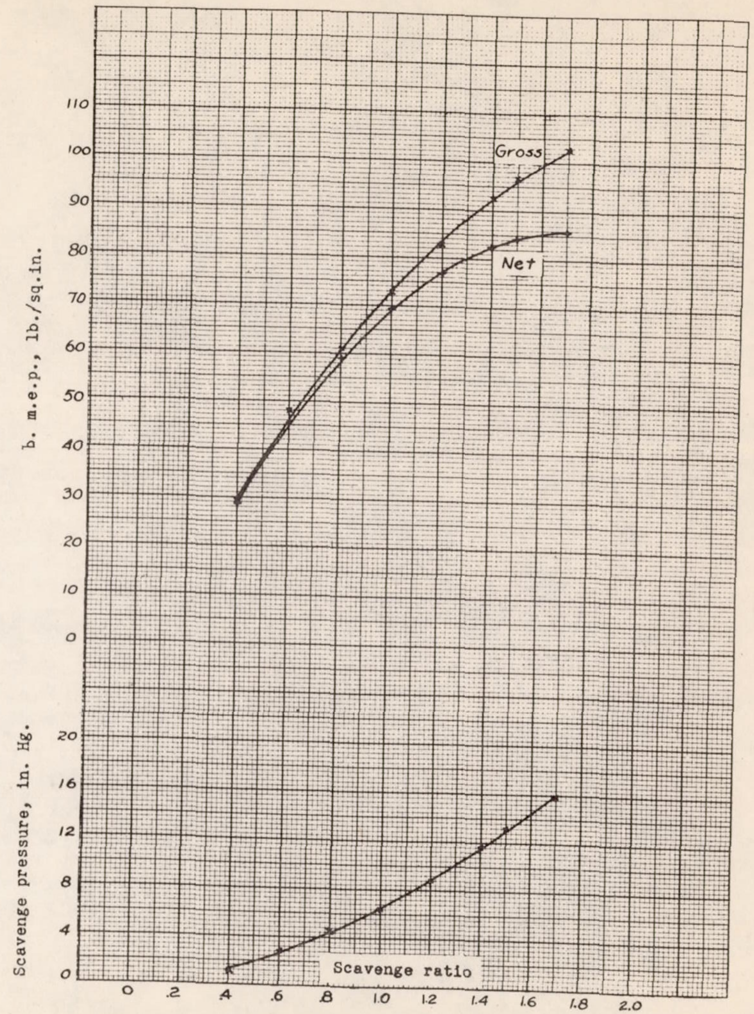


Figure 23. - The effect of scavenge ratio on scavenge pressure and brake mean effective pressure. Port arrangement, D; inlet-port opening, 47° B.B.C.; exhaust-port opening, 65° B.B.C.; engine speed, 1,800 r.p.m.; spark-ignition; illuminating gas.

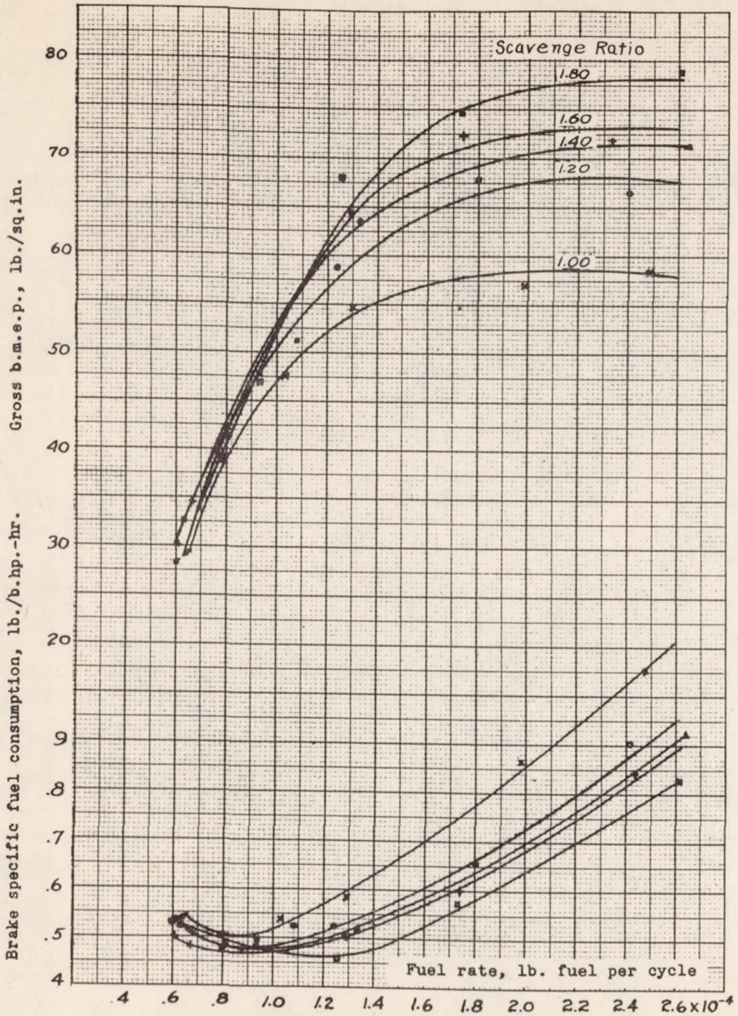


Figure 24. - The effect of fuel rate on brake specific fuel consumption and brake mean effective pressure. Port arrangement, E; inlet-port opening, 55° B.B.C.; exhaust-port opening, 76° B.B.C.; engine speed, 1,800 r.p.m.; compression-ignition.

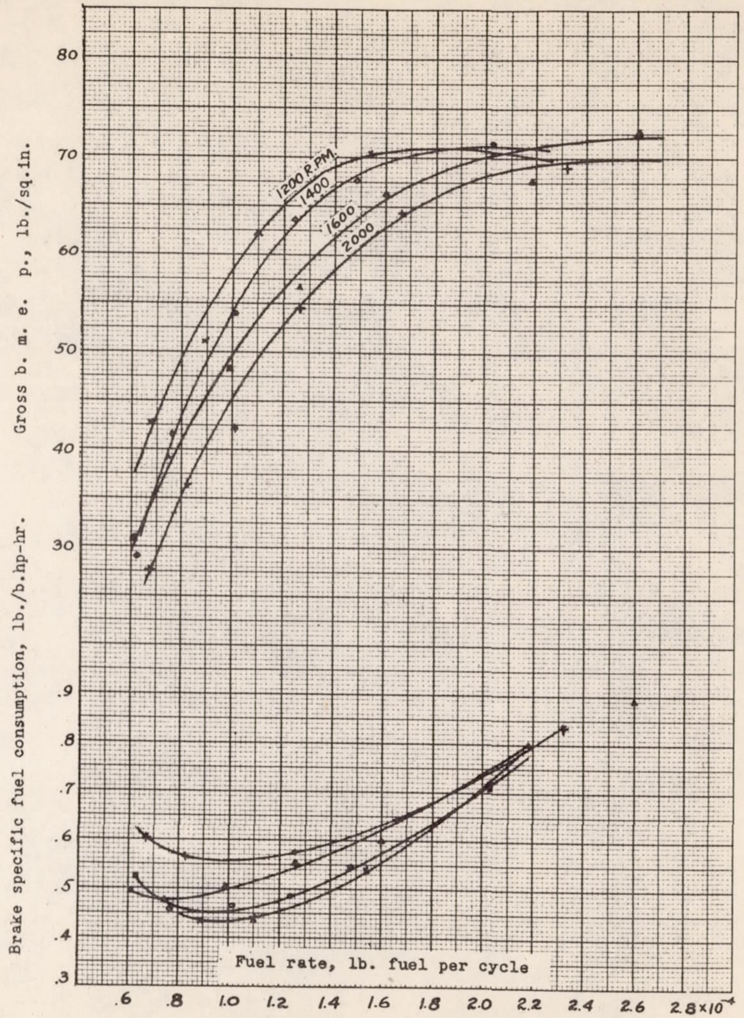


Figure 25. - The effect of fuel rate on brake specific fuel consumption and brake mean effective pressure. Port arrangement, E; inlet-port opening, 55° B.B.C.; exhaust-port opening 76° B.B.C.; scavenge ratio, 1.4; compression-ignition.

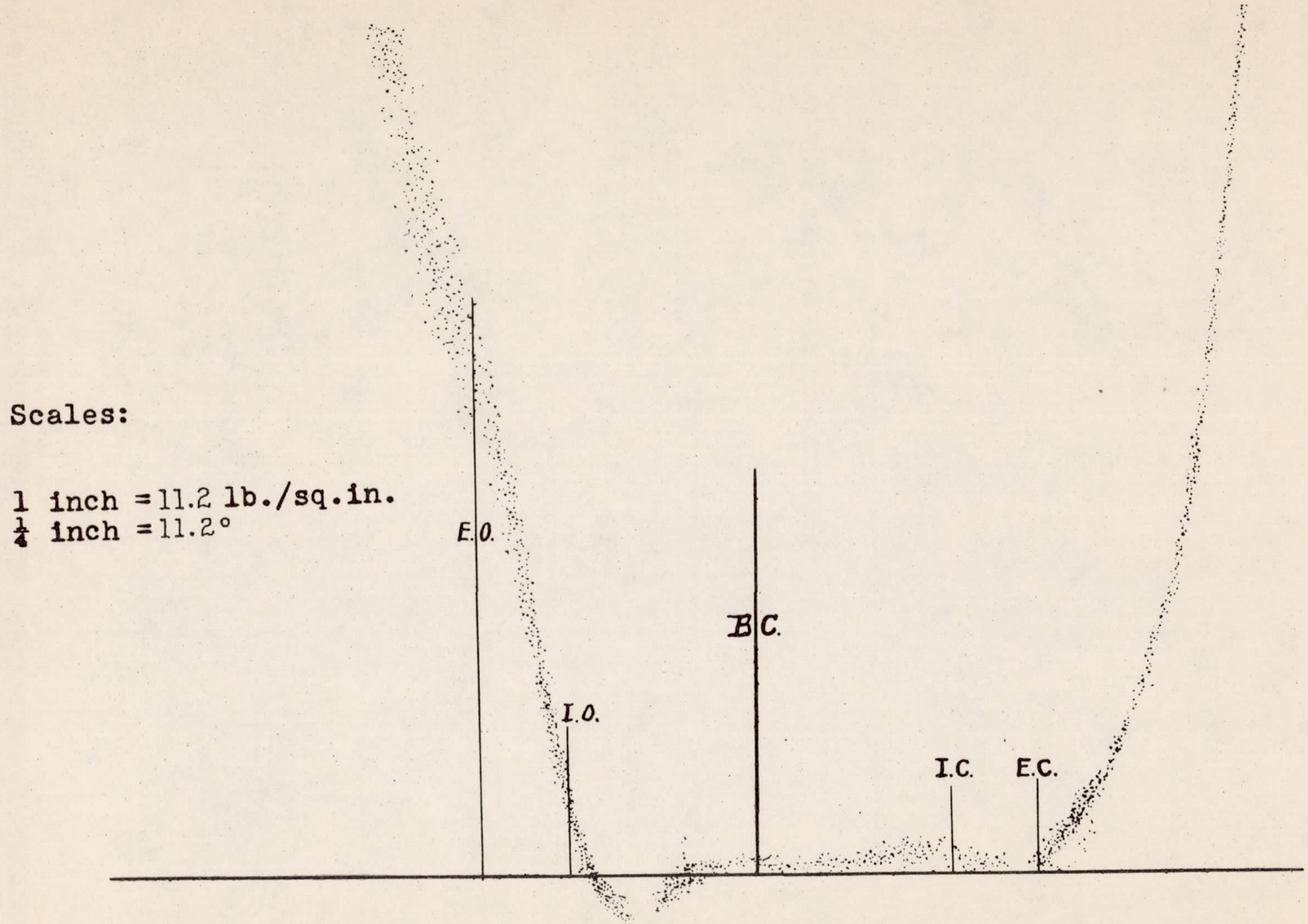


Figure 26.- Light spring indicator diagram for exhaust-port opening of 76° B.B.C. and inlet-port opening of 52° B.B.C. Compression-ignition; engine speed, 1,800 r.p.m.

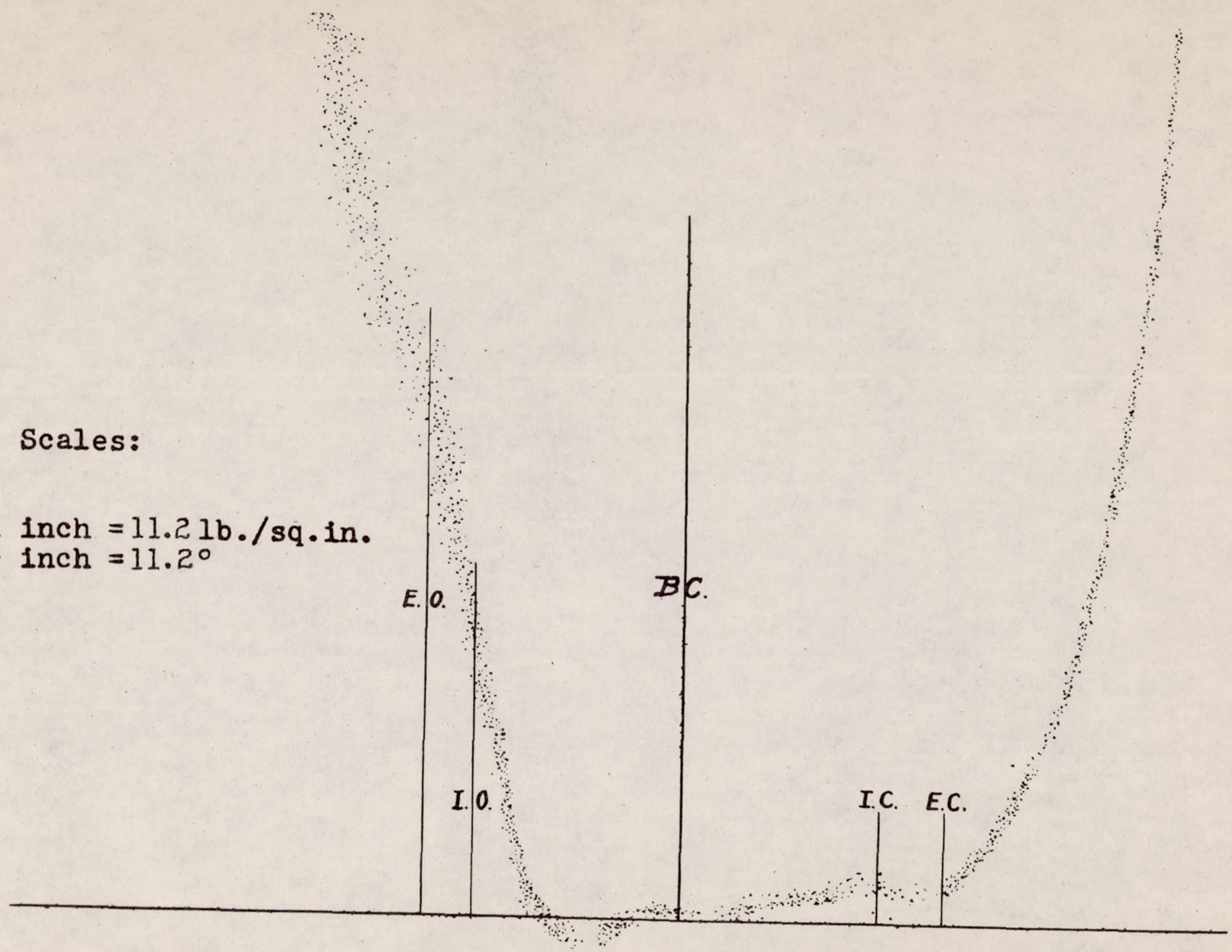


Figure 27.- Light spring indicator diagram for exhaust-port of 70° B.B.C. and inlet-port opening of 52° B.B.C. Compression-ignition; engine speed, 1,800 r.p.m.