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EFFECT OF CHANGING THE STROKE ON AIR CAPACITY, POWER OUTPUT, AND DETONATION OF A SINGLE-CYLINDER ENGINE

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EFFECT OF CHANGING THE STROKE ON AIR CAPACITY, POWER OUTPUT, AND DETONATION OF A SINGLE-CYLINDER ENGINE

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SUMMARY

For this investigation a special single-cylinder test engine was constructed with provision for operation with three different lengths of piston stroke. The cylinder diameter was constant and equal to the intermediate length of piston stroke. The compression ratio and ratio of crank radius to connecting rod length were held constant. Performance tests were made over a wide range of piston speeds and inlet pressures. Detonation limits were established for increasing inlet pressures.

It was found that the air capacity and volumetric efficiency of the engine were independent of the stroke for any particular piston speed at any particular inlet pressure within the range investigated. This result might be altered if dynamic disturbances develop in the inlet or exhaust systems.

With the particular cylinder and operating conditions used, and at one fixed piston speed, the engine could be more highly supercharged without detonation when the shorter strokes were employed.

The ratio of indicated power output to air capacity over a wide range of air capacities did not show any significant differences for the three strokes.

Brake mean effective pressure at a constant inlet pressure was found to be progressively lower with the shorter piston strokes for any particular piston speed within the range covered. This is attributed to the greater mechanical

friction losses accompanying the higher engine revolutions per minute of the shorter strokes. Calculated friction losses in the main journal and crankpin bearings accounted in part for the differences noted.

The differences in brake mean effective pressure with the three strokes were found to be substantially independent of the inlet pressure when supercharging at one fixed piston speed.

No differences in pumping losses were noted between the three strokes at any piston speed.

INTRODUCTION

Bore-stroke ratio for many years has been a subject of controversy among engine designers. One of the important elements of this controversy has been the question whether two engines of the same cylinder bore, but of differing strokes, would have the same air capacity, efficiency, power output, and detonation limit when operated under comparable conditions.

The application of dimensional analysis to engines of geometrically similar design, but of differing size, shows that the indicated power output would be expected to be proportional to the square of a characteristic length, provided the piston speed is kept constant. When such a result is interpreted as meaning that the indicated power output is proportional to the square of the bore, for constant piston speed, it must be remembered that the bore and the stroke bear a definite relationship because of the geometric similarity. For engines in which geometric similarity is not maintained because of a variation in bore-stroke ratio, dimensional analysis reveals that the indicated power output at constant piston speed may be dependent on the bore-stroke ratio as one of the design factors. Since the influence of bore-stroke ratio on engine performance is not easy to predict on theoretical grounds, it is best determined by experiment.

In reference 1 a report was made of a previous investigation at the Massachusetts Institute of Technology of the effect of inlet valve design, size, and lift on the volumetric efficiency of an engine. In that report it was shown that, with short inlet and exhaust pipes, the volumetric efficiency of the engine investigated could be expressed as a function of the parameter,

$$\frac{\text{piston speed} \times \text{piston area}}{\text{inlet air sound velocity} \times \text{inlet valve area} \times \text{average valve flow coefficient}}$$

provided other operating conditions and other design ratios were kept constant. It may be expected that the volumetric efficiency of any engine with short inlet and exhaust pipes will likewise be a function of the same parameter, although not necessarily the same function. Effects due to inlet and exhaust pipes of various lengths are most conveniently handled as a separate problem.

If the length of stroke of an engine is changed, the revolutions per minute associated with a given value of piston speed changes in inverse proportion. This change introduces a variation in the frequency of the pulsations which exist in the air flow. This varying departure from the conditions of steady flow (assumed in the derivation of the parameter above) might be suspected to affect:

(1) The actual dynamic flow coefficient of the inlet valve

(2) The magnitude of the inertia effects in the valve port, cylinder, inlet and exhaust pipes during the charging process

The first of these possibilities was investigated by Waldron (reference 2), who found that dynamic flow coefficients with pulsating flow agreed well with the steady flow values. The second effect was shown to be of negligible importance in the report of Reynolds, Schecter, and Taylor (reference 3), provided very short inlet and exhaust pipes were used. For these reasons it would appear that the length of stroke of an engine would not materially affect the air capacity if the piston speed is kept the same and if large changes in the dynamics of the inlet and exhaust gas columns are avoided by using very short inlet and exhaust pipes.

The effect of changing the length of stroke on combustion efficiency is difficult to predict since there are changes in the combustion chamber size and shape as well as in the engine revolutions per minute.

With regard to detonation, if the stroke of an engine is decreased and compression ratio kept constant, the size of the combustion chamber is decreased with the resulting possibility that the tendency to detonate will be decreased. The fact that the revolutions per minute for a given piston speed is increased with decreasing stroke may further decrease the likelihood of detonation.

This investigation was carried out in order to provide experimental answers to those uncertainties connected with the effects of changing the length of stroke. The specific objectives were:

(1) To determine the effect of changing length of stroke on the air capacity, efficiency, and power output of a four-stroke engine

(2) To determine the effect of changing length of stroke on the detonation tendencies of the engine

With such information, engine designers will be able to evaluate the effects of bore-stroke ratio on performance and other characteristics of a projected design.

This investigation, conducted at the Massachusetts Institute of Technology, was sponsored by and conducted with the financial assistance of the National Advisory Committee for Aeronautics.

DESCRIPTION OF APPARATUS

Engine

The engine used in this investigation was specially designed and built to accommodate a 5.25-inch-bore, liquid-cooled Lycoming cylinder. The cylinder was mounted on a heavy crankcase. One of three cylinder adapter plates could be used allowing a constant compression ratio of 6 to be maintained when crankshafts of different stroke were installed. The three crankshafts provided strokes of 4.25, 5.25, and 6.25 inches, and three connecting rods were used in order to maintain the ratio of crank radius to connecting-rod length constant at 0.25. A special camshaft was driven from an accessory shaft by a bevel and spur gear train. A schematic diagram of the engine setup is shown in figure 1.

Air Inlet System

Air supplied to the engine was passed through an orifice meter into a large steam-jacketed vaporizing tank, where fuel was introduced. The resulting dry fuel-air mixture was then passed through a short inlet pipe to the inlet port of the cylinder. The air used could be taken either from the atmosphere or from a laboratory air line which supplied compressed air from which practically all water vapor had been removed by a drying tower containing activated alumina.

Fuel and Fuel System

The fuel used was taken from the laboratory storage tank of high octane aviation gasoline, except for the detonation limit determinations. For those tests a supply of 87-octane fuel was obtained and carefully stored in order that all detonation limits could be determined with the same batch of fuel.

The fuel was supplied by a small pressure pump through a rotameter and a manually adjusted needle valve to a point in the air supply pipe just before it entered the steam-jacketed vaporizing tank. The tank walls were heated sufficiently to insure complete evaporation of the fuel.

Exhaust System

The exhaust gases were passed through a short pipe into a large water-jacketed surge tank. From this tank the exhaust gases passed through a throttle valve into the laboratory exhaust system.

Cooling System

A centrifugal pump circulated water through the cylinder coolant jacket. Sufficient cold water from the city main was added to control the jacket outlet temperature. After considerable difficulty was experienced from cylinder barrel fatigue, which was attributed to corrosion, a closed system was adopted. In this arrangement approximately five gallons of water containing a corrosion inhibitor were circulated by the pump through the jacket and through a heat exchanger which was cooled by city water.

Lubrication

Lubricating oil was supplied under pressure to the engine by means of a separately driven gear pump. The relatively large amount of oil bypassed by a relief valve was passed through a heat exchanger and returned to the oil supply tank. Regulation of the oil supply temperature was obtained by control of steam and water admitted to the heat exchanger. Oil was collected in the engine sump and returned to the supply tank by a separately driven gear pump.

MEASURING INSTRUMENTS

Load was provided by an eddy-current dynamometer. Torque was measured by means of a manometer connected to a cylinder having a hydraulically balanced piston attached to the dynamometer arm. Speed was determined by a mechanical tachometer in conjunction with a strobotac operating from a 60-cycle alternating-current line and illuminating stripes painted on the flywheel.

Air flow was measured by a sharp-edged orifice installed in a 3-inch pipe line in accordance with the A.S.M.E. Fluid Meters Committee Report (reference 4).

Air temperatures were measured with mercury-in-glass thermometers or iron-constantan thermocouples in conjunction with a potentiometer and sensitive galvanometer. All other temperatures were measured with vapor pressure thermometers.

Fuel flow was measured by a calibrated rotameter.

Atmospheric pressure was measured with a mercury barometer; inlet pressure was measured in the vaporizing tank with a mercury manometer; and exhaust pressure was measured in the exhaust surge tank by the same means.

Spark advance was measured with a neon light, revolving with the crankshaft adjacent to a stationary protractor. The spark discharge excited a bright flash in the lamp, which was shielded to make the flash appear as a thin red line.

Detonation was measured with a Sperry vibration-type pickup designed for this purpose. It was mounted on the cylinder head near the inlet port. This unit was connected to the Y-axis input of a DuMont type 208 oscillograph. When detonation occurred, typical "splashes" of high frequency vibration patterns could be observed on the oscillograph screen. These splashes increased rapidly in amplitude and occurred more often as detonation intensity increased. It was not difficult for an experienced observer to duplicate measurements made by this method.

Both heavy spring and light spring indicator cards were taken with the M.I.T. balanced pressure diaphragm indicator unit.

PROCEDURE

Each of the three crankshafts was installed in the engine in succession. With each installation air capacity tests were run at piston speeds varying from 1350 to 3150 foot per minute in 300-foot-per-minute increments, with the exception that the short stroke was run only to 2550 feet per minute since the eddy current dynamometer was unsuitable for speeds in excess of 3600 rpm. Spark advance was varied to obtain best power in all cases. The quantities held constant are listed in table 1.

TABLE 1

Inlet mixture temperature, °F	120
Vaporizing tank pressure, in. Hg abs	27.4
Fuel-air ratio	0.075
Oil inlet temperature, °F	150
Jacket water outlet temperature, °F	180
Exhaust surge tank pressure, in. Hg abs	31.4
Valve timing: Inlet opens, deg B.T.C.	20
Inlet closes, deg A.B.C.	56
Exhaust opens, deg B.B.C.	64
Exhaust closes, deg A.T.C.	20

Additional tests were made with each stroke at a constant piston speed of 2400 feet per minute over a range of inlet pressures obtained by means of a separately driven supercharger.

During these runs determinations of the detonation-limited performance were made by setting the inlet pressure at various values in small increments and varying the spark advance to obtain incipient detonation. The quantities held constant during these tests appear in table 2.

TABLE 2

Piston speed, ft/min	2400
Fuel-air ratio	0.075
Oil inlet temperature, °F.	160
Jacket water outlet temperature, °F	180
Exhaust surge tank pressure, in. Hg abs.	31.4
Fuel	aviation 87-octane
Valve timing: See table 1.	

RESULTS AND DISCUSSION

The results of the investigation are incorporated in figures 2 to 9.

Figure 2 shows clearly that the air capacity of the engine tested is very definitely a function of the piston speed, and that changing the stroke of the engine does not change this function. In other words, at any particular value of piston speed the air capacity is independent of stroke. The volumetric efficiency of the engine over the piston speed range is represented by a single curve for all strokes as shown in figure 3. Figure 4 shows further that air capacity at constant piston speed is independent of stroke over a wide range of inlet pressures.

Figure 5 indicates the extent to which the spark must be retarded in order to prevent detonation as the inlet pressure is increased. The curves represent the borderline of detonation for each stroke. From any point on any of the three curves, the region above the line is a region of detonation, while below the curve there will be no detonation. These curves might be changed considerably if operating conditions or the fuel should be changed, but it is unlikely that their relative positions would be reversed. The fact that detonation occurs at higher inlet pressures as the stroke is shortened may be due to either or both of the following two factors:

(1) Increased flame speed. - Flame speeds are known to be related closely to the degree of turbulence existing within the cylinder at the time of combustion. In the present tests the average gas velocity through the inlet valve is the same for all strokes at the same piston speed, since air capacity is unaffected by stroke. It would be reasonable, therefore, to expect the same degree of turbulence to be produced during the intake process for each of the three strokes, at constant piston speed. It is possible, however, that with the higher revolutions per minute associated with the shorter stroke, at constant piston speed, the turbulence created has less time to decay before combustion is completed, thus giving an increased flame speed with the shorter stroke. An increase in flame speed will give a greater time rate of pressure rise during combustion. It was demonstrated in reference 5 that increasing the time rate of compression allowed higher pressures and temperatures to be reached before detonation.

The indicator cards were examined for evidence of differences in rate of pressure rise for the three strokes, but it was found that wide variations in the combustion line from cycle to cycle made it impossible to determine accurately the rate of pressure rise.

(2) Smaller combustion chamber. - With decreasing stroke the combustion chamber becomes smaller in volume and slightly more compact if the compression ratio is held constant. This difference might be expected to reduce the tendency toward detonation, since it is well recognized that, other conditions being the same, reducing the maximum distance for flame travel reduces the tendency to detonate. However, the changes in combustion chamber volume were accomplished with only a very small change in maximum distance for flame travel.

A single straight line passing through the origin in figure 6 represents a good correlation of indicated horsepower against air capacity for all strokes with the exception of three points taken for the 6.25-inch stroke. The indicator diagrams corresponding to these three points differ from the others. Maximum pressure appears nearer top center and the pressures measured during expansion are lower. The differences are such as might have been caused by a phase shift between the top center line and the measured pressure. These three indicated horsepower determinations, therefore, are believed to be in error. Disregarding these three points, the data indicate that there are no significant differences in indicated horsepower for the different strokes over the range of air capacities covered. These results lead to the conclusion that the combustion efficiency is unaffected by the changes in piston stroke covered by this investigation.

The brake mean effective pressure curves shown in figure 7 reveal progressively lower values as the stroke is shortened for any particular piston speed. The pumping mean effective pressure, also shown in figure 7, reveals no variation in pumping losses for the three strokes. The differences in brake mean effective pressure therefore are attributed to differences in mechanical friction with the three strokes. In order to maintain a constant piston speed as the stroke is made shorter, it is necessary to operate the engine at higher revolutions per minute. Since the main journal and the crankpin bearing dimensions were the same for all three strokes, the higher revolutions per minute associated with the shorter stroke gives a higher bearing friction loss. Thus the brake mean effective pressure for any piston speed is lowest for the 4.25-inch stroke and highest for the 6.25-inch stroke.

Values of journal and crankpin bearing friction mean effective pressure have been calculated by Petroff's equation (appendix 1), assuming oil-film temperatures of 160°, 190°, and 220° F. The results are presented in table 3. These values are only approximations of the actual bearing friction, as the effects of bearing eccentricity and distortion are not included in Petroff's equation.

The calculated journal and crankpin friction mean effective pressure values for 160° and 220° F assumed oil-film temperatures have been plotted in figure 8 for the three strokes.

Adjusted brake mean effective pressure curves, obtained by adding calculated journal and crankpin bearing friction mean effective pressure to brake mean effective pressure values for corresponding piston speeds, are also shown in figure 8. Two families of adjusted brake mean effective pressure curves are obtained, one based on 160° F and the other on 220° F assumed oil-film temperature in the bearing.

If the actual oil-film temperature is near 160° F, then the good correlation of the adjusted brake mean effective pressure curves of figure 8 for this temperature indicates that the differences noted between brake mean effective pressure curves for the three strokes at a given piston speed (fig. 7) can be accounted for by the differences in journal and crankpin bearing friction mean effective pressure for the three strokes. However, if the actual bearing oil-film temperature is near or above 220° F, the adjusted brake mean effective pressure curves of figure 8 show that differences in journal and crankpin bearing friction mean effective pressure will account for only part of the differences in brake mean effective pressure curves for the three strokes.

If distortion of the connecting rod affected the crankpin friction, the friction would tend to be greater with the short stroke at a given piston speed, due to the higher inertia forces. This factor may account for differences not covered by the calculations based on Petroff's equation.

The differences in brake mean effective pressure for the three strokes are shown in figure 9 to be substantially the same over a wide range of inlet pressures when operating at a constant piston speed. This fact gives further confirmation of the theory that the differences in brake output are due to differences in mechanical friction.

If this result is applied to a multicylinder engine, it must be remembered that there are fewer crankshaft journals per cylinder than in a single-cylinder engine. Consequently the mechanical efficiency is higher and the changes in journal friction caused by changes in revolutions per minute will represent a smaller fraction of the power output.

CONCLUSIONS

The following conclusions seem justified by this investigation:

1. The air capacity and volumetric efficiency of the engine are independent of the stroke for any particular piston speed and any particular inlet pressure within the ranges covered.

2. With the particular cylinder and operating conditions used and at one piston speed the engine can be more highly supercharged without detonation as the stroke is shortened.

3. The ratio of indicated horsepower to air capacity shows no significant variation with change in stroke over a wide range of air capacities.

4. Values of brake mean effective pressure are progressively lower for shorter strokes when operating at constant inlet pressure and varying piston speed and also when operating at constant piston speed and varying inlet pressure. Calculations indicate that the differences in brake mean effective pressure for the three strokes are largely accounted for by the differences in journal and crankpin friction mean effective pressure.

5. No differences in pumping losses at the same piston speed for the three strokes were noted over a wide range of piston speeds.

CONCLUDING REMARKS

Although not the subject of this report, it may be well to note here some basic mechanical problems which arise as the stroke is decreased at a given piston speed.

If an engine with a short stroke is run at the same piston speed as one with the same bore but a long stroke, the ratio of the accelerations of the piston and connecting rod in the two engines will be inversely proportional to the stroke. Thus in this particular case, the relative accelerations of the piston are shown in the following tabulation:

Stroke (in.)	$\frac{\text{Piston speed}}{\text{Piston speed with 6.25-in. stroke}}$	$\frac{\text{Rpm}}{\text{Rpm with 6.25-in. stroke}}$	$\frac{\text{Max. acceleration of piston}}{\text{Max. acceleration of piston with 6.25-in. stroke}}$
4.25	1.00	1.47	1.47
5.25	1.00	1.19	1.19
6.25	1.00	1.00	1.00

Since the pistons are identical, the inertia forces on the piston with the short stroke are 1.47 times the inertia forces with the long stroke at the same piston speed. This computation points out one of the basic problems of the short-stroke engine.

Since the valve mechanism is identical for the long- and short-stroke engines, the inertia forces here would be proportional to $(\text{rpm})^2$ or $\left(\frac{1}{\text{stroke}}\right)^2$. Consequently, the valve of the short-stroke engine would have 2.16 times the acceleration of the valve of the long-stroke engine at the same piston speed, and would present a more difficult design problem for a short-stroke engine. Present aircraft engines, where the stroke-bore ratio is unity or more, generally operate near their valve-gear speed limit at take-off. It may be noted here that difficulty was experienced with the valve gear of the engine used in these tests. A special spring was necessary in order to insure that the valve followed the cam above 3300 rpm. When the speed of a valve gear of given size is increased, the problem of avoiding critical spring vibration becomes more difficult. Valve mechanisms which do not rely on spring return, such as the sleeve valve, may be a possible way out of this difficulty.

For the same propeller revolutions per minute and the same power output, short-stroke engines require a larger propeller-reduction-gear ratio.

Another difficulty which may be associated with the operation of short-stroke engines at high piston speed is preignition. During the course of the tests described in the present report, severe preignition was encountered at high output (supercharged) when the engine was run with the short stroke. The "coldest" standard aircraft spark plug was found unsatisfactory under the combination of high inlet pressure and high revolutions per minute. A special spark plug (Champion RJ-24) eliminated the trouble. The probable reason for preignition with the short stroke, but not with the longer strokes, lies in the fact that at a given piston speed the revolutions per minute for the short stroke is greater. High revolutions per minute results in more frequent "scrubbing" of hot gases in and out of the spark plug cavity with resultant increased rate of heat transfer to the spark plug electrodes.

Massachusetts Institute of Technology,
Cambridge, Mass., May 24, 1944.

APPENDIX

In noting the progressively lower values of brake mean effective pressure with shorter stroke at a constant piston speed, figure 7, it was realized that such a trend could be caused by an increase in the mechanical friction losses in the main journal and crankpin bearings due to the higher engine revolutions per minute required with the shorter strokes in maintaining constant piston speed. It was decided, therefore, to calculate the friction in the main journal and crankpin bearings.

The friction losses in the bearings were calculated by means of Petroff's equation for fluid friction. This equation is based on the assumption that there is a complete film of lubricant between the journal and bearing and that the bearing loading does not cause any eccentricity. No information was available on the temperature distribution in the oil film. Based on the fact that the entering oil temperature was 150° F, the bearing friction was calculated for assumed oil film temperatures of 160°, 190°, and 220° F. The bearing clearances were not known very precisely, but were taken as the minimum probable values based on the measurements available in order to give maximum probable friction losses.

From Petroff's equation,

$$\text{friction horsepower} = 3.75 \times 10^{-13} \mu N^2 \frac{L}{C} D^3$$

where

μ viscosity, centipoises

N revolutions per minute

L length of bearing, inches

D diameter of bearing, inches

C diametral clearance of bearings, inches

This equation was converted to friction mean effective pressure on the basis of the 5.25-inch piston diameter as follows:

$$\text{friction mean effective pressure} = 8.22 \times 10^{-8} \mu \frac{S}{s^2} \frac{L}{C} D^3$$

where

S piston speed, feet per minute

s piston stroke, inches

The friction mean effective pressure was calculated for the two main journal bearings and the one crankpin bearing for the following values of dimensions and operating variables:

	Main journal bearing	Crankpin bearing
L, inches	3.50	2.375
D, inches	3.25	2.875
C, inches	0.008	0.005
μ, centipoises	57.5 at 160° F 31.5 at 190° F 19.0 at 220° F	

Calculated values of the friction mean effective pressure for each of the three strokes over the piston speed range of the investigation are listed in table 3.

Calculations then were made to determine the eccentricity of the bearings and the influence of the eccentricity on the power losses. The calculations were made for the maximum gas pressure loading and for the maximum inertia force loadings on the bearings using the method of A. E. Norton, "Lubrication," chapter VII. These calculations showed that the magnitude of the error introduced by neglecting eccentricity of the bearings was considerably less than the deviations introduced by not knowing the exact oil film temperature. Consequently, it was decided to make no correction for eccentricity. In using the friction mean effective pressure to obtain values of adjusted brake mean effective pressure (fig. 8), the calculations for 160° F oil film temperature were used in order to establish a probable upper limit to the values of friction mean effective pressure.

TABLE 3

For 160° F oil film temperature fmep (lb/in.²)

Piston speed, S (ft/min)	s_1 = 4.25 in.	s_2 = 5.25 in.	s_3 = 6.25 in.
1350	14.6	9.6	6.8
1650	17.9	11.7	8.3
1950	21.1	13.8	9.8
2250	24.4	15.9	11.3
2550	27.6	18.1	12.8
2850	30.8	20.2	14.3
3150	34.1	22.3	15.8

For 190° F oil film temperature fmep (lb/in.²)

Piston speed, S (ft/min)	s_1 = 4.25 in.	s_2 = 5.25 in.	s_3 = 6.25 in.
1350	8.0	5.2	3.7
1650	9.8	6.4	4.5
1950	11.6	7.6	5.3
2250	13.3	8.7	6.2
2550	15.1	9.9	7.0
2850	16.9	11.1	7.8
3150	18.7	12.2	8.6

For 220° F oil film temperature fmep (lb/in.²)

Piston speed, S (ft/min)	s_1 = 4.25 in.	s_2 = 5.25 in.	s_3 = 6.25 in.
1350	4.8	3.2	2.2
1650	5.9	3.9	2.7
1950	7.0	4.6	3.2
2250	8.0	5.3	3.7
2550	9.1	6.0	4.2
2850	10.2	6.7	4.7
3150	11.3	7.4	5.2

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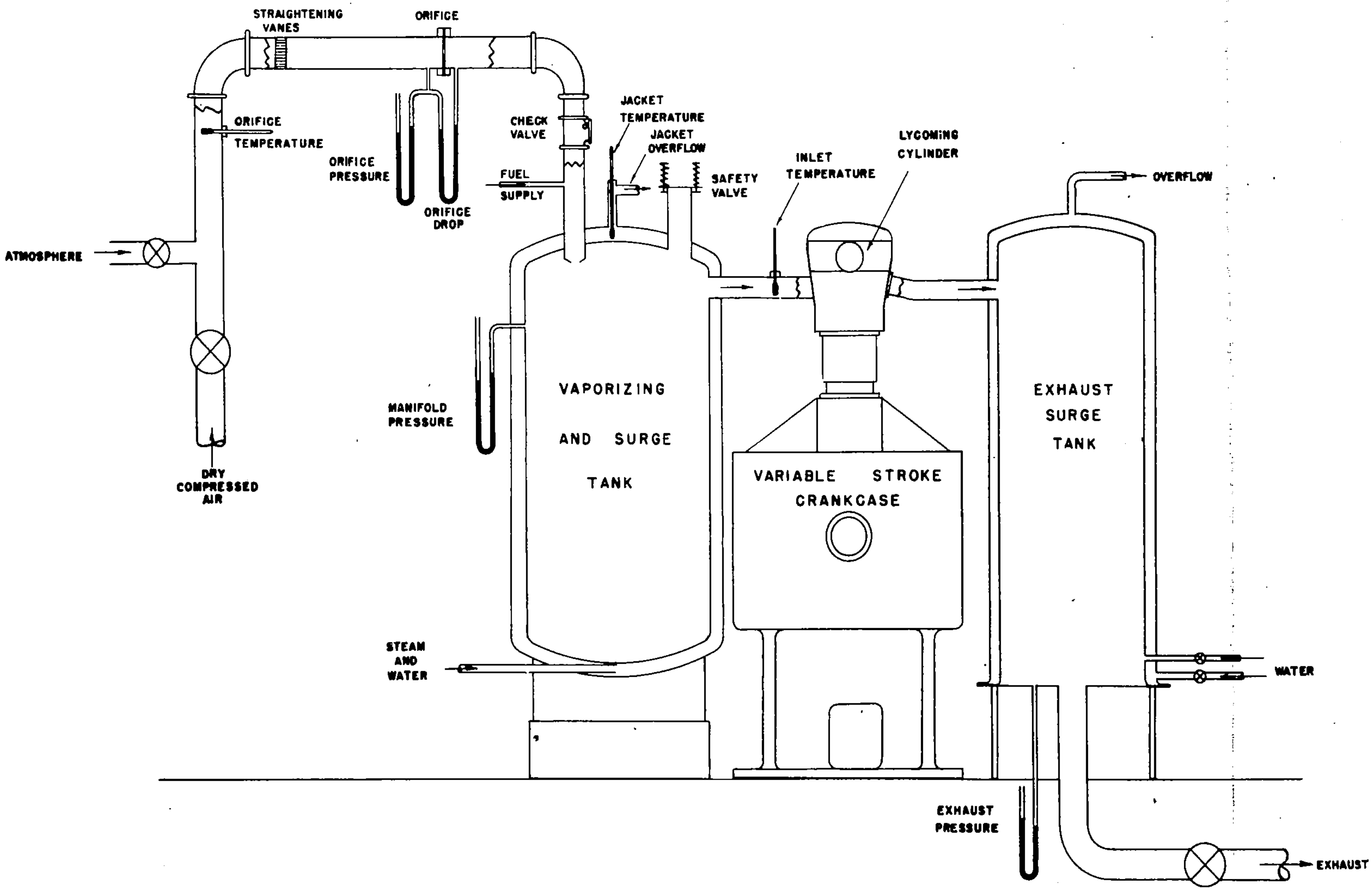

FIG. I — DIAGRAMATIC SKETCH OF ENGINE SET-UP

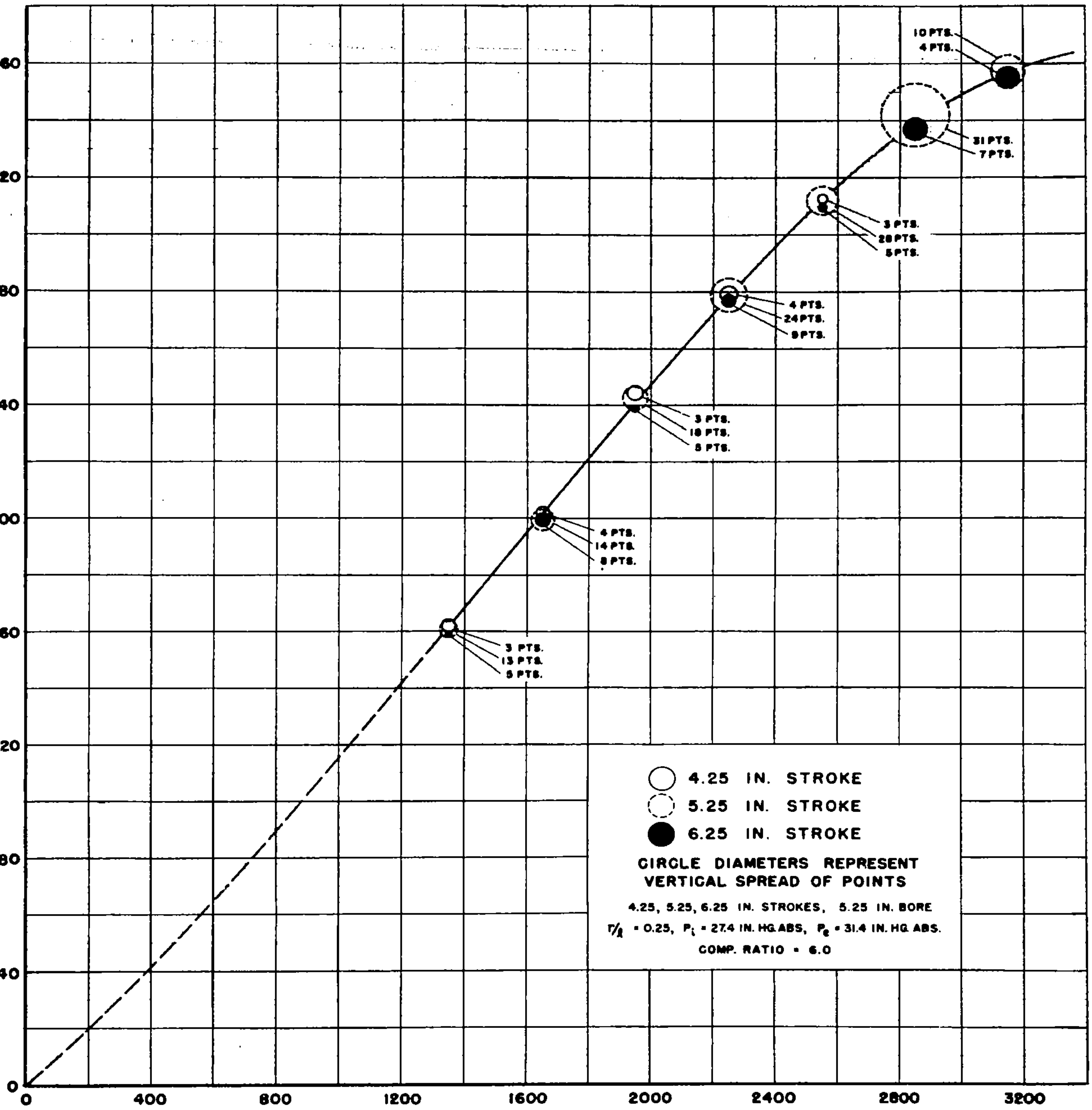

FIG. 2 — THE EFFECT OF CHANGING STROKE ON AIR CAPACITY FOR VARIOUS PISTON SPEEDS

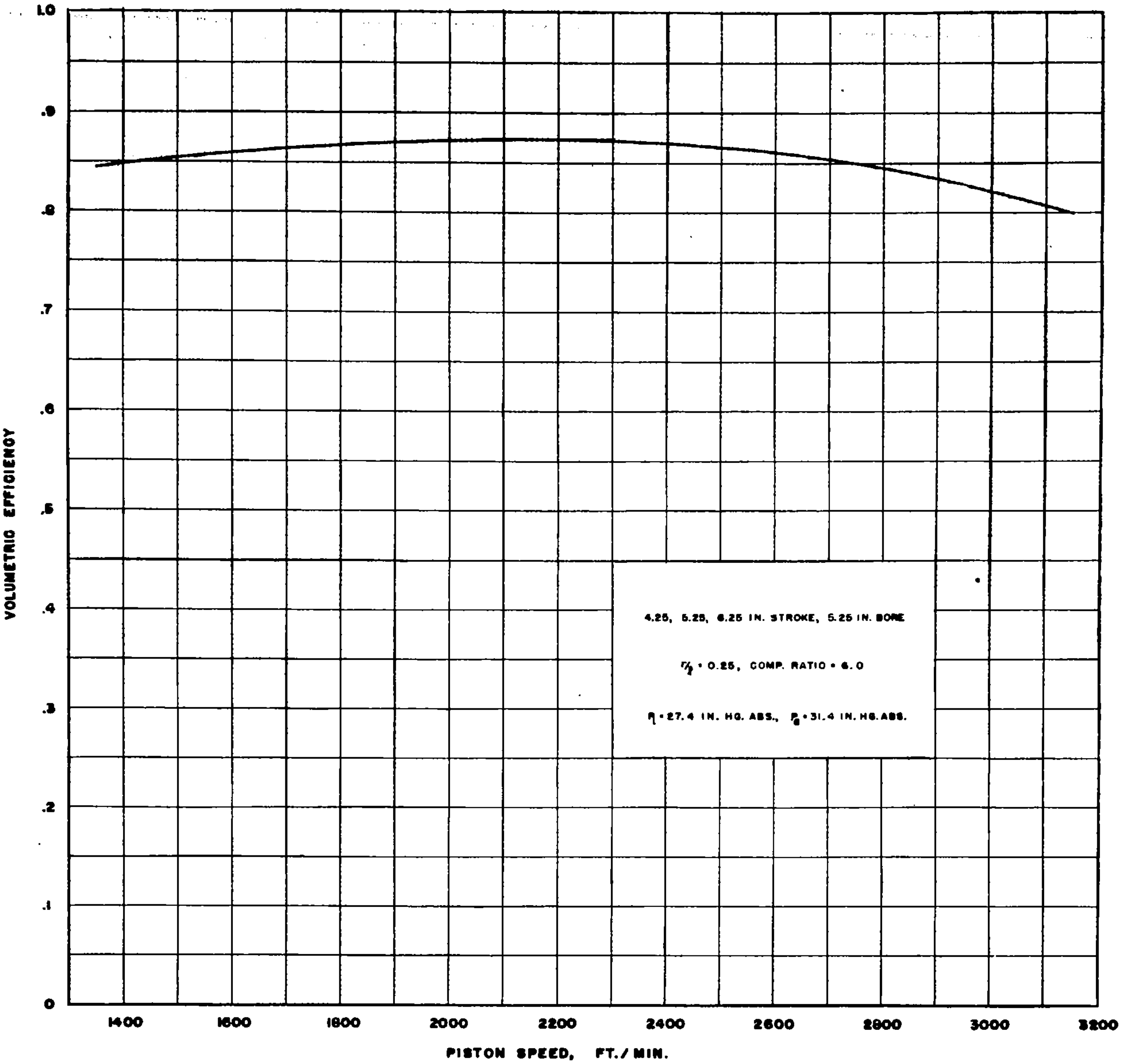

FIG. 3 — VOLUMETRIC EFFICIENCY FOR ALL STROKES COMPUTED FROM AIR CAPACITY CURVE, FIG. 2

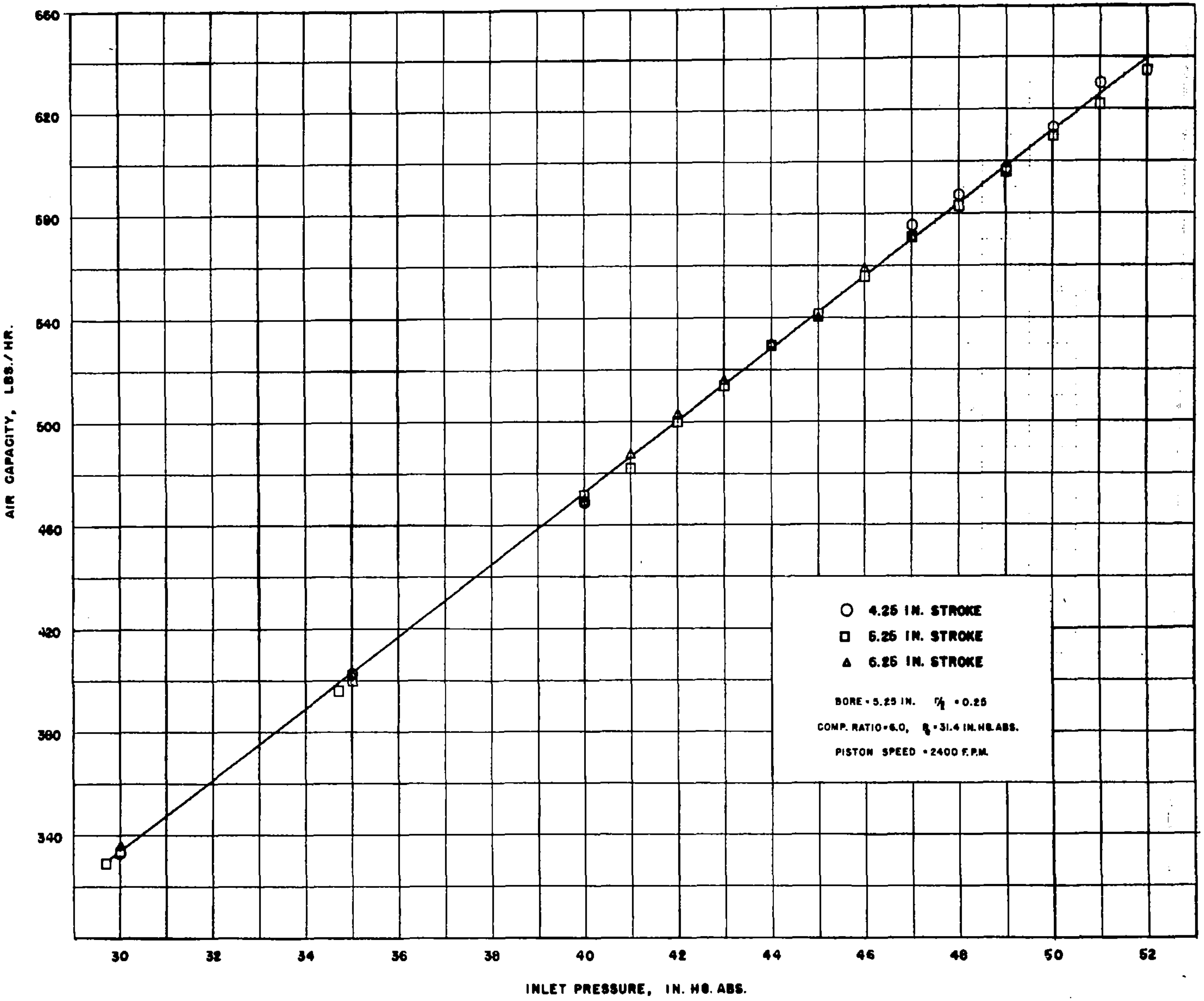

FIG. 4 — THE EFFECT OF CHANGING STROKE ON AIR CAPACITY

FOR VARIOUS INLET PRESSURES

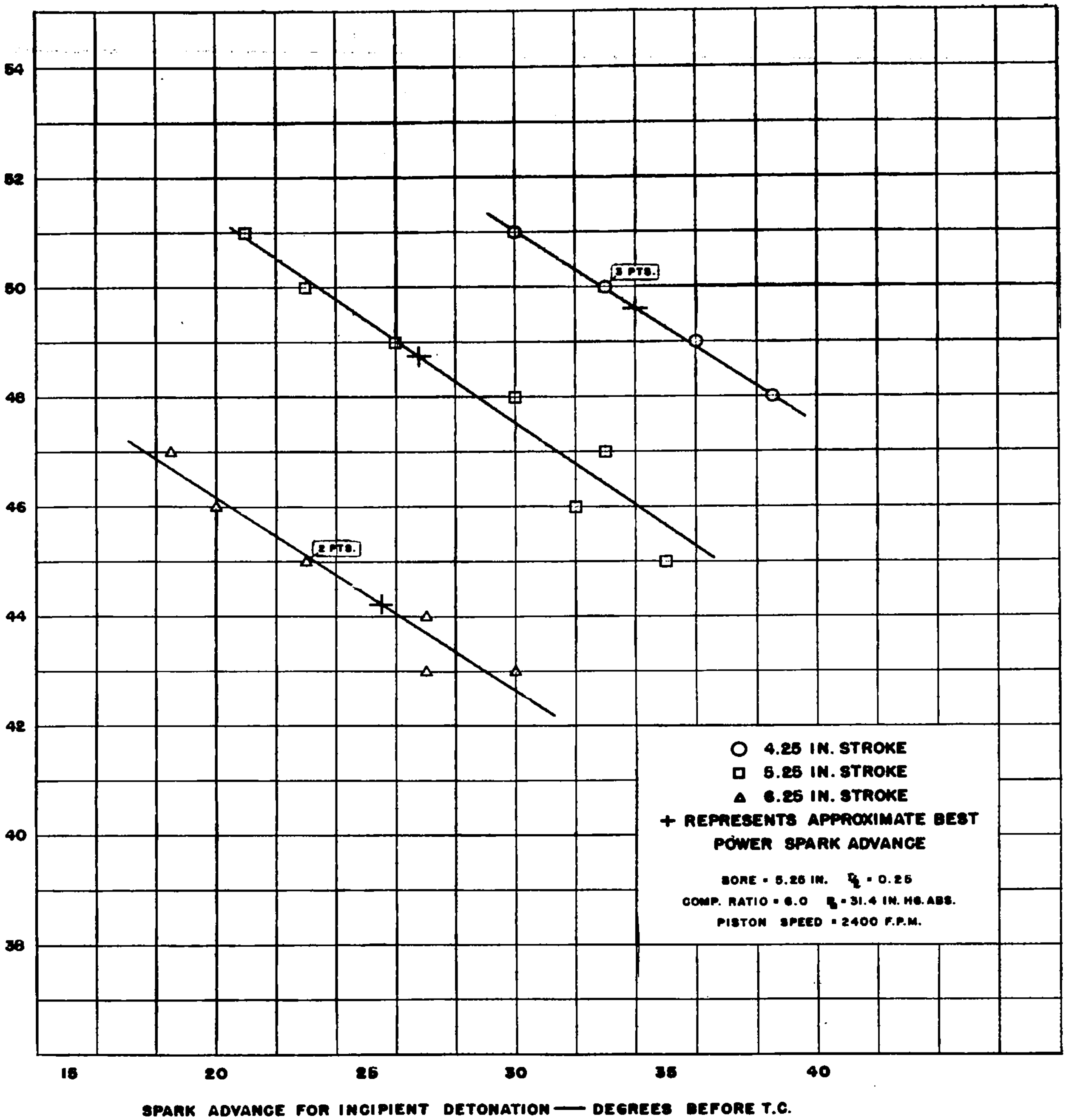

FIG. 5 — CURVES OF CONSTANT DETONATION FOR THREE STROKES

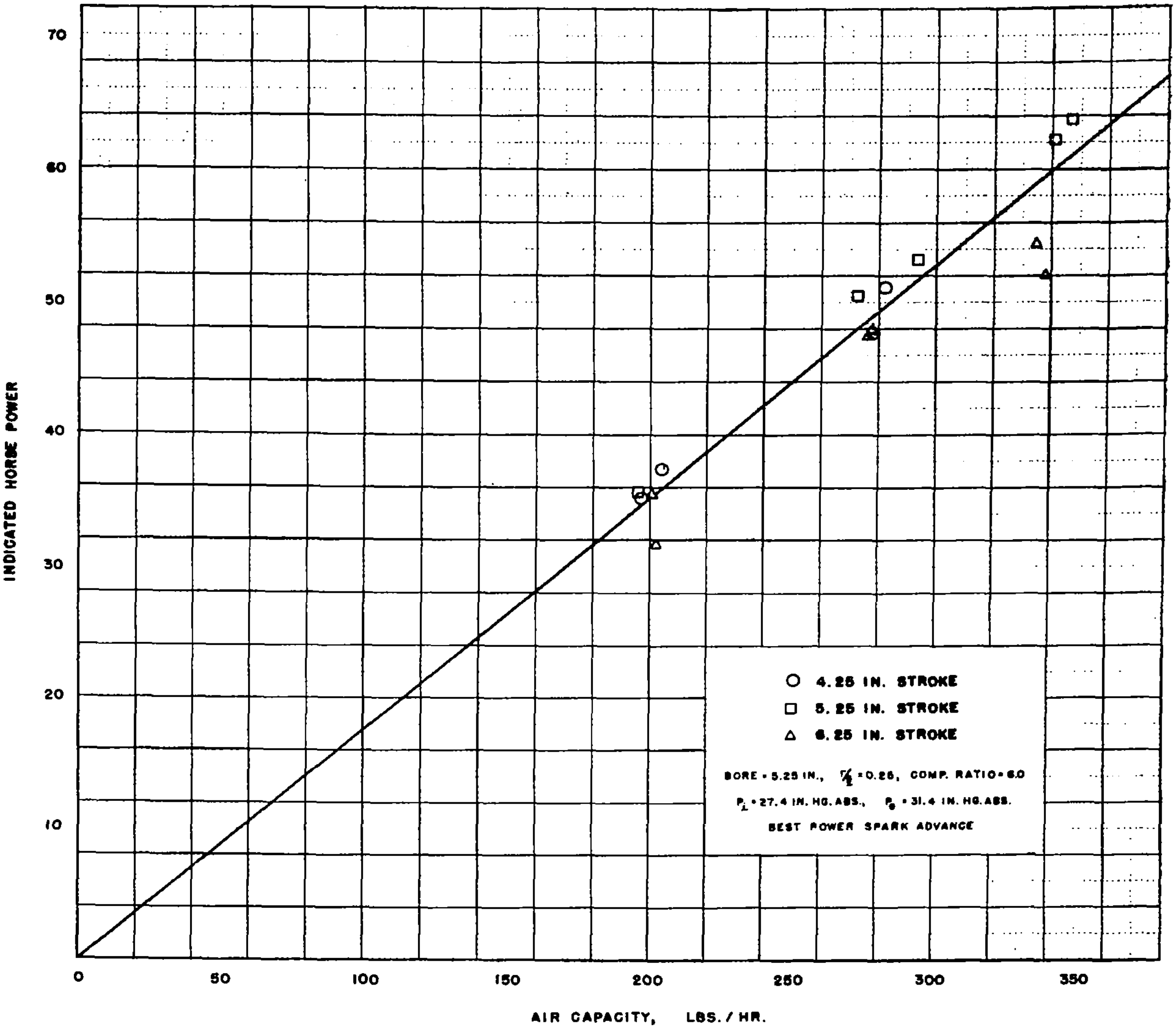

FIG. 6 — THE EFFECT OF CHANGING STROKE ON INDICATED H. P. FOR VARIOUS AIR CAPACITIES

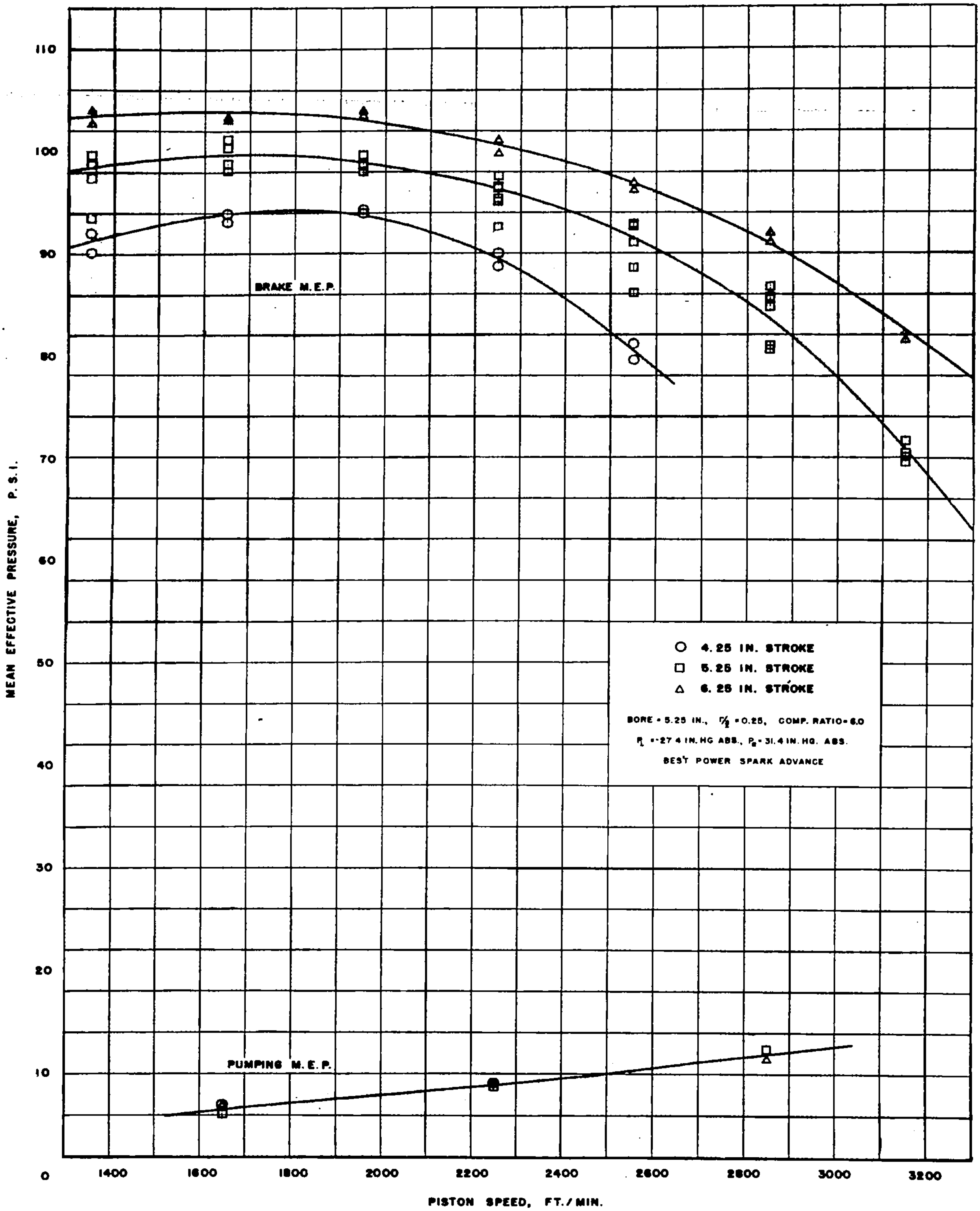

FIG. 7 — THE EFFECT OF CHANGING STROKE ON BRAKE M.E.P. AND PUMPING M.E.P. FOR VARIOUS PISTON SPEEDS

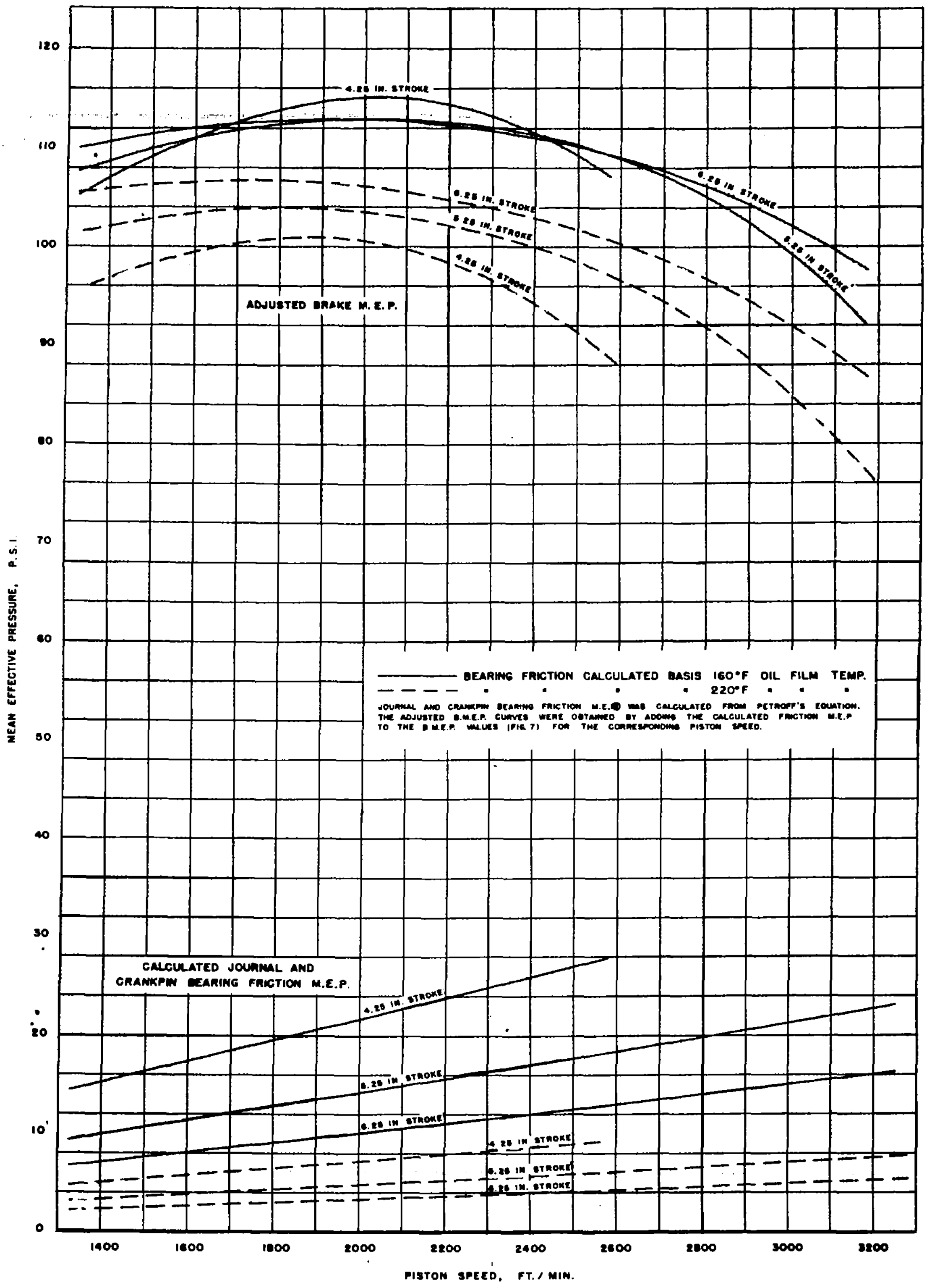

FIG. 8 — ADJUSTMENT OF BRAKE M.E.P. CURVES OF FIG. 7
FOR CALCULATED JOURNAL AND CRANKPIN BEARING FRICTION

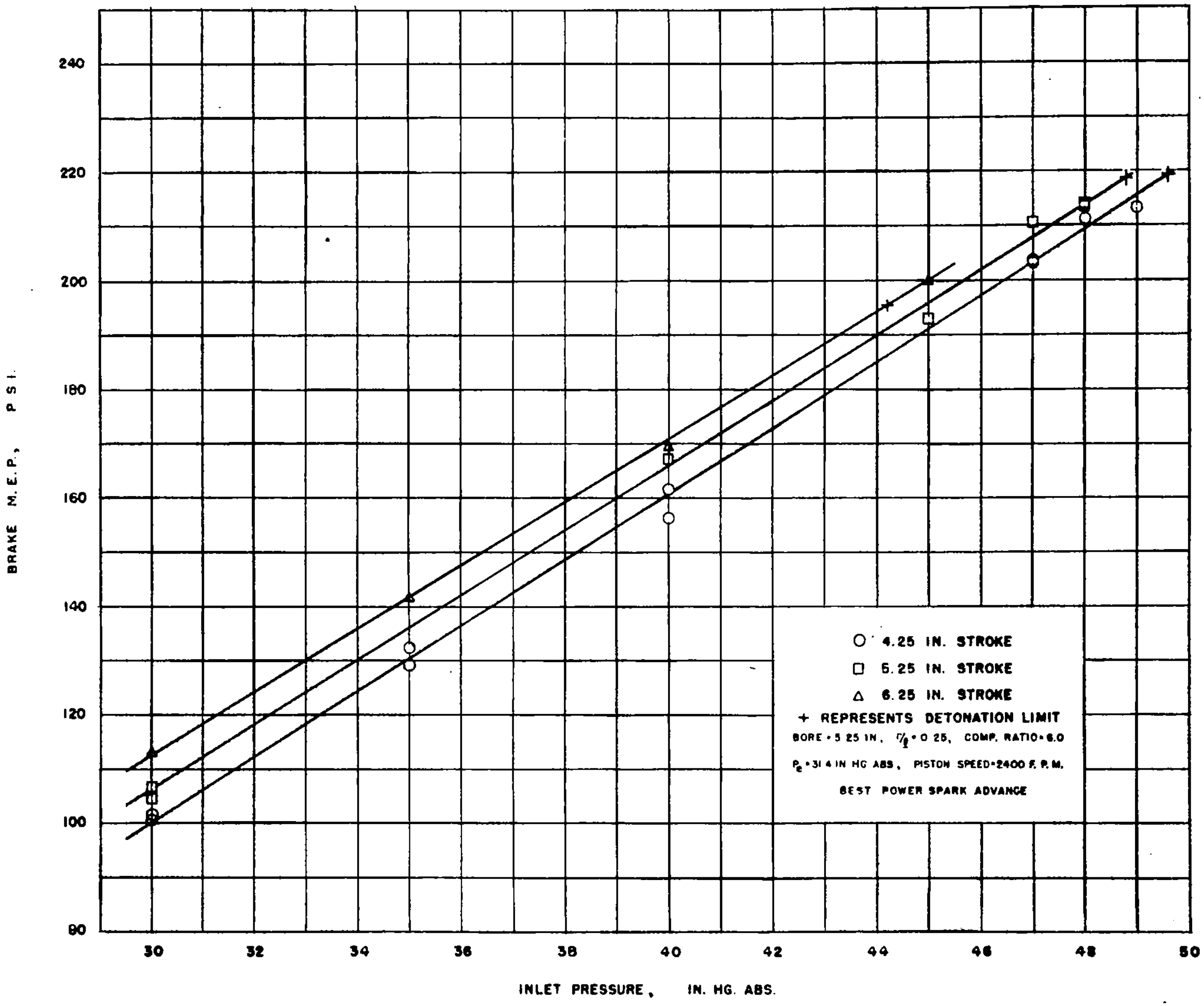

FIG. 9 — THE EFFECT OF CHANGING STROKE ON BRAKE M.E.P. FOR VARIOUS INLET PRESSURES