

Automobiles, Gasoline - Motors

Full Auto motive engine
bearing loads

A STUDY OF
AUTOMOTIVE ENGINE BEARING LOADS

Thesis
Submitted as Part of the Requirements
For the Master of Science Degree

By

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THESIS

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OBJECT

The object of this work was to study automotive engine bearing loads and to determine various constants relative to bearing design from graphic results of computations from data supplied by many leading automobile manufacturers.

INTRODUCTION

The automotive engineers in keeping up with the public demand for automobiles capable of higher speeds are again facing an old problem - that of bearing failure. In the early days of the industry, bearing failures were not at all an uncommon occurrence. In fact almost every driver could expect one or more bearing failures during the life of the vehicle. This problem was gradually worked out through better design, improvement in bearing metals, precision manufacture and pressure lubrication. These improvements took place over a period of years. The new problem of bearing failure will undoubtedly be solved in a short time and the industry will again experience a period during which bearing failures will be infrequent. Although several methods have been brought forth to lengthen bearing life, many indications point to inherent design as the factor that will solve the problem.

With the increasing importance of accuracy in design, it will be evident to any one familiar with such work and its difficulties that simplified methods giving reasonably accurate results will be very useful tools in overcoming bearing failures.

PROCEDURE

In determining the various constants, the work was confined to six and straight eight cylinder engines equipped with aluminum alloy pistons. Engines of these types are used by many leading manufacturers in the industry and usually are not radical in design. The reason for confining the work to engines equipped with aluminum alloy pistons was that these pistons are better suited to high speed work than are cast iron pistons. Since advancement in speed is the factor which is bringing about bearing failures, it is logical that data for aluminum alloy pistons will be of more value at a later date than would data for cast iron pistons. Also it might be added that all data supplied and used in this work was for the 1934 models in production at the time this work was done.

The first problem was to determine the ratio, if any, or a representative graph of the ratio, of the mean load to the maximum load on the crank pin bearing. It has been determined that the characteristic curve of this ratio plotted against speed gives a fairly constant value for the ratio at speeds above 3500 r.p.m. At and above this speed the inertia and centrifugal forces are much more important than the gas forces. In fact designers may disregard the gas forces and deal only with the inertia and centrifugal

forces which would represent the maximum conditions in such a connecting rod linkage. It would also be desirable to determine whether or not the constant value or the graph of the ratio could be applied to conditions where various empirical formulas had been used to design the different engines.

To determine the factors in question force diagrams representing the resultant force on the crank pin bearing were drawn for each engine at a speed of 4000 r.p.m. This speed is one which is near the upper limit of present production models and is within the speed range for which the force ratio is fairly constant. In determining the diagrams, the crank interval used was 30 degrees. To find the gas pressures for each interval a theoretical indicator diagram¹ was drawn for each engine. Referring to print No. 1, P_a was taken as 13.5 pounds per square inch abs. The values of (n) were taken as 1.3 for compression and 1.4 for expansion². $P_{(max)}$ was taken as 4.65 times P_c . $P_{(max)}$ actual was taken as 79% of $P_{(max)}$ and this pressure is developed at 20 degrees from top dead center on the explosion stroke³. The pressures at the various intervals were converted from absolute values to pressures above

1. See print No. 1.
2. Janeway, Automotive Industries, Vol. 64.
3. Janeway, Automotive Industries, Vol. 64.

atmosphere. Using these values, the gas forces were found. The inertia forces of the reciprocating parts were determined for the various intervals, the reciprocating weight being taken as the sum of the weight of the piston assembly and the weight of the upper end of the connecting rod. The centrifugal force which is constant for any given speed was determined using the weight of the lower end of the connecting rod as the rotating mass. A line diagram¹ of the engine was laid out to a convenient scale. The radius of the crank was taken as the length of the vector representing the constant centrifugal force. The value of the centrifugal force divided by the radius therefore is the force which is represented by unit length vector of the diagram. The other forces were drawn to this scale and combined graphically.

The area of the force diagram was found by means of a planimeter and from this area a mean circle was determined which represents the mean force on the crank pin bearing. The ratio was then found between the mean force and the maximum force on the bearing.

The force ratios for the 6 cylinder engines were constant at .530. In plotting the force ratios of 8 cylinder engines against different factors, it was found that the force ratio increased very slightly with an increase in

1. See print No. 14.

piston displacement¹. The ratio, however, is fairly constant at .550.

Other curves to be plotted were total weight of the piston assembly against the bore² and the ratio of the weight of the connecting rod to the length of the connecting rod against the bore³.

The main bearing loads were then examined. It would be desirable to know if it was possible to determine a set of constants whereby the mean load on any main bearing of an engine of that type could be determined in terms of an easily calculated load. In attempting to solve this problem, it was necessary to resolve the forces on the crank pin into resultant forces at the main bearings. The method of resolving the forces in the inverse ratio of distances was examined. Also the method of resolving the forces by considering the crankshaft a continuous beam was considered. It was decided to consider the shaft a continuous beam and to determine the reactions at the main bearings in terms of fractions of the force at any crank pin. The sum of the fractions of the forces at any one main bearing is the resultant at that bearing due to the forces on all the cranks. In applying this method to the actual crankshaft, the force vector on each crank can be

1. See print No. 13.
2. See print No. 11.
3. See print No. 12.

resolved into horizontal and vertical components. The components of the resultants at the main bearings can be combined giving the total resultant. The actual calculations of this work were not completed.

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DISCUSSION

A study of automotive engine bearings shows that the life of a bearing is determined to a great extent by one thing, the so-called "pv" factor. For this reason it would be well to discuss the causes of bearing failures and see how the "pv" factor regulates the life of bearings.

The most frequent cause for bearing failure is high operating temperature. Three factors of bearing life affected by high temperatures are the melting point of the metal, the strength of the metal at that temperature and the load-carrying capacity of the oil film¹. It might be said that the first factor is not important since the melting point of the metal is never reached until the bearing has begun to fail due to other factors. The strength of the bearing metal at high temperatures is important, especially if there is a high specific load caused by large inertia forces or deflection of the shaft. In this case, if the conditions are not of extreme magnitude the bearing will fail gradually by the breaking away of small particles. The load-carrying capacity of the oil film is of course very important. Once the film has broken down, a bearing failure is forthcoming.

In the explanation of bearing temperatures, it is known that the work due to frictional losses must be

1. Janeway, Automotive Industries, Vol. 64.

dissipated as heat. It is necessary, therefore, to have a fairly good estimation of the coefficient of friction in the bearing. The data on this subject is very limited, also the various authorities have different ideas as to what actually happens. The opinion of R.H. Janeway is that in the upper range of operating speeds the coefficient of friction is nearly constant, also that the oil film is not continuous. He bases his opinion upon results of examination of bearings of engines which have been run continuously in the upper range of operating speeds and upon Tower's experiments which are representative of the work on lubrication. Although automotive engine bearings are considered well lubricated bearings, it is evident that, when the results of Tower's experiments are applied to the conditions in an automotive engine, a continuous oil film is not maintained. His experiments show that with loads up to 350 pounds per square inch and with velocities up to 500 feet per minute the coefficient of friction varied inversely with the pressure. This undoubtedly indicates that a continuous oil film is maintained within this range. Above this the coefficient of friction remained practically constant over a wide range of pressures until seizure was approached. Upon examination of bearing pressures in engines that have given good results in service, it is evident that at high speeds the pressures are well within the range during which the coefficient of friction is constant. Since

a bearing is designed for maximum operating conditions, the coefficient of friction may be taken as constant.

P. M. Heldt¹ questions the statement that the coefficient of friction is constant. He bases his opinion on the fact that automotive engine bearings are well lubricated and for this reason the coefficient of friction is not constant, but is inversely proportional to the load and increases with speed. It will be granted that such is the case at slow speeds, but from previous statements it can be seen that this factor is not important in the design of a bearing for a high speed automotive engine.

Knowing the coefficient of friction to be fairly constant, it can be said that work equals pressure times velocity times coefficient of friction. It is evident that this work must be dissipated as heat and that although a small amount of it may be dissipated to the block, the most of the heat is transferred to the oil, thereby raising the temperature of the oil. Janeway then shows by a very unique proof² that the temperature difference between the bearing surface and the oil is proportional to the product of the unit pressure on the bearing and the surface velocity which is called the "pv" factor. It must be added that this is true for given coefficients friction and heat transfer.

1. Heldt, Automotive Industries, Vol. 66.

2. Janeway, Automotive Industries, Vol. 64.

The proof is:

$$W = fpv$$

w = work done in unit time

f = coefficient of friction

p = bearing load in pounds

v = journal velocity in feet per minute.

Also:

$$W = \text{Heat} = KS (T_b - T_o) = \frac{fpv}{778}$$

K = coefficient of heat transfer in Btu./sq.in./
1°F/min.

S = projected area of bearing in sq. in.

T_b = mean temperature of bearing

T_o = mean temperature of oil.

Then:

$$T_b - T_o = \frac{1}{778} \left(\frac{f}{K} \right) \frac{p}{S} v$$

Therefore:

$$T_b - T_o = \frac{p}{S} v$$

$$\text{Where } p = \frac{p}{S}$$

= unit pressure on the bearings.

There may be several objections¹ to this derivation from the point of view of oil film, but it is sufficiently correct for practical use.

1. Puvrey, Automotive Industries, Vol. 65.

From Janeway's discussion it is obvious that the safe value of "pv" decreases as the oil temperature rises. Also, it is known that an increase in "pv" creates a viscous circle since it increases the temperature differential between the bearing and the oil and also raises the temperature of the oil due to the greater amount of work to be dissipated as heat. Therefore the "pv" factor may be taken as a factor effecting bearing life.

In design work the "pv" factor must be associated with the diameter and length of the bearing which in turn are associated with the stiffness of the shaft and the overall length of the engine. The designer must "juggle" the various factors involved so as to keep within the boundary conditions of each factor. Data on "pv" factors are limited. Janeway is of the opinion that "pv" values around 30,000 may be used with good results, but that values of 40,000 and above should not be used.

In discussion of methods of calculating crank pin bearing loads, the old tedious method of dividing the engine cycle up into many intervals and calculating the instantaneous forces at each interval is of course the old "stand-by" for accurate results. Janeway's method¹ is a great improvement over the old method and gives fairly accurate results. However, the results of this work cannot be used in

1. Janeway, Automotive Industries, Vol. 64.

calculations of shaft design since the forces found are mean forces over an interval and not directional force vectors for each interval. Thus, if only the mean force on a crank pin bearing is needed, it is much easier to multiply the maximum force by a constant corresponding to the engine to determine that mean force. The maximum force on the crank pin bearing is the resultant on that bearing when the crank is at upper dead center after the exhaust stroke. This force is easily calculated.

The force diagram ratio for 8 cylinder engines when plotted against piston displacement per cylinder showed a slight increase with an increase in displacement.

The range of piston displacement covered by the 6 cylinder engines was too small to be represented on a graph. As a result the force diagram ratio was taken as having a constant value of .530. However, it might be said that the ratio would undoubtedly increase slightly with an increase of displacement providing the range of displacement was great enough.

The explanation of the fact that the ratio for 6 cylinder engines is smaller than that for 8 cylinder engines is that the maximum forces are greater in the case of the 6 cylinder engine due to the greater reciprocating mass of the larger piston.

The main bearing loads also came into the field of this work. Two methods of determining the main bearing

loads are to equal to the resultant at the main bearings with the shaft considered a continuous beam¹ and to divide the forces at the crank pin bearings inversely as the distances between bearings.

A crank shaft is a continuous beam of varying section, but it may be considered a continuous beam of constant section if, under test for deflection, it is found to deflect nearly the same as its equivalent straight shaft.

An attempt was made to determine constants whereby the mean loads on the main bearings could be easily calculated, but due to lack of time these calculations could not be completed. However, it might be added that the accuracy of these constants would be questionable when applied to other motors than the one for which they were determined. Deflection of the shaft and unequal spacing of the counterweights tend to make that would seem a fairly simple very complex. It might be added that the better way to design the main bearings is to use actual experimental data allowing various factors of safety and then approve the engine only after it has undergone a very thorough test.

1. Root, Dynamics of Engine and Shaft.

CONCLUSIONS

1. The ratio between the mean force and the maximum force on a crank pin bearing for straight 6 cylinder engines equipped with aluminum alloy pistons increases slightly with an increase in piston displacement, but may be taken as .550 with accuracy well within practical limits.

2. The ratio between the mean force and the maximum force on a crank pin bearing for 6 cylinder engines equipped with aluminum alloy pistons was found to be .530 over the range of engines for which data was available.

3. A crankshaft may be considered a continuous beam of constant cross section if, when tested for deflection, it is found to deflect the same as an equivalent shaft.

DATA AND RESULTS

Engine	No. of Cylinders	Bore (in.)	Stroke (in.)	Comp. Ratio	Wt. of piston (lbs.)	Wt. of oil (lbs.)
A	6	3-5/16	5	6.00	1.52	2.52
B	6	3-1/2	5	6.00	1.78	2.64
C	6	3-1/16	4-1/4	6.30	.86	1.96
D	6	3-1/16	3-5/8	6.30	.80	1.78
E	6	3-5/8	5	5.45	1.44	2.62
F	6	3-1/4	4-5/8	5.60	1.59	1.90
G	6	3-3/8	5	5.45	1.42	2.62
H	6	3-1/2	3-7/8	5.32	1.72	2.18
I	6	3-1/4	4-1/8	6.30	1.22	2.10
J	6	3-1/2	4-7/8	5.75	1.70	2.52

DATA AND RESULTS

Engine	Length of Rod (in.)	Ratio of $\frac{V}{\text{Con. Rod. L}}$	Piston Disc/cyl. (sq.in.)	Force Ratio
A	10-7/8	.237	39.95	.575
B	10-7/8	.231	43.05	.555
C	8	.233	31.30	.555
D	8-1/4	.217	37.35	.534
E	10-1/2	.249	44.75	.585
F	7-15/16	.239	36.30	.496
G	10-1/2	.249	44.75	.585
H	8-7/16	.260	37.33	.532
I	8-1/4	.255	34.22	.509
J	8-1/4	.231	40.20	.507

EQUATIONS AND CALCULATIONS

(Calculations for Engine "H")

Indicator Diagram:

$$P_2 = P_1 \left(\frac{V_1}{V_2} \right)^n$$

P_2, P_1 = pressures at the 2 points in lbs./sq.in.

V_2, V_1 = volumes at the 2 points

n = exponent - for compression 1.3

for expansion 1.4*

$$P_c = 15.5 \left(\frac{5.73}{1} \right)^{1.3}$$
$$= 118.7 \text{ lbs./sq.in.}$$

Where P_c = compression pressure

$$P_{(\max)} = 4.65 P_c$$

$P_{(\max)}$ = maximum pressure of theoretical indicator card.

$$P_{(\max)} = 4.65 \times 118.7$$
$$= 552 \text{ lbs./sq.in.}$$

$$P_{(\max)} \text{ actual} = .79 P_{(\max)}$$

$P_{(\max)} \text{ actual}$ = actual maximum pressure developed in cylinder at 20° past t.d.c. on the explosion stroke.

$$P_{(\max)} \text{ actual} = .79 \times 552$$
$$= 436 \text{ lbs./sq.in.}$$

$$F_g = A(P - 15)$$

F_g = gas force along centerline of engine in lbs.

A = area of piston in sq.in.

(P-15) = gas pressure above atmosphere

$$F_G = 9.61 (118.7-15) \\ = 997 \text{ lbs.}$$

Centrifugal force:

$$F_C = .0000284 W_C r N^2$$

F_C = centrifugal force in lbs.

W_C = weight of lower end of connecting rod in lbs.

r = radius of crank in inches

N = r.p.m. of engine

$$F_C = .0000284 \times 1.60 \times 1.9375 \times (4000)^2 \\ = 1433 \text{ lbs.}$$

Inertia force:

$$F_I = .0000284 W_I r N^2 \left(\cos \alpha + \frac{r}{L} \cos 2\alpha \right)$$

F_I = inertia force in lbs.

W_I = weight of piston assembly - weight of upper end of connecting rod in lbs.

r = radius of the crank in inches.

N = r.p.m. of engine

L = length of connecting rod in inches

α = crank angle from t.d.c.

$$F_I = .0000284 \times 3.20 \times 1.9375 \times (4000)^2 \\ \left(\cos 360^\circ + \frac{1.9375}{3.438} \cos 720^\circ \right) \\ = 3400 \text{ lbs.}$$

Maximum force on crank pin:

The maximum force on the crank pin bearing is

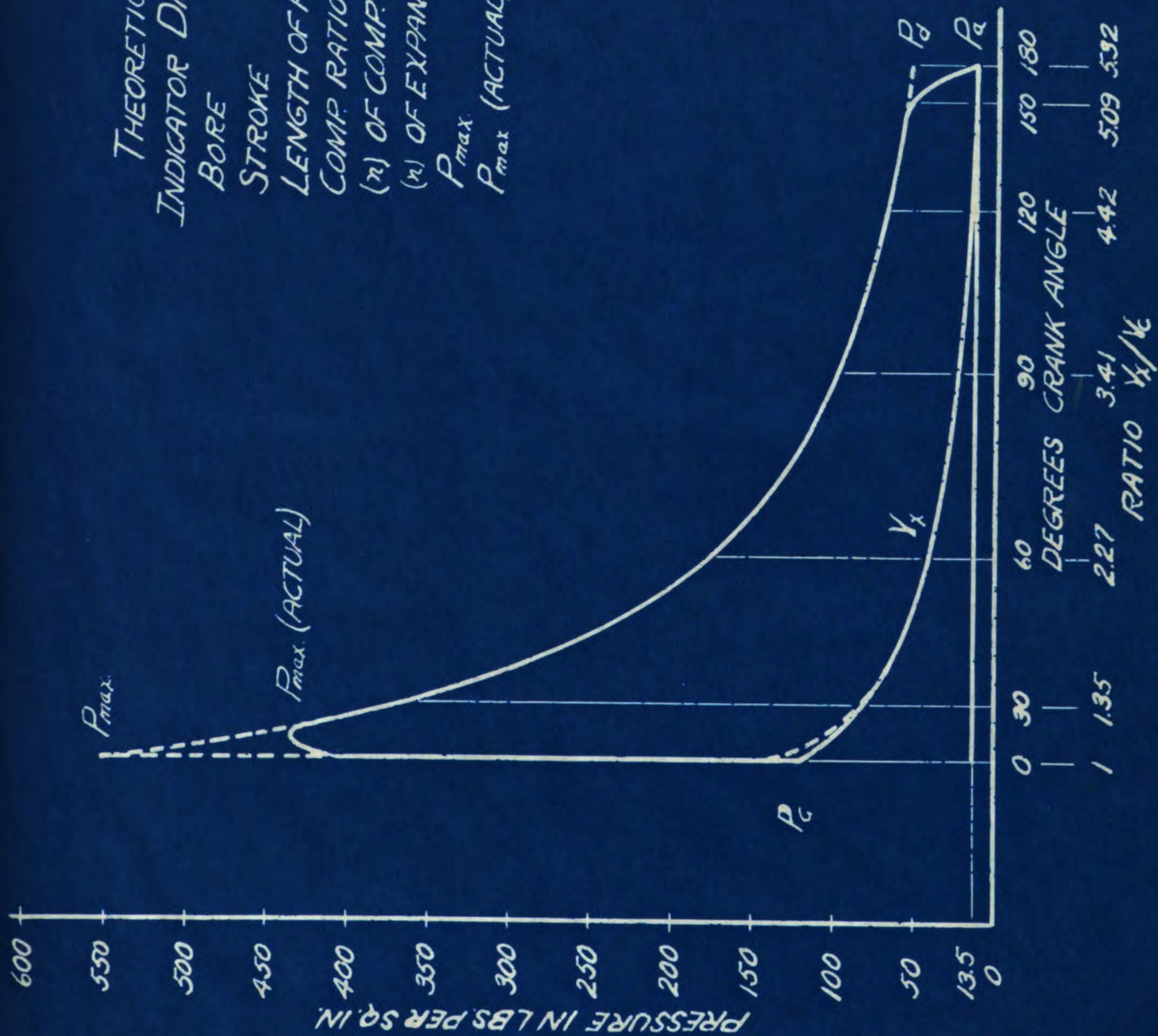
the sum of the centrifugal forces and the inertia forces at a crank angle of 0° or 360° .

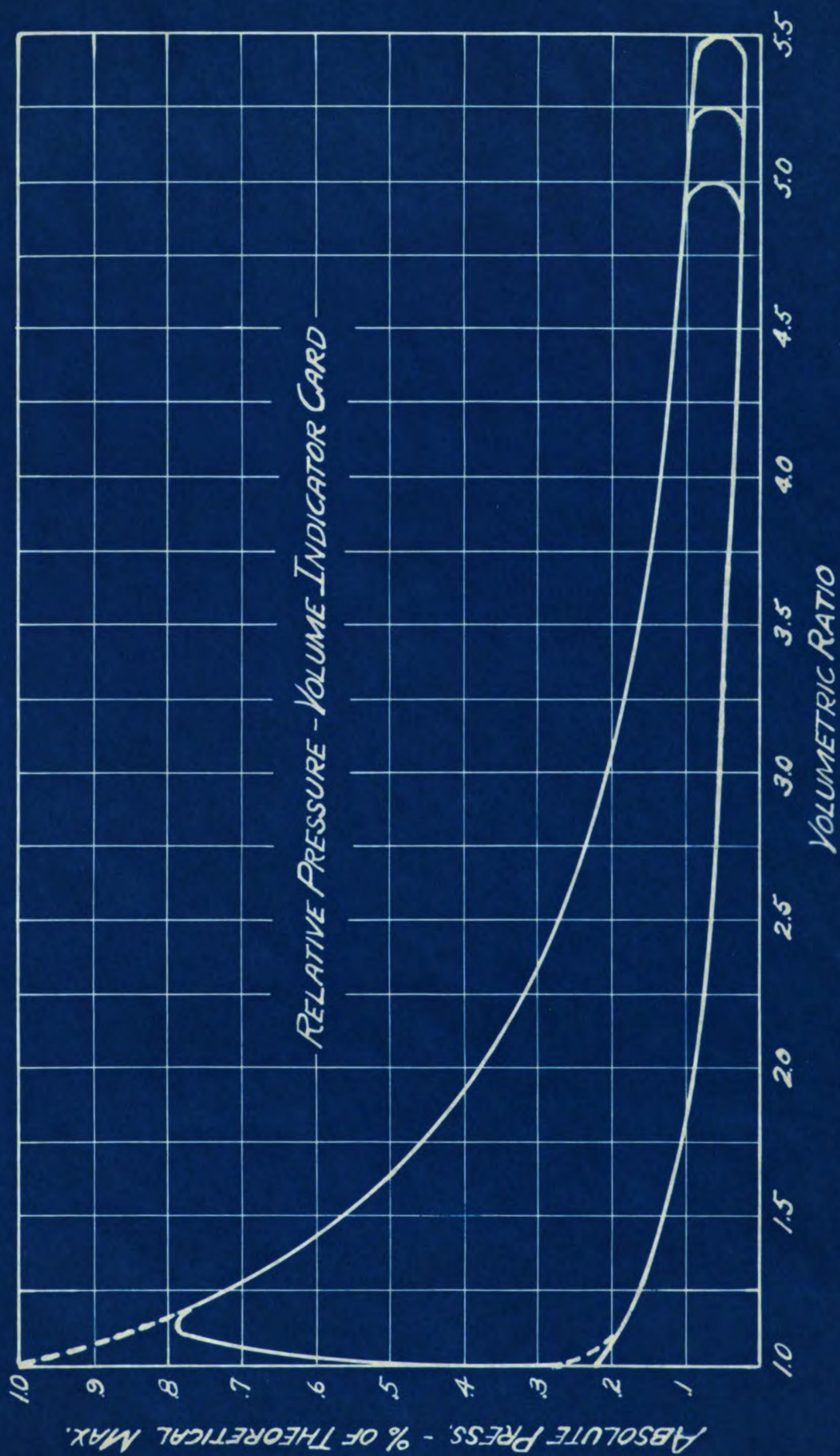
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1. Janeway, R. N. Automotive Industries, Vol. 64
2. Hildt, P. H. Automotive Industries, Vol. 66
3. Purney, Paul Automotive Industries, Vol. 65
4. Root, R. E. Dynamics of Engine and Shaft.

THEORETICAL
INDICATOR DIAGRAM

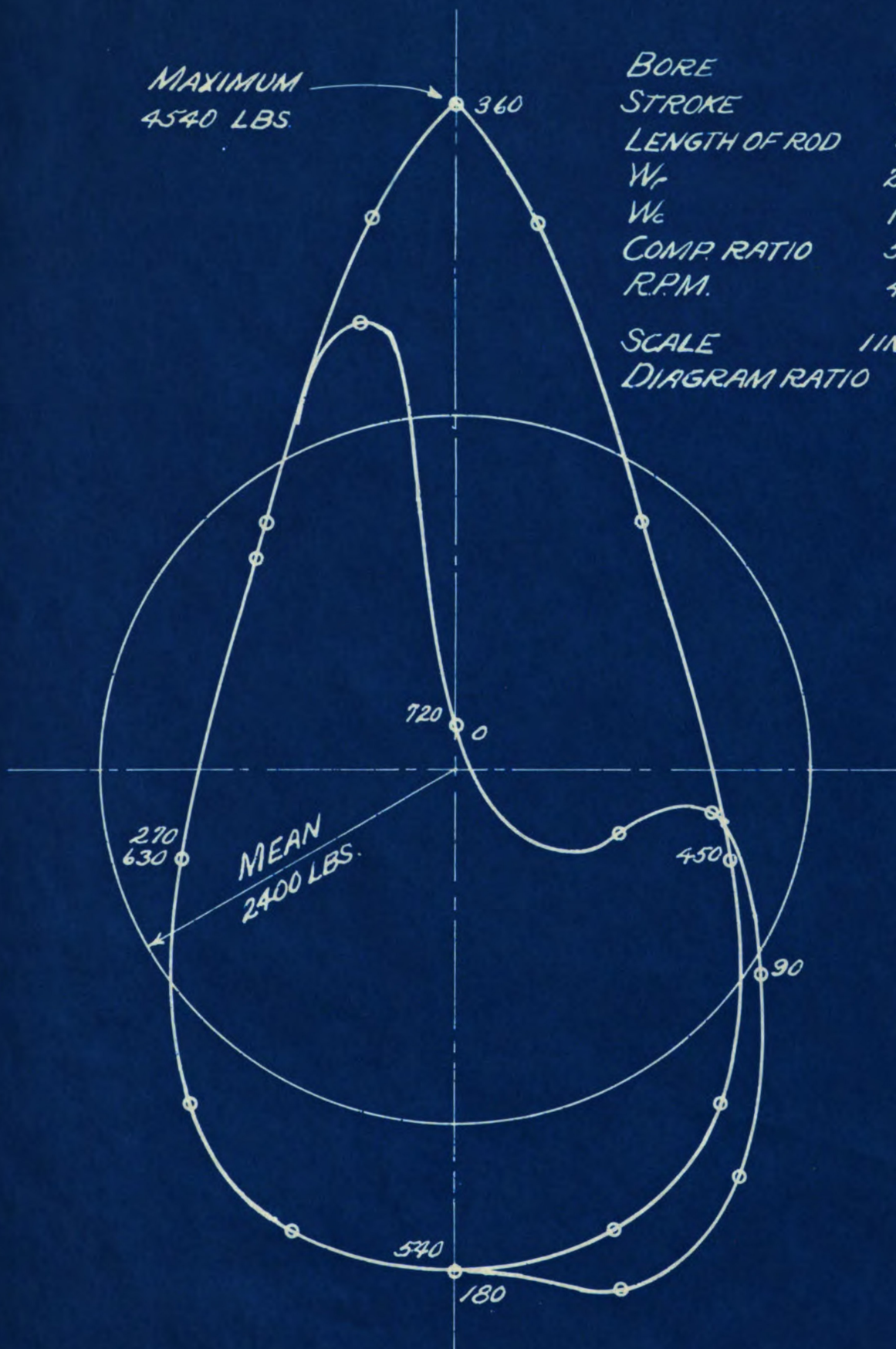
BORE $3\frac{1}{2}$ IN.
STROKE $3\frac{3}{8}$ IN.
LENGTH OF ROD $8\frac{1}{2}$ IN.
COMP. RATIO 5.32-1
(r) OF COMP. 1.3
(r) OF EXPAN. 1.4
 P_{max} 4.65 P_c
 P_{max} (ACTUAL) .79 P_{max}





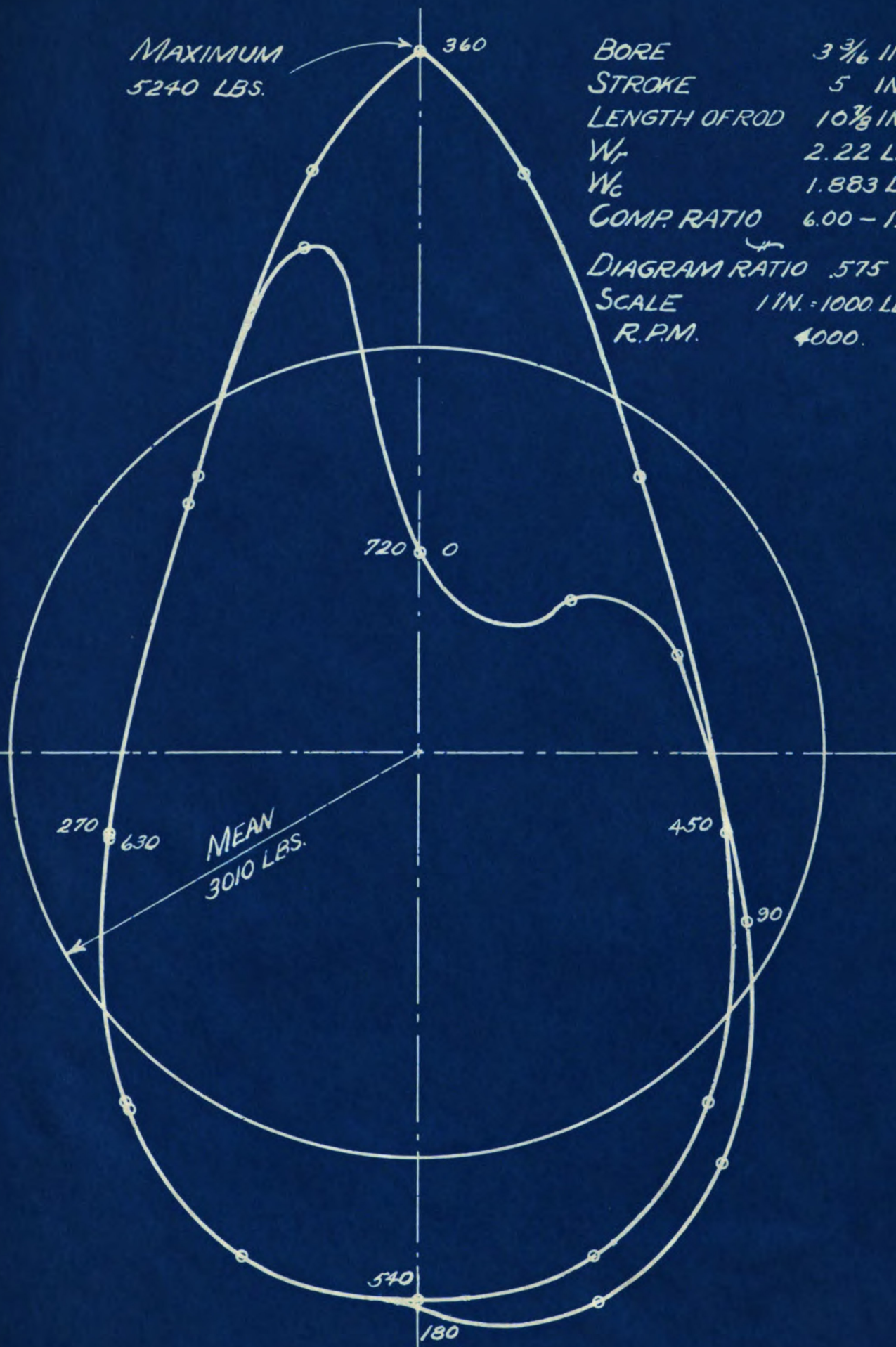
MAXIMUM
4540 LBS.

BORE $3\frac{1}{2}$ IN.
STROKE $4\frac{1}{4}$ IN.
LENGTH OF ROD $8\frac{1}{4}$ IN.
 W_r 2.34 LBS.
 W_c 1.77 LBS.
COMP. RATIO 5.75 - 1
R.P.M. 4000.
SCALE 1 IN. = 1000 LBS.
DIAGRAM RATIO .527



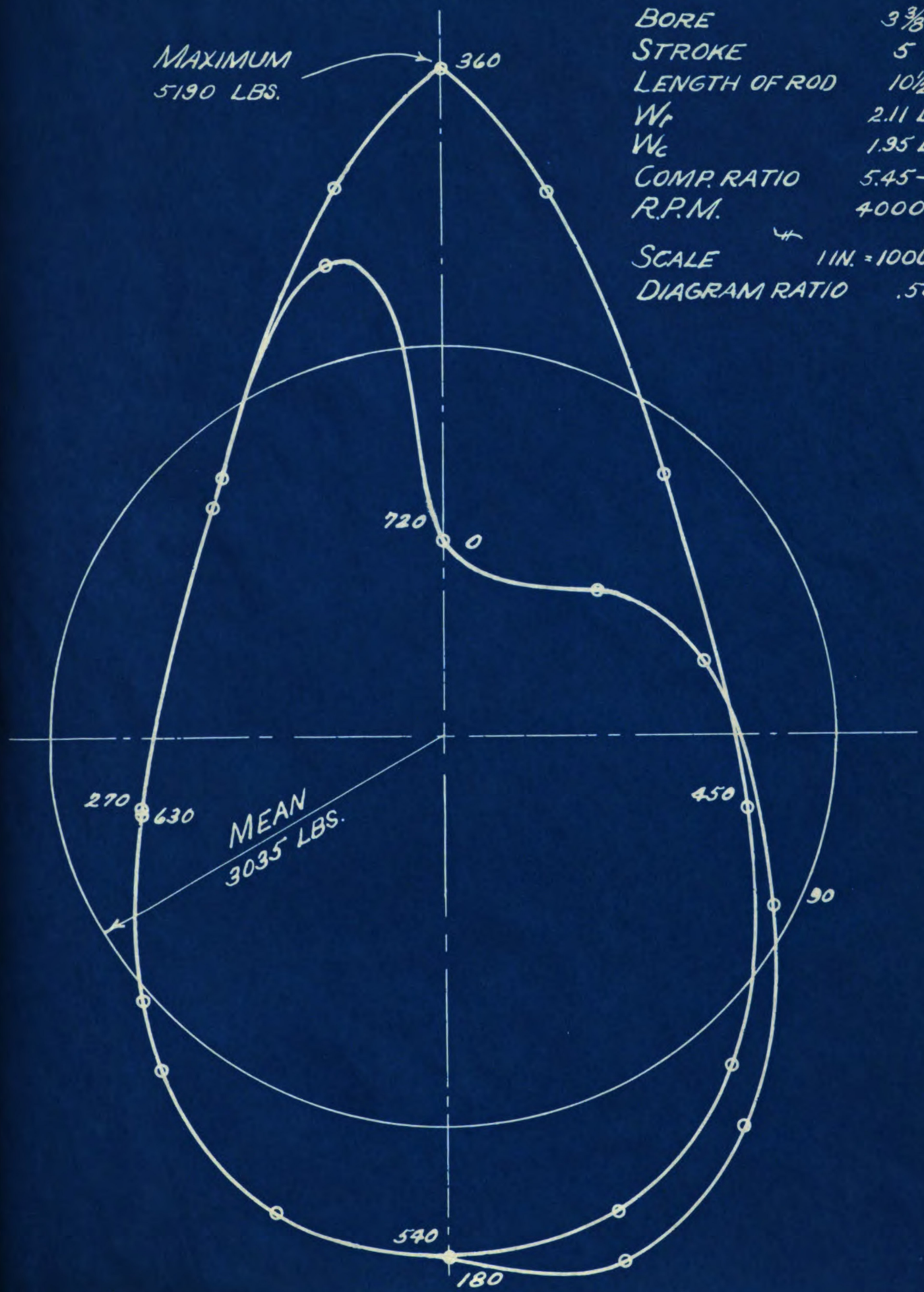
MAXIMUM
5240 LBS.

BORE $3\frac{3}{8}$ IN.
STROKE 5 IN.
LENGTH OF ROD $10\frac{7}{8}$ IN.
 W_r 2.22 LBS.
 W_c 1.883 LBS.
COMP. RATIO 6.00-1.
DIAGRAM RATIO .575
SCALE 1 IN. = 1000 LBS.
R.P.M. 4000.



MAXIMUM
5190 LBS.

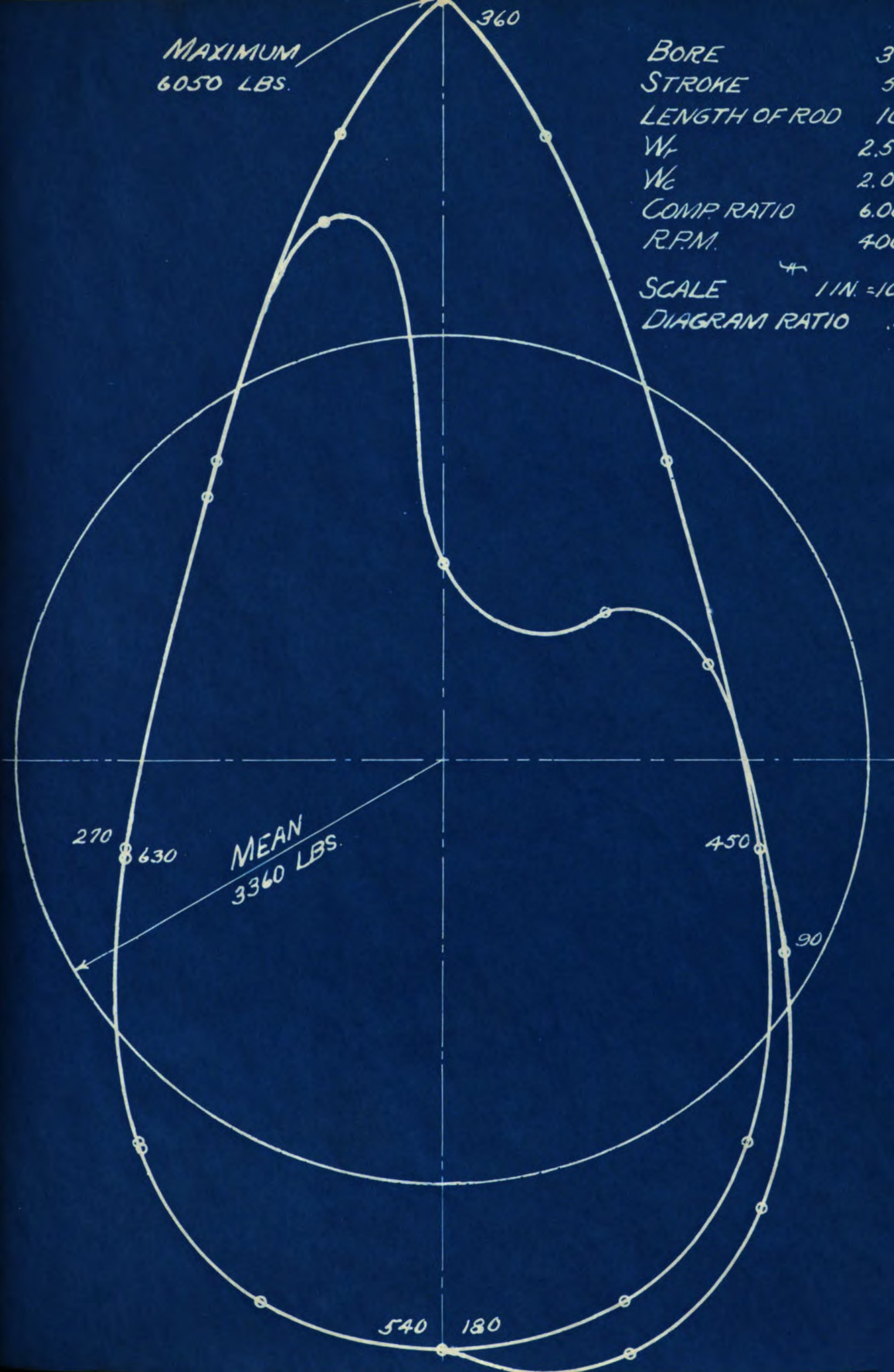
BORE $3\frac{3}{8}$ IN.
STROKE 5 IN.
LENGTH OF ROD $10\frac{1}{2}$ IN.
 W_f 2.11 LBS.
 W_c 1.95 LBS.
COMP. RATIO 5.45-1
R.P.M. 4000.
SCALE $1\text{ IN.} = 1000\text{ LBS.}$
DIAGRAM RATIO .585

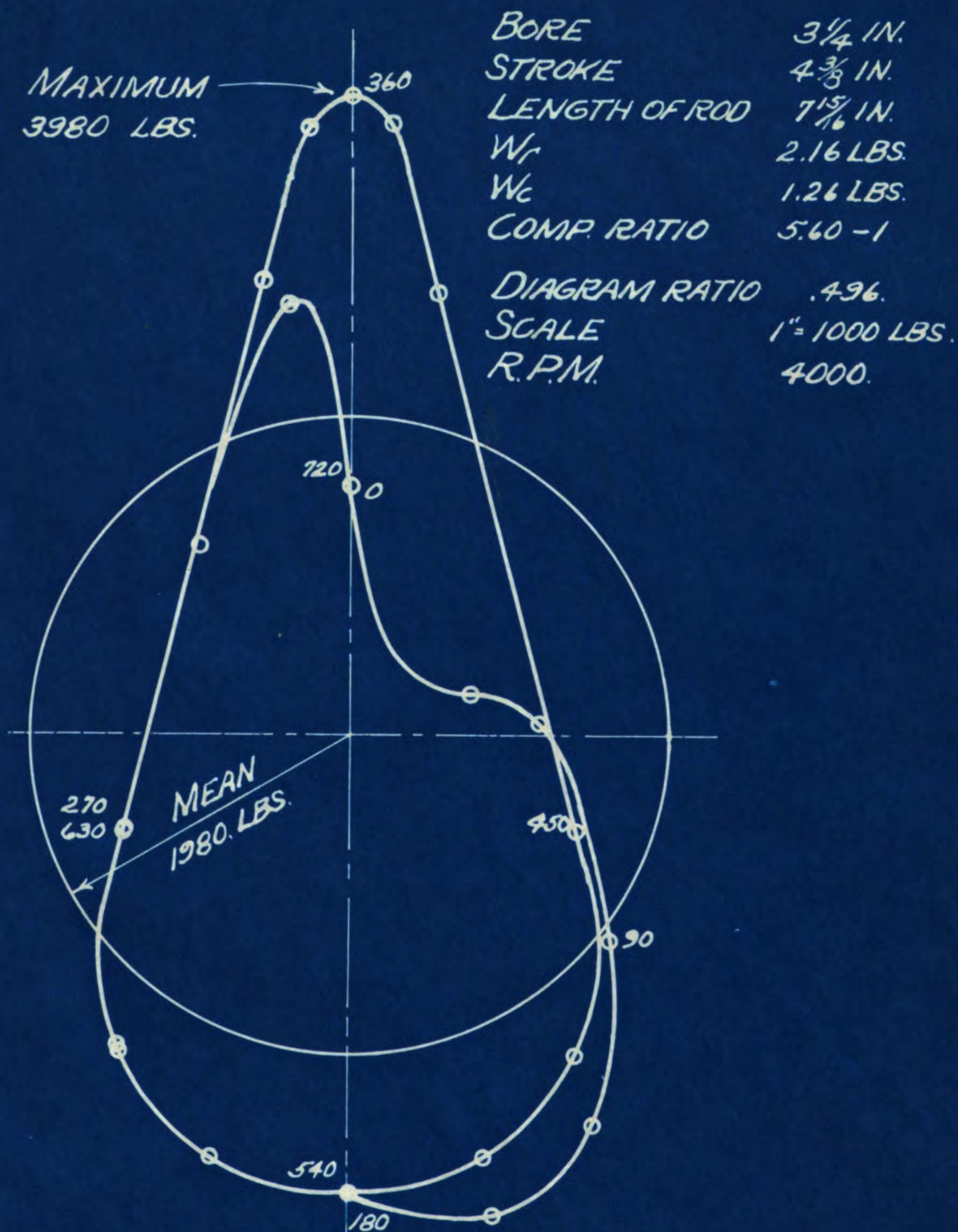


MAXIMUM
6050 LBS.

BORE $3\frac{1}{2}$ IN.
STROKE 5 IN.
LENGTH OF ROD $10\frac{7}{8}$ IN.
 W_f 2.56 LBS.
 W_c 2.05 LBS.
COMP. RATIO 6.00 - 1
R.P.M. 4000.

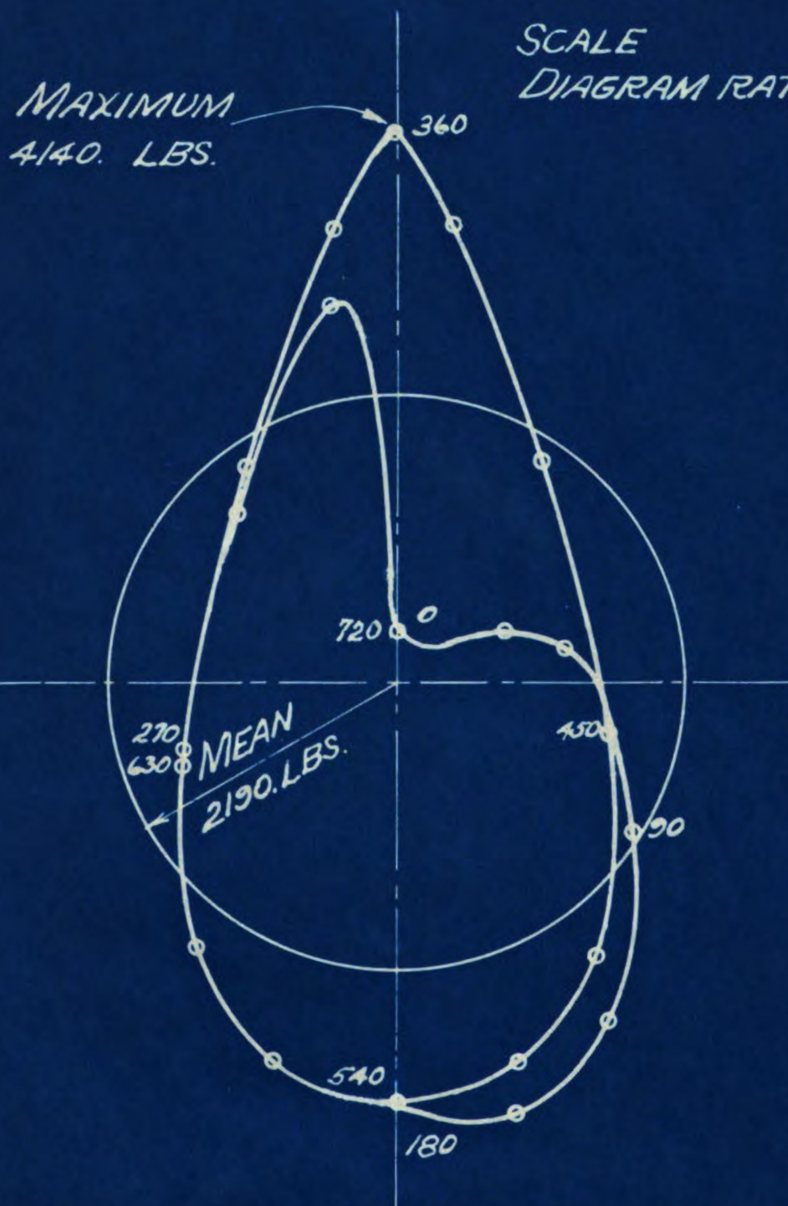
SCALE $\frac{1}{4}$ 1 IN. = 1000 LBS.
DIAGRAM RATIO .555



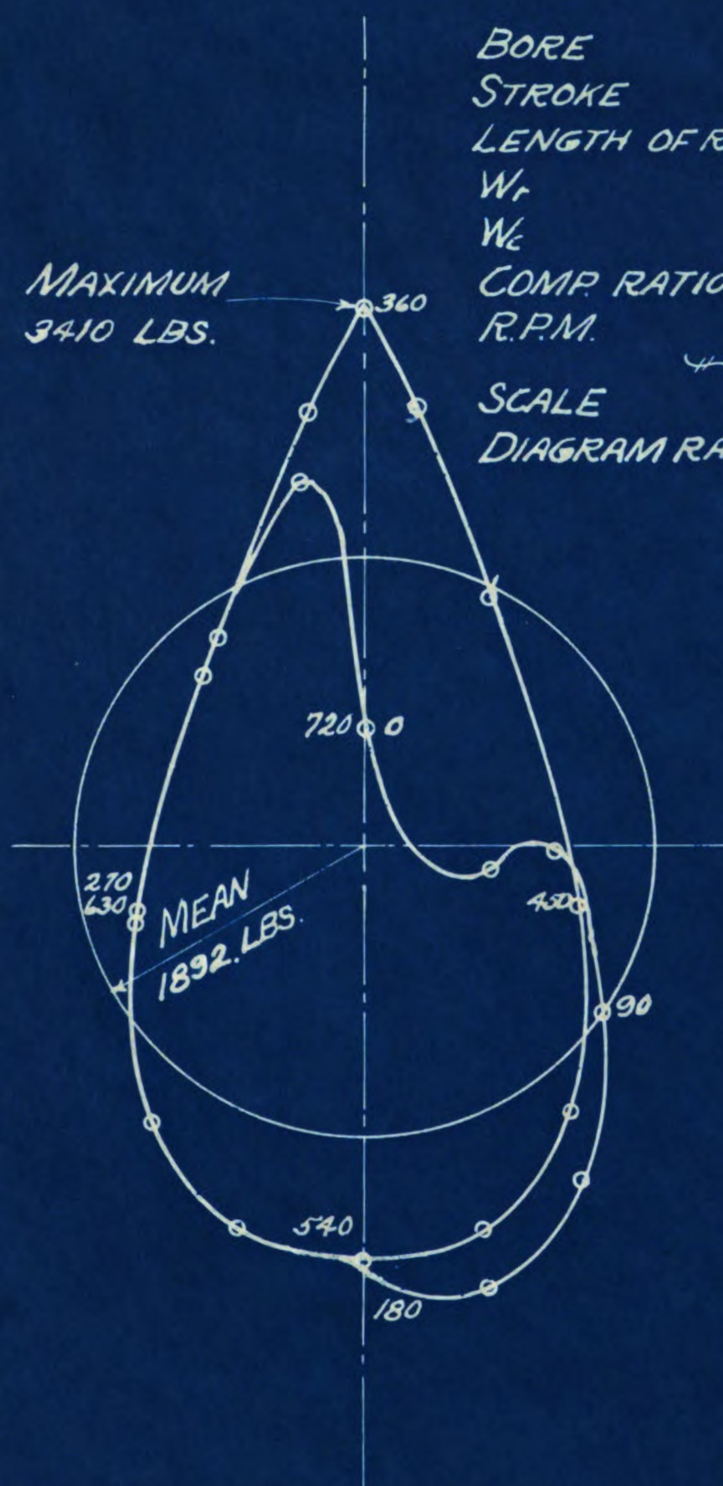


BORE	3 1/4 IN.
STROKE	4 1/8 IN.
LENGTH OF ROD	8 1/4 IN.
W_r	1.92 LBS.
W_c	1.56 LBS.
COMP. RATIO	6.3 - 1
R.P.M.	4000.

SCALE	1 IN. = 1415 LBS.
DIAGRAM RATIO	.529



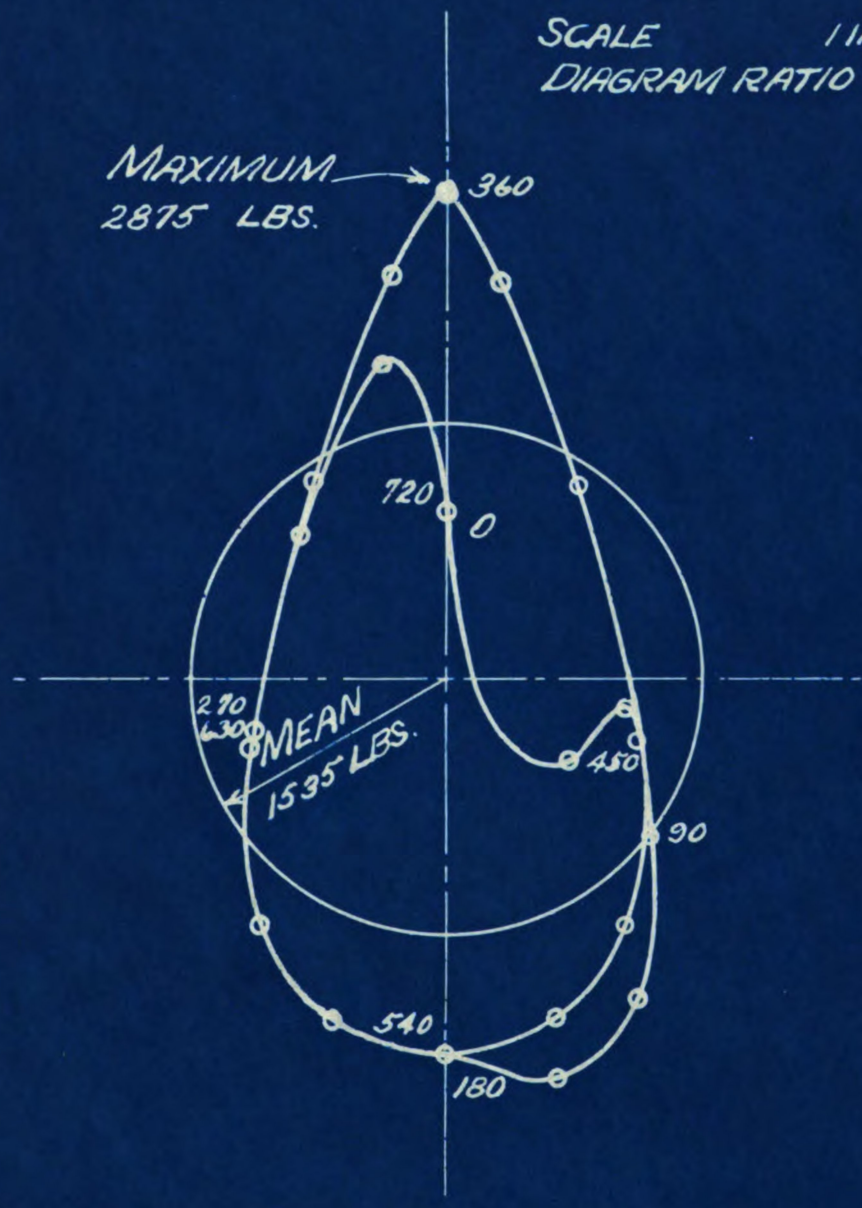
MAXIMUM
3410 LBS.



BORE $3\frac{1}{16}$ IN.
 STROKE $4\frac{1}{4}$ IN.
 LENGTH OF ROD 8 IN.
 W_r 1.49 LBS.
 W_c 1.33 LBS.
 COMP. RATIO 6.3 - 1
 R.P.M. 4000.
 SCALE 1 IN. = 1210 LBS.
 DIAGRAM RATIO .555

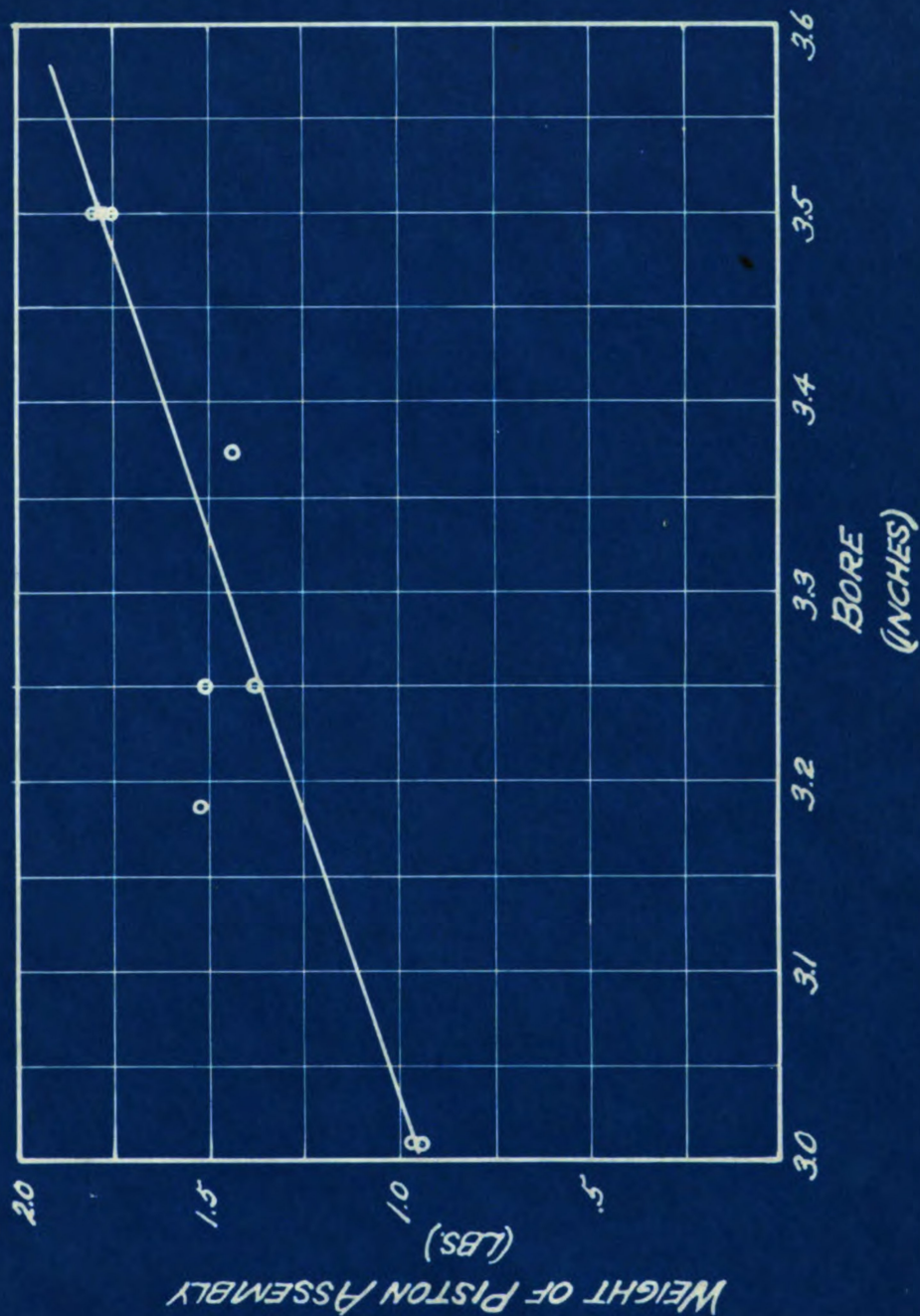
BORE $3\frac{1}{16}$ IN.
 STROKE $3\frac{3}{4}$ IN.
 LENGTH OF ROD $8\frac{1}{4}$ IN.
 W_f 1.45 LBS.
 W_c 1.30 LBS.
 COMP RATIO 6.3-1
 R.P.M. 4000.

SCALE 1 IN. = 1118 LBS.
 DIAGRAM RATIO .534

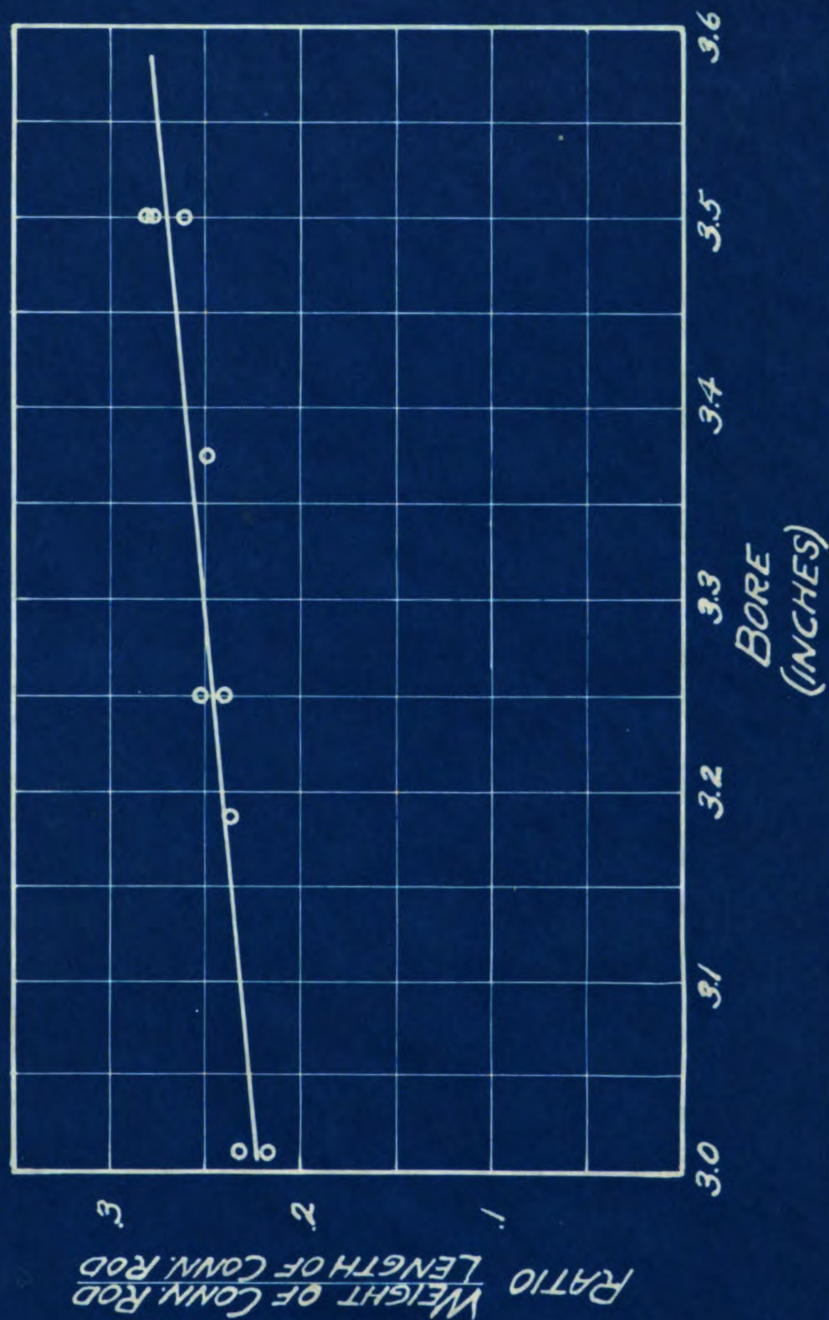


PISTON WEIGHT GRAPH

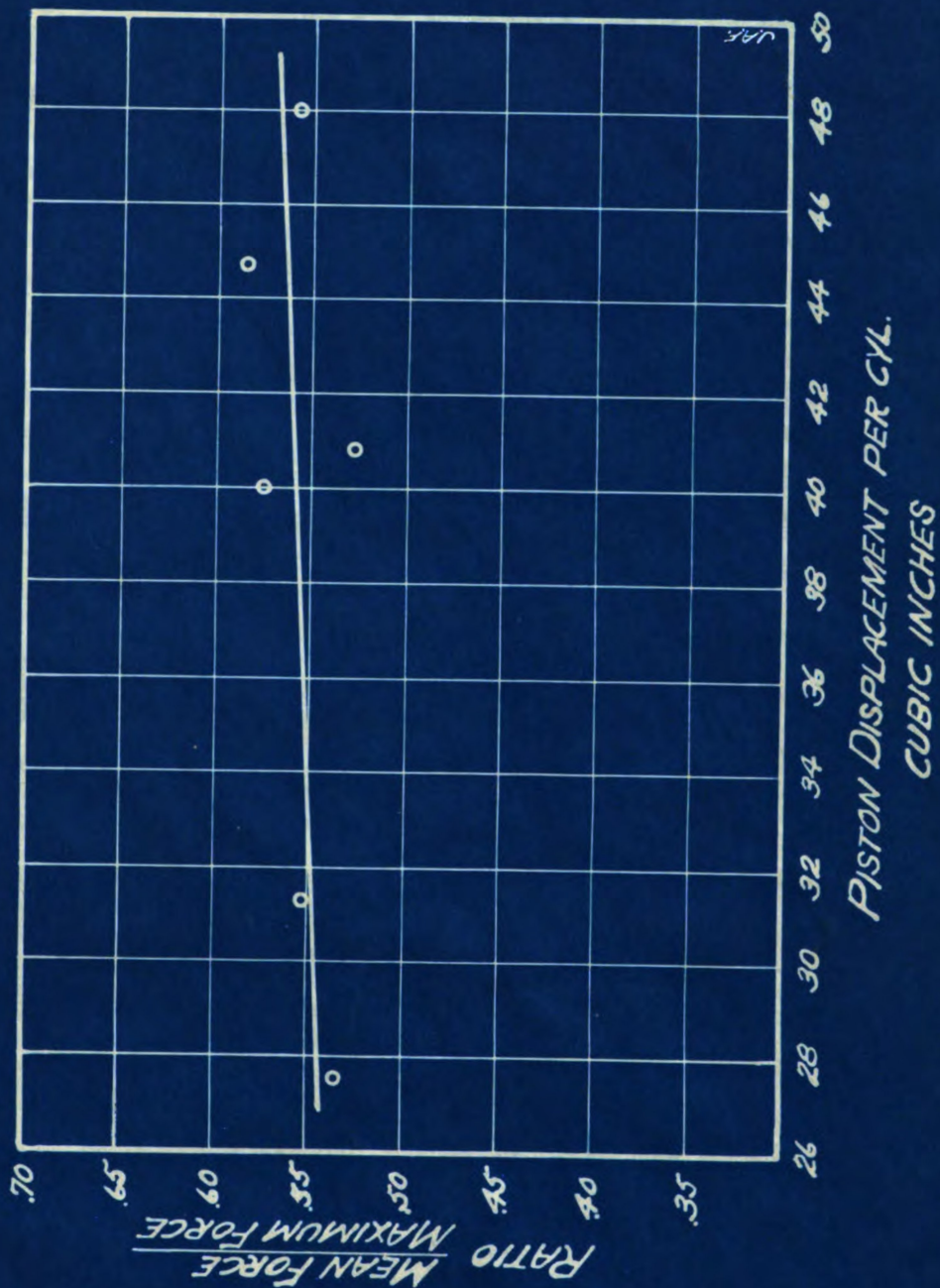
(688 CYLINDER ENGINES)

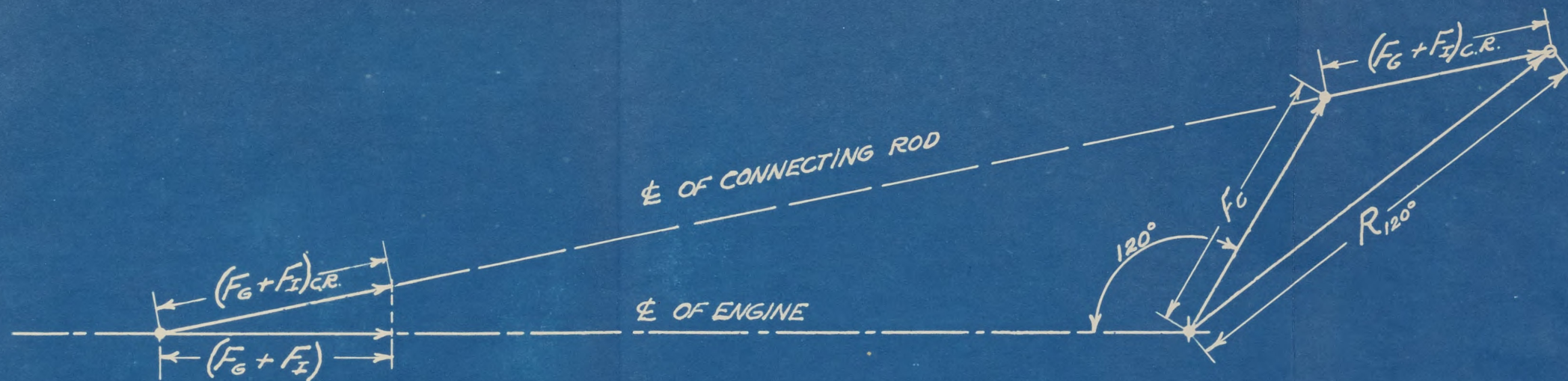


CONNECTING ROD RATIO GRAPH
(688 CYLINDER ENGINES)



FORCE RATIO GRAPH 8 CYLINDER ENGINES



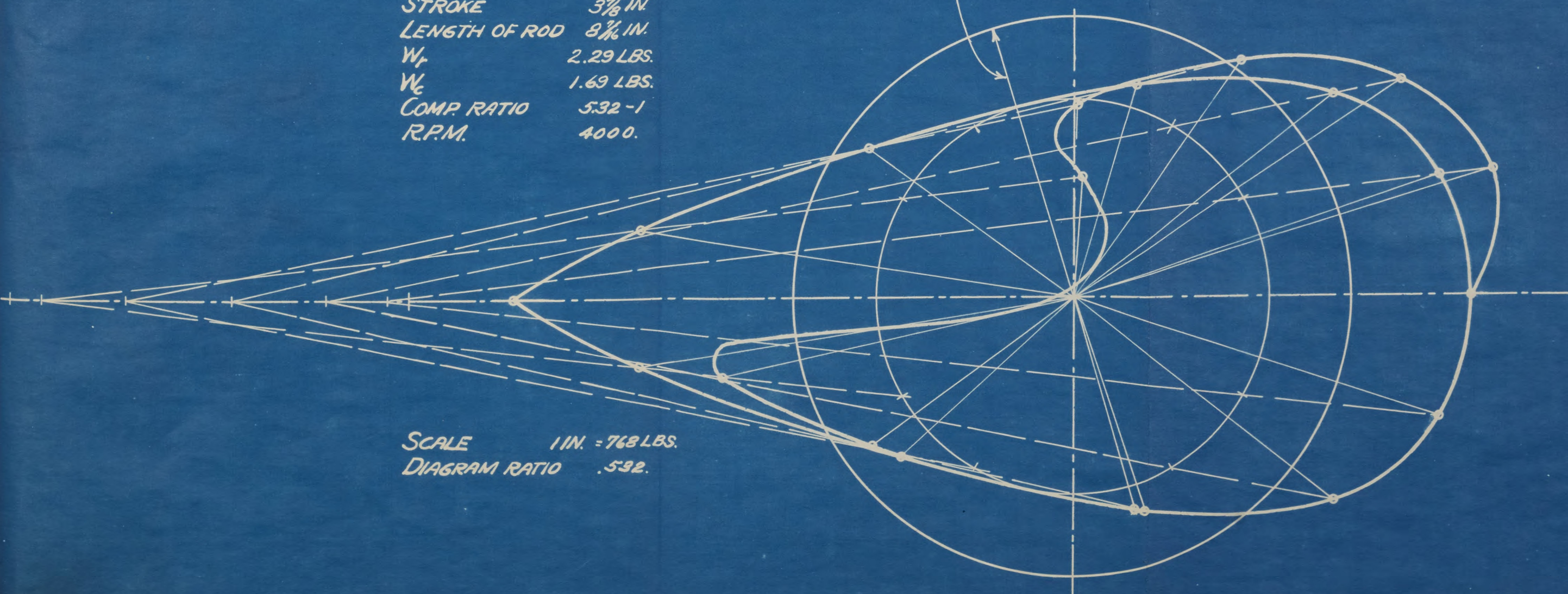


FORCE DIAGRAM

BORE	3 1/2 IN.
STROKE	3 7/8 IN.
LENGTH OF ROD	8 1/4 IN.
W_f	2.29 LBS.
W_c	1.69 LBS.
COMP. RATIO	5.32-1
R.P.M.	4000.

MEAN CIRCLE
R = 2110 LBS.

SCALE 1 IN. = 768 LBS.
DIAGRAM RATIO .592.



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