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IN-LINE AIRCRAFT-ENGINE BEARING LOADS

I - CRANKPIN-BEARING LOADS

By Milton C. Shaw and E. Fred Macks

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NACA

WASHINGTON

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

ADVANCE RESTRICTED REPORT

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SUMMARY

A method of generalizing the results of a relatively few conventional bearing-load analyses has been developed. When dimensional analysis is applied, a smooth curve is obtained when the quantity $(\text{rpm})^2/(\text{imep})$ is plotted against (bearing load at a particular crank angle)/(imep). Such a family of curves, using crank angle as the parameter, may be obtained from the results of as few as six conventional bearing-load analyses. This family of curves enables charts to be constructed that give the maximum and mean crankpin-bearing loads for any combination of engine speed and indicated mean effective pressure. This method is applicable to the principal bearings of both in-line and radial engines.

From an application of this analysis to crankpin-bearing loads of V-type engines, it was found that optimum combinations of engine speed and indicated mean effective pressure exist for which mean and maximum crankpin-bearing loads are minima for a given power; such combinations lie in the practical region of operation. The best dive throttle setting with regard to crankpin-bearing load is one that will produce an indicated mean effective pressure slightly less than that corresponding to this optimum combination of speed and indicated mean effective pressure at dive speed. At a given power level the optimum maximum crankpin-bearing load varies directly with the compression ratio. The ratio of connecting rod length to crank throw does not appreciably influence the mean or maximum crankpin-bearing load.

Charts are presented from which the maximum and the mean crankpin-bearing loads for a production, 12-cylinder, V-type engine can be determined for all values of engine speed to 5000 rpm and for all values of indicated mean effective pressure to 500 pounds per square inch. The maximum crankpin-bearing load for this engine

is shown to occur in the crank-angle region of 20°, 120°, or 680° depending upon the relative values of engine speed and indicated mean effective pressure employed.

INTRODUCTION

An exact knowledge of the instantaneous magnitude and direction of the load acting upon aircraft-engine bearings is important, particularly when an attempt is being made to increase the power output of an engine. The effect of an increased power level on the maximum load, mean load, rubbing velocity, and other criteria of bearing operation is an important consideration for the engine designer. In order to determine this effect, the various bearing loads must be analyzed and computed.

The fundamental method of determining bearing loads for internal-combustion engines by means of polar diagrams was introduced by Burkhardt in 1919. (See references 1 and 2.) This method of analysis provides for combining, at specific crank-angle positions, the vectors of the gas force and the inertia forces of the reciprocating and rotating masses. The principles of Burkhardt's method were standardized in 1923 by Caminez and Iseler. Examples of the application of Burkhardt's method of analysis to aircraft-engine bearings were published in references 3 and 4.

A few attempts have been made to simplify the computation of aircraft-engine bearing loads. In 1931 Janeway (reference 5) proposed a simplified method of obtaining the mean load acting upon a crankpin bearing. The horizontal and vertical bearing-load components were averaged to obtain a mean resultant for a selected crank-angle interval. The mean bearing load was considered to be equal to the average of the individual resultant values. Samuels (reference 6) described in 1935 a simplified method of determining the mean crankpin-bearing load. Prescott and Poole (reference 4) also presented a simplified computation of bearing loads.

At the request of the Air Technical Service Command, Army Air Forces, an investigation was conducted at the NACA laboratory at Cleveland from September 1943 to March 1944 to develop a general method of determining bearing loads. A number of bearing loads were determined by Burkhardt's method, the results of which led to the development of a general solution for determining bearing loads over a wide range of engine conditions. The analyses and the generalized method are described herein, and the computation of maximum and mean crankpin-bearing loads for a production V-type engine is presented to demonstrate this method.

THEORY

The dimensional method of reasoning employed by Buckingham (reference 7) has been used in other fields of research will considerable success. The first step in applying this technique to the determination of the loads acting in an in-line V-type aircraft engine, is to list all the variables. These variables are tabulated as follows:

Symbol	Variable	Dimensional formula	Relation
N	Engine speed, rpm	T^{-1}	Independent
p	Indicated mean effective pressure, pounds per square inch	FL^{-2}	Independent
L_S	Stroke, inches	L	Independent
W	Crankpin-bearing load, pounds	F	π_a
M_1	Reciprocating mass per crankpin, slugs	FT^2L^{-1}	π_b
M_c	Rotating mass per crankpin, slugs	FT^2L^{-1}	π_c
D	Diameter of bore, inches	L	π_d
L_R	Length of connecting rod, inches	L	π_e
p_m	Manifold pressure, pounds per square inch absolute	FL^{-2}	π_f
r	Compression ratio	None	π_g
θ	Crank angle, degrees	None	π_h
γ	Angle between cylinder center lines, degrees	None	π_i

When the three independent variables are speed, indicated mean effective pressure, and stroke, all the unknown, nondimensional π quantities may be expressed in terms of the independent variables by the conventional method. The results are as follows:

$$\pi_a = \frac{W}{L_S{}^2 p}$$

$$\pi_b = \frac{M_1 N^2}{L_S p}$$

$$\pi_c = \frac{M_c N^2}{L_S p}$$

$$\pi_d = \frac{D}{L_S}\ .$$

$$\pi_e = \frac{L_R}{L_S}$$

$$\pi_f = \frac{p_m}{p}$$

When Buckingham's π theorem is applied (explained and simplified in reference 8), the following equation is obtained:

$$W = L_S{}^2 p\Omega\left[\frac{M_r N^2}{L_S p}, \frac{M_c N^2}{L_S p}, \frac{D}{L_S}, \frac{L_R}{L_S}, \frac{p_m}{p}, \theta, r, \gamma\right] \tag{1}$$

where Ω is some function of the several nondimensional π quantities. If the indicated mean effective pressure is assumed to be proportional to the manifold pressure, equation (1) simplifies to the following expression for a specific engine:

$$W = p\Omega'\left[\frac{N^2}{p}, \theta\right] \tag{2}$$

Equation (2) establishes the fact that, if W/p is plotted against N^2/p at a constant value of crank angle, a smooth curve will be obtained. Equation (2) is applicable to the principal bearings of both in-line and radial engines.

CONVENTIONAL COMPUTATION OF CRANKPIN-BEARING LOADS

The significance of equation (2) will be demonstrated by applying it to the analysis of the maximum and the mean crankpin-bearing loads of a V-type engine under various engine speeds and indicated mean effective pressures. In order to apply equation (2) to a specific engine and thereby obtain values of W/p, the bearing loads for a number of representative engine operating conditions must be computed. These computations are made in the usual manner prior to generalization.

The symbols and conventions used hereinafter are defined in the preceding section and in figure 1. Specifications for the V-type engine investigated herein are given in the appendix, and sketches of the connecting-rod, blade-bearing, and crankpin-bearing arrangement are shown in figures 2 and 3. Throughout this report, a crank angle θ of 0° refers to the top-center position of cylinder 1L at the beginning of the expansion stroke.

Power conditions. - The specific conditions of speed and indicated mean effective pressure used for application of the analysis are given in the following table. The intake-manifold pressure was assumed to increase proportionately with indicated mean effective pressure.

Power condition	Engine speed (rpm)	Engine speed (percent rated)	imep (lb/sq in.)	imep (percent rated)	Intake-manifold pressure (in. Hg absolute)	ihp
1	3000	100	182	75	39	1170
2	3000	100	242	100	52	1570
3	3000	100	303	125	65	1960
4	3000	100	363	150	78	2350
5	3300	110	242	100	52	1720
6	3600	120	242	100	52	1880

Method and construction of diagrams. - Standard cycle diagrams were constructed according to the method given in reference 3 for each of the foregoing conditions using an exponent of 1.30 for both the expansion and the compression curves. An indicator-diagram factor of 0.90 was employed, and the maximum gas pressure was made 75 percent of the computed maximum. A representative indicator diagram for power condition 2 (in the foregoing table) is given in figure 4.

The gas force at any crank angle is equal to the product of the corresponding value of the pressure from the indicator diagram and the piston area. The reciprocating inertia force at any crank angle is equal to the product of the reciprocating mass and the piston acceleration. Values of acceleration may be found in Smith's compilation of piston accelerations (reference 9). The resultant load acting on the piston pin parallel to the cylinder axis is obtained by algebraically combining the gas force and the reciprocating inertia force. This resultant load, when multiplied by the secant of the angle ϕ (fig. 1), gives the force acting along the connecting-rod axis.

The centrifugal force acting on the crankpin may be computed from the values of the rotating mass, the crank throw, and the engine speed. The resultant load acting on the crankpin at any particular crank angle is obtained by the vector addition of the centrifugal force, the force along the fork-rod axis, and the force along the blade-rod axis, account being taken of the firing order of the engine.

A representative polar diagram of forces acting upon the crankpin bearing with respect to the engine axis for power condition 2 (engine speed, 3000 rpm; imep, 242 lb per sq in.) is shown in figure 5. The three individual vectors constituting the resultant crankpin load at a crank angle of 20° are shown to illustrate the method of vector addition.

A polar diagram with respect to the crank axis is more useful than a diagram with respect to the engine axis in defining loads acting on the crankpin. Polar diagrams with respect to the crank axis may be obtained by rotating each resultant vector of figure 5 counter to the direction of rotation through an angle corresponding to the number of crank-angle degrees indicated at the terminal end of the vector. The polar diagrams with respect to the crank axis, for each of the six power conditions, are given in figures 6 and 7 in terms of crank-angle degrees.

The polar diagrams with respect to the fork-rod axis are also of interest with regard to loads acting on the bearing shell. These diagrams are obtained by rotating the diagram with respect to the crank axis in the direction of crankshaft rotation through an angle of 180° + α_1, where α_1 is the angle defined in figure 1. The polar diagrams with respect to the fork-rod axis, for each of the six power conditions, are given in figures 8 and 9 in terms of crank-angle degrees.

APPLICATION OF THE DIMENSIONAL-ANALYSIS METHOD

Generalized Load Charts

Maximum bearing loads. - The resultant maximum crankpin forces shown in figures 5 to 9 can be generalized by means of equation (2). If W/p is plotted against N^2/p for each of the six power conditions at constant values of crank angle (fig. 10), the maximum values of W/p occur at crank angles of approximately 20°, 120°, or 680°. Additional points were computed for the crank angles of 20°, 120°, and 680° in order to extend these curves beyond the region covered by the six power conditions. The solid portion of each curve corresponds very closely to the maximum value of W/p over the particular range of N^2/p concerned. All plots of W/p against N^2/p are portions of hyperbolic-type curves. The solid portions of each of the three curves lie sufficiently far from their respective vertices to be considered linear.

A convenient chart (fig. 11(a)) for determining maximum crankpin loads is obtained from the curves of figure 10. In order to facilitate visualization of figure 11(a), a topographic representation of the surface defined by the load-contour lines of this figure is presented in figure 11(b). Constant indicated-horsepower curves have been included for convenience. The line OA represents the locus of optimum combinations of speed and indicated mean effective pressure for which the maximum bearing load at a given power level is a minimum.

The use of the constant load chart is illustrated in the following example. The bearing loads for three combinations of speed and indicated mean effective pressure producing 2000 indicated horsepower are given in the following table. The first column corresponds to a point on the line OA (fig. 11) for an optimum combination of speed and indicated mean effective pressure. Values of maximum load for indicated mean effective pressures 10 percent above and 10 percent below the optimum are given in columns 2 and 3, respectively.

	1	2	3
Indicated mean effective pressure, pounds per square inch	292	321	263
Engine speed, rpm	3,180	2,890	3,520
Maximum bearing load, pounds	17,400	21,650	20,600
Maximum unit bearing load[a], pounds per square inch	2,995	3,730	3,545

[a]The effective projected bearing area was taken as 5.81 square inches.

Mean bearing loads. - The mean load $\bar{W}$ acting on the crankpin can be determined from a rectangular plot of load W against crank angle using a planimeter to obtain the average height of the curve. The resultant gas force, the resultant inertia force, and the resultant total force W are plotted in figure 12 for an engine speed of 3000 rpm and an indicated mean effective pressure of 242 pounds per square inch.

The results of the dimensional treatment were again utilized to generalize the mean-load analysis. In figure 13, $\bar{W}/p$ is plotted against N^2/p. The equation of the straight line representing the plotted points is:

$$\frac{\bar{W}}{p} = 965 \times 10^{-6} \left(\frac{N^2}{p}\right) + 13.2 \tag{3}$$

Equation (3) may be used to obtain the mean crankpin-bearing loads for all practical combinations of engine speed and indicated mean effective pressure.

A useful chart is obtained from equation (3) by plotting engine speed against indicated mean effective pressure for constant values of mean load. Such a family of curves is given in figure 14. The optimum-maximum-load curves shown for four compression ratios were included to permit comparison with each other and with the mean-load curves and will be discussed in the following section under the heading Compression ratio. Constant indicated-horsepower curves have been included for convenience.

Optimum combinations of indicated mean effective pressure and engine speed. - Families of constant indicated-horsepower curves (fig. 15) for mean and maximum crankpin-bearing loads were obtained from figures 11 and 14. (Points beyond the range of these charts were obtained from figs. 10 and 13.) The loci of optimum combinations of engine speed and indicated mean effective pressure for maximum and mean crankpin-bearing loads are represented by the lines CC and DD, respectively.

For a given indicated horsepower, the optimum maximum-load and the optimum mean-load combinations of speed and indicated mean effective pressure differ considerably. The optimum mean-load combinations (line OC of fig. 14) fall in an impracticable operating region. It is possible to approach this optimum mean-load condition only by operating at low engine speed and high indicated mean effective pressure.

A closed throttle setting in a dive is therefore desirable from a consideration of only the mean bearing load. The rate of change of maximum bearing load with indicated mean effective pressure at constant speed is large for points above line OA of figure 11 but negligible for points below OA. The indicated mean effective pressure will therefore affect the maximum bearing load very little if the point representing the dive speed and the indicated mean effective pressure lies below line OA. Inasmuch as difficulty is frequently experienced owing to increased oil pumping during a dive with closed throttle, it appears advisable to operate with the throttle partly closed in order that the combination of indicated mean effective pressure and speed lies close to, but below, line OA. Although such a throttle setting will give a mean load greater than the optimum, it appears to be the best solution with regard to crankpin-bearing loads.

Rubbing factor. - The "rubbing factor," as defined by Prescott and Poole in reference 4 (p. 310), may be obtained for any combination of speed and indicated mean effective pressure from the value of the mean load obtained from figure 14 and the equation

$$F = \left(2.25 \times 10^{-3}\right) N\bar{W}$$

where

F rubbing factor, (ft-lb)/(sq in.)(sec)

N engine speed, rpm

$\bar{W}$ mean crankpin-bearing load, pounds

Although the rubbing factor is not generally considered a good criterion for the severity of bearing operating conditions, it is given for what it may be worth.

Verification of the Generalized Load Charts

As a means of checking the results of the generalized analysis, a polar diagram was constructed employing an extreme combination of engine speed and indicated mean effective pressure: engine speed of 3600 rpm and indicated mean effective pressure of 182 pounds per square inch. The resulting polar diagram is given in figure 16.

The maximum load in figure 16 is 20,100 pounds at a crank angle of 130°. The maximum load from figure 11 is 19,600 pounds at a crank angle of approximately 120°. This close agreement of values is considered an excellent check of the accuracy of the computations as well as of the generalization method.

Effect of Engine Dimensions upon Crankpin-Bearing Loads

The bearing-load charts presented herein are directly applicable only to an engine having the dimensions given in the appendix. Attempts to make the load charts applicable to any in-line engine have not been entirely successful because no simple method has been found by which a change in the magnitude of the reciprocating and rotating weights may be taken into consideration by the application of dimensional analysis. The changes in the load charts brought about by changes in the connecting-rod length and the compression ratio, however, have been determined.

Connecting-rod length. - The connecting-rod length plays an unimportant part in the development of bearing loads. This length affects only the ratio of the connecting-rod length to the crank throw, which in turn influences the acceleration of the piston and thus the magnitude of the reciprocating force. Practical values of the ratio of the connecting-rod length to the crank throw for in-line aircraft engines might be considered to lie in the range from 3 to 4. It has been found that changes in the rod length-crank throw ratio within this range have no measureable effect upon the load magnitudes given in the polar diagrams.

Compression ratio. - The compression ratio affects the shape of the indicator diagram and therefore the gas force developed in the engine cylinder. The influence of compression ratio upon gas force during the exhaust stroke, the intake stroke, and most of the compression stroke is quite small. The compression ratio has a considerable effect upon the gas force, however, during that portion of the expansion stroke when the piston is near the top-center position.

The compression ratio will influence the mean crankpin-bearing load very little at any given engine speed-power combination inasmuch as the compression ratio affects the gas force significantly only during a small portion of the cycle, and part of this effect is compensatory. The mean-load diagrams shown in figures 13 and 14 are applicable for all values of compression ratio from 5.50 to 8.50.

The maximum crankpin-load curves in figure 11 for a compression ratio of 6.65 are supplemented by figure 17 in which the compression ratios are 5.50, 7.50, and 8.50. The maximum loads may occur in the crank-angle region of 20°, 120°, or 680°, depending upon the value of N^2/p. These figures show that the resultant crankpin-bearing loads in the region of 20° are considerably influenced by compression ratio.

The location of the curves OA, which indicate optimum combinations of engine speed and indicated mean effective pressure, changes with compression ratio. The OA curves for the four compression ratios considered herein were included in figure 14 for comparative purposes. The following table shows the variation of maximum and mean loads with compression ratio at 2000 indicated horsepower for optimum maximum-load combinations of indicated mean effective pressure and engine speed.

	Compression ratio			
	5.50	6.65	7.50	8.50
Engine speed, rpm	2,960	3,180	3,300	3,440
Indicated mean effective pressure, pounds per square inch	313	292	281	270
Optimum maximum load, pounds	16,100	17,400	18,150	19,400
Mean load, pounds	12,600	13,600	14,200	15,000

Even though the mean load is independent of compression ratio for a given combination of indicated mean effective pressure and engine speed, both the optimum maximum and the corresponding mean crankpin-bearing loads are seen to increase directly with compression ratio at a constant power level.

DISCUSSION

The dimensional treatment has enabled a generalization of the findings from a relatively few bearing-load analyses to be extended to any power condition. This dimensional method is applicable to both in-line and radial engines. Although specific numerical values given in this report are applicable only to the production V-type engine herein considered, the qualitative conclusions apply to any in-line, V-type engine.

Two important bearing-operating criteria are bearing running temperature and fatigue strength of the bearing metal. If a bearing tends to overheat, two general remedies are available:

1. The design of the bearing may be changed to increase the rate of oil flow by increasing the oil-inlet pressure, the clearance, the engine speed, or by introducing a circumferential oil groove. Such changes as these may also influence engine operating variables other than oil flow and, therefore, may not offer a satisfactory solution.

2. The bearing operating temperature may be lowered by decreasing the heat generated in the bearing by reducing the mean bearing load.

The bearing-load analysis shows that a considerable decrease in mean bearing load is realized with a reduction of engine speed. If an engine must operate at the optimum combination of speed and indicated mean effective pressure for maximum load at a given power output and overheating of the crankpin bearing develops, little can be done to decrease the heat generated and some means of increasing the rate of heat dissipation must be employed.

If the range of stress is taken as the criterion of fatigue severity, then, as a first approximation, the difference between the maximum and the minimum bearing loads will be a measure of the tendency for a bearing to fail from fatigue. Because the minimum bearing load is close to 0 pound for the wide range of power conditions considered (figs. 6 and 7), the difference between the maximum bearing load and zero bearing load may be considered as representative of the stress range. If failures from crankpin-bearing fatigue are experienced, operation at a combination of indicated mean effective pressure and engine speed corresponding to the optimum line OA in figure 11 or CC in figure 15 would be advantageous.

Representative values of crankpin-bearing operating characteristics obtained from the maximum- and mean-load charts of this report are given in table I.

CONCLUSIONS

A method of computing maximum and mean bearing loads of an aircraft engine that is applicable to both radial and V-type engines was developed by dimensional analysis. From a series of computations using this method of analyzing bearing loads of a V-type engine, the following conclusions were drawn.

For V-type engines:

1. Optimum combinations of engine speed and indicated mean effective pressure exist for which the mean and maximum crankpin-bearing loads are minima for a given power.

2. At a given power level the optimum maximum crankpin-bearing load varies directly with the compression ratio.

3. The ratio of connecting-rod length to crank throw does not appreciably influence the mean or the maximum crankpin-bearing load.

For the production V-type engine herein considered:

1. The combinations of speed and indicated mean effective pressure corresponding to minimum values of mean crankpin-bearing loads lie in an impractical operating region; whereas the maximum-load optimum combinations fall in a practical operating region.

2. The maximum crankpin-bearing load occurs in the crank-angle region of 20°, 120°, or 680° depending upon the relative values of engine speed and indicated mean effective pressure employed.

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APPENDIX - SPECIFICATIONS OF A PRODUCTION,

12-CYLINDER, V-TYPE ENGINE

Number of cylinders	12
Arrangement of cylinders	Two blocks at an angle of 60°
Method of numbering cylinders from antipropeller end, both blocks	1, 2, 3, 4, 5, 6
Firing order	1L, 2R, 5L, 4R, 3L, 1R, 6L, 5R, 2L, 3R, 4L, 6R
Direction of crankshaft rotation, viewing antipropeller end	Counterclockwise
Bore, in.	5.50
Stroke, in.	6.00
Piston area, sq in.	23.75
Rated engine speed at take-off, rpm	3000
Rated indicated mean effective pressure at take-off, lb/sq in.	242
Rated brake mean effective pressure at take-off, lb/sq in.	186
Manifold pressure at take-off, in. Hg absolute	52
Assumed mechanical efficiency at take-off, percent	77
Compression ratio	6.65
Fork-rod length, in.	10.00
Blade-rod length, in.	10.00
Ratio of connecting-rod length to crank throw	3.33
Spark advance, deg B.T.C.:	
Intake	28
Exhaust	34
Valve timing:	
Intake valve opens, deg B.T.C.	48
Intake valve closes, deg A.B.C.	62
Exhaust valve opens, deg B.B.C.	76
Exhaust valve closes, deg B.T.C.	26
Crankpin diameter, in.	2.998
Effective length of crankpin bearing, in.	1.94
Projected area of crankpin bearing, sq in.	5.81
Reciprocating and rotating weights:	
Weight of piston assembly, lb	5.31
Average weight of upper end of blade or fork rod, lb	1.41
Reciprocating weight per cylinder, lb	6.72
Weight of crankpin bearing, lb	2.21
Weight of lower end of fork rod, lb	3.10
Weight of lower end of blade rod, lb	2.60
Rotating weight per crankpin, lb	7.91

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TABLE I - CRANKPIN-BEARING OPERATING CHARACTERISTICS OF THE PRODUCTION

V-TYPE ENGINE HEREIN CONSIDERED

Power condition	Engine speed, N (rpm)	imep, p (lb/sq in.)	ihp	Maximum bearing load[a], W (lb)	Maximum unit-bearing load[b] (lb/sq in.)	Location of maximum bearing load (crank-angle deg)	Mean bearing load[a], $\overline{W}$ (lb)	Mean unit-bearing load[b] (lb/sq in.)	Rubbing factor, RF (ft lb)/(sq in.)(sec)
1	3000	182	1170	14,800	2540	120	11,100	1910	75,000
2	3000	242	1570	15,500	2670	120	11,800	2030	79,600
3	3000	303	1960	19,100	3300	20	12,600	2170	85,000
4	3000	363	2350	25,100	4320	20	13,400	2300	90,400
5	3300	242	1720	18,100	3120	120	13,500	2320	100,000
6	3600	242	1880	21,000	3610	120	15,500	2660	125,000
7	3175	298	2060	17,900	3080	20 and 120	13,900	2390	99,000

[a]The bearing-load data were taken from figures 11 and 14 and deviate slightly from the values shown on the polar diagrams (figs. 5 to 9).

[b]The effective projected bearing area is taken as 5.81 square inches.

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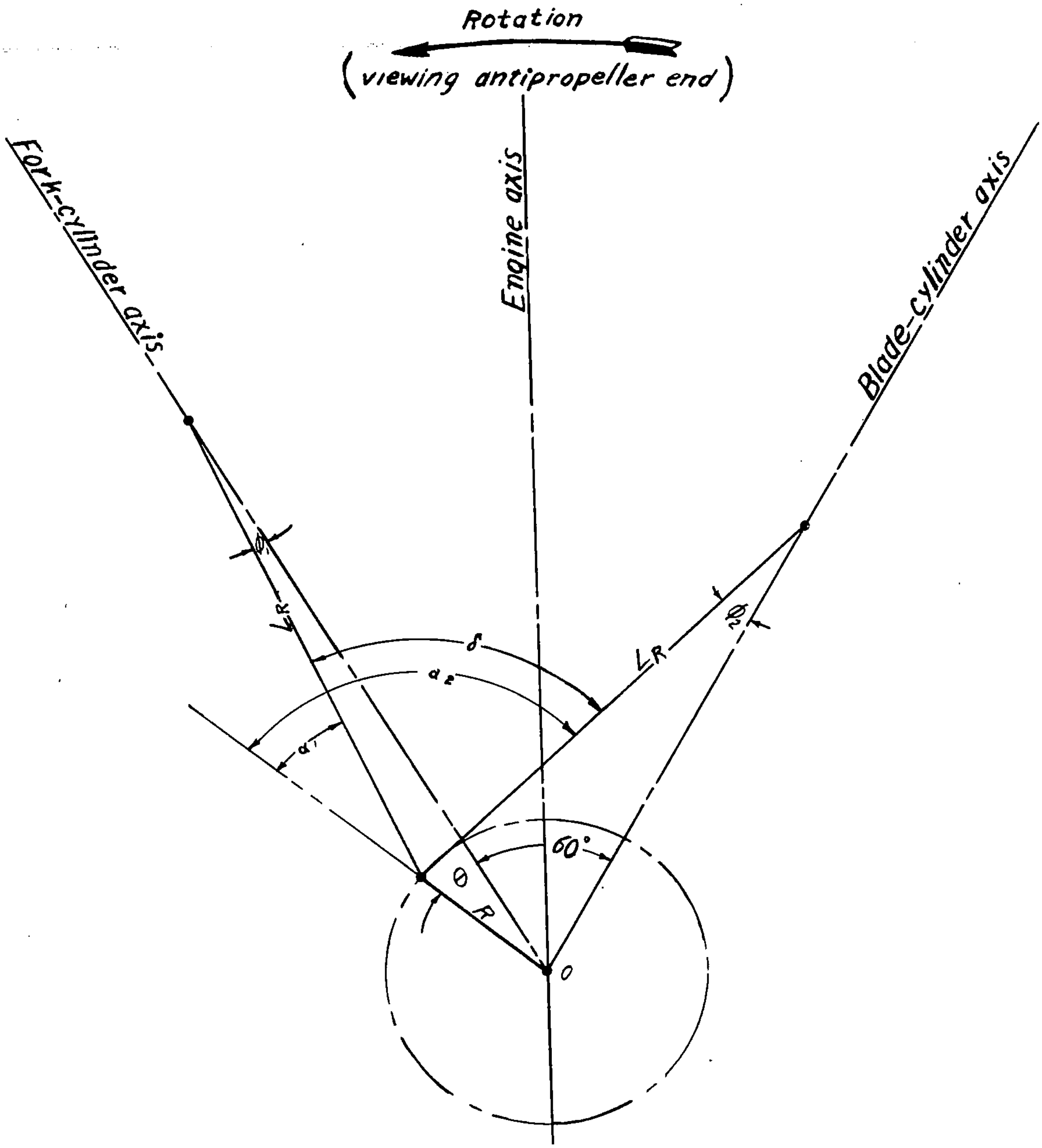

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Figure 1. - Schematic diagram of the mechanism of a 12-cylinder V-type engine.

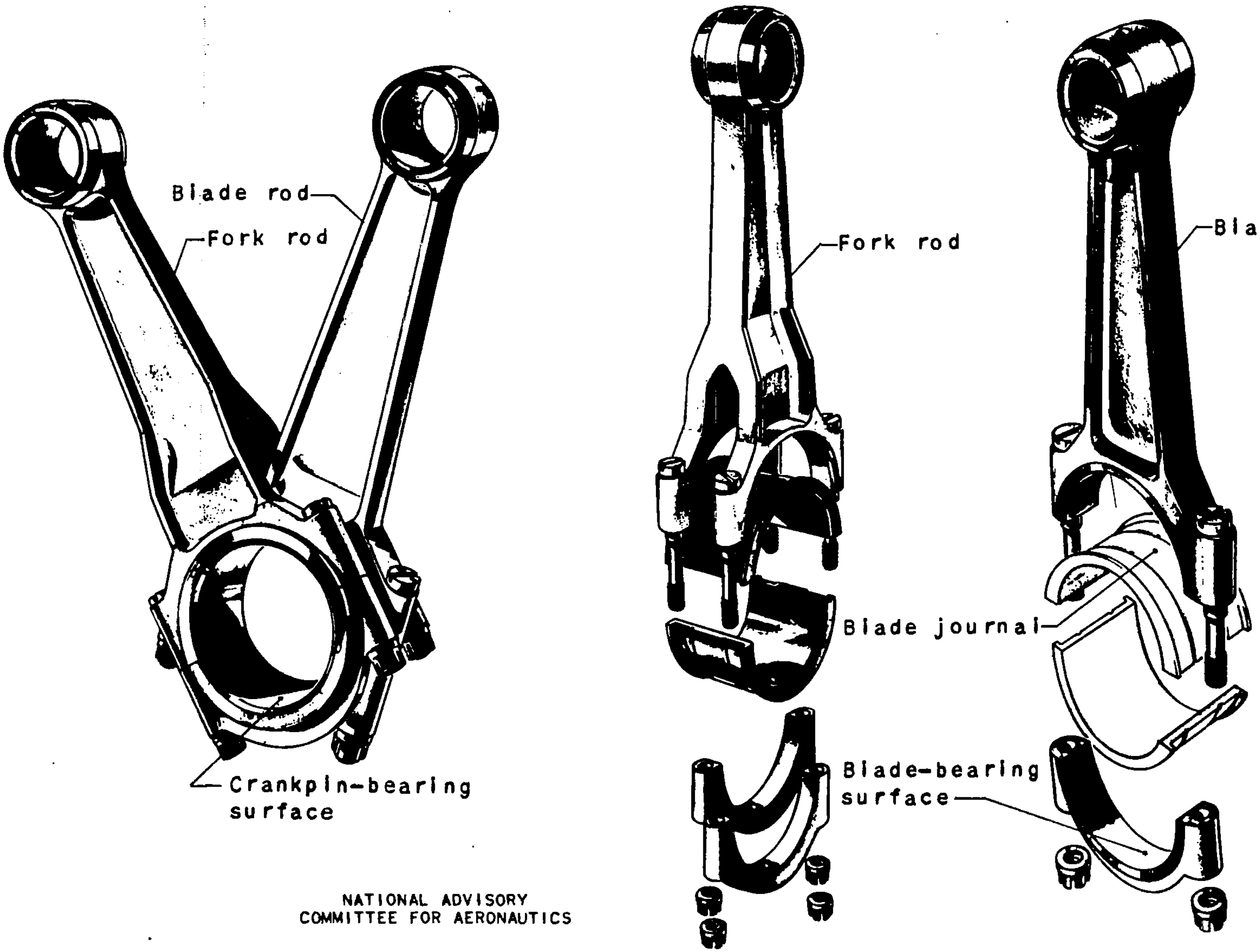

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Figure 2. - Connecting-rod and blade-bearing arrangement for a V-type engine.

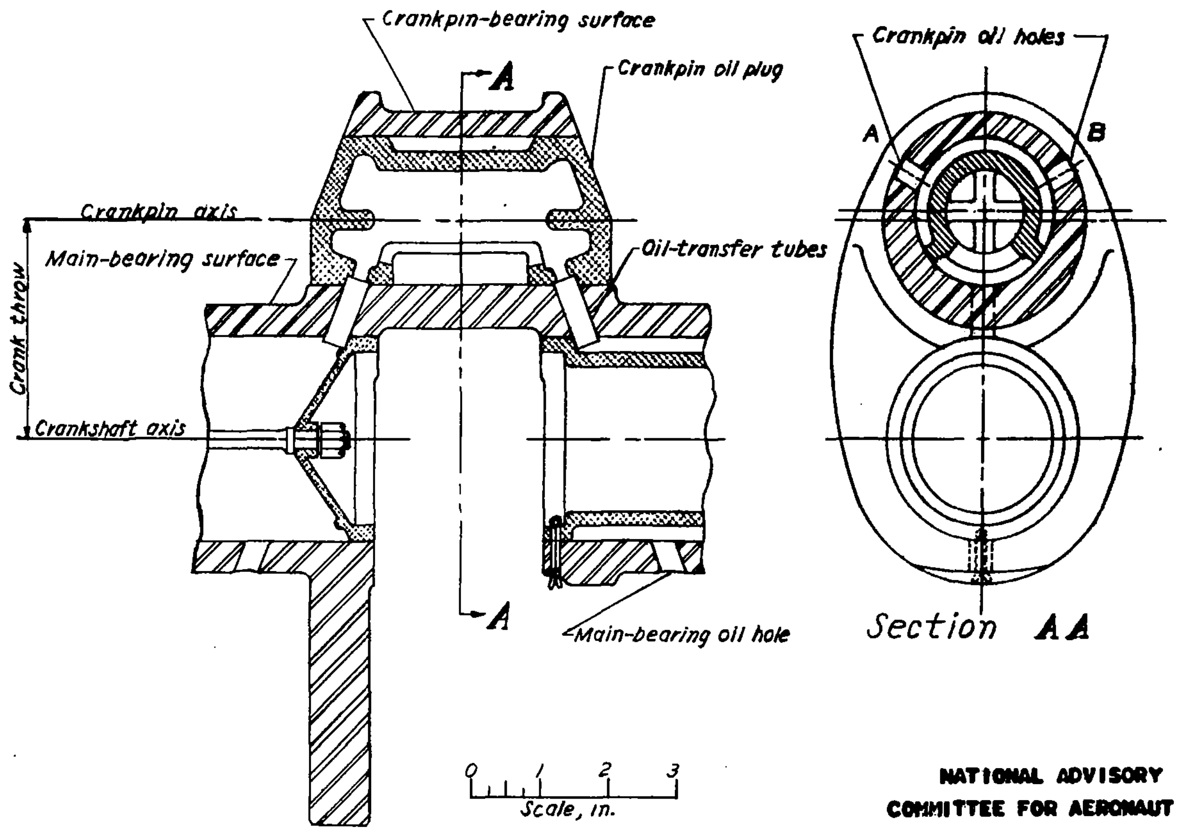

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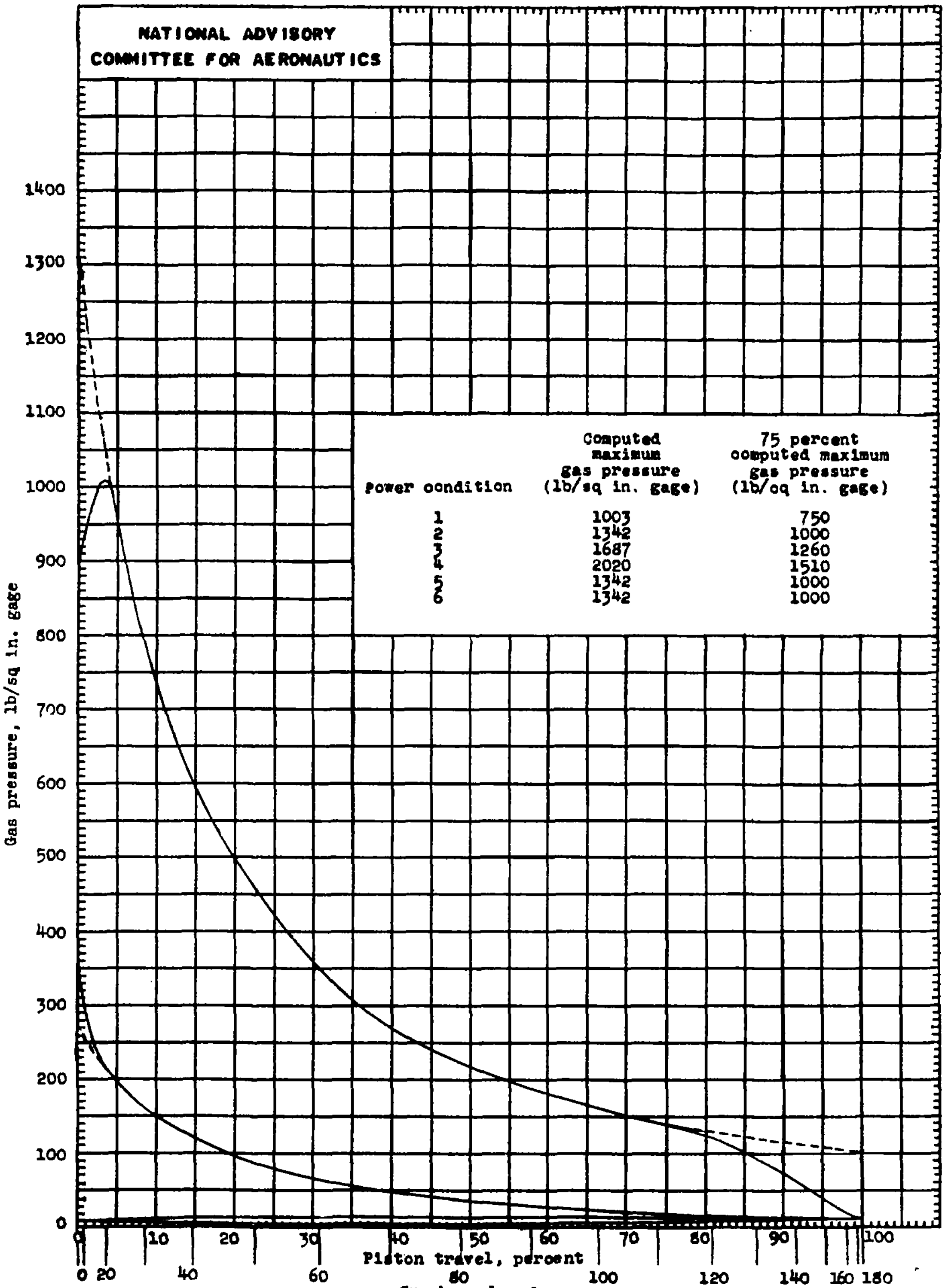
Figure 3. - Schematic diagram of crankpin arrangement for a V-type engine.



Power condition	Computed maximum gas pressure (lb/sq in. gage)	75 percent computed maximum gas pressure (lb/sq in. gage)
1	1003	750
2	1342	1000
3	1687	1260
4	2020	1510
5	1342	1000
6	1342	1000

Figure 4. - Theoretical indicator diagram for a V-type engine. Indicated mean effective pressure, 242 pounds per square inch; engine speed, 3000 rpm; indicator-diagram factor, 0.90; compression ratio, 6.65; exponent of expansion and compression curves, 1.30; maximum pressure, 75 percent of computed maximum.

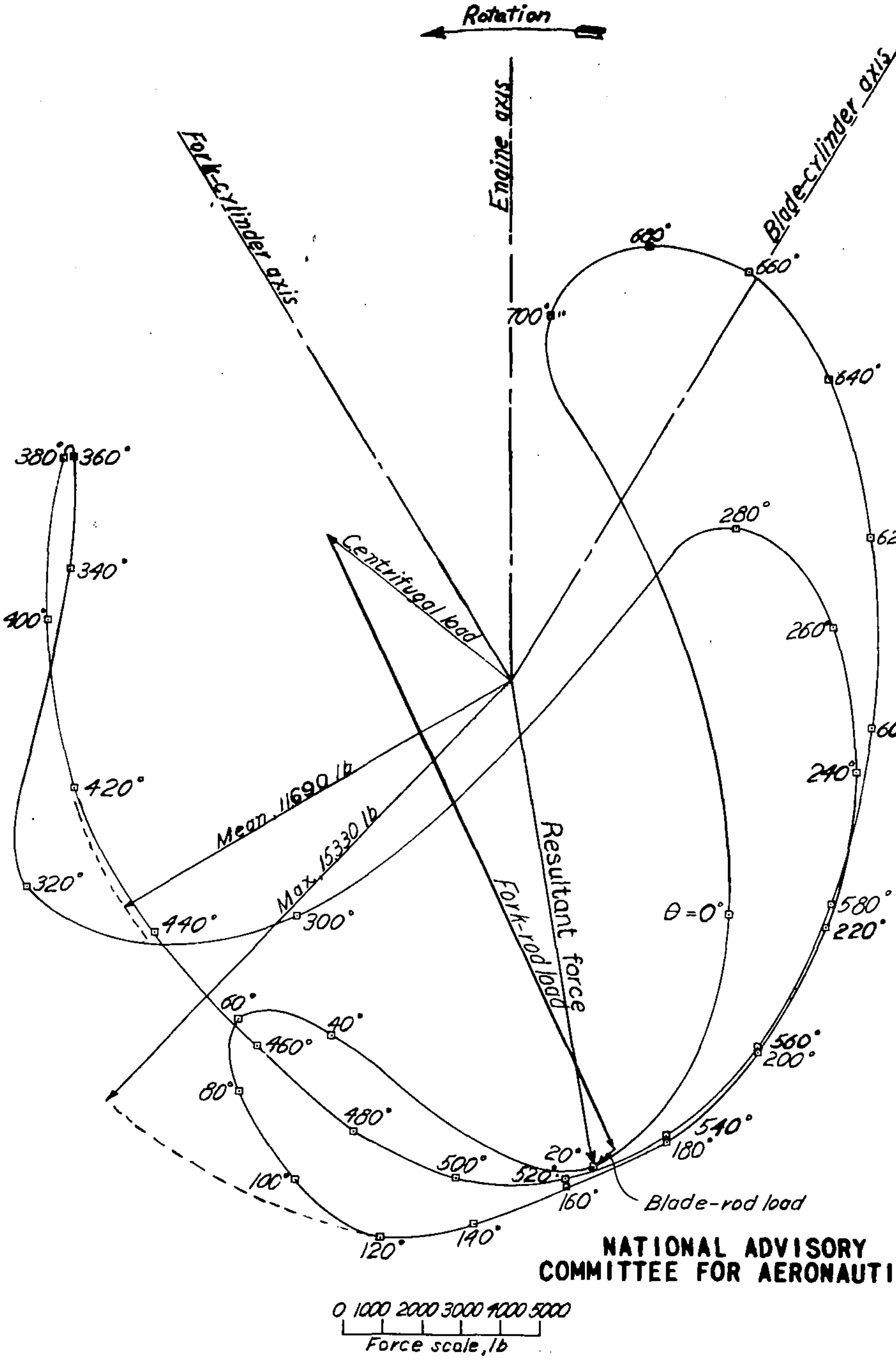


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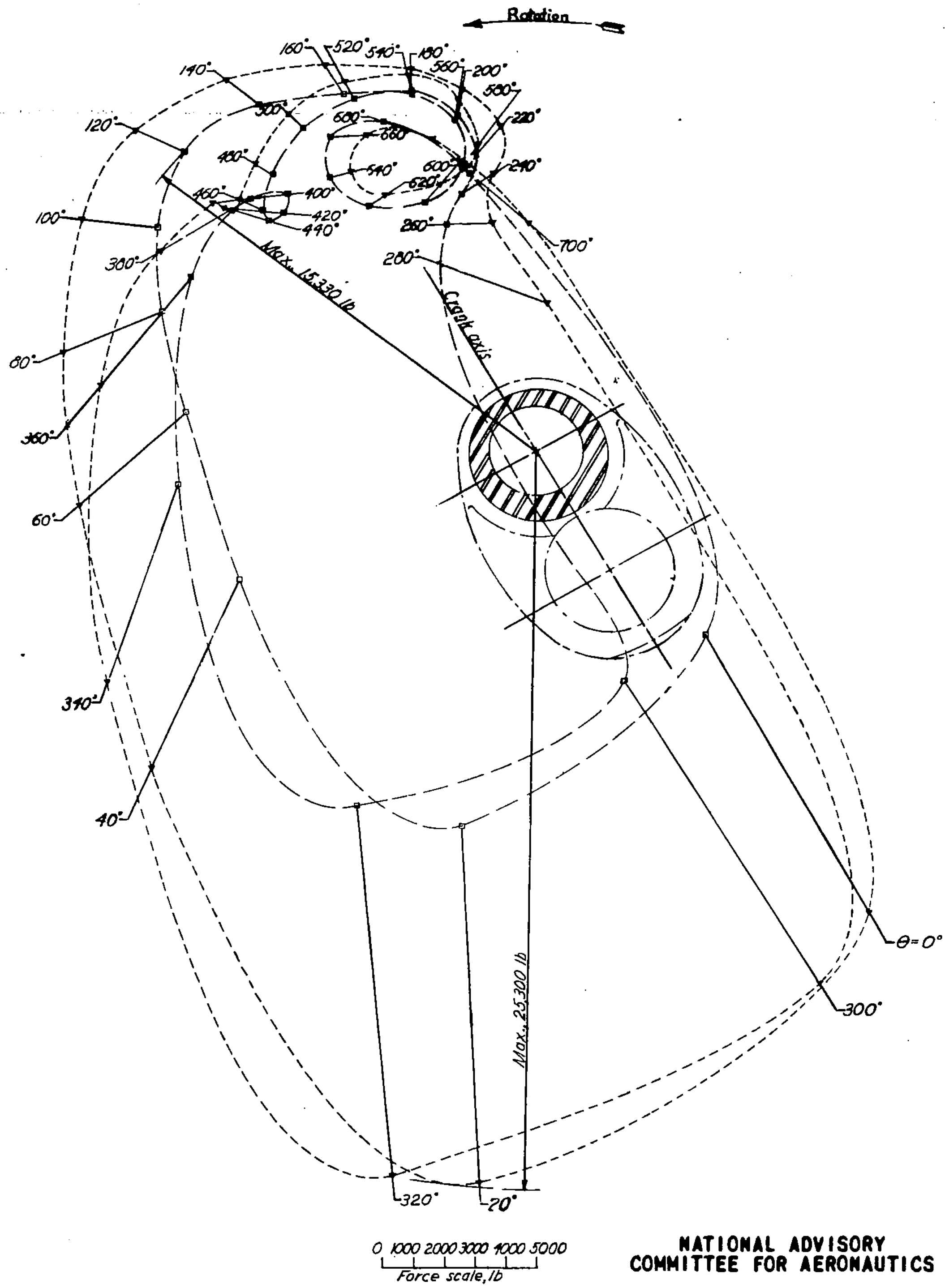
Figure 5. - Polar diagram showing the magnitude of the resultant force on the crankpin of a V-type engine and its direction with respect to the engine axis. Engine speed, 3000 rpm; indicated mean effective pressure, 242 pounds per square inch.



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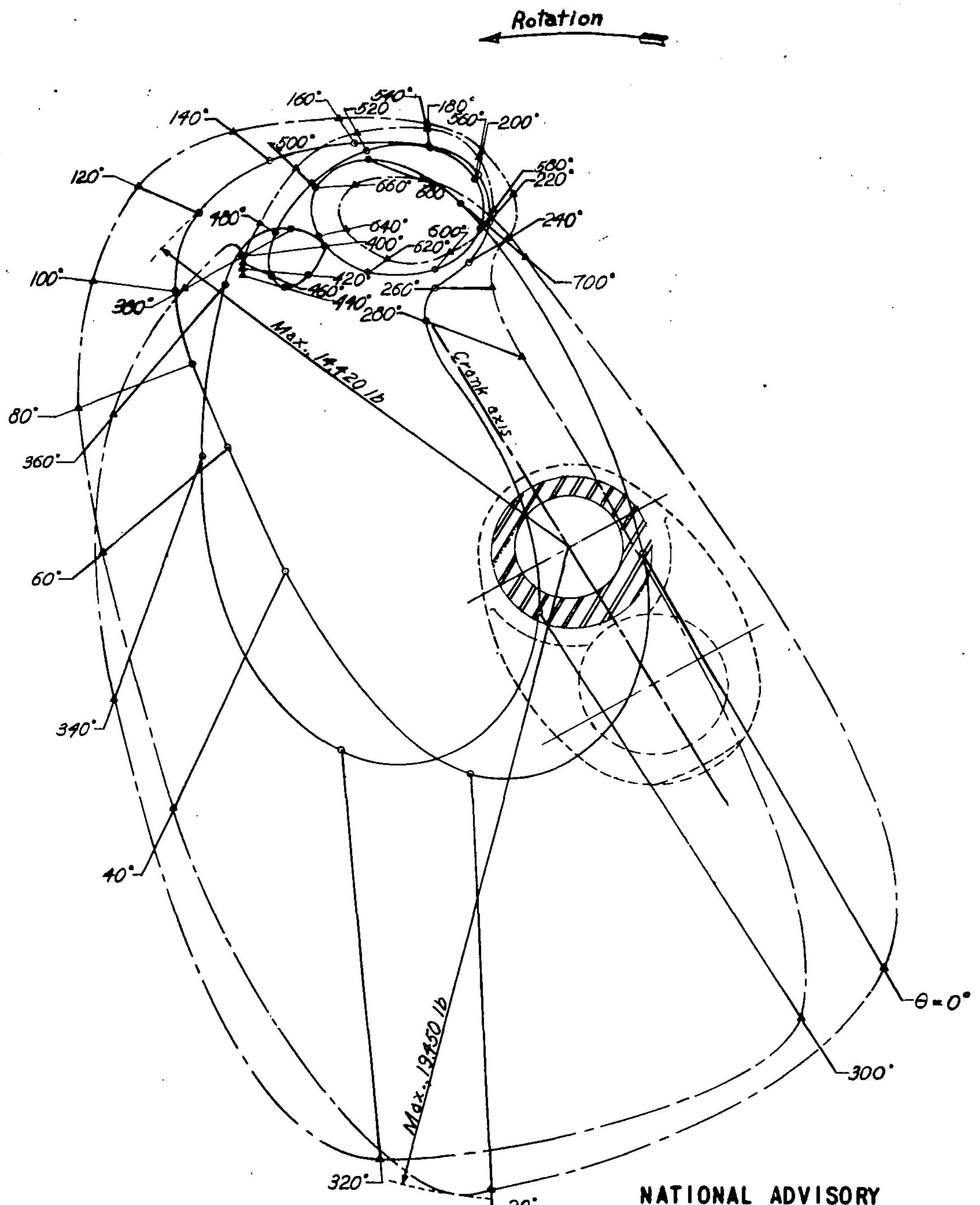
(a) Indicated mean effective pressures, 242 and 363 pounds per square inch.

Figure 6. - Polar diagrams showing the magnitude of the resultant force on the crankpin of a V-type engine and its direction with respect to the crank axis at an engine speed of 3000 rpm and different indicated mean effective pressures.



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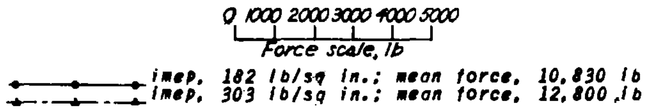

(b) Indicated mean effective pressures, 182 and 303 pounds per square inch.

Figure 6. - Concluded.

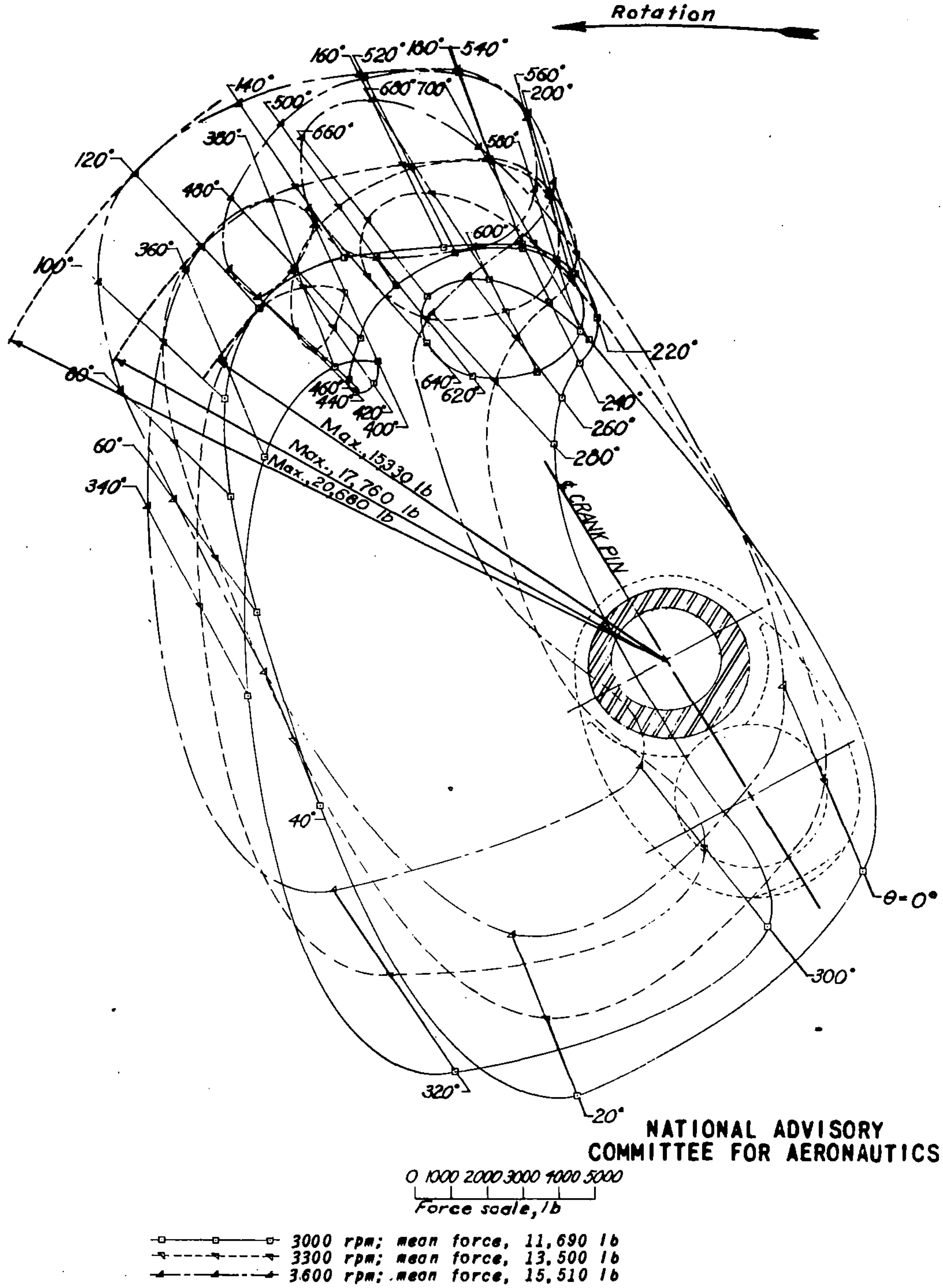

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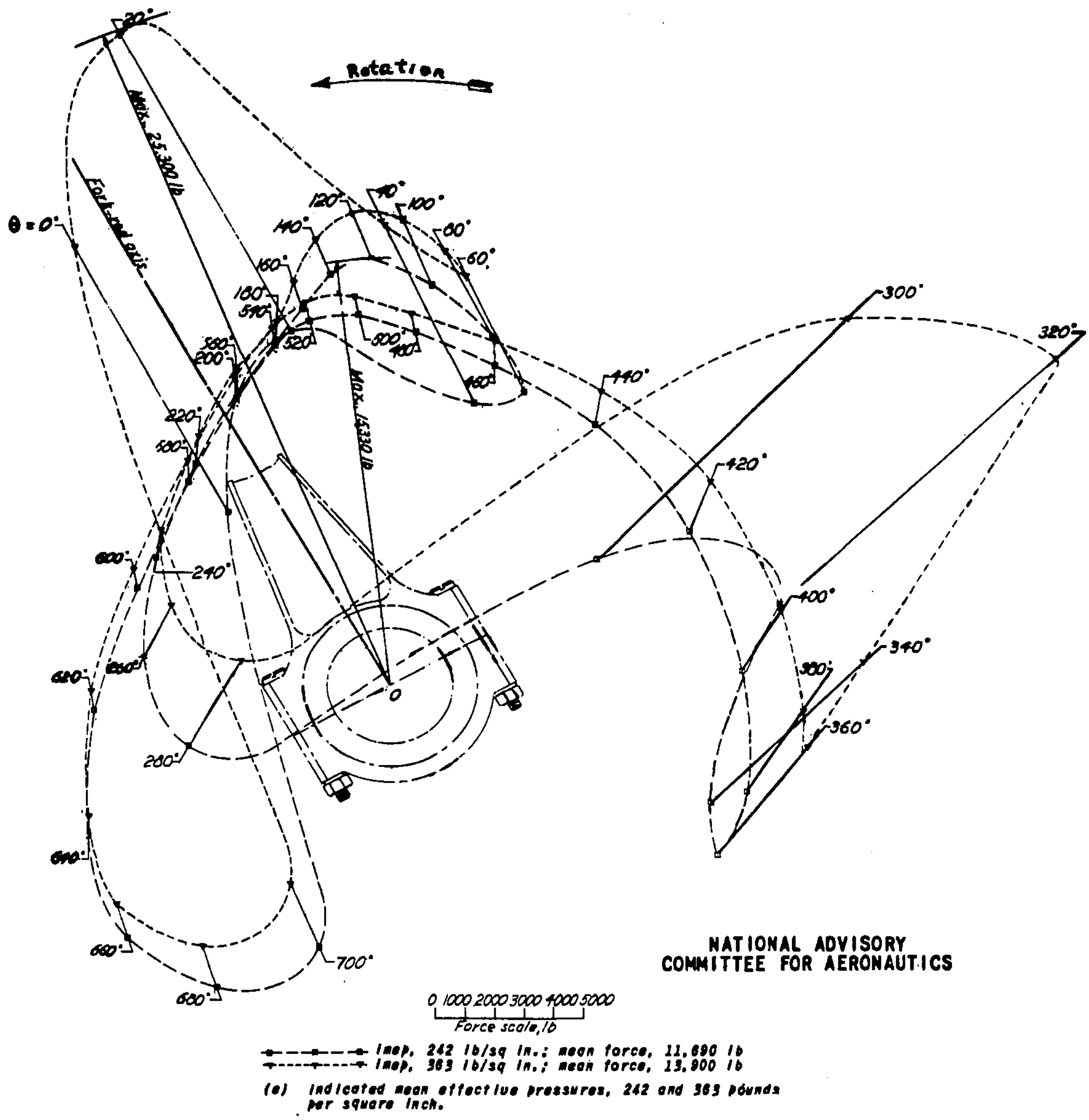
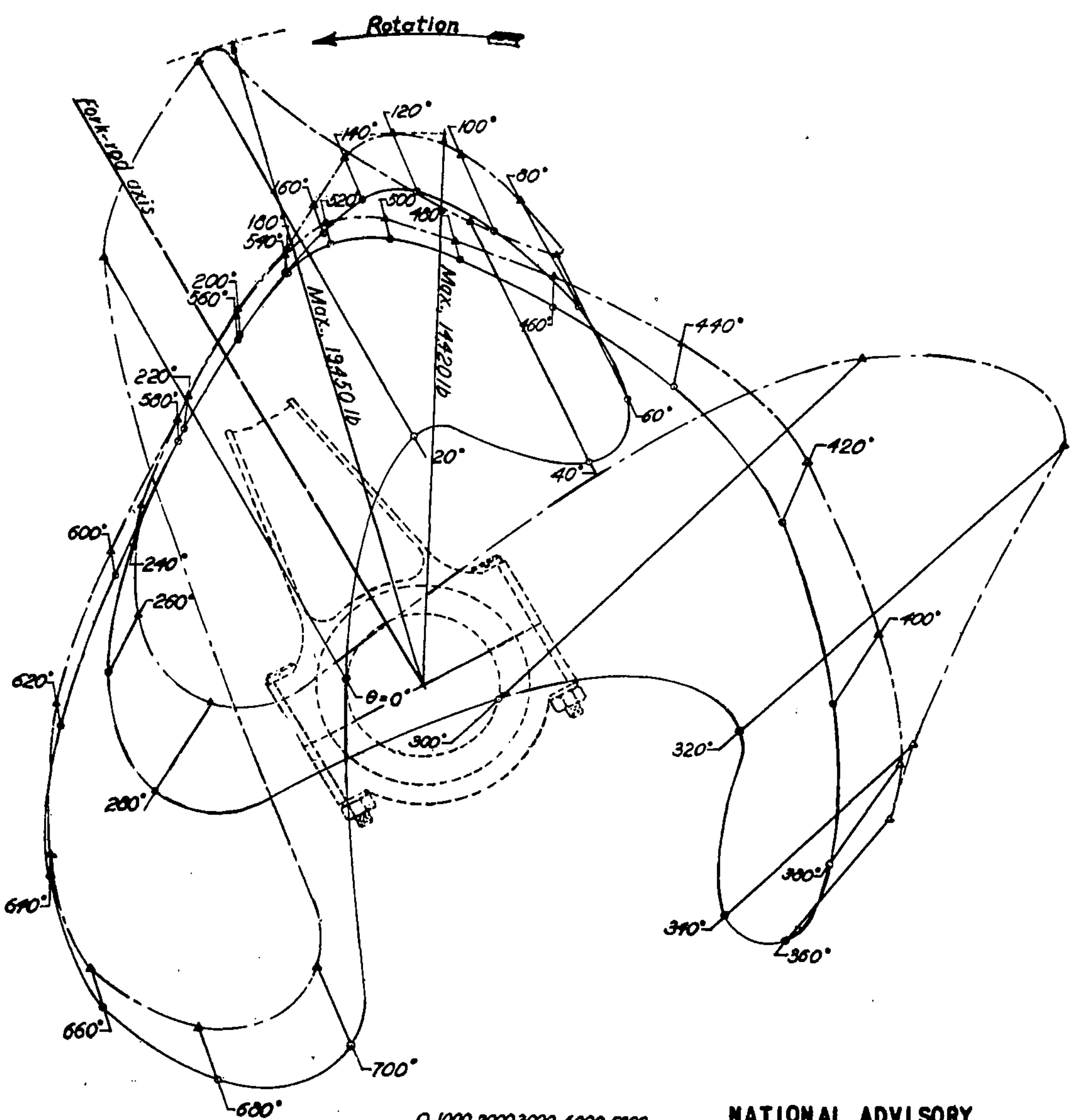
Figure 7. - Polar diagrams showing the magnitude of the resultant force on the crankpin of a V-type engine and its direction with respect to the crank axis at an indicated mean effective pressure of 242 pounds per square inch and different engine speeds.



(a) Indicated mean effective pressures, 242 and 363 pounds per square inch.

Figure 8. - Polar diagrams showing the magnitude of the resultant force on the crankpin bearing of a V-type engine and its direction with respect to the fork-rod axis at an engine speed of 3000 rpm and different indicated mean effective pressures.



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imep, 182 lb/sq in.: mean force, 10,830 lb
imep, 303 lb/sq in.: mean force, 12,800 lb

(b) Indicated mean effective pressures, 182 and 303 pounds per square inch.

Figure 8. - Concluded.

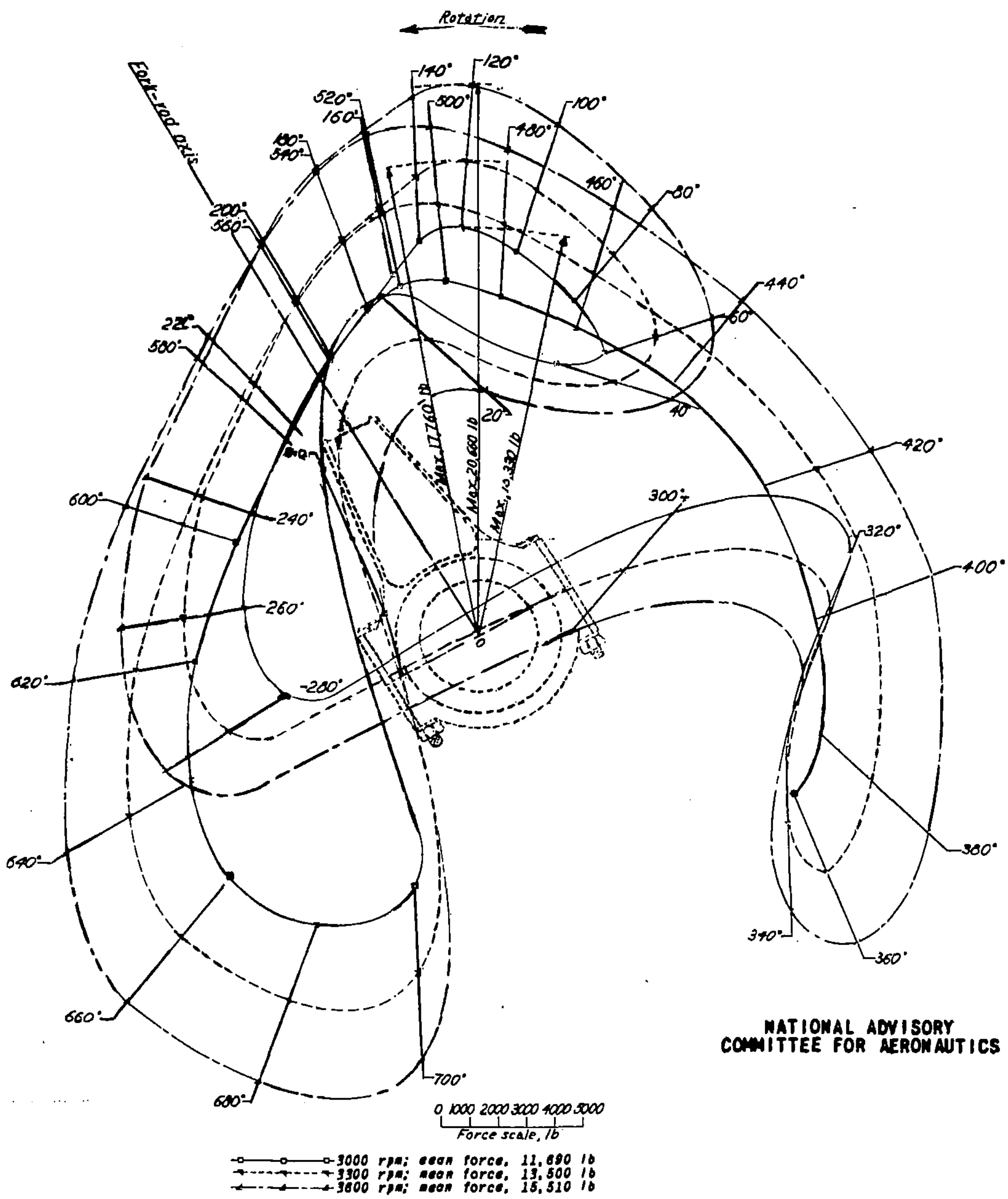


Figure 9. - Polar diagrams showing the magnitude of the resultant force on the crankpin bearing of a V-type engine and its direction with respect to the fork-rod axis at an indicated mean effective pressure of 242 pounds per square inch and different engine speeds.

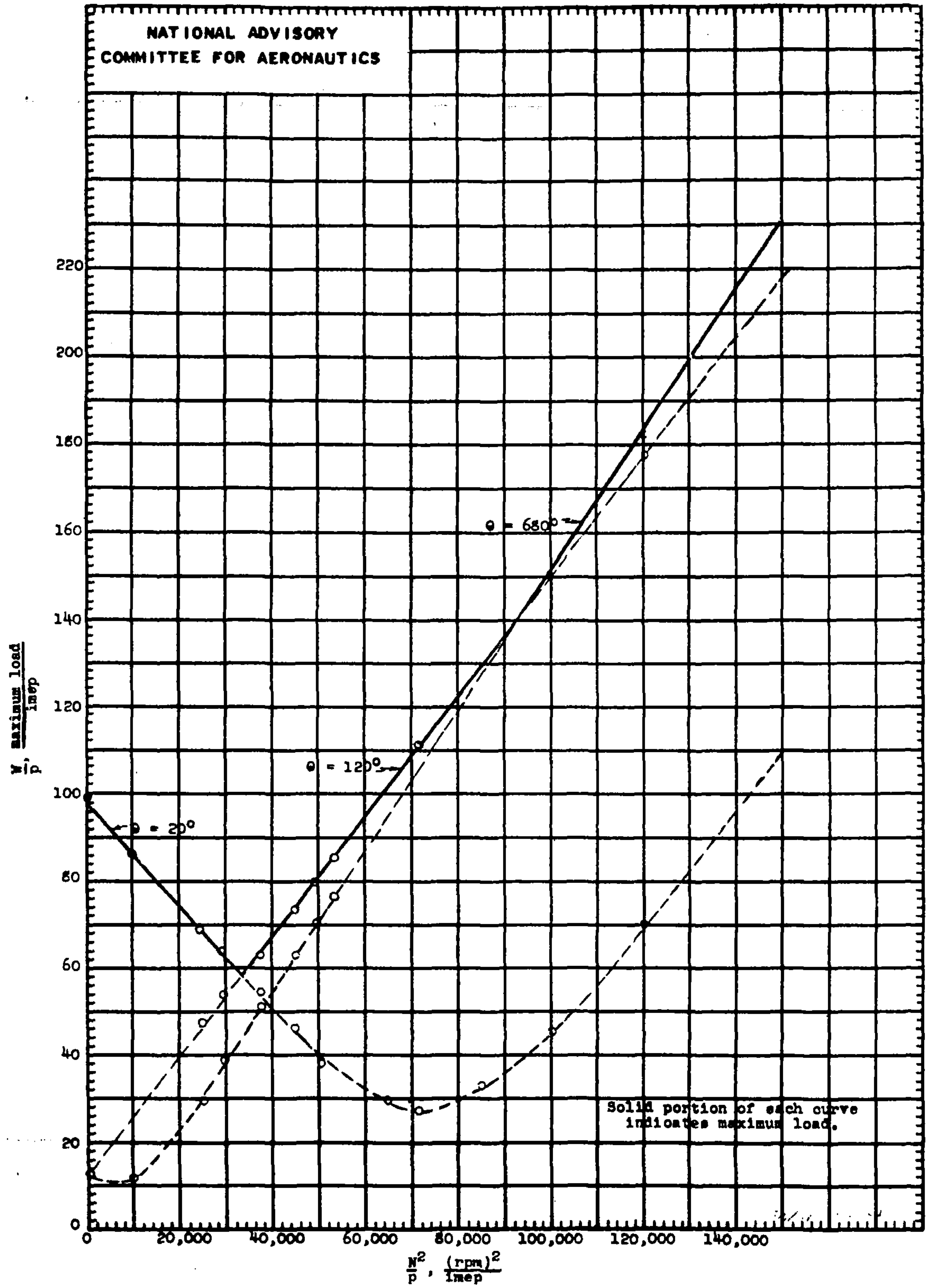
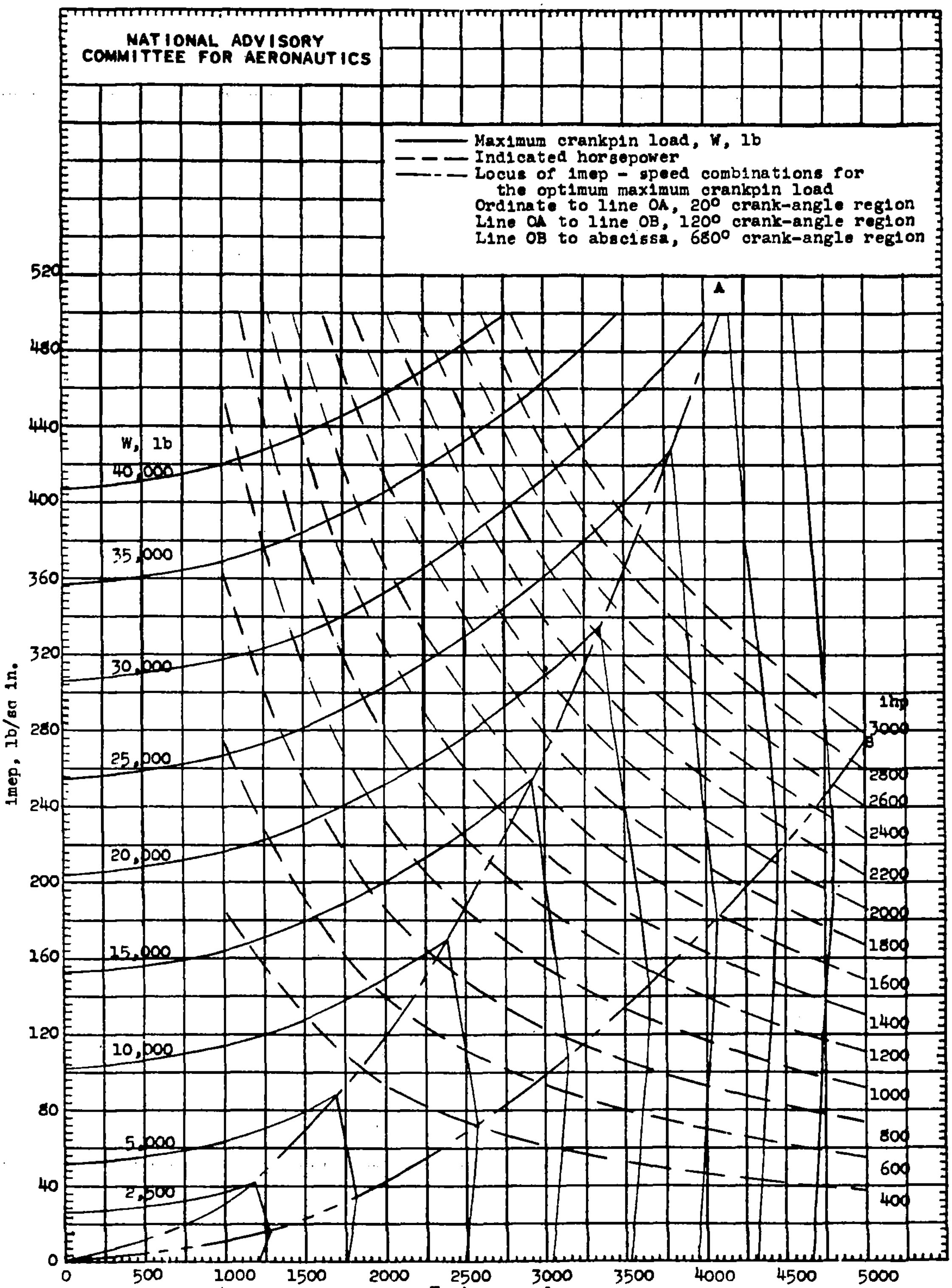
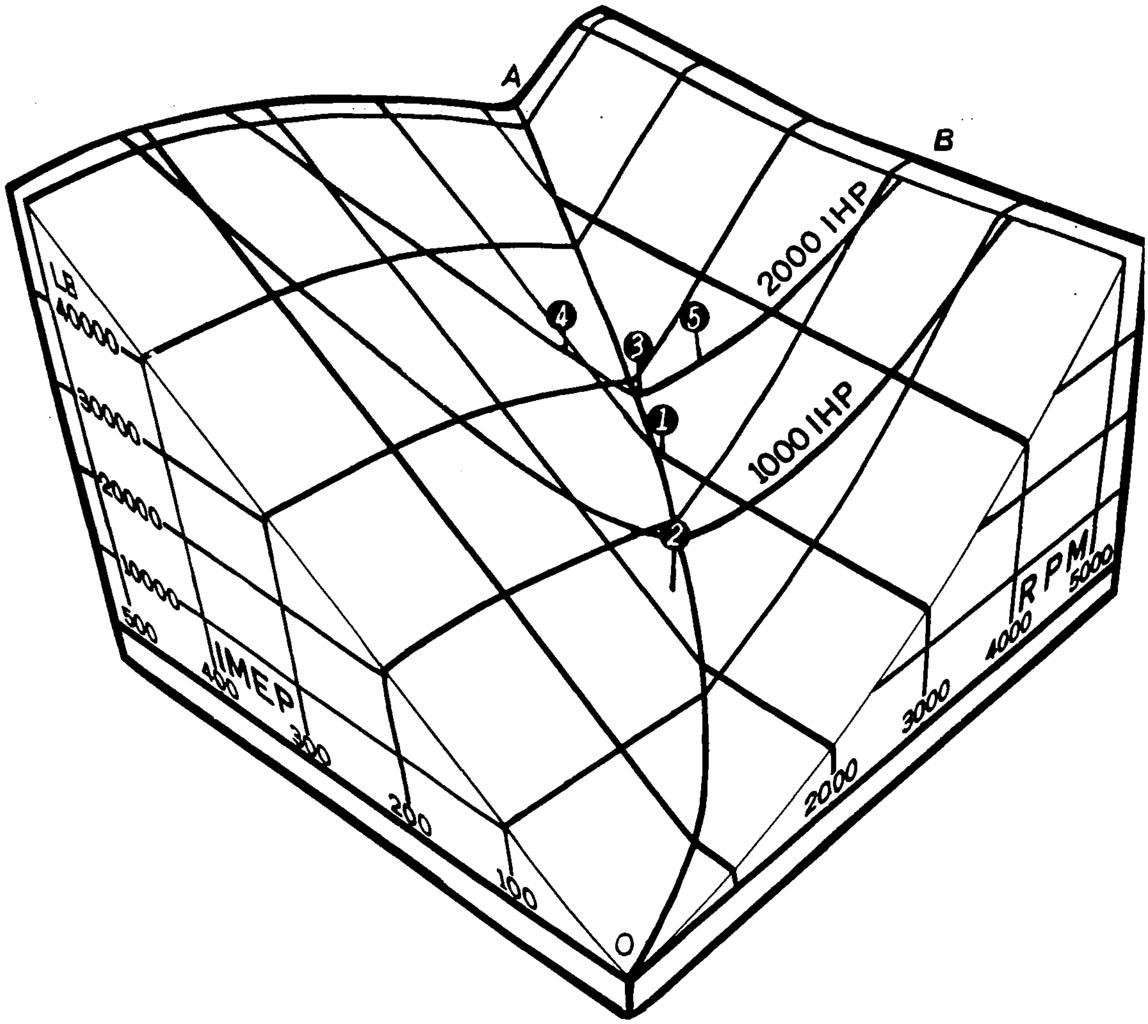


Figure 10. - Variation of W/p with N^2/p at crank angles of 20°, 120°, and 680° for a V-type engine.



(a) Contour plot.

Figure 11. - Maximum load on the crankpin of a production V-type engine for all values of indicated mean effective pressure and engine speed at compression ratio of 6.65. Effective bearing area, 5.81 square inches.



(b) Solid model.

Figure 11. - Concluded.

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Point	Engine speed (rpm)	Indicated mean effective pressure (lb/sq in.)	Indicated horsepower	Maximum bearing load (lb)	Remarks
1	3000	242	1570	15,500	Take-off
2	2190	158	750	10,000	60 percent cruise
3	3180	292	2000	17,400	Optimum at 2000 ihp
4	2890	321	2000	20,450	imep 10 percent above optimum at 2000 ihp
5	3520	263	2000	20,000	imep 10 percent below optimum at 2000 ihp

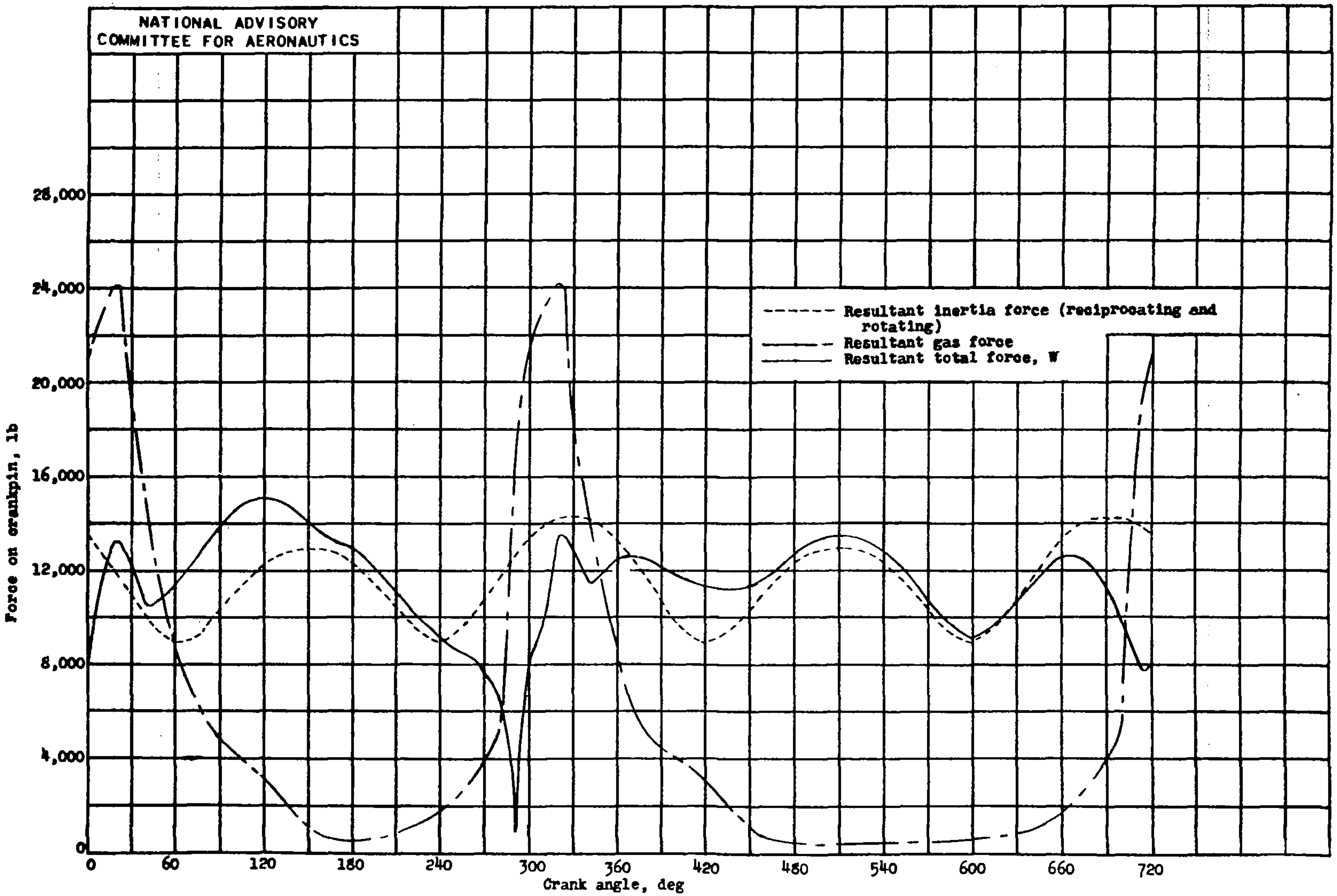

Figure 12. - Variation of resultant crankpin gas force, resultant crankpin inertia force, and resultant crankpin total force with crank angle for a V-type engine. Engine speed, 3000 rpm; indicated mean effective pressure, 242 pounds per square inch.

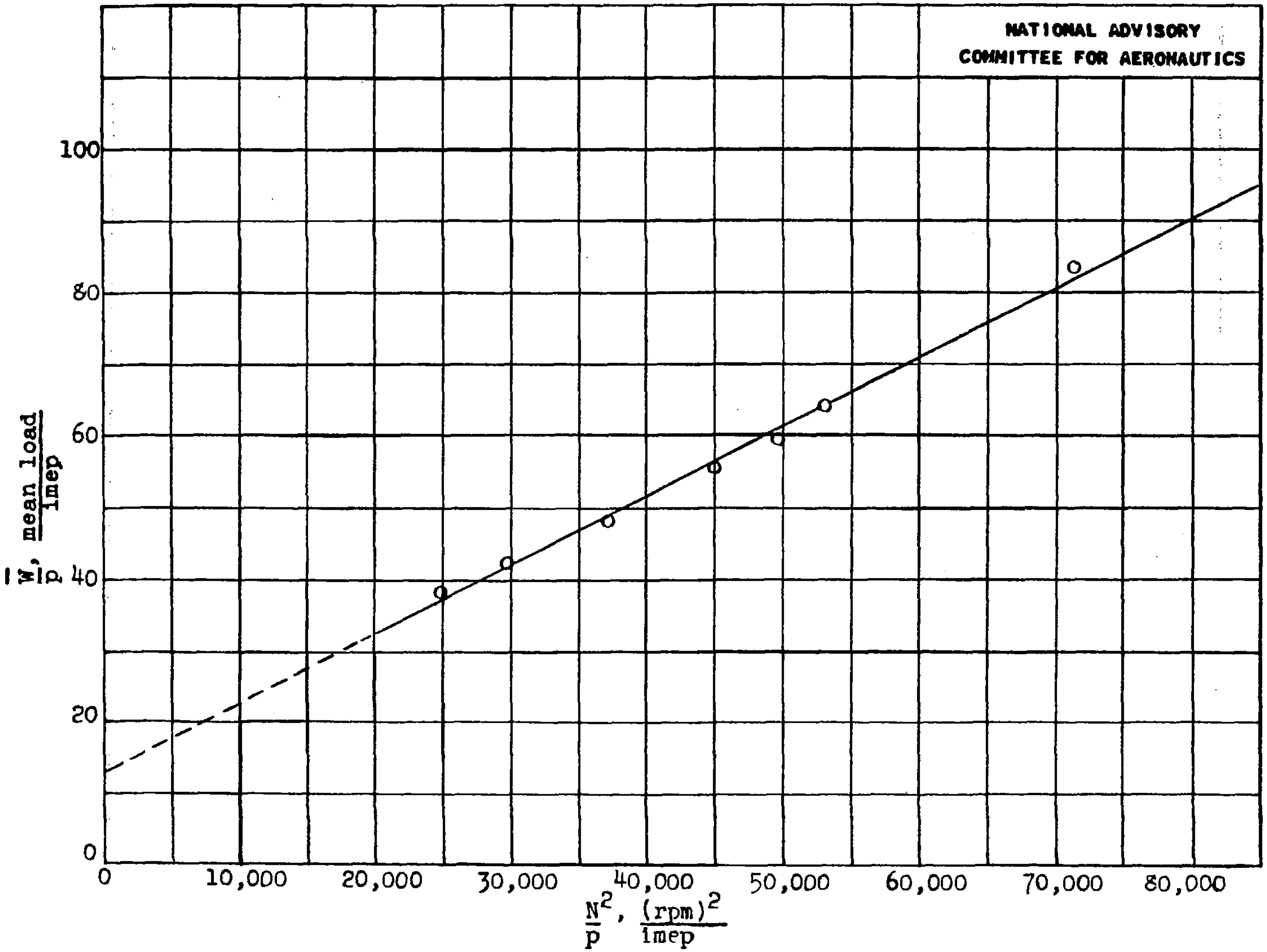

Figure 13. - Variation of $\overline{W}/p$ with N^2/p for a V-type engine.

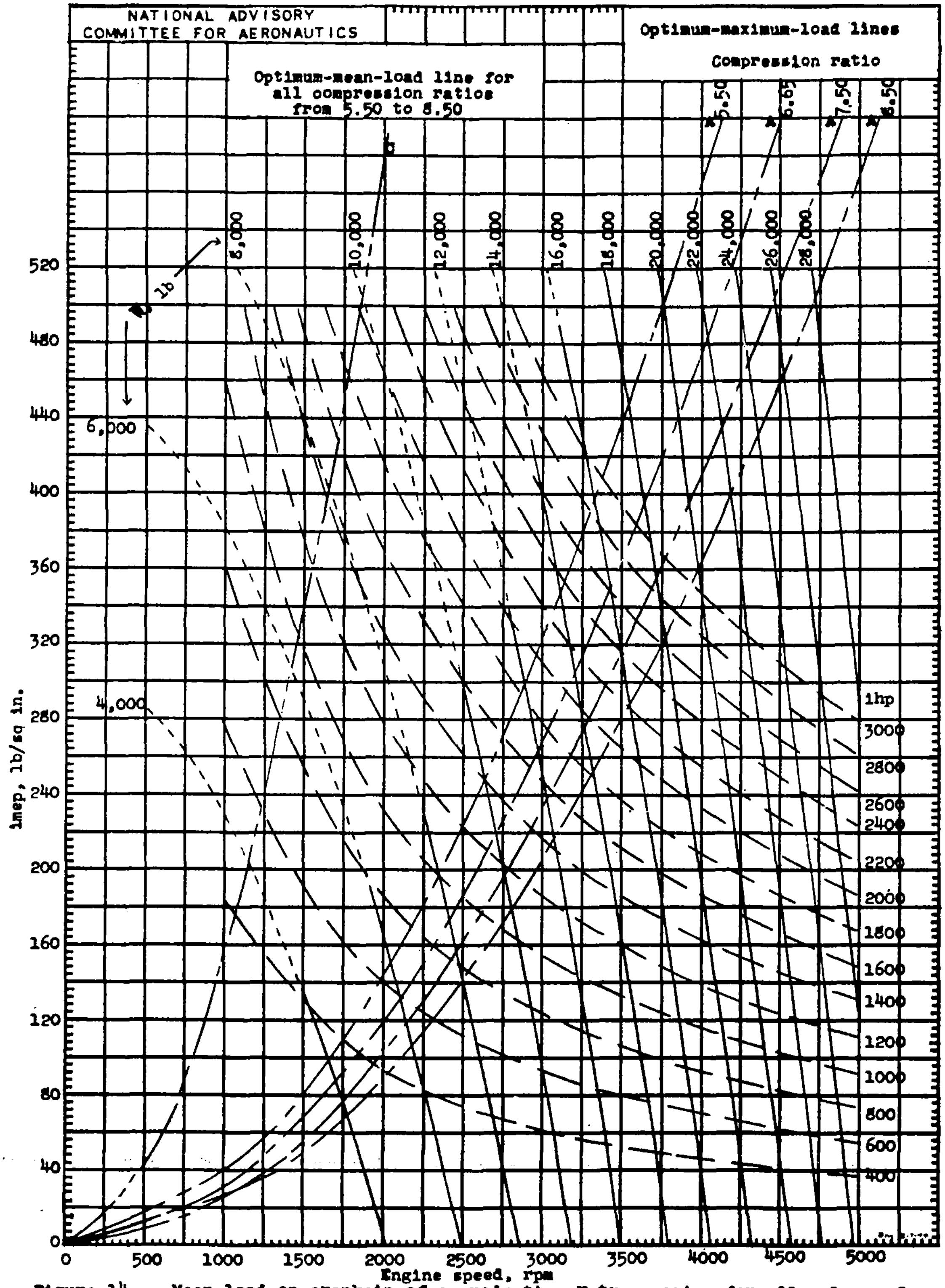

Figure 14. - Mean load on crankpin of a production V-type engine for all values of indicated mean effective pressure and engine speed. (Constant mean-load curves.) Effective bearing area, 5.81 square inches.

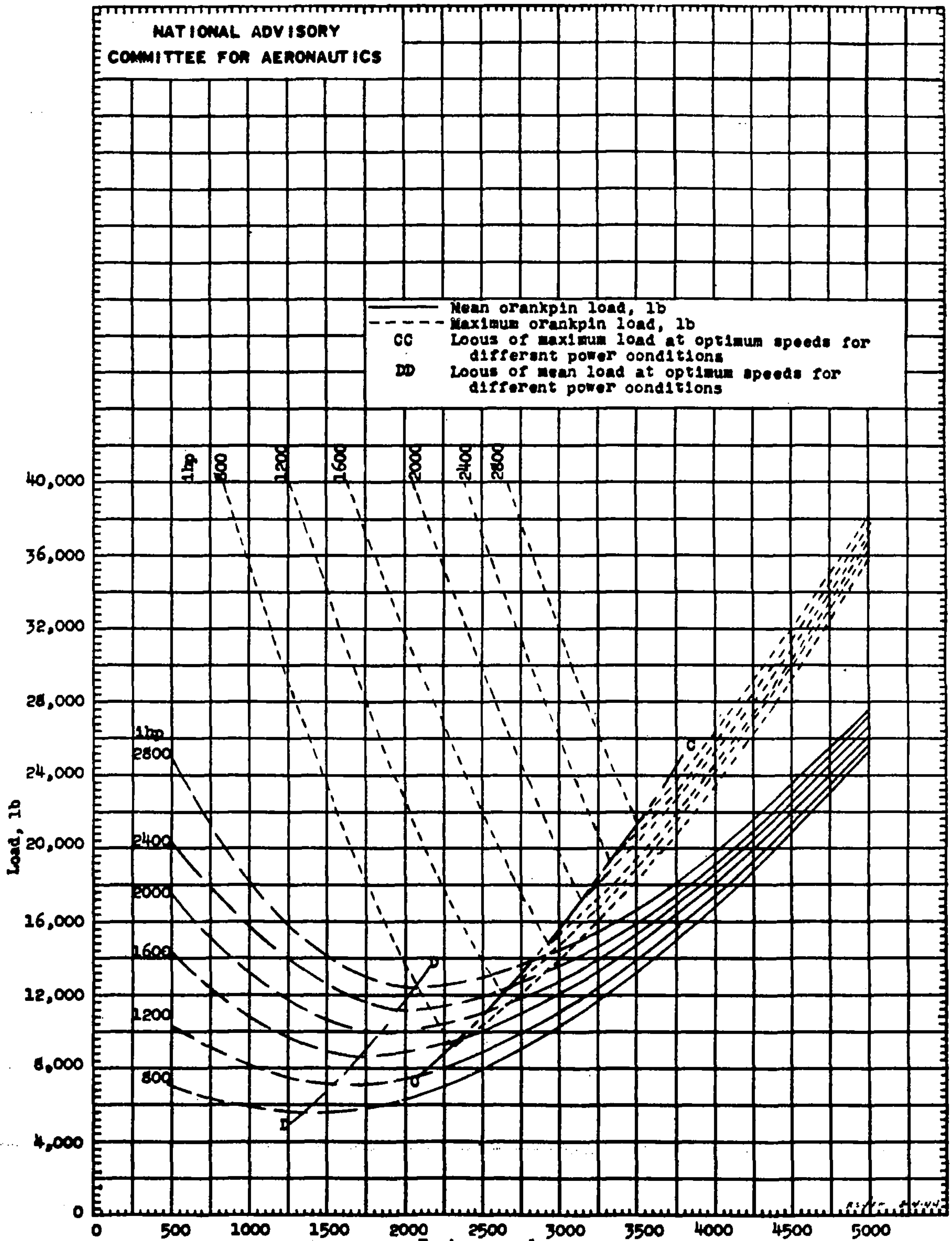

Figure 15. - Mean and maximum load on crankpin of a V-type engine for all values of indicated horsepower and engine speed at a compression ratio of 6.65. (Constant indicated-horsepower curves.) Effective bearing area, 5.81 square inches.

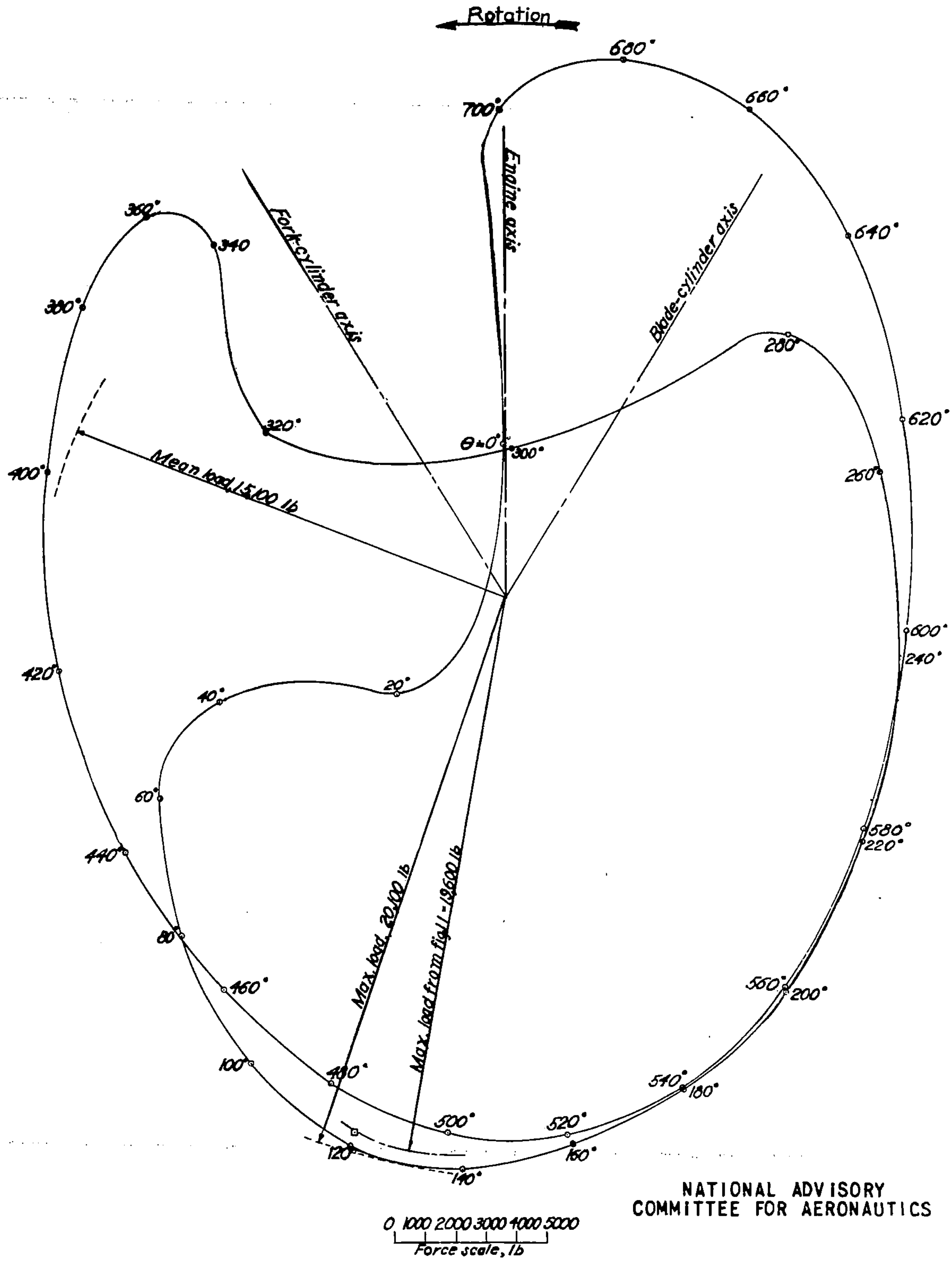

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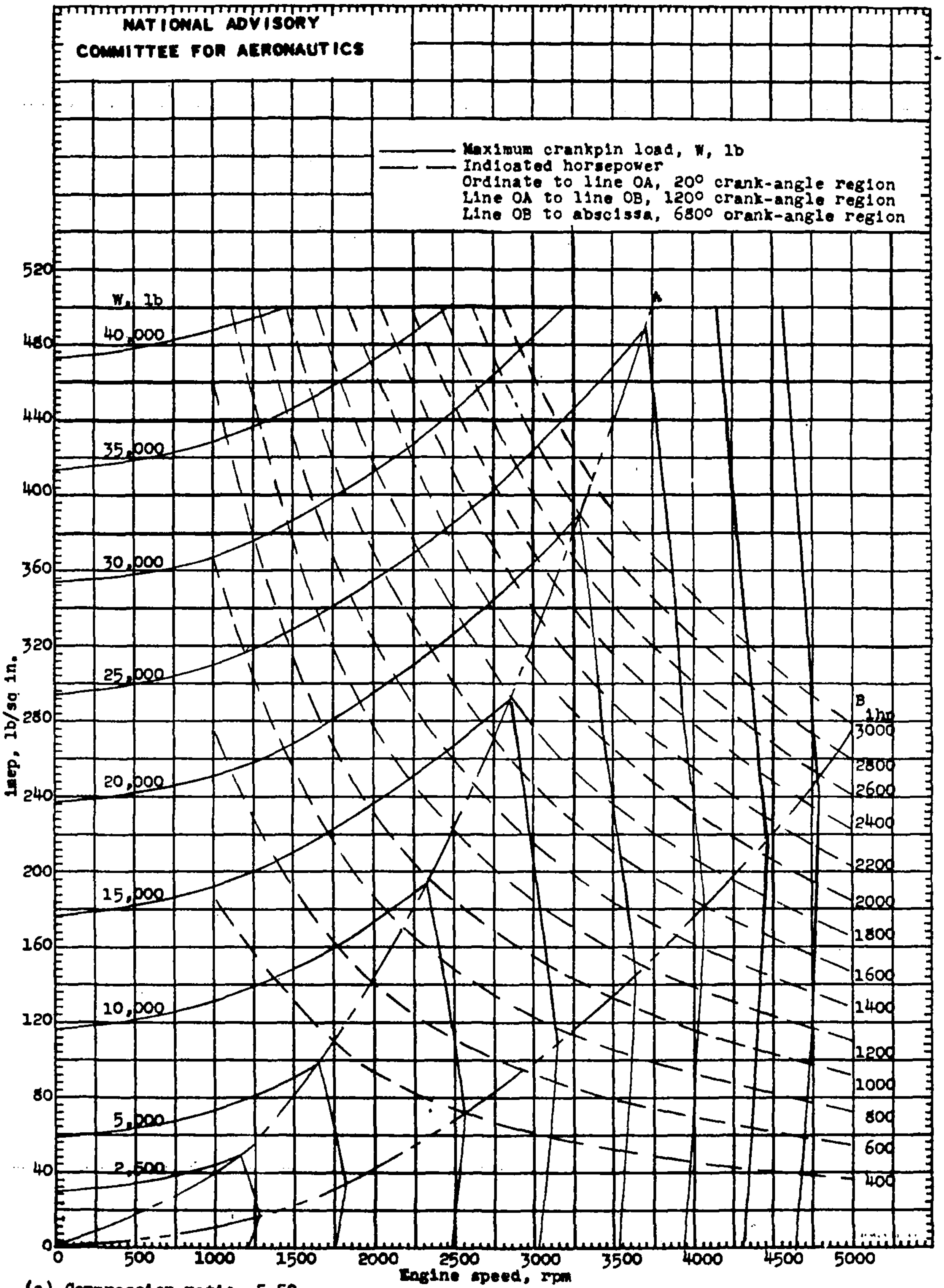
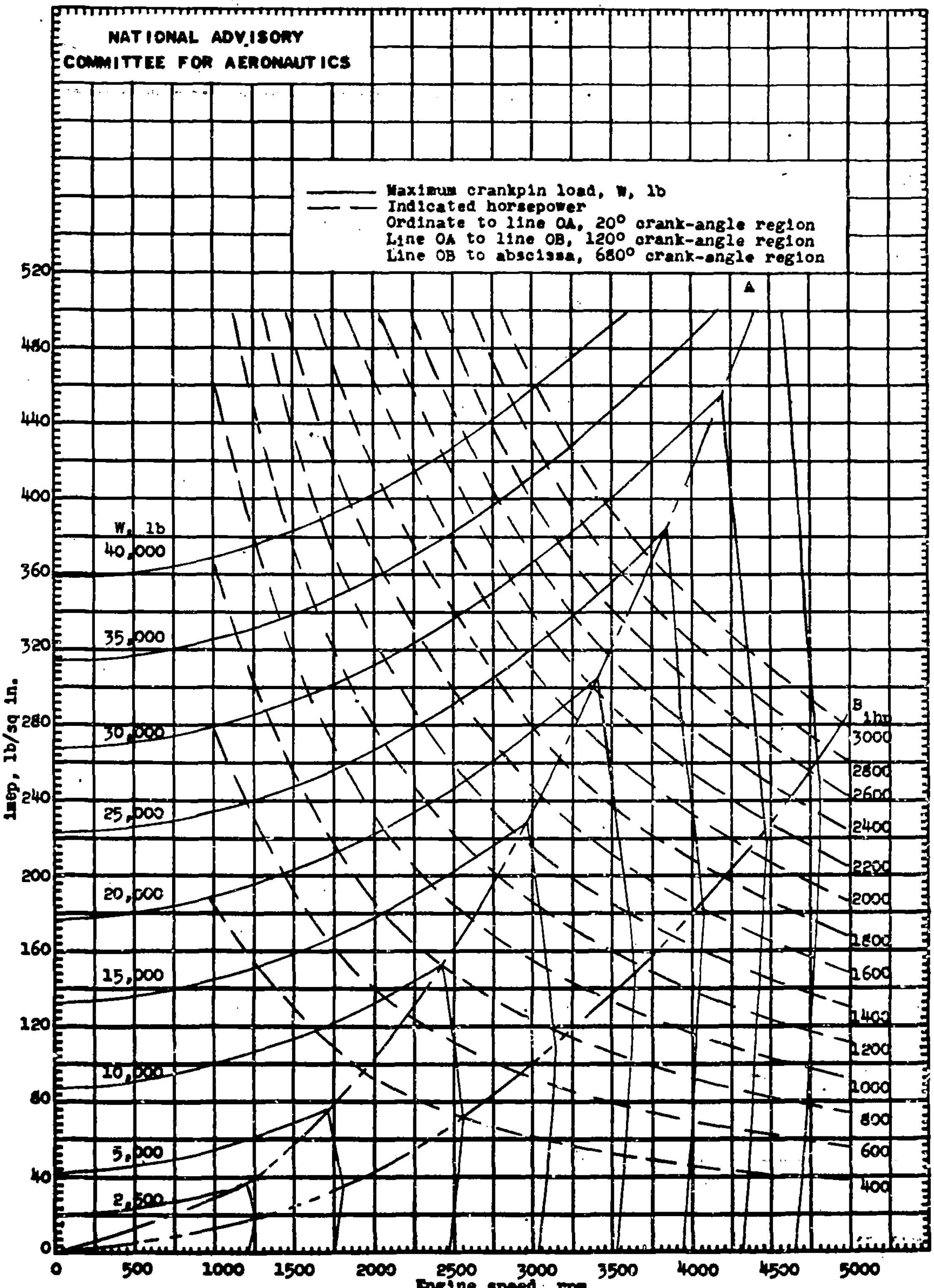
Figure 16. - Polar diagram showing the magnitude of the resultant force on the crankpin of a V-type engine and its direction with respect to the engine axis. Engine speed, 3600 rpm; indicated mean effective pressure, 182 pounds per square inch.

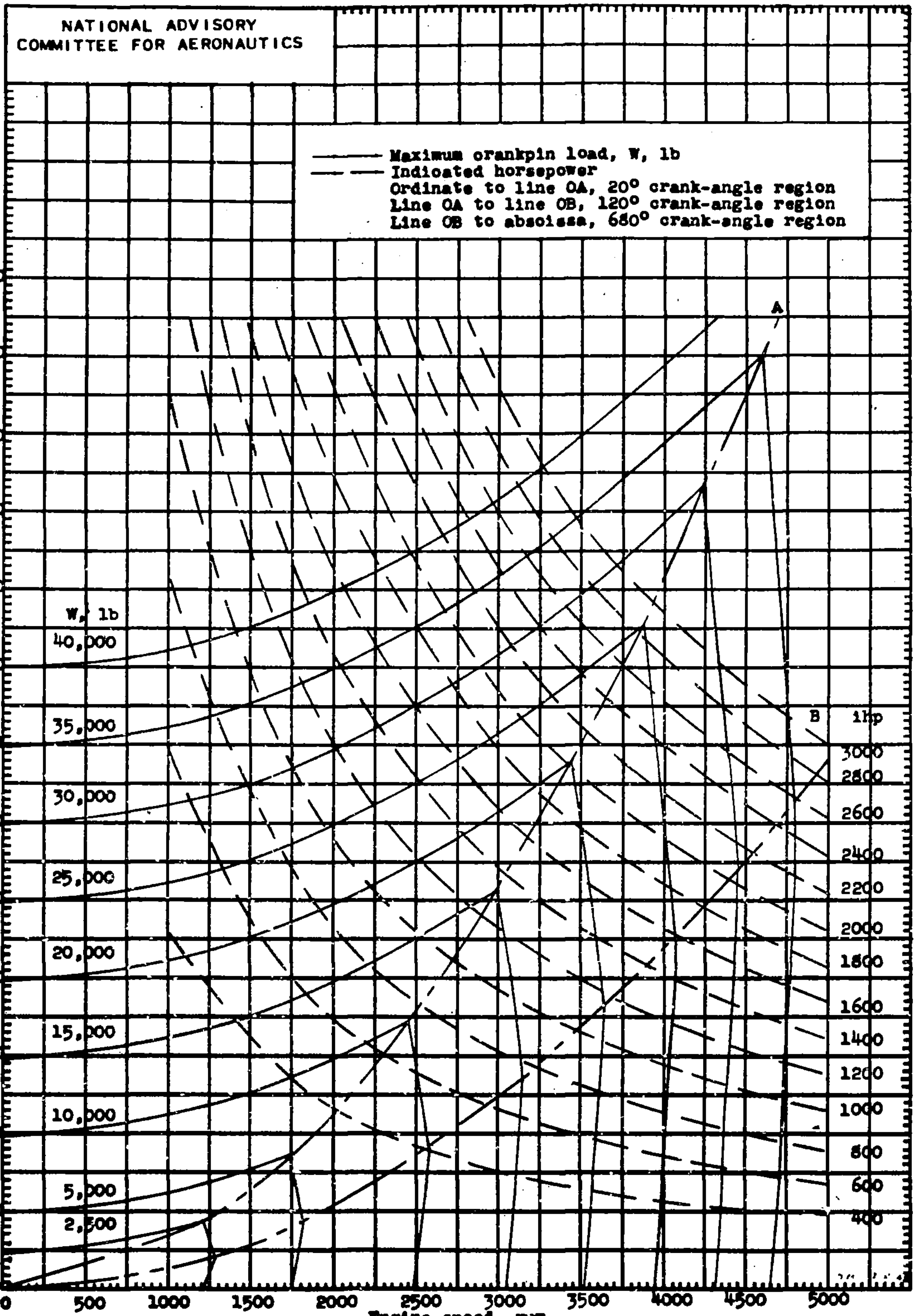


(a) Compression ratio, 5.50.

Figure 17. - Maximum load on crankpin of a production V-type engine for all values of indicated mean effective pressure and engine speed at various compression ratios. (Constant maximum-load curves.) Effective bearing area, 5.81 square inches.



(b) Compression ratio, 7.50.
Figure 17. - Continued.



(c) Compression ratio, 8.50.
Figure 17. - Concluded.

ROLL FORM (18 MAR 47)

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ABSTRACT

Method of generalizing results of a relatively few conventional bearing-load analyses has been developed. Application to crankpin-bearing loads of V-type engines showed that optimum combinations of engine speed and indicated mean effective pressure exist for which mean and maximum crankpin-bearing loads are a minima for a given power. Maximum crankpin-bearing load for a 12-cylinder V-type engine is shown to occur in the crank-angle region of 20°, 120°, or 680° depending upon relative values of engine speed and indicated mean effective pressure applied.

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