CONTINENTAL AIRCRAFT ENGINE CO.

DETROIT, MICHIGAN.

DESIGN REPORT #59
Date-April 17, 1934
5 Sheets
Curves C-18
C-19
C-20
C-21
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CONTINENTAL O-1430-1 ENGINE

CRANKSHAFT STRESS ANALYSIS.

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CRANKSHAFT STRESS ANALYSIS

Reference:- A.S.I.C. No. 421

It is impossible to determine the exact nature and magnitude of all the stresses in the crankshaft, since the deflection of the engine members and the torsional vibration in the crankshaft produce complicated strains which are difficult, if not impossible, to calculate. The 0-1430-1 crankshaft has been analyzed in the following manner; first, according to the standard method as described in A.S.I.C. report #421, considering the crankthrow nearest the propeller as transmitting the most torque and erroneously deducting that this crankthrow is subjected to the greatest maximum torque. Few, if any, crankshaft failures have occurred in the cheek nearest the propeller. An investigation of cumulative torque shows that, while this cheek transmits the greatest mean torque, its maximum torque is far below that carried in other cheeks. It is also more important to investigate the stress range to which the material is subjected rather than to consider only the actual maximum stress.

The method of determination and the value of stresses which are believed to exist in the 0-1430-1 crankshaft will be found in the second part of this report.

PART ONE

T_m = max. inst. torque, lb. in. $= \dfrac{63000 \times BHP \times K}{N}$

$$T_m = \frac{63000 \times 1000 \times 1.2}{3000} = 21000 \times 1.2 \quad 25200 \text{ lbs. in.}$$

$$F_t = \frac{T_m}{r} = \frac{25200}{2.5} = 10080 \text{ lbs.}$$

$$F_n = 13800$$

$$F_b = \sqrt{F_t^2 + F_n^2} = \sqrt{(10080)^2 + (13800)^2} = 17000$$

a = Distance from center of crankpin to center of bearing bolts = 3.593 in.

r = Crank radius = 2.500 in.

c = Distance from centerline of crankpin to tip of journal = .937 in.

e = Distance from center of crankpin to center of crankcheek = 1.8125 in.

f = ½ crankpin length = 1.250 in.

JOURNAL

$$Z = \frac{\pi}{32} \times \frac{95.25 - 25.60}{3.125} = .098 \times 22.3 = 2.19$$

$$Z_p = 2 \times 2.19 = 4.38$$

A = 3.693 sq. in.

CRANKPIN.

Moment of inertia of section $I_y = 1/64\ D^4 - (1/64\pi d^4 + \pi r^2 h^2)$

$I_y = 2.506$

$Z = \frac{I}{C} = \frac{2.506}{1.4375} = 1.74$

Polar section modulus $Z_p = 2Z = 2 \times 1.74 = 3.48$

$S_b = \frac{Me}{Z} = \frac{20350}{3.48} = 5850$

CRANKCHEEK

$Z_p = kbt^2 \qquad k = .265 \qquad b = 3.875 \qquad t = 1.125$

$Z_p = .265 \times 3.875 \times 1.265 = 1.29$

Z_b about HH is $\frac{b^2 t}{6} = 2.82\ in^3$

Z'_b about EE is $\frac{bt^2}{6} = .82\ in^3$

Comparative data on crankshaft stresses on crank throw nearest propeller.

ENGINE	RATING H.P.	SPEED R.P.M.	CRANKPIN DIMENSIONS			JOURNAL DIMENSIONS			CHEEK	
			I.D.	O.D.	Lg.	I.D.	O.D.	Lg.	Thickness	Width
LIBERTY	420	1700	1.250	2.375	2.500	1.375	2.625	2.00	1.000	3.470
V-1570	630	2400	1.250	2.500	1.920	2.750	3.500	1.830	.387	3.790
0-1430-1	1000	3000	2.000	2.875	2.500	2.250	3.125	2.437	1.125	3.890

ENGINE	JOURNAL		CRANKPIN		CRANKCHEEK	
	Max Tension S_b	Max Shear S_s	Max Tension S_b	Max Shear S_s	Direct Tensile Stress S_t	Max Equiv. Shear Stress S_e
LIBERTY	8000	6500	11200	8700	6730	14000
V-1570	5250	4670	9340	7770	4550	11555
0-1430-1	10200	7620	5850	9310	4940	16870

PART TWO

In order to determine the magnitude and location of the maximum and instanteous torque, single cylinder torque curves were plotted in their various phase relations, thus giving the total torque curve for each journal. It may be seen from the accompanying curves, C-19, C-20 and C-21, that the maximum does not occur in journal #7, nearest the propeller, but rather in journal #5.

TORQUE VARIATION IN EACH JOURNAL

JOURNAL	MAX. TORQUE LBS. INCHES	MIN. TORQUE LBS. INCHES	MEAN TORQUE LBS. INCHES	TORQUE RANGE LBS. INCHES
2	16,500	-9,000	4,500	25,500
3	27,500	-10,000	9,000	37,500
4	36,500	-6,000	13,500	42,500
5	43,000	-3,000	18,000	46,000 (MAX)
6	40,000	10,000	22,500	30,000
7	33,000	22,000	27,000	11,000

The axial component of force acting in the direction of the crank throw at the time that the torque ia at its maximum is obtained from the polar diagram of resultant forces acting on the crankpin, curve #C-22, showing the direction of force with respect to the crankpin.

Max. F_t = Maximum tangential force from the maximum torque

Min. F_t = Minimum tangential force from the minimum torque

Max. F_n = Corresponding axial force parallel to crankthrow = 8500 lbs.

Min. F_n = Corresponding axial force parallel to crankthrow = 7700 lbs.

$$F_t = \frac{Tm}{r}$$

$$\text{Max } F_t = \frac{43000}{2.5} = 17200 \text{ lbs.}$$

$$\text{Min } F_t = \frac{-3000}{2.5} = -1200 \text{ lbs.}$$

F_b = Resultant of F_t and F_n

F_b max = 23,850 lbs. F_b min. = 7300 lbs.

STRESSES IN JOURNAL

The bending moment in the journal, M_b, due to the force F_b is

$$M_b = \tfrac{1}{4} F_b a$$

STRESSES IN JOURNAL, cont.

$$M_b \text{ max} = \frac{23850}{4} \times 3.593 = 21{,}400 \text{ lbs.}$$

$$M_b \text{ min} = \frac{7300}{4} \times 3.593 = 6{,}560 \text{ lbs.}$$

The equivalent bending moment M_e resulting from Mb and the tortional moment in the crankshaft T_m is

$$Me = \tfrac{1}{2} M_b + \tfrac{1}{2}\sqrt{M_b^2 + T_m^2}$$

From the above, M_e max = 34,700 M_e min = 10,490

The rectangular section modulus for the journal Z = 2.19

The polar section modulus Z_p = 2 x 2.19 = 4.38

Area of section = 3.35

The maximum fiber stress $S_b = \frac{Mc}{Z}$

$$S_b \text{ max} = \frac{34700}{2.19} = \underline{15{,}850} \text{ lbs/sq in.}$$

$$S_b \text{ min} = \frac{10490}{2.19} = \underline{4800} \text{ lbs/sq in.}$$

Due to the fact that the minimum force under consideration is acting in the opposite direction to the maximum, the stress range is

15,850 + 4,800 = <u>20,650</u> lbs/sq in.

The total shear stress is $S_s = \frac{Tm}{Zp} + \frac{Fn}{2A}$

Max S_s = <u>10,950</u> lbs/sq in.

Min S_s = <u>1,835</u> lbs/sq in.

Stress range = <u>12,785</u> lbs/sq in.

STRESSES IN CRANKPIN

Using the load values above and following the general method as described in A.S.I.C #421, the maximum fiber stress in the section is

S_b max <u>20,000</u> lbs/sq in.

S min <u>4,140</u> lbs/sq in.

STRESSES IN CRANKPIN, cont.

Stress range = 24.140 lbs/sq in.

The total shear stress is

S_s max = 13,620 lbs/sq in.

S_s min = 2,010 lbs/sq in.

Stress range = 15,630 lbs/sq in.

STRESSES IN CRANK CHEEK

Following the standard method using the load values as given before, the direct tensile stress is

S_t max = 6680 lbs/sq in.

S_t min = 1180 lbs/sq in.

Stress range = 7860 lbs/sq in.

The shear stress is

S_s max = 24,900 lbs/sq in.

S_s min = 1,690 lbs/sq in.

The maximum equivalent stree resulting from S_t and S_s is

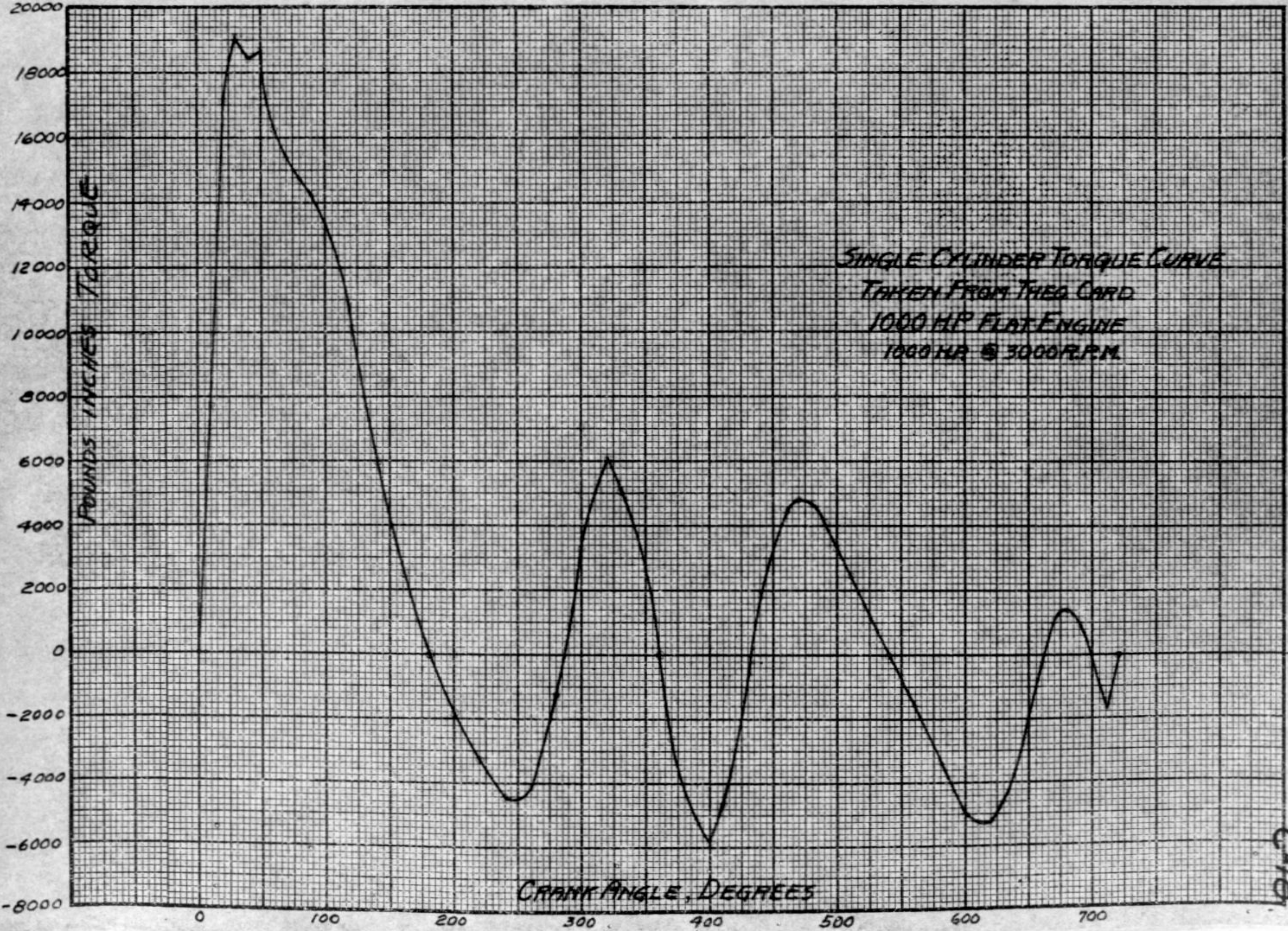
S_e max = 28,460 lbs/sq in.

S_e min = 2,380 lbs/sq in.

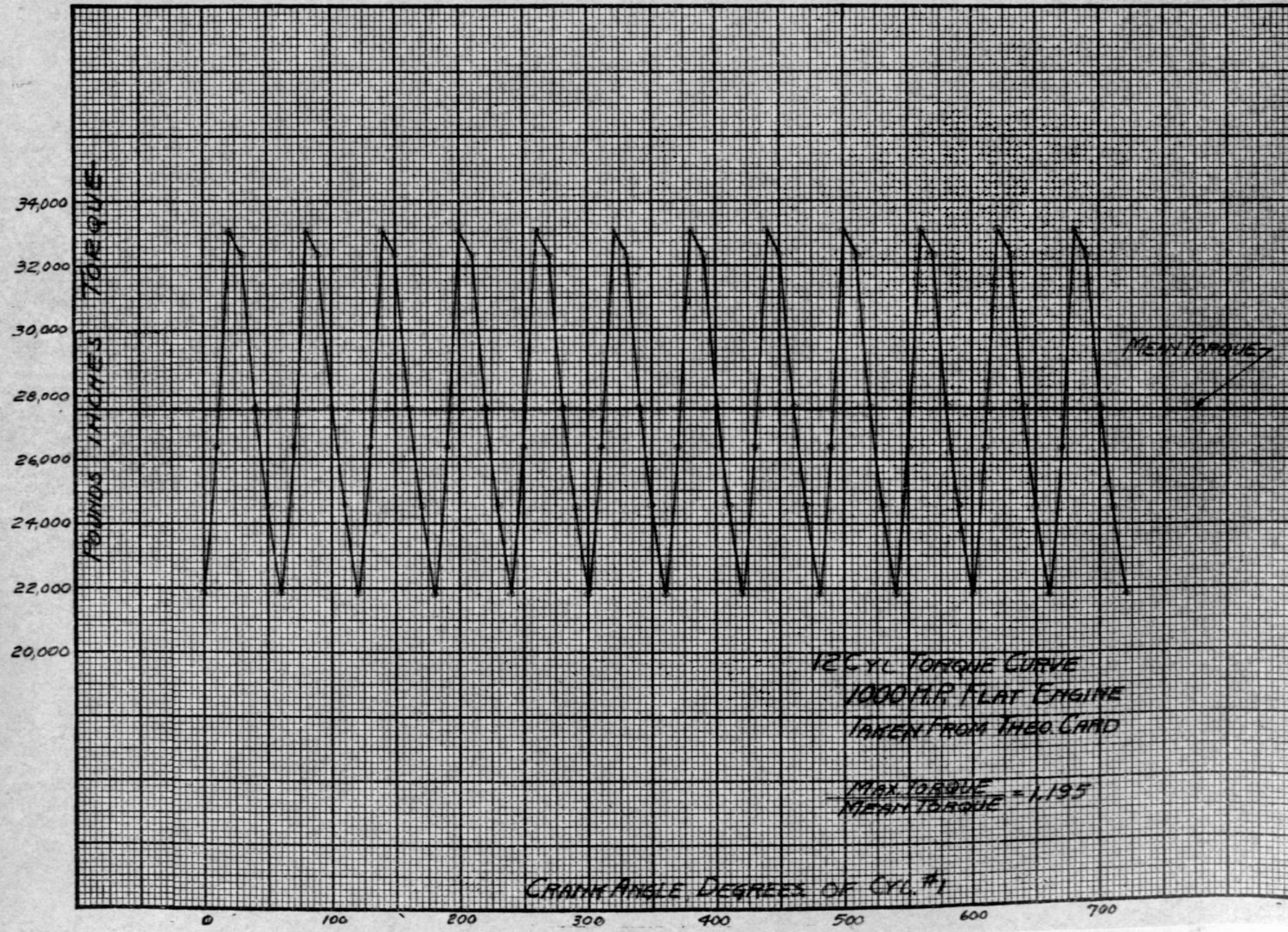
Stress range = 30,840 lbs/sq in.

The above method does not take into account the overlap in the crankpin and journal sections, which materially adds to the stiffness. The maximum equivalent stress range in the crankcheek is undoubtedly decreased by this overlap. The elastic limit for the crankshaft steel will not be less than 120,000 lbs/sq.in., which gives a safety factor of four.

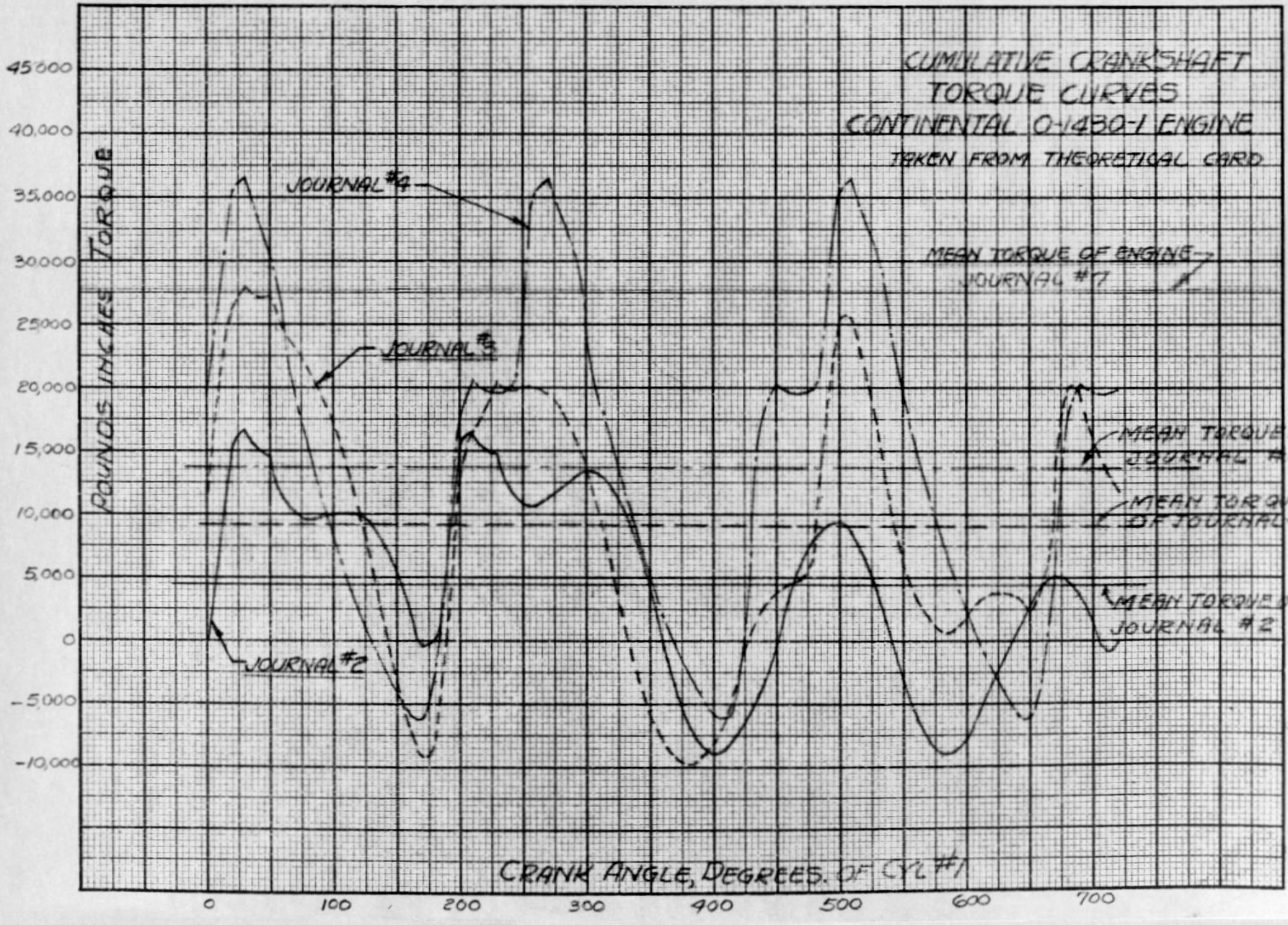
Complete information is not at hand to calculate comparable data, using the known maximum torque values, as they appear in successive cheeks of the crankshaft. It would be interesting to make such a calculation on existing crankshafts, where failures have been encountered, and undoubtedly some valuable information for use in future designs could be compiled.


SINGLE CYLINDER TORQUE CURVE
TAKEN FROM THEO CARD
1000 H.P. FLAT ENGINE
1000 H.P. @ 3000 R.P.M.
POUNDS INCHES TORQUE
CRANK ANGLE, DEGREES
20000
18000
16000
14000
12000
10000
8000
6000
4000
2000
0
-2000
-4000
-6000
-8000
0
100
200
300
400
500
600
700

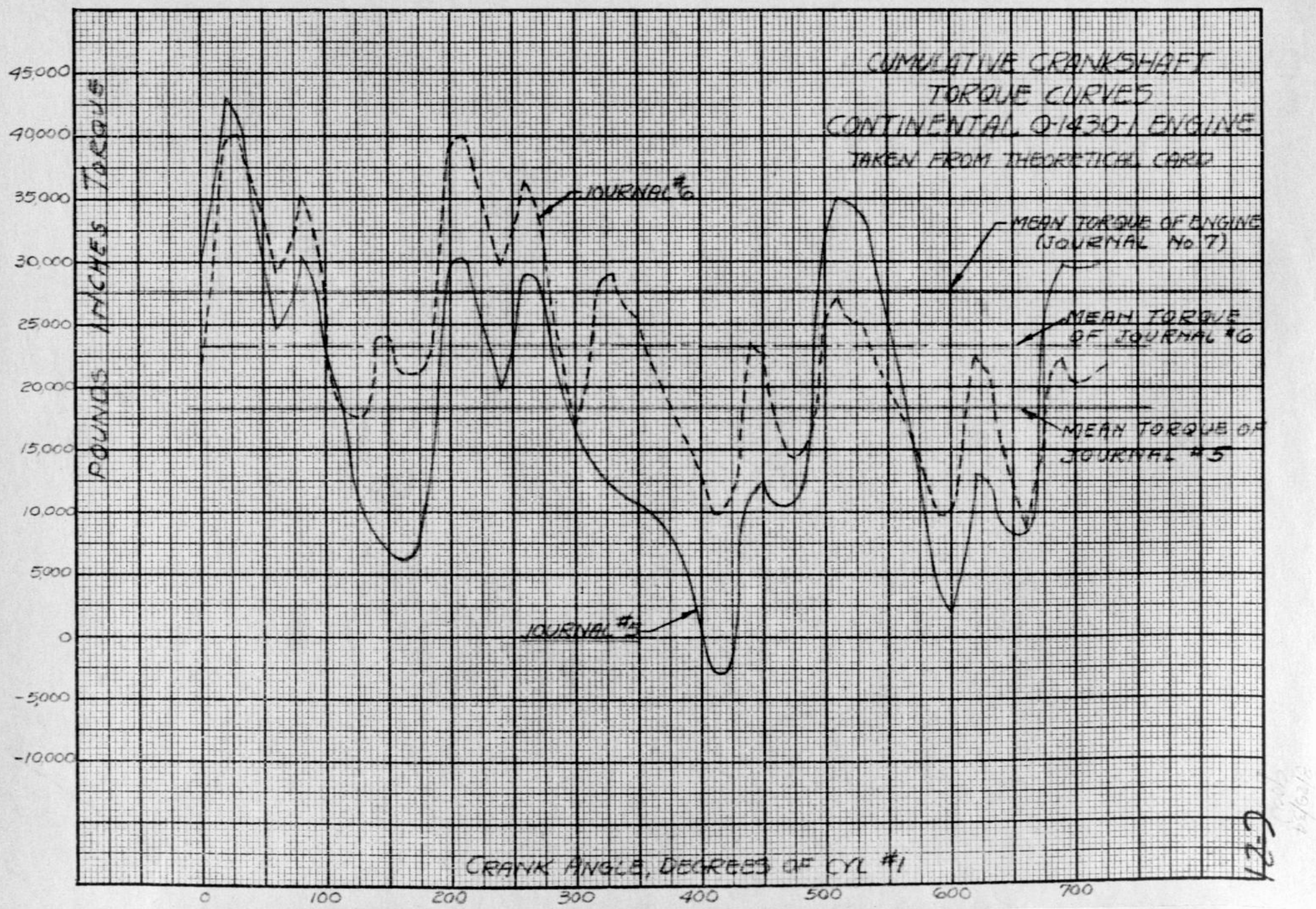
C-18


34,000
32,000
30,000
28,000
26,000
24,000
22,000
20,000
POUNDS INCHES TORQUE
MEAN TORQUE
12 CYL TORQUE CURVE
1000 H.P. FLAT ENGINE
TAKEN FROM THEO. CARD
$\frac{\text{MAX. TORQUE}}{\text{MEAN TORQUE}} = 1.195$
CRANK ANGLE, DEGREES OF CYL. #1
0
100
200
300
400
500
600
700

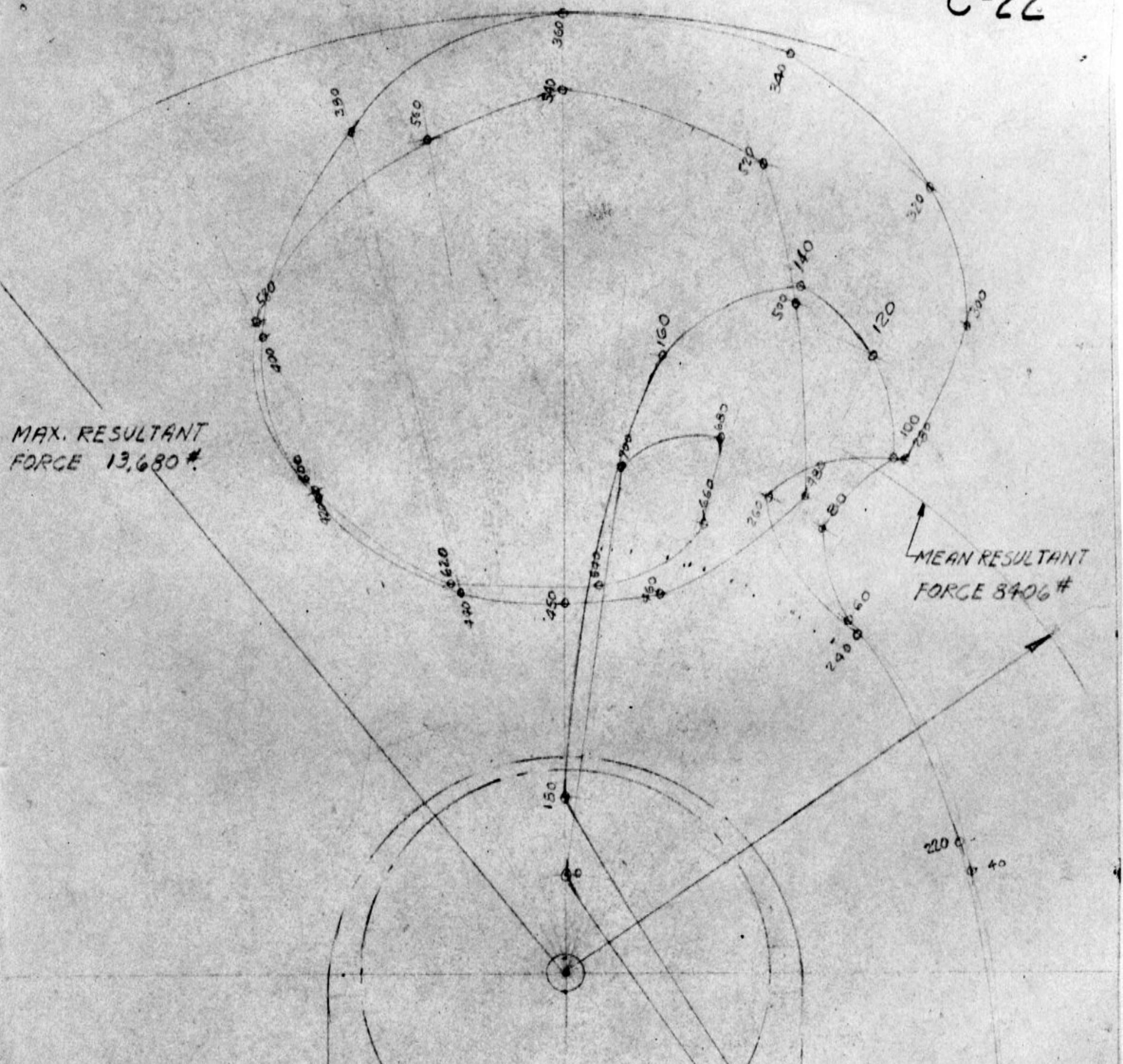
C-19


CUMULATIVE CRANKSHAFT TORQUE CURVES
CONTINENTAL O-1430-1 ENGINE
TAKEN FROM THEORETICAL CARD
POUNDS INCHES TORQUE
CRANK ANGLE, DEGREES. OF CYL #1
JOURNAL #4
JOURNAL #3
JOURNAL #2
MEAN TORQUE OF ENGINE - JOURNAL #7
MEAN TORQUE OF JOURNAL #4
MEAN TORQUE OF JOURNAL #3
MEAN TORQUE OF JOURNAL #2
45,000
40,000
35,000
30,000
25,000
20,000
15,000
10,000
5,000
0
-5,000
-10,000
0
100
200
300
400
500
600
700

C-20



CUMULATIVE CRANKSHAFT
TORQUE CURVES
CONTINENTAL O-1430-1 ENGINE
TAKEN FROM THEORETICAL CARD
JOURNAL #6
MEAN TORQUE OF ENGINE
(JOURNAL No 7)
MEAN TORQUE
OF JOURNAL #6
MEAN TORQUE OF
JOURNAL #5
JOURNAL #5
POUNDS INCHES TORQUE
CRANK ANGLE, DEGREES OF CYL #1
45,000
40,000
35,000
30,000
25,000
20,000
15,000
10,000
5,000
0
-5,000
-10,000
0
100
200
300
400
500
600
700



POLAR DIAGRAM FORCES ON CRANKPIN RELATIVE TO CRANKTHROW

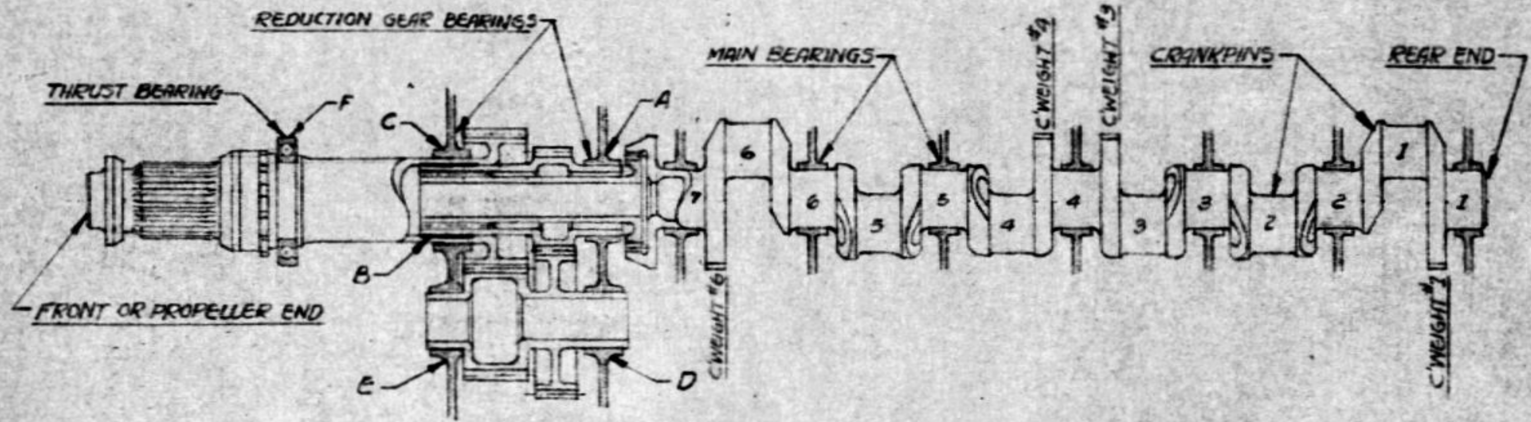
INENTAL O-1430-1 ENGINE (3000 R.P.M.) SCALE 1" = 2000 LBS

ALE 1" = 2000 #

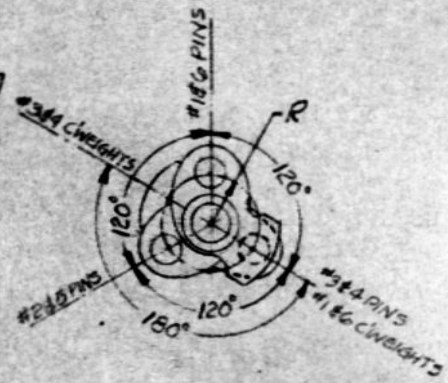
J.H.S.

APR-11-34

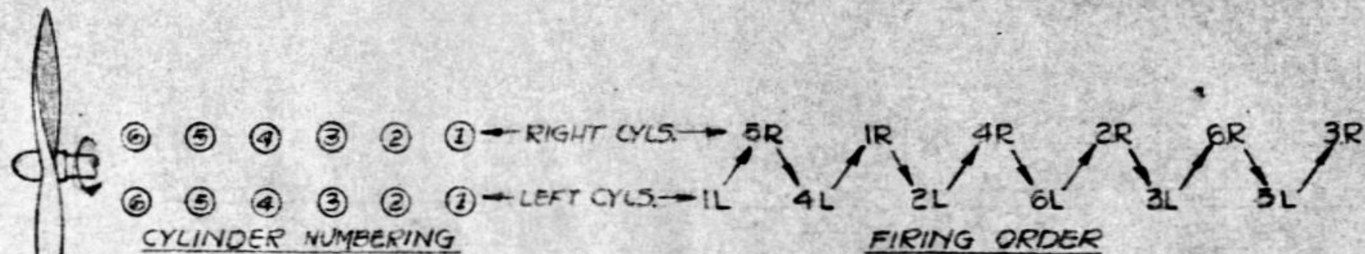
(PRELIMINARY ONLY)



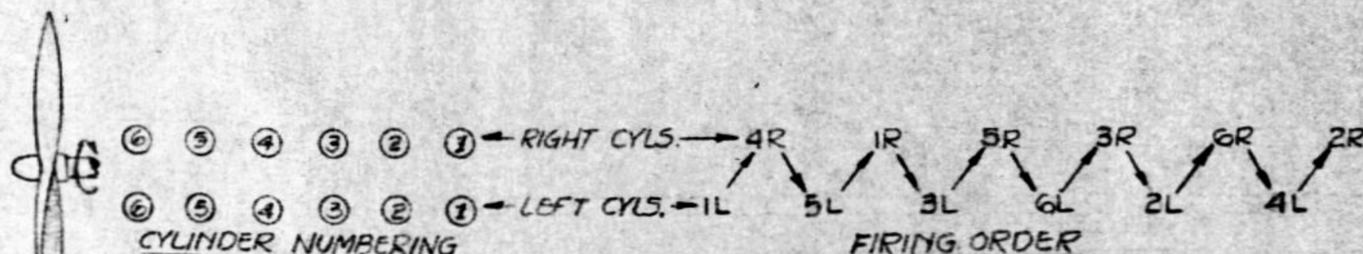
LOCATION OF CRANKPINS, MAIN BEARINGS, AND REDUCTION GEAR BEARINGS



CRANK & COUNTERWEIGHT ARRANGEMENT FACING REAR OF ENGINE



COUNTER-CLOCKWISE ROTATION



CLOCKWISE ROTATION

CRANK ARRANGEMENT, CYLINDER NUMBERING AND FIRING ORDER - CONTINENTAL 0-1430-1 ENGINE