COEFFICIENT OF FRICTION OF BALL BEARINGS

BY

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ARMOUR INSTITUTE OF TECHNOLOGY

1919

23657 / 22 Sawyh

The Coefficient of Friction of Ball Bearings and Horse Power to Drive

A THESIS

PRESENTED BY

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TO THE

PRESIDENT AND FACULTY

OF

ARMOUR INSTITUTE OF TECHNOLOGY

FOR THE DEGREE OF

BACHELOR OF SCIENCE

IN

MECHANICAL ENGINEERING

MAY 29, 1919

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APPENDIX

DETERMINATION OF THE COEFFICIENT OF FRICTION OF BALL BEARINGS AND THE HORSE POWER REQUIRED TO DRIVE

The object of this investigation is the determination of the coefficient of friction of ball bearings and the horse power required to drive them, under various loads and speeds.

The machine for making this investigation was designed by Prof. G. E. Gebhardt of the Armour Institute of Technology and built in the shops of the school. It consists essentially of four ball bearings and a shaft, mounted as shown in figure 1. The two outer bearings support the shaft, while the two inner ones are merely hung on it at equal distances from the outer bearings. The machine is so made as to permit the placing of axial as well as radial loads on all four bearings, either independently or in any combination of the two.

The details of the machine will not be taken up. It consists of a base A, figure 1, on which are

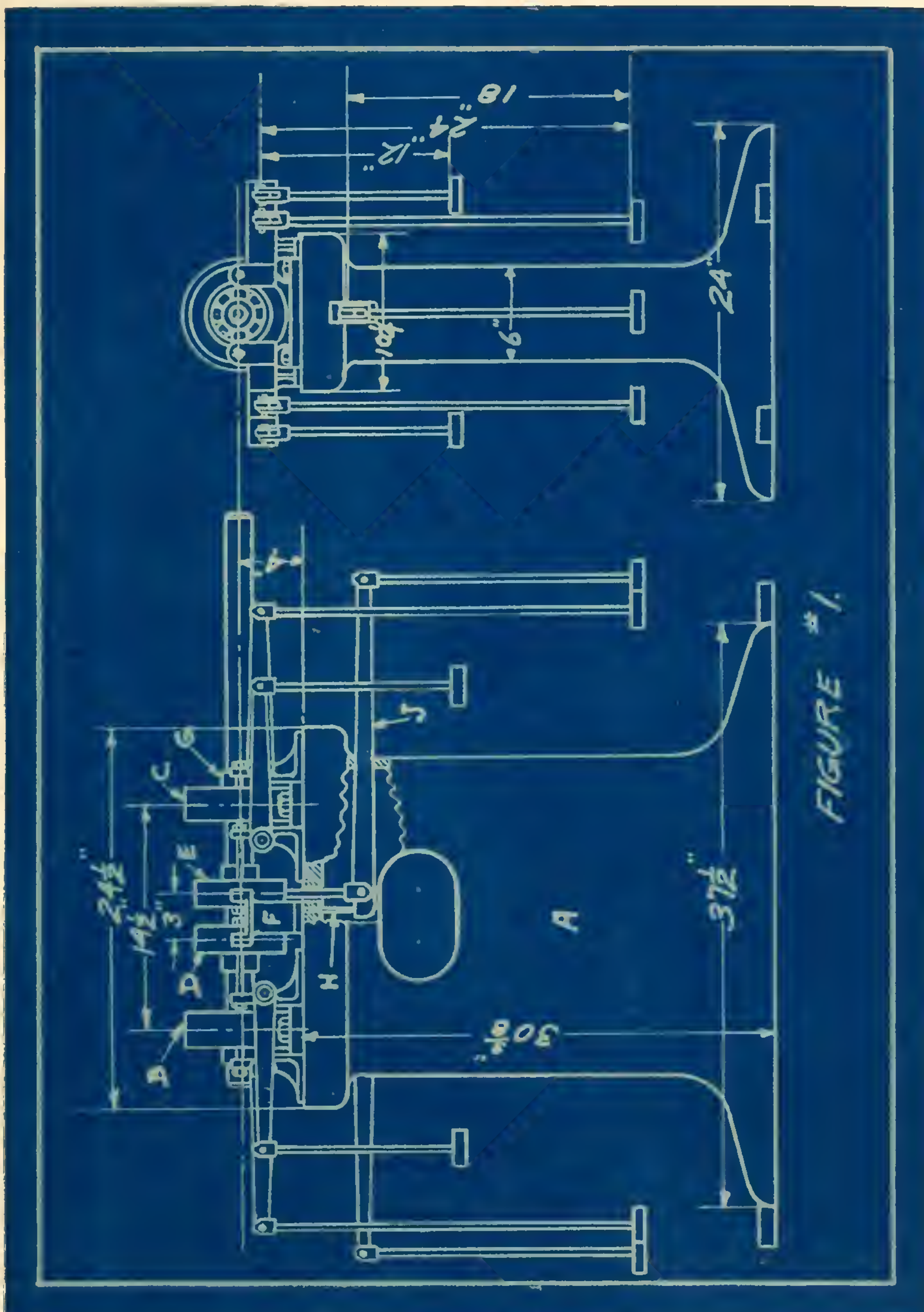


FIGURE #1.

mounted two Hess-Bright #6309 ball bearings, B and C. The two central bearings, also Hess-Bright #6309, D and E, are mounted on the shaft, but not on the base; the housing being kept from rotating by the casting F, shown more clearly in figure 2. The shaft is connected by means of a flange coupling to a sensitive electric cradle dynamometer. The lock rings G-G keep the bearings in a fixed position relative to the shaft.

The method of applying the radial load can be seen in figure 1. The pin H is screwed into the base, and is the fulcrum for the lever J. The ratio of the lever arms is 20 to 1, therefore any load applied on the end of the lever will induce a load 20 times this on the bearing. The method of fastening the rod K into the housing can be seen in figure 2. The rod L passes through the housing of the bearing and also through the short arm of the

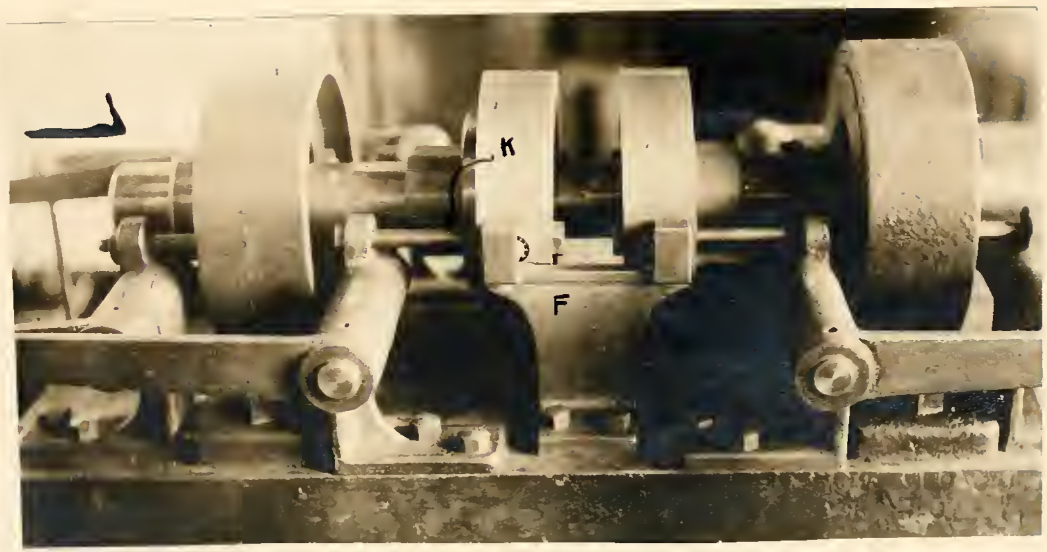


Figure 2.

lever M, being held in place by two conical headed nuts. Thus a load applied at the end of the lever M will be transmitted to the housing of the bearing, which in turn presses against the outer race of the ball bearing, the inner race being pressed against the collar on the shaft. This latter arrangement can be followed out quite easily in figure 3. The ratio of the arms on lever M is 10 to 1, thus any load placed on the end of the lever will transmit a load ten times as great to the bearing.

In conducting this investigation, the machine had to be calibrated before any tests could be made, which consisted of finding the dead weights of the levers (this includes the weight rods on the ends of the levers.) To accomplish this a bell crank was made, as shown in figure 4. This was placed on the end of the base with the aid of two small brackets, as shown in figure 5, so that the hole B would line up with the bolt C. A wire was then

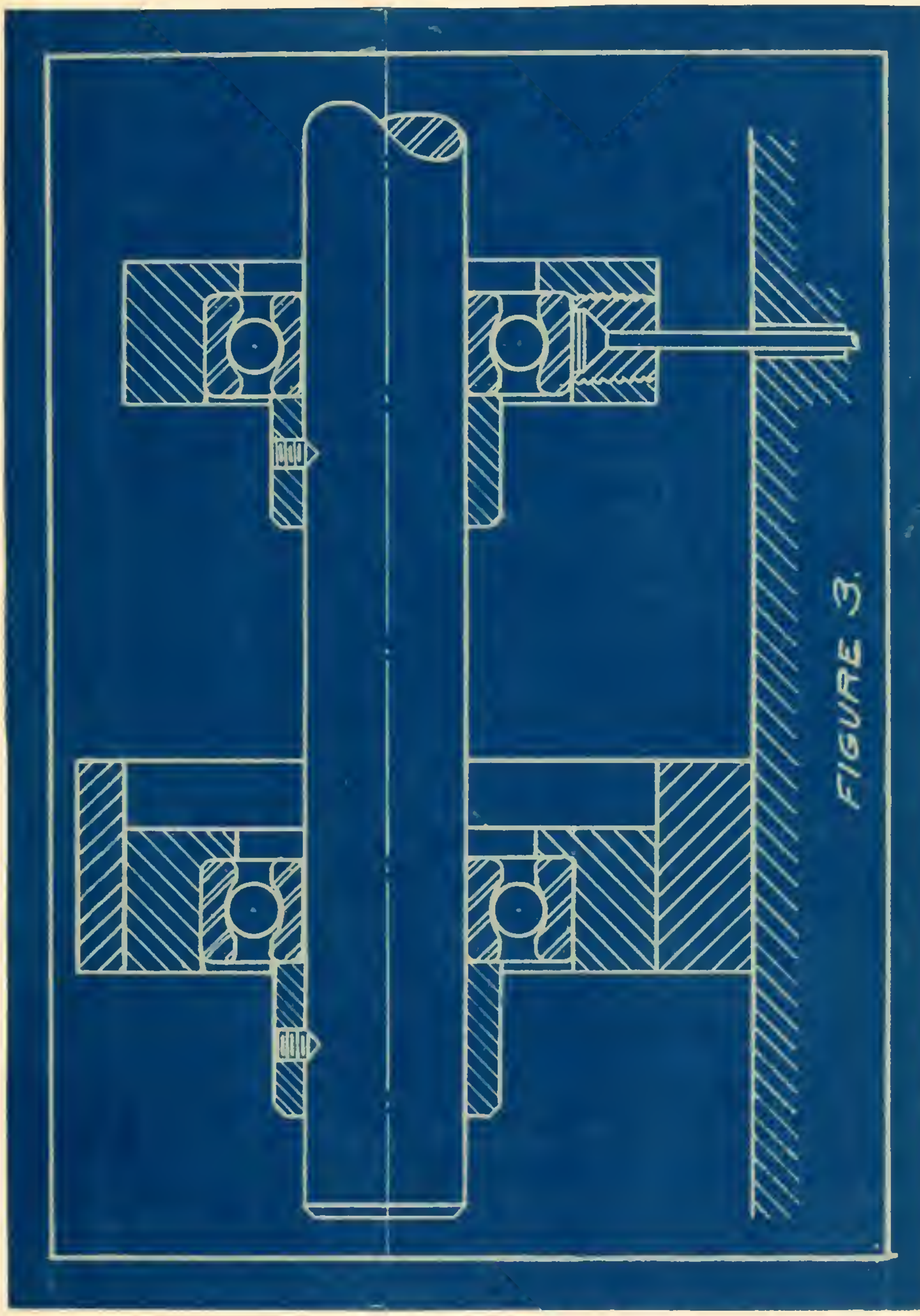


FIGURE 3.

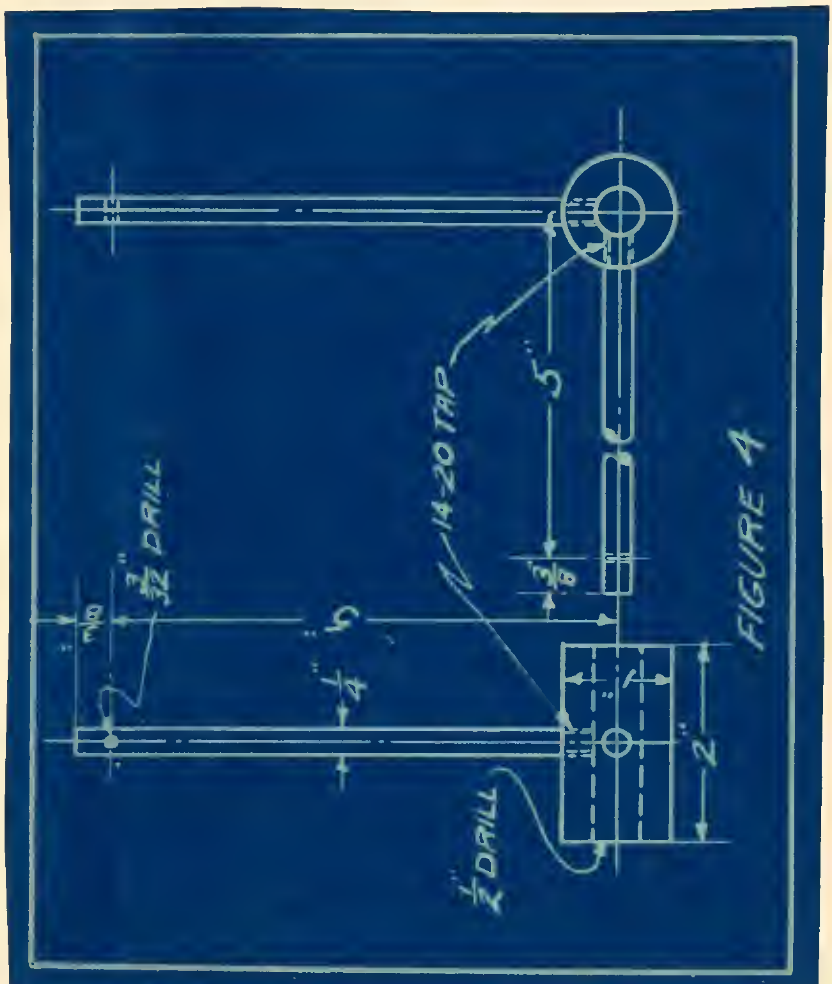


FIGURE A

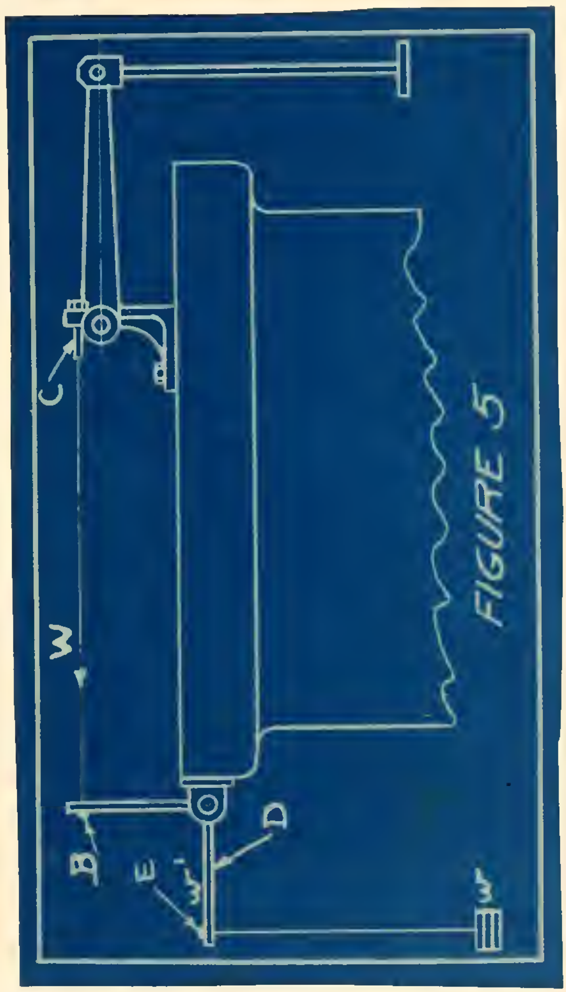


FIGURE 5

passed through the hole B, and fastened to C. Another wire was passed through the hole E, so that the weights could be placed on the end of it. Weights were now placed on the end of the arm D, until a balance accurred.

Knowing the weight of the arm D, and the load on it, a moment equation can be set up as follows:

$$5\ W = 5w + 2\ \frac{15}{16}\ w$$

$$W = w + .588\ w \quad (1)$$

Since w is so small compared with the other weights, the term .588w may be omitted leaving $W = w$.

In following out the above procedure it was found that the levers with 12 inch weight rods gave an 11 pound pull at the bearing, while the levers with 24 inch weight rods produced a 15 pound pull at the bearing. Enough weights were added to the weight rods to produce a 100 pound axial load on the bearings.

The radial load dead weights were found by weighing the radial load levers and weight rods, and by finding the center of gravity of the levers. Knowing these values, the value of P, figure 6, can be calculated from the following equation:

$$P \times l = R \times W + 10 \times w \quad (2)$$

In which -

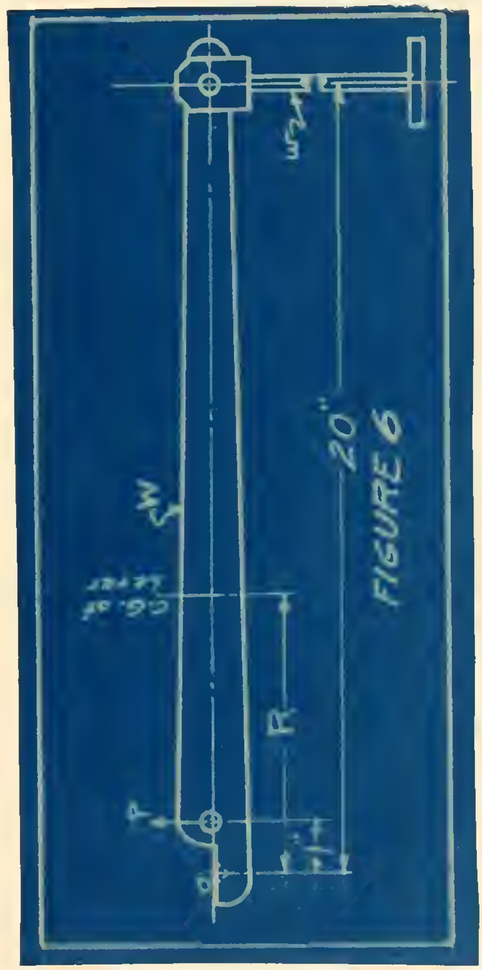
P - pull on the bearing
R - distance of pivot to center of gravity
W - weight of lever
w - weight of weight rod.

For one lever the following values were obtained, R = 8 5/8", W = 2.6#, w = 1#

Substituting these values in equation (2) we get P = 42.4#.

For the other lever the following values were obtained R = 8 1/4", W = 2.3#, w = 1#, P = 39#.

Enough weights were added to the weight rods to produce a 50 pound radial load on each of the bearings.


20"
R
FIGURE 6

The dynamometer was uncoupled from the machine, and its zero load determined, which was made 0.5 pounds. The length of the dynamometer arm was found to be $31\frac{1}{2}$". The dynamometer was recoupled to the machine, the apparatus now being ready for the first run.

Before entering into the actual testing let us see what determines the coefficient of friction of a ball bearing. First, we must have a definition of the coefficient of friction, which is the ratio of the force required to slide one surface over another, to the total load on the surfaces. In a ball bearing, or any bearing for that matter, it is a torque which tends to do the sliding. The torque required to drive a ball bearing is constand for a fixed load and speed, but the size of the force producing this torque varies according to the distance from the shaft. Since the co-

efficient of friction depends directly on the force, and only indirectly on the torque, some means should be made for determining this force which would be universal.

It can easily be seen that if the torque arm say, is measured to the surface of the outer race, the force, and therefore the coefficient of friction would be smaller than if measured say to the surface of the innter race. This question of measuring the torque arm offers a subject of much discussion and confusion among ball bearing manufacturers, and also between the manufacturers and buyers.

Let us assume that all manufactuers based their coefficients of friction on the distance from the center of the shaft to the surface of the inner race. This still would not solve the problem, because some manufacturers may make a thick inner race, while others would make a very thin one.

This, then would be very impracticable.

The only practical method left, and the one to solve all difficulties is to base the coefficient of friction on the radius of the shaft. It will be said that this will give a larger coefficient of friction than really exists, which is true, but, if all ball bearing manufacturers based their results on this, comparative values would be obtained which would be just as good as the actual coefficients for determining the merits of one bearing over another. Another advantage of this method of determining the coefficient of friction is that it can be compared directly to bearings of the roller and babbit type. Therefore, in the investigation, all coefficients of friction will be based on the radius of the shaft.

In conducting this thesis the following method of procudure was adopted:

A zero load run was made first. This was made with all of the levers removed, therefore, the only load on the bearings is that produced by the weight of the shaft, which amounts to about 10# per bearing. The torque necessary to drive under no load was found to 0.16 pounds for speeds as high as 2500 R.P.M. Under these conditions the coefficient of friction is constant and the horse power to drive varies directly with the speed.

The radial load test was made next. Starting with as low a speed as possible and keeping it con - stant, the bearings were loaded by increments of 100 pounds, up to 2060 pounds, the torque being noted at each new load.

Knowing the toruqe, speed and load on the bearing the horse power to drive and coefficient of driction may be calculated.

In testing out the bearings under axial loads

it was found that the two bearings supporting the shaft could not be loaded axially as originally intended, because the bearing housing would not slide freely in the main support. This necessitated the following procedure:

The two, center bearings were removed, and the machine run at different speeds under no load the torque being noted at each speed. The two center bearings were then replaced, and axial loads applied to them at different speeds, (starting with a zero load, and increasing to 800 pounds by increments of 100 pounds.) The torque was noted at each changed of speed or load, from which was subtracted the torque produced by the two outer bearings. This then gave the true torque due to the axial load on the bearings, from which may be calculated the coefficient of friction and horse power to drive.

It is not thw scope of this investigation to

determine the best shape of the races, and the number and size of balls, for the least frictional losses, but a word may be said about these factors. It has been found that the frictional resistance was least for balls rolling between straight line sections, or perfectly flat surfaces, giving two points of contact. Increasing the points of contact to three and four produced higher frictional resistance, without materially affecting the carrying capacity. Curving the race resulted in an important increase in carrying capacity, with a barely measurable increase in friction. The spacing of the balls by a separator has proven to be more satisfactory, allowing them to rest one against the other.

Stribeck developed from his experiments the following equation for the carrying capacity of an annular or radial bearing:

$$L = Knd^2 \quad \text{in which}$$

L = load capacity in pounds

d = ball diameter in eights of an inch, e.g., $\frac{1}{2}$ inch diameter ball, d = 4

n = number of balls

k = a constant dependent upon the material the shape of the ball - supporting surface and the speed.

An attempt was made at getting the coefficient of friction and horse power to drive under a combination of radial and axial loads, but the results obtained were inconsistent with the results obtained when the loads were applied individually. In most cases no change was noted between combination loads and radial loads alone. An explanation of this arises from the fact that the ball bearing did not slide freely in the housing as soon as a radial load was applied. It was therefore necessary to redesign the two center housings, as shownnin figure 7. Time did not permit the making of the housings and therefore presentable data was not obtained for combination.

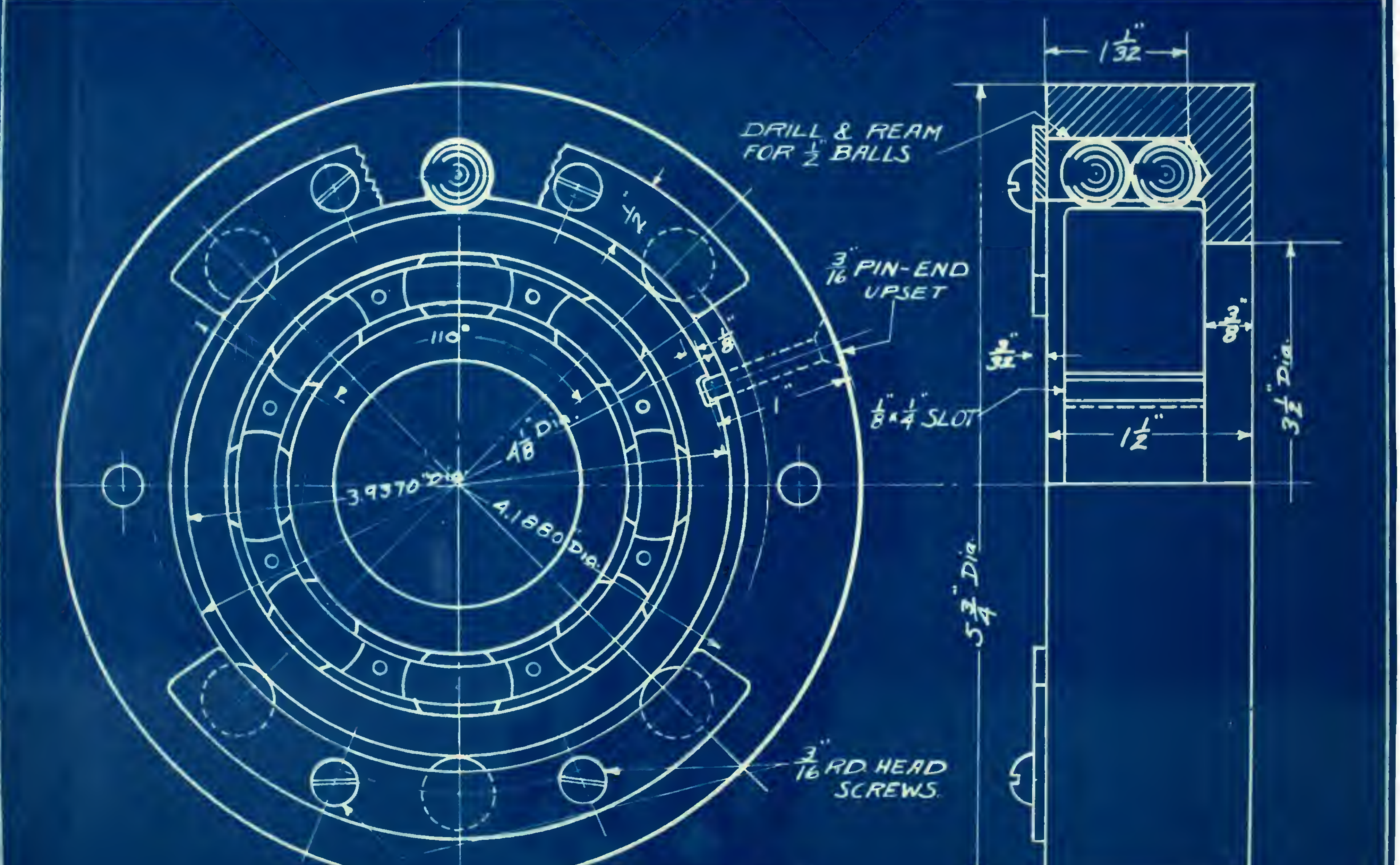

DRILL & REAM FOR 1/2" BALLS
3/16" PIN-END UPSET
1/8" x 1/4" SLOT
3/16" RD. HEAD SCREWS.
3.9370" DIA.
4.1880" DIA.
110°
5 3/4" DIA.
3 1/2" DIA.
1 1/32"
1 1/2"
3/32"

FIGURE 7.

loads.

If now we refer to the various curves we can study the action of the bearings. Referring to the radial load curve (horse power vs. speed) we see that the horse power to drive varies directly with the speed and increases with the load on the bearing. Since these are straight lines the equation of them will be in the form of $y = m x + b$, but b is zero for all of these, therefore the equation will reduce $y = m x$ or H.P. = m×R.P.M. wherein m is the slope of the line and has the following values:

Radial Load in pounds.	m	Radial load in pounds.	m
200	.0000375	1200	.0000545
400	.000040	1400	.0000605
600	.0000430	1600	.0000672
800	.0000465	1800	.0000740
1000	.000050	2000	.0000835

For intermediate loads we can get m by interpolation.

Referring to the axial load curve (horse power vs. speed) we see that the curves are straight lines up to 1600 R.P.M. Thus the same equation will hold here, as was used for radial loads, H.P. = m R.P.M. in which m has the following value as determined from the curves:

Axial load in pounds	m
200	.00002
400	.0000280
600	.0000343
800	.00004075

For intermediate loads get m by interpolation.

If we turn to the coefficient of friction curves we see that the two curves have practically the same form, the only difference being that one is drawn closer to the y axis than the other. They show that the coefficient of friction is less for axial loads, varying from 0 to 1000 pounds, than for this range in radial loads. Beginning with 1000 pounds and up the coefficient of friction is the same for both types of loads.

APPENDIX

Sample Calculations

Radial Load

Let C - coefficient of friction
T - total torque (torque of 4 bearings) in inch Pounds
R - Radius of shaft in inches
L - Load on bearing in pounds

H.P. - horse power to drive one bearing
R - radius of dynamometer arm - $31\frac{1}{2}$"
N - R. P. M.
W - net scale reading in pounds

Calculations made for 1260# load and 1000 R.P.M.

$$C = \frac{T}{4\ r\ L} = \frac{.46 \times 31.5}{4 \times .8858 \times 1260} = \underline{\underline{.00324}}$$

$$H.P. = \frac{2\ \pi\ R\ N\ W}{33,000 \times 4} = \frac{2\ \pi \times 31.5 \times 1000 \times .46}{12 \times 33,000 \times 4} = \underline{\underline{.0576}}$$

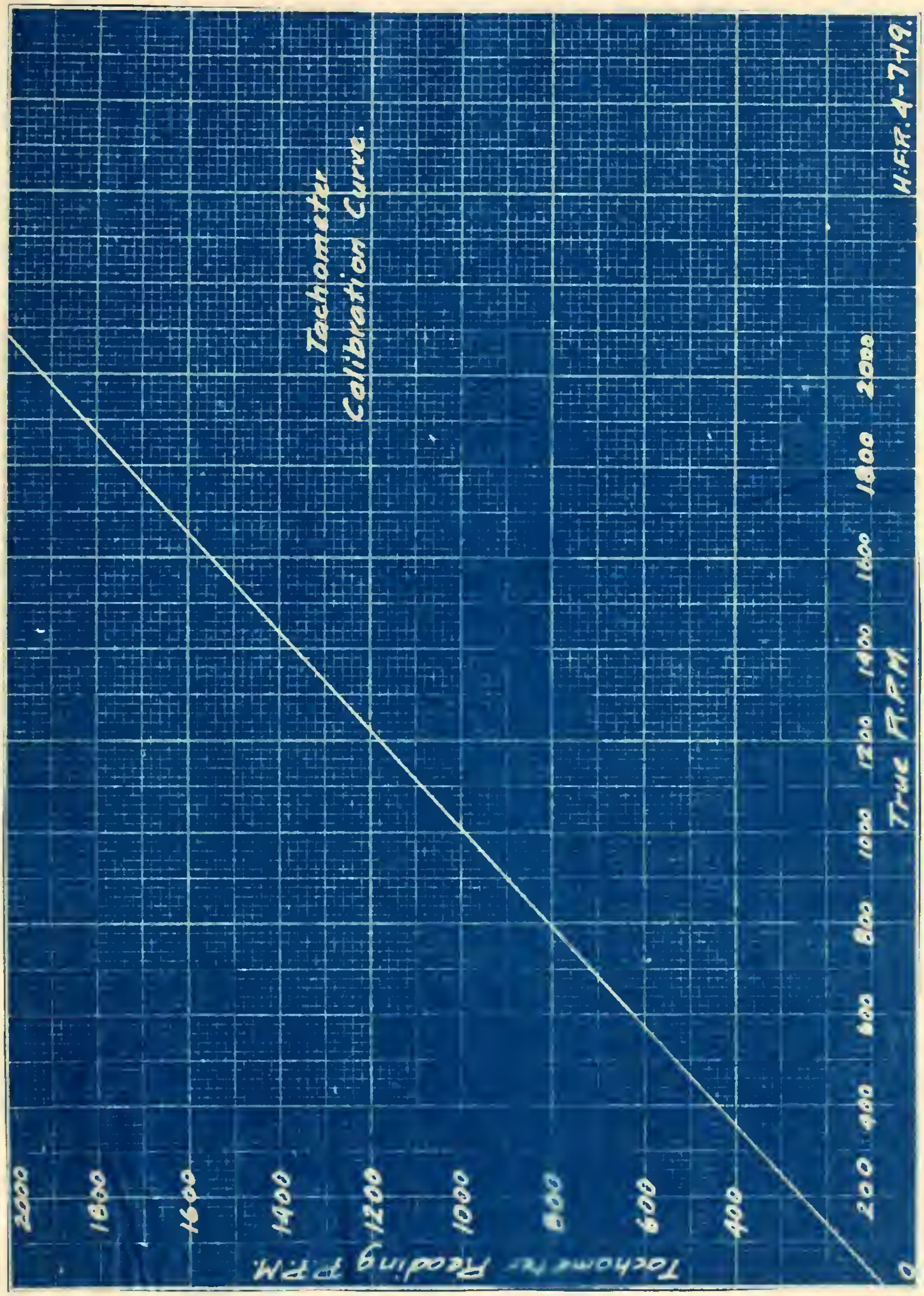
Axial Load

Notations same as above.

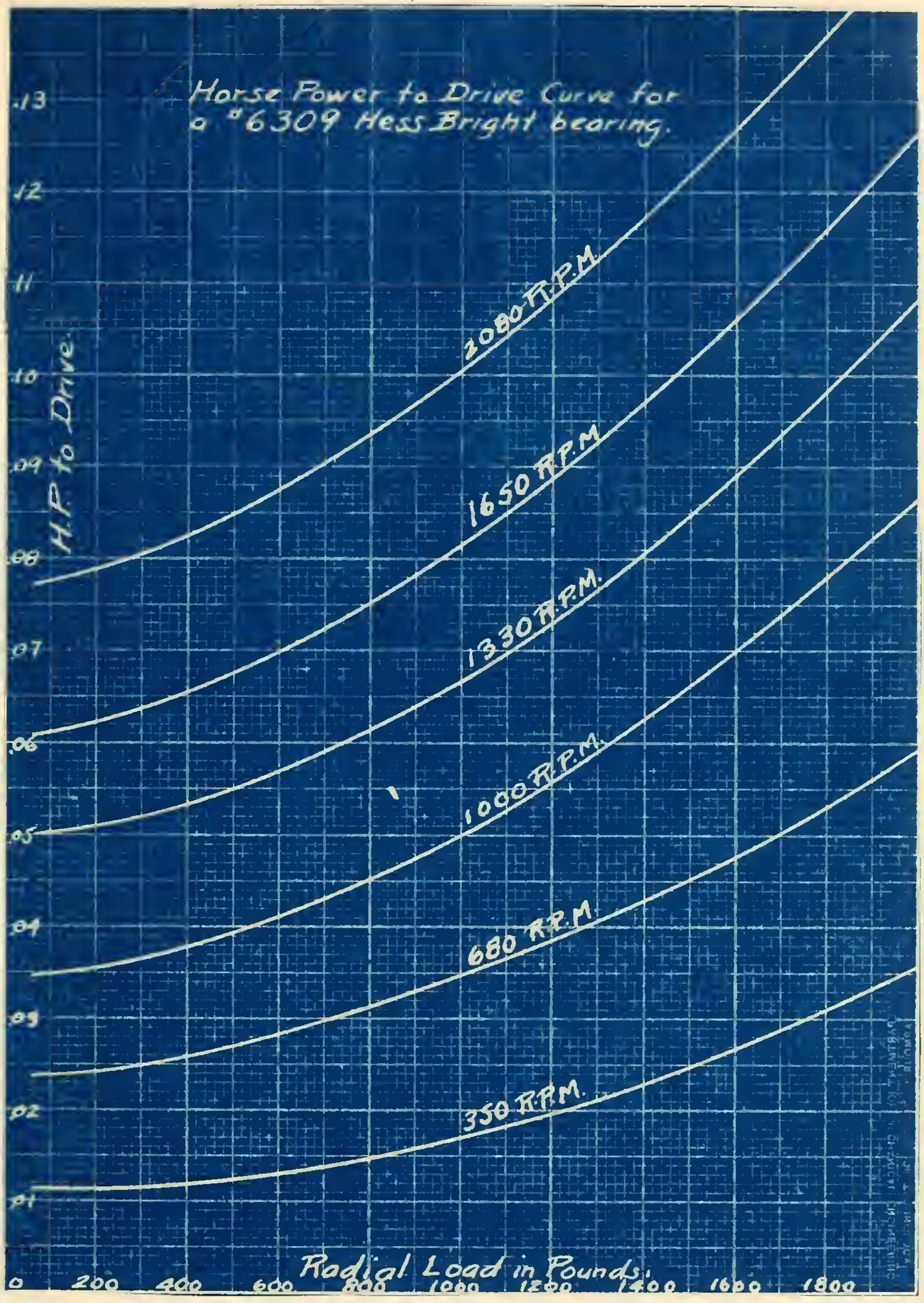
Calculations made for 400# load and 1010 R. P. M.

$$C = \frac{T}{2\ r\ L} = \frac{.12 \times 31.5}{2 \times .8858 \times 400} = \underline{\underline{.00533}}$$

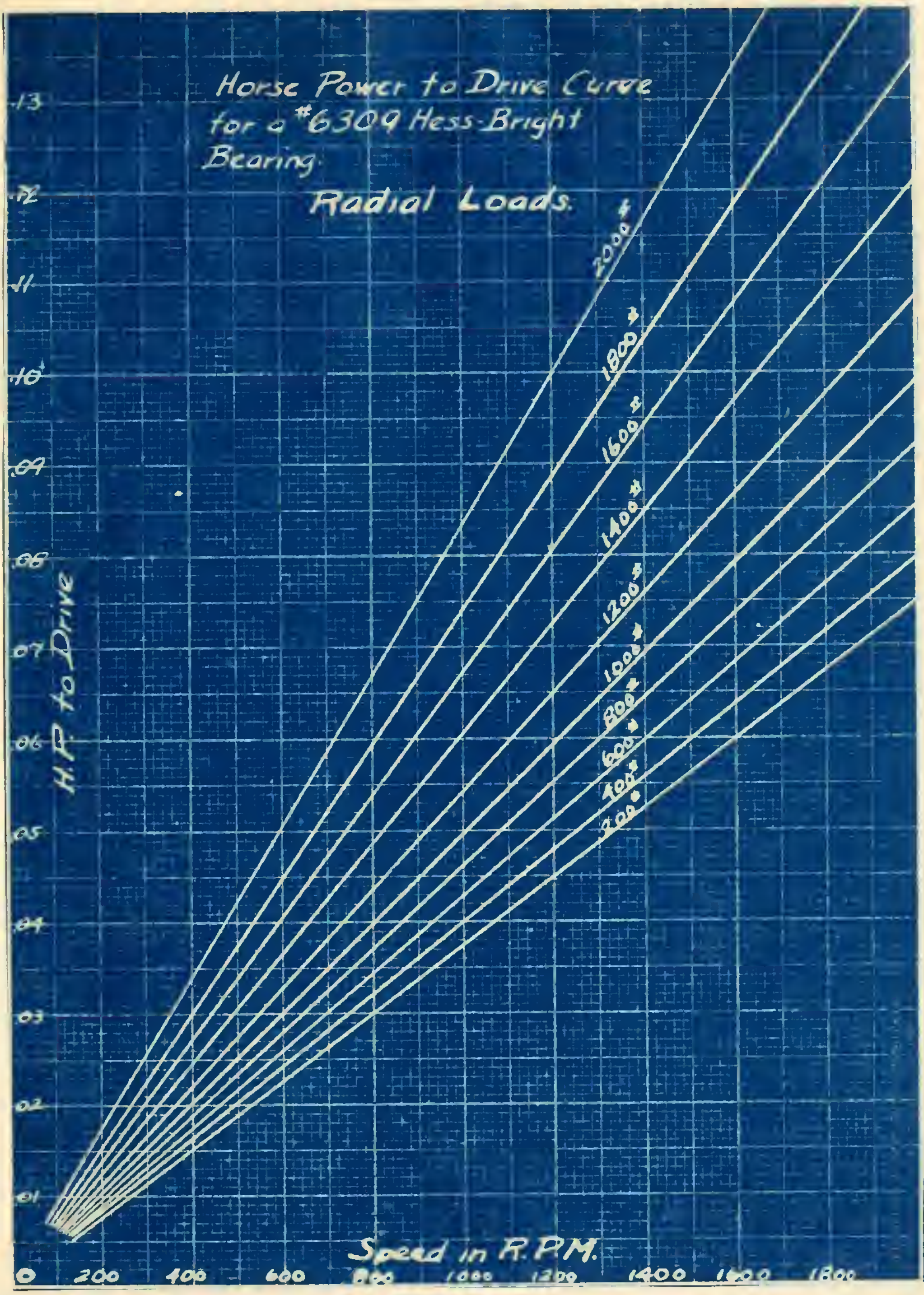
$$H.\ P. = \frac{2\ \pi\ R\ N\ W}{33,000 \times 2} = \frac{2\ \pi \times 31.5 \times 1010 \times .15}{12 \times 33,000 \times 2} \doteq \underline{\underline{.0304}}$$

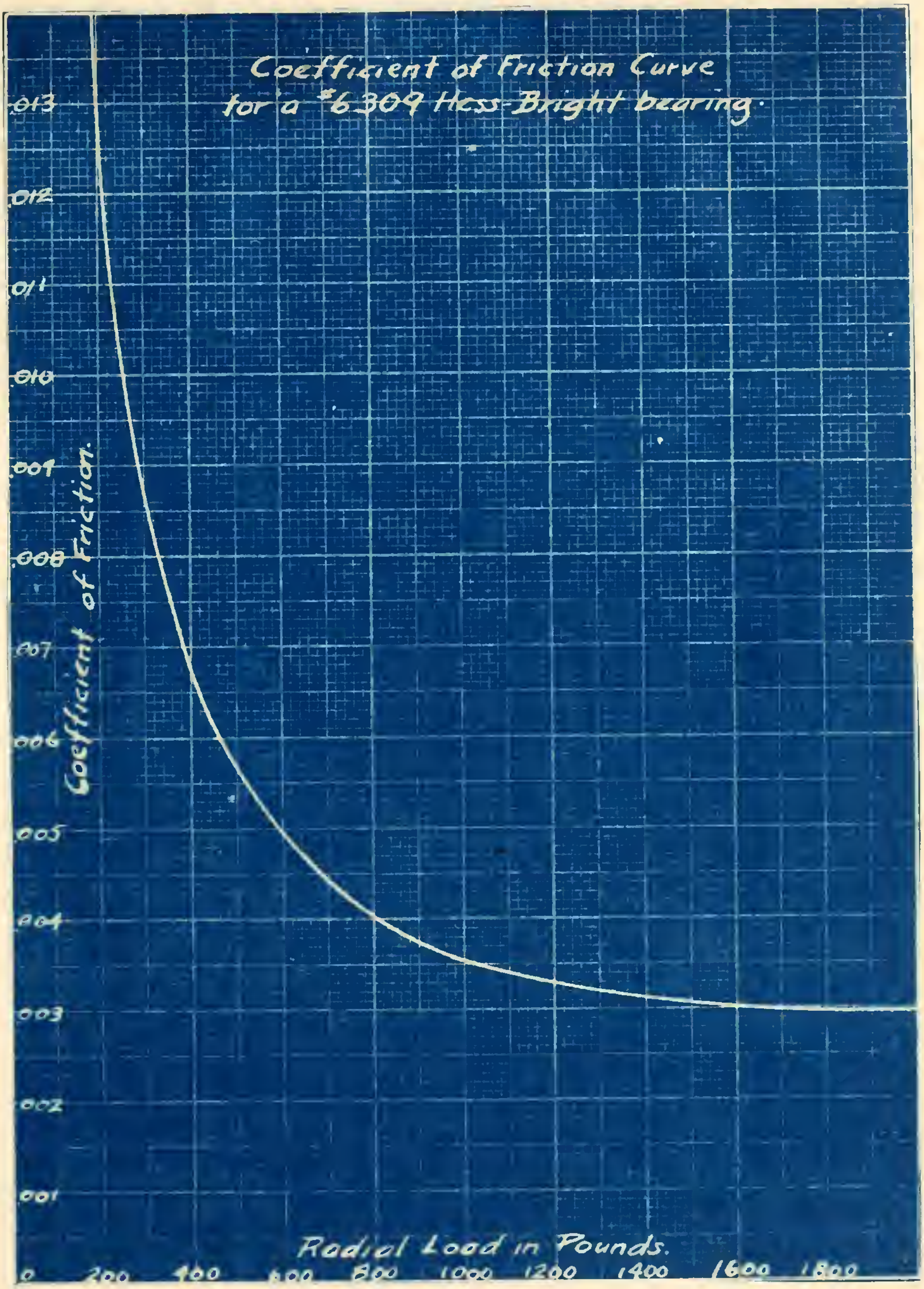


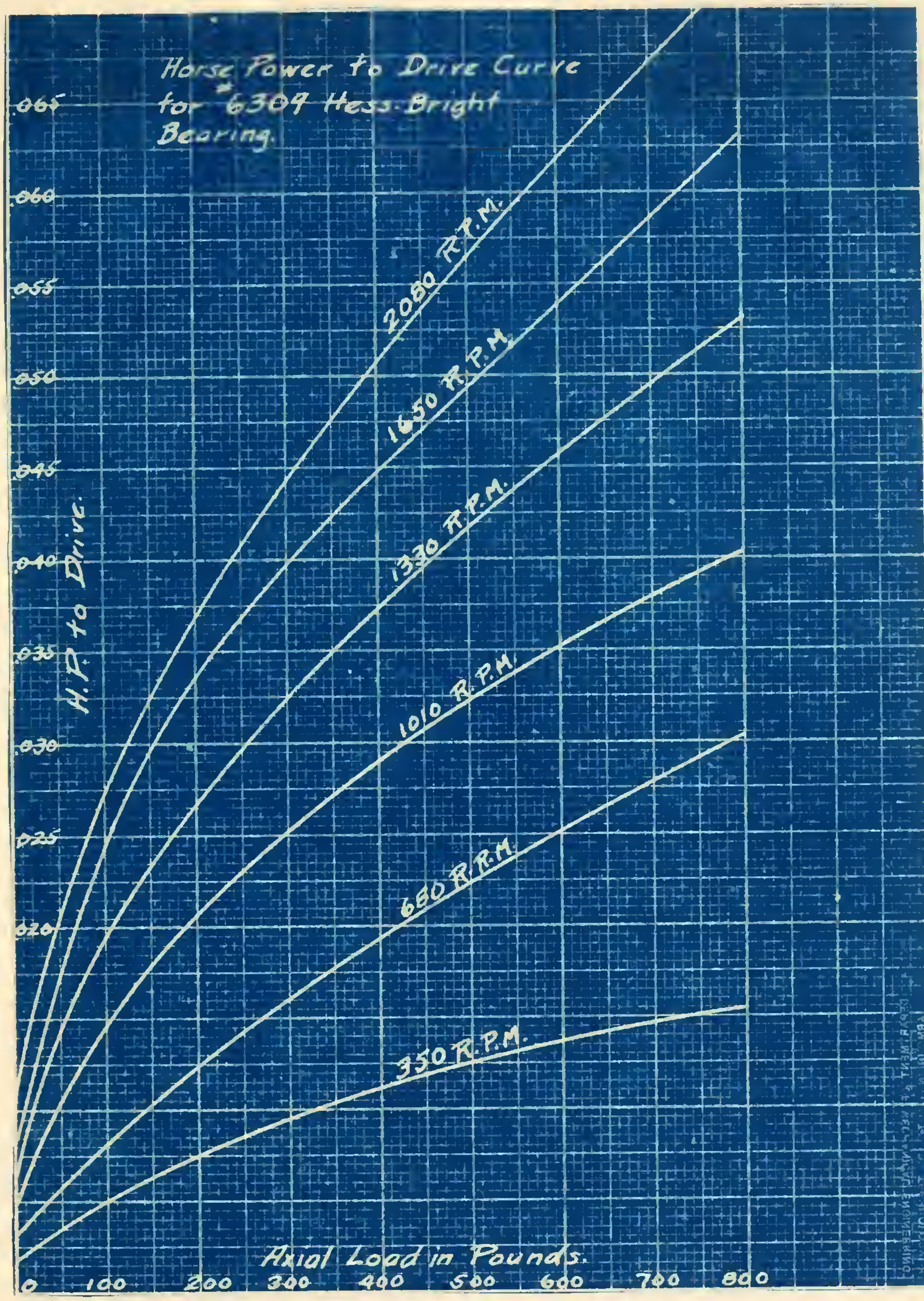
Tachometer
Calibration Curve.
H.F.R. 4-7-19.
True R.P.M.
0
200
400
600
800
1000
1200
1400
1600
1800
2000
Tachometer Reading R.P.M.
400
600
800
1000
1200
1400
1600
1800
2000



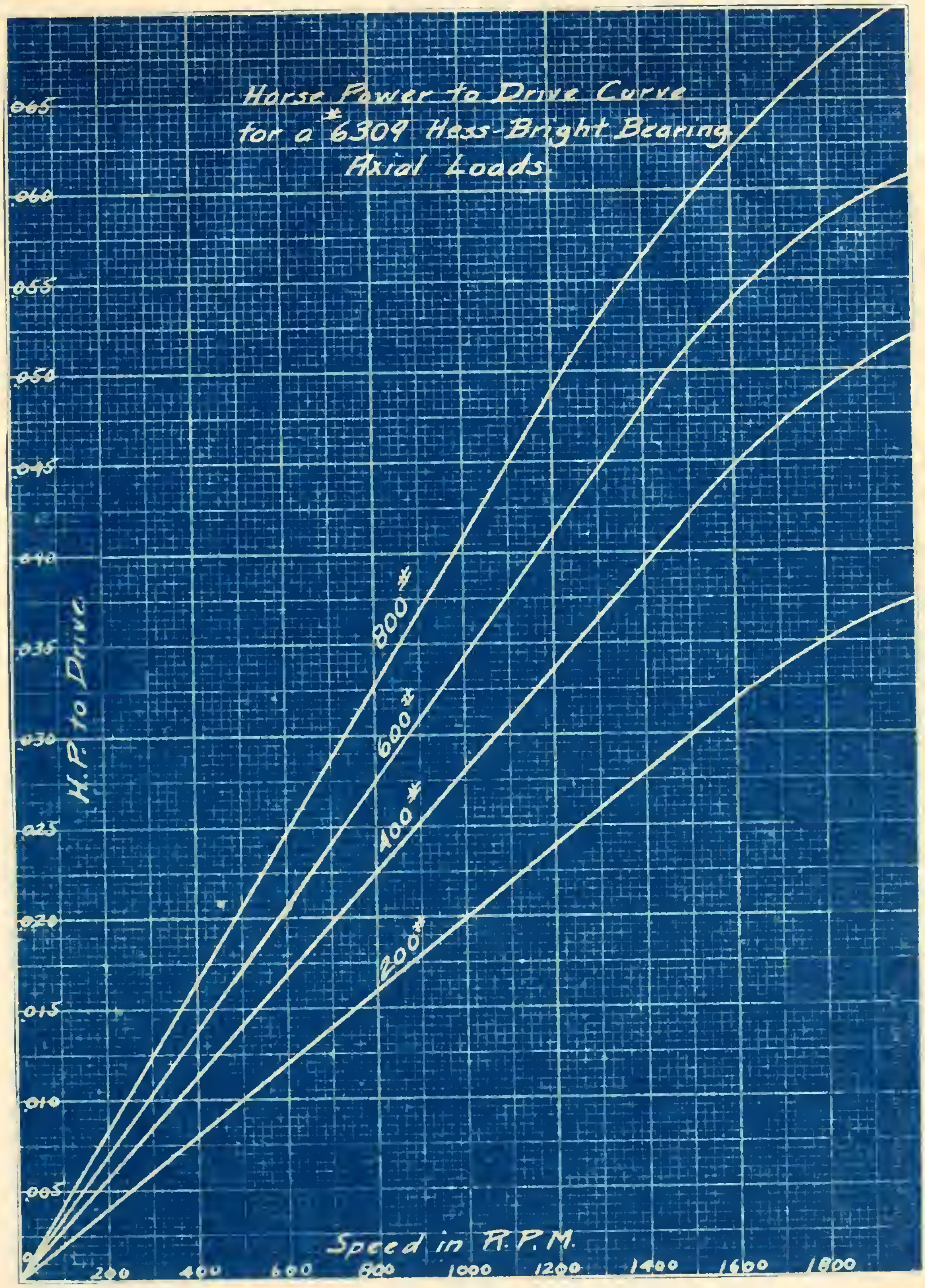
Horse Power to Drive Curve for
a #6309 Hess Bright bearing.
H.P. to Drive.
.13
.12
.11
.10
.09
.08
.07
.06
.05
.04
.03
.02
.01
0
2080 R.P.M.
1650 R.P.M.
1330 R.P.M.
1000 R.P.M.
680 R.P.M.
350 R.P.M.
Radial Load in Pounds.
200
400
600
800
1000
1200
1400
1600
1800


Horse Power to Drive Curve
for a #6309 Hess-Bright
Bearing.
Radial Loads.
2000#
1800#
1600#
1400#
1200#
1000#
800#
600#
400#
200#
H.P. to Drive
.13
.12
.11
.10
.09
.08
.07
.06
.05
.04
.03
.02
.01
Speed in R.P.M.
0
200
400
600
800
1000
1200
1400
1600
1800

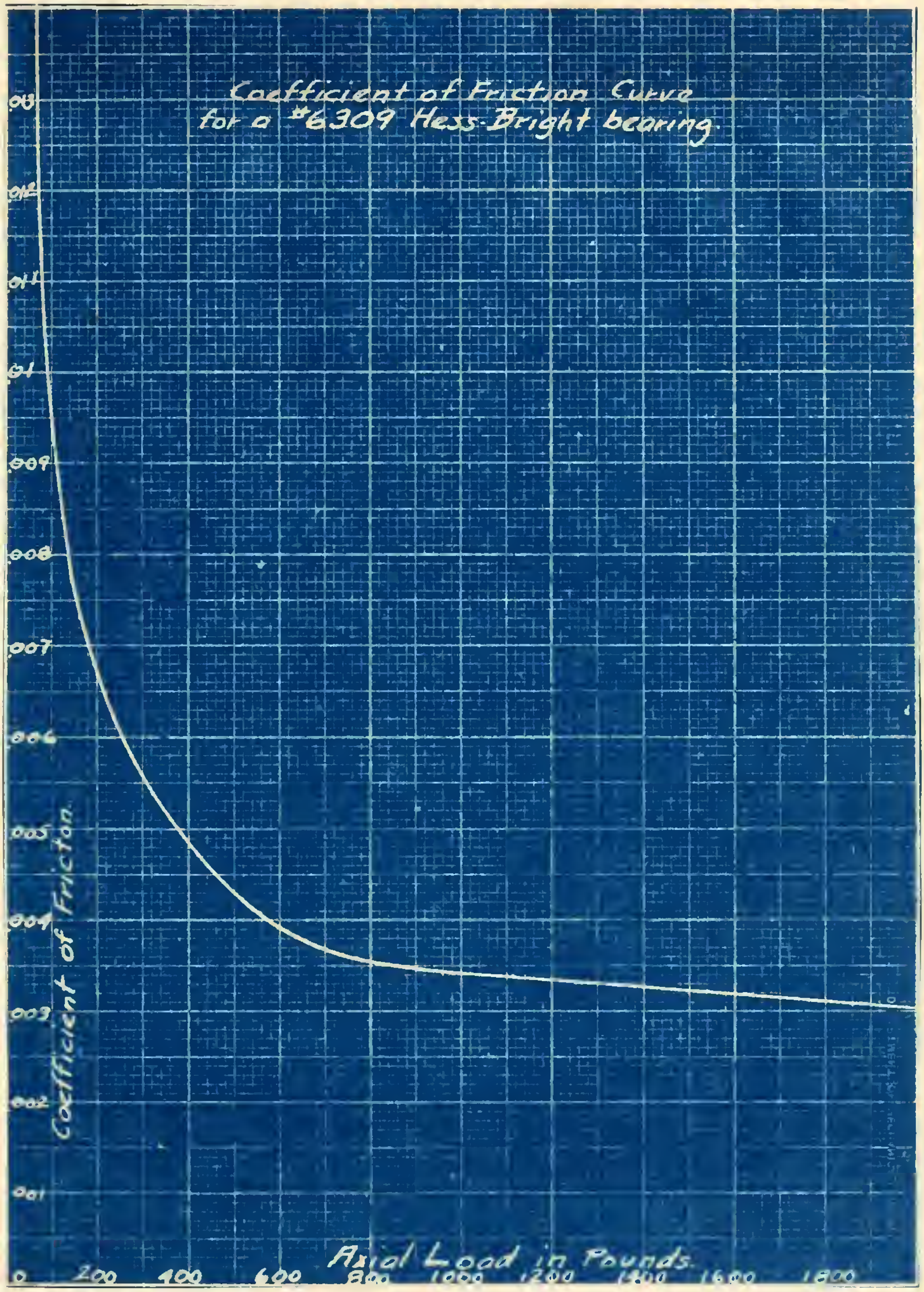

Coefficient of Friction Curve
for a #6309 Hess-Bright bearing.
Coefficient of Friction.
.013
.012
.011
.010
.009
.008
.007
.006
.005
.004
.003
.002
.001
Radial Load in Pounds.
0
200
400
600
800
1000
1200
1400
1600
1800



Horse Power to Drive Curve
for #6309 Hess-Bright
Bearing.
H.P. to Drive.
.065
.060
.055
.050
.045
.040
.035
.030
.025
.020
2080 R.P.M.
1650 R.P.M.
1330 R.P.M.
1010 R.P.M.
680 R.P.M.
350 R.P.M.
Axial Load in Pounds.
0
100
200
300
400
500
600
700
800



Horse Power to Drive Curve
for a #6309 Hess-Bright Bearing.
Axial Loads.
H.P. to Drive
Speed in R.P.M.
.065
.060
.055
.050
.045
.040
.035
.030
.025
.020
.015
.010
.005
200
400
600
800
1000
1200
1400
1600
1800
800#
600#
400#
200#


Coefficient of Friction Curve
for a #6309 Hess-Bright bearing.
Coefficient of Friction
Axial Load in Pounds.
.013
.012
.011
.01
.009
.008
.007
.006
.005
.004
.003
.002
.001
0
200
400
600
800
1000
1200
1400
1600
1800

RADIAL LOAD DATA.

350 R.P.M.				
L	W	T	C	H.P.
60	0.26	8.2	.0384	.0114
160	.26	8.2	.01735	.0114
260	.26	8.2	.00888	.0114
360	.26	8.2	.00840	.0114
460	.29	9.16	.00526	.0177
560	.30	9.47	.004760	.0131
660	.32	10.10	.00430	.0140
760	.32	10.10	.00374	.0140
860	.34	10.70	.0035	.0149
960	.38	11.96	.0035	.0166
1060	.40	12.6	.00334	.0175
1160	.40	12.6	.00305	.0175
1260	.48	15.1	.00337	.0210
1360	.54	17.0	.0035	.0733
1460	.58	18.3	.00352	.0254
1560	.62	19.5	.0035	.0272
1660	;64	20.4	.00346	.0280
1760	.68	21.4	.00342	.0298
1860	.72	22.7	.00342	.0318
1960	.76	23.9	.00342	.0333
2060	.82	25.8	.0035	.0359

680 R.P.M.				
L	W	T	C	H.P.
60	0.28	8.84	.0399	.0238
160	.28	8.84	.0155	.0238
260	.28	8.84	.00955	.0238
360	.30	9.47	.00746	.0255
460	.32	10.10	.00586	.0272
560	.34	10.7	.00534	.0289
660	.34	10.7	.00455	.0289
760	.36	11.35	.00420	.0306
860	.38	11.96	.00391	.0323
960	.40	12.6	.00366	.0340
1060	.42	13.25	.00352	.0357
1160	.44	13.9	.00368	.0374
1260	.46	14.5	.00352	.0392
1360	.50	15.8	.00352	.0426
1460	.52	16.4	.00315	.0442
1560	.56	17.65	.00318	.0476
1660	.58	18.3	.00309	.0493
1760	.60	18.9	.00302	.0511
1860	.66	20.8	.00311	.0561
1960	.68	21.4	.00303	.0578
2060	.68	21.4	.00291	.0578

1000 R.P.M.				
L	W	T	C	H.P.
60	0.28	8.84	.0399	.0351
160	.28	8.84	.0155	.0351
260	.30	9.47	.0102	.0376
360	.32	10.10	.00791	.0401
460	.32	10.10	.00586	.0401
560	.34	10.7	.00534	.0416
660	.34	10.7	.00455	.0416
760	.36	11.35	.00420	.0451
860	.38	11.96	.00391	.0476
960	.38	11.96	.0035	.0476
1060	.40	12.6	.00345	.0501
1160	.42	13.25	.00330	.0526
1260	.46	14.5	.00324	.0576
1360	.48	15.1	.00312	.0601
1460	.50	15.8	.00303	.0626
1560	.54	17.0	.00306	.0676
1660	.58	18.3	.0031	.0726
1760	.60	18.9	.00306	.0751
1860	.64	20.4	.00308	.0801
1960	.66	20.8	.00298	.0826
2060	.68	21.4	.00291	.0852

1330 R.P.M.				
L	W	T	C	H.P.
60	0.30	9.47	.0444	.0500
160	.30	9.47	.0161	.0500
260	.32	10.10	.012	.0534
360	.32	10.10	.00861	.0534
460	.34	10.7	.00617	.0567
560	.34	10.7	.00501	.0567
660	.36	11.35	.00483	.0600
760	.36	11.35	.00409	.0600
860	.36	11.35	.00371	.0600
960	.39	12.3	.00360	.0650
1060	.40	12.6	.00333	.0667
1160	.42	13.25	.0032	.0700
1260	.44	13.9	.00311	.0733
1360	.46	14.5	.00300	.0766
1460	.48	15.1	.0029	.0800
1560	.52	16.4	.00295	.0867
1660	.56	17.65	.00398	.0935
1760	.60	18.9	.00302	1.000
1860	.62	19.5	.00293	1.032
1960	.66	20.8	.00298	1.100
2060	.68	21.4	.00291	1.130

1650 R.P.M.				
L	W	T	C	H.P.
60	0.28	8.84	.0399	.0578
160	.28	8.84	.0155	.0578
260	.28	8.84	.0954	.0578
360	.30	9.47	.0074	.0620
460	.30	9.47	.00547	.0620
560	.32	10.10	.00505	.0661
660	.34	10.7	.00456	.0702
760	.36	11.35	.0042	.0742
860	.36	11.35	.0037	.0742
960	.38	11.96	.0035	.0785
1060	.40	12.6	.00344	.0826
1160	.42	13.25	.00319	.0868
1260	.44	13.9	.0031	.0909
1360	.48	15.1	.00311	.0991
1460	.50	15.8	.00303	.1.030
1560	.50	15.8	.00284	1.030
1660	.52	16.4	.00276	1.070
1760	.56	17.65	.00278	1.155
1860	.58	18.3	.00278	1.195
1960	.60	18.9	.0027	.1235
2060	.64	20.4	.00276	.1320

2080 R.P.M.				
L	W	T	C	H.P.
60	0.30	9.47	.0443	.0782
160	.30	9.47	.0161	.0782
260	.30	9.47	.01015	.0782
360	.32	10.10	.00185	.0835
460	.32	10.10	.00558	.0835
560	.32	10.10	.00503	.0835
660	.36	11.35	.00483	.0939
760	.36	11.35	.00419	.0939
860	.36	11.35	.00371	.0939
960	.38	11.96	.00345	.0990
1060	.40	12.6	.00322	.1040
1160	.42	13.25	.00295	.1092
1260	.42	13.25	.00288	.1092
1360	.44	13.9	.00268	.1145
1460	.44	13.9	.00562	.1145
1560	.46	14.5	.00664	.1198
1660	.50	15.8	.00262	.1300
1760	.52	16.4	.00262	.1352
1860	.54	17.0	.00256	.1405
1960	.58	18.3	.00261	.1510
2060	.62	19.5	.0266	.1610

AXIAL LOAD DATA

350 R. P. M.						
L	Scale Reading	mScale e_0	W	t	C	H P
0	.16	.14	.06	.315		.00115
100	.22	.14	.08	1.26	.0142	.00702
200	.22	.14	.08	1.26	.0711	.00702
300	.26	.14	.12	1.89	.00711	.0105
400	.28	.14	.14	2.205	.00622	.0123
500	.28	.14	.14	2.205	.00498	.0123
600	.28	.14	.14	2.205	.00414	.0123
700	.30	.14	.16	2.52	.00406	.0140
800	.32	.14	.18	2.835	.00400	.0158

680 R. P. M.						
L	Scale Reading	Scale 0	W	t	C	H P
0	20	.18	.02	.315		.0034
100	22	.18	.04	.630	.00711	.0068
200	24	.18	.06	.945	.00533	.0104
300	26	.18	..08	1.260	.00474	.0136
400	28	.18	.10	1.575	.00445	.01700
500	30	.18	.12	1.890	.00426	.0204
600	32	.18	.14	2.205	.00413	.0233
700	34	.18	.16	2.520	.00406	.0276
800	36	.18	.18	2.835	.00483	.0306

1010 R. P. M.						
L	Scale Reading	Scale 0	W	T	C	HP
0	20	10	.02	.315		,00506
100	24	10	.06	.945	.01065	.0152
200	26	10	.08	1.260	.00711	.0202
300	28	10	.10	1.575	.00597	.0253
400	30	10	.12	1.890	.00533	.0304
500	30	10	.12	1.890	.00427	.0304
600	32	10	.14	2.205	.00414	.0354
700	32	10	.14	2.205	.00457	.0354
800	34	10	.16	2.520	.00356	.0404

1350 R. P. M.						
L	Scale Reading	Scale 0	W	T	C	HP
0	.22	.20	.06	.315		.00666
100	.26	.20	.06	.945	.01065	.0247
200	.28	.20	.08	1.26	.00711	.03300
300	.30	.20	.10	1.575	.00597	.0412
400	.30	.20	.10	1.575	.00501	.0412
500	.32	.20	.12	1.89	.00427	.0495
600	.32	.20	.12	1.89	.00356	.0495
700	.34	.20	.14	2.205	.00356	.0577
800	.36	.20	.16	2.52	.00356	.066

1650 R. P. M.						
L	Scale Reading	Scale 0	W	t	C	H P
0	.22	.20	.02	.315		.00825
1000	.26	.20	.06	.945	.01065	.0247
200	.28	.20	.08	1.26	.00711	.03300
300	.30	.20	.10	1.575	.00597	.0412
400	.30	.20	.10	1.575	.00501	.0412
500	.32	.20	.12	1.89	.00427	.0495
600	.32	.20	.12	1.89	.00356	.0495
700	.34	.20	.14	2.205	.00356	.0577
800	.36	.20	.16	2.52	.00356	.066

2000 = 2080 R. P. M.						
L	Scale Reading	Scale 0	W	t	C	H P
0	24	22	.02	.315		.0104
100	26	22	.04	.630	.00711	.0208
200	28	22	.06	.945	.00533	.0316
300	30	22	.08	1.260	.00414	.0416
400	30	22	.08	1.260	.00356	.0416
500	32	22	.10	1.575	.00356	.0520
600	32	22	.10	1.575	.00098	.0520
700	34	22	.12	1.890	.00305	.0624
800	36	22	.14	2.205	.00311	.728