DEPARTMENT OF COMMERCE

TECHNOLOGIC PAPERS
OF THE
BUREAU OF STANDARDS

S. W. STRATTON, DIRECTOR

No. 201

FRICTION AND CARRYING CAPACITY OF BALL AND ROLLER BEARINGS

BY

H. L. WHITTEMORE, Mechanical Engineer
S. N. PETRENKO, Assistant Mechanical Engineer
Bureau of Standards

OCTOBER 6, 1921

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ABSTRACT

The experiments were undertaken by the Bureau of Standards to determine the maximum safe load and the static friction under load of ball and flexible roller bearings.

Tests were made on balls of 1.00, 1.25 and 1.50 inches diameter in grooved races and on rollers 1.25 inches in diameter and 5.25 inches long in flat and cylindrical races.

The total deformation and area of contact of bearings and races were measured and compared with Hertz's theory.

Conclusions.—1. The results agree roughly with Hertz's theory. The differences are ascribable to inhomogeneity of the material.

2. The ratio of friction to load is practically constant and equal to 0.00055 for all three sizes of balls up to a "critical" load, which varies with the diameter of ball: 1300 pounds for 1.00-inch, 1700 pounds for 1.25-inch, and 2200 pounds for 1.5-inch balls.

3. A similar "critical" load, 25 000 pounds, was found for the roller bearings with a ratio of friction to load equal to 0.00075.

4. This "critical" load at which the friction began to increase more rapidly was in all cases lower than the safe load as determined by permanent deformation and as calculated from Stribeck's law.

CONTENTS

I. INTRODUCTION

In order to facilitate the training of large guns, it is very desirable to reduce the friction at the trunnion bearings. These bearings are moved infrequently and at very low speeds. They may be, however, subjected to great loads when the gun is fired. These conditions are very different from those usual for bearings in engineering work. For the latter the speed is much greater and the periods of operation much longer. They, however, are not often subject to great loads or to impact.

The use of ball and roller bearings for line shafts, vehicle wheels, etc., has become quite extensive, due to their high efficiency. The results obtained from service tests of this kind give very little data for the design of ordnance bearings.

These tests were undertaken by the Bureau of Standards, at the request of the Navy Department, to obtain experimental data on the frictional resistance of both ball and roller bearings at very low speeds and also the loads which they will safely sustain.

The tests may be listed as follows: 1, Static friction test on ball bearing; 2, static friction test on roller bearing; 3, compression test on ball bearing; and 4, compression test on roller bearing.

II. APPARATUS

The special apparatus required for these tests was designed and built by the Navy Department in consultation with the Bureau of Standards. The balls and rollers were obtained from commercial manufacturers and were such as were considered suitable for this use.

1. BALLS

The hardened steel balls were 1.00, 1.25 and 1.50 inches diameter. Four of each size were provided.

2. BALL RACES

The cost of making complete bearings was prohibitive. If, however, complete bearings had been tested, the results could not be used for a bearing having a different diameter, due to the impossibility of measuring the load on the individual balls. Sections of a complete race, only, were represented by small rectangular steel blocks. These are shown in Figs. 1 and 4. Each block had a cylindrical groove on one face, parallel to the opposite face, having a radius slightly greater than that of the ball with which it was to be used. These races were hardened and the groove ground to the required radius. In an actual ball bearing, the axis of the groove would be an arc of a circle about the axis of rotation.

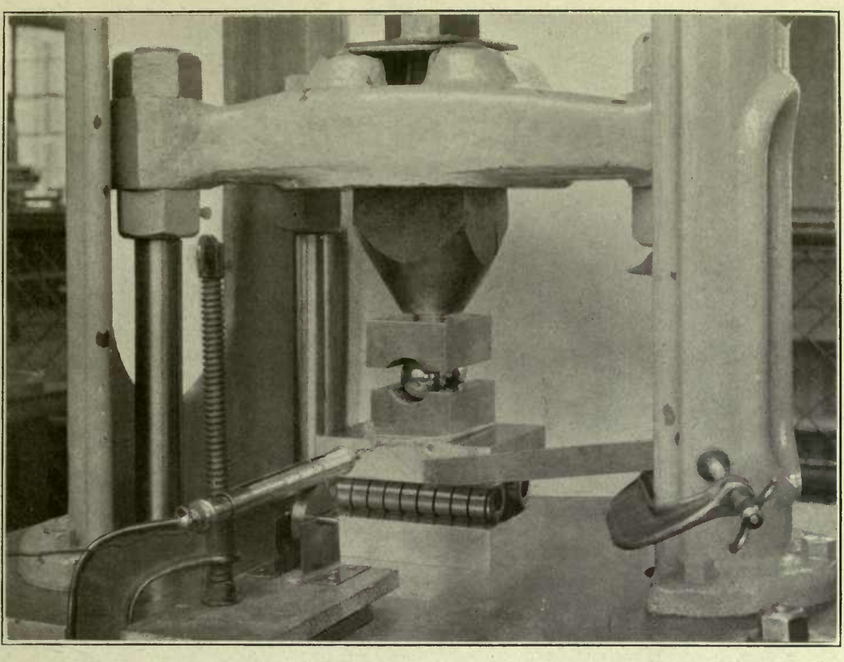

FIG. 1.—*Measuring the static friction of a ball bearing*

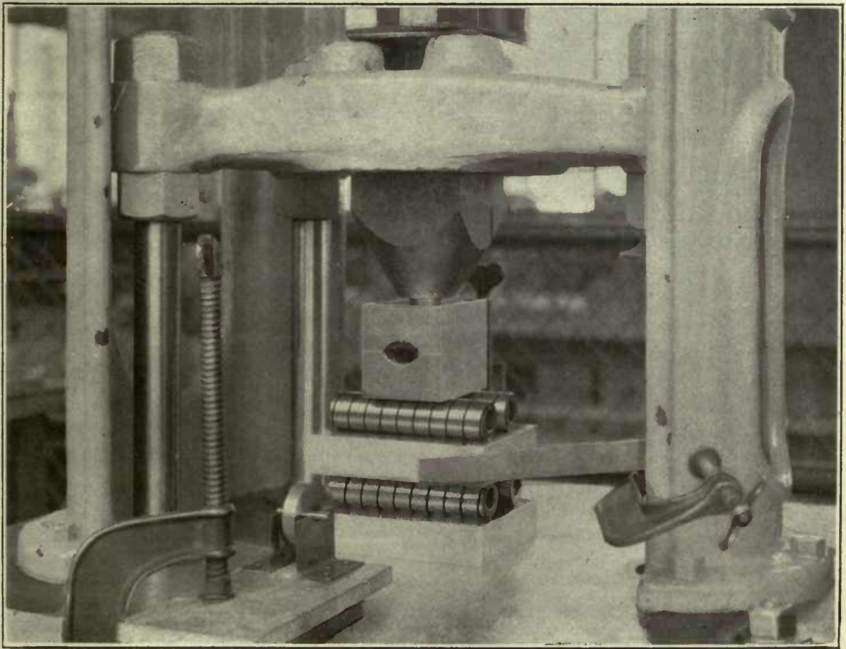

FIG. 2.—*Measuring the static friction of a roller bearing*

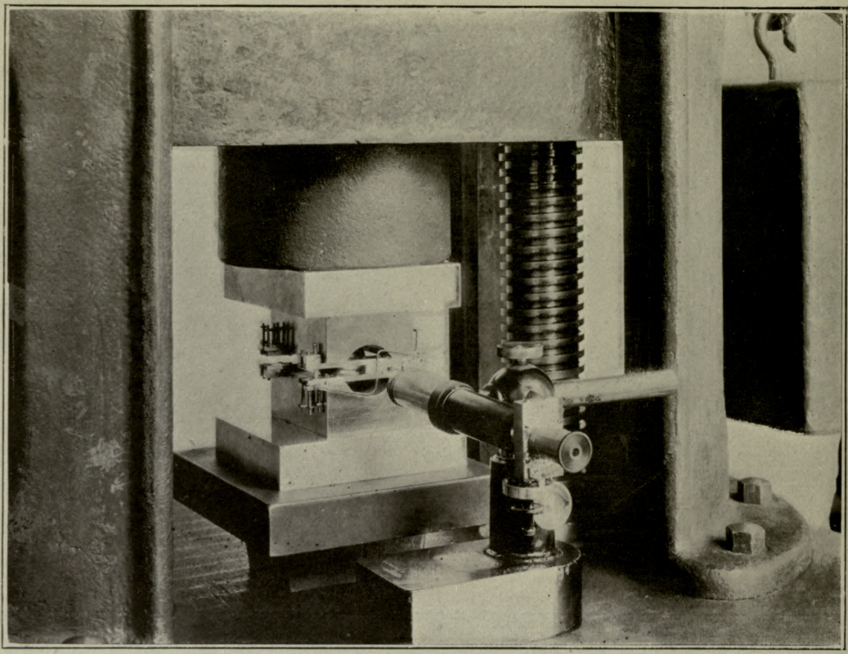

Fig. 3.—*Measuring the deformation of a ball and race under load*

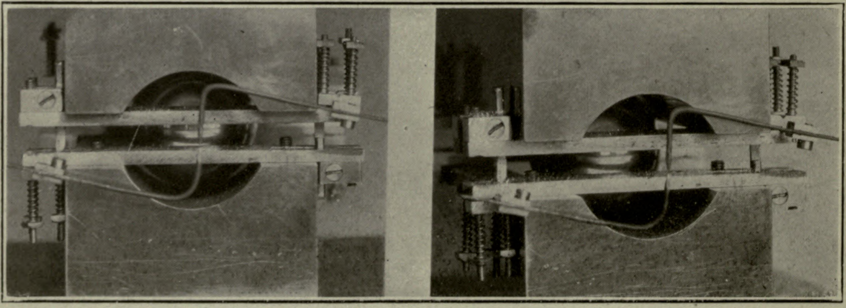

Fig. 4.—*Apparatus for measuring deformation of a ball and race*

The experimental work was much easier because races having straight grooves were used and it is believed that the results apply with reasonable accuracy to bearings having a large diameter such as are used for ordnance work.

Grooved races are used in practice as with them the area of contact between the ball and the race is greater than is obtained with plane races, and therefore the allowable load on the bearing is increased. The load is without doubt a maximum for races grooved to the same diameter as the ball; the friction, however, would be excessive in a bearing of this kind. Two pairs of races were therefore made for each size of ball. One had, perhaps, the smallest practicable radius and the other was somewhat greater. The ratios of groove radii to ball radii are given in Table 1. These races were used both for the friction and the load tests.

TABLE 1.—Ratio of Groove Radii to Ball Radii

Ball diameter, inches	Small groove	Large groove
1.00	1.03	1.10
1.25	1.04	1.12
1.50	1.04	1.12

3. ROLLERS

The rollers were of the flexible roller type. They were closed helices made from steel bars of about 0.52 by 0.30 inch in cross section. The length was about 5.25 inches and the internal diameter about 0.65 inch. They were hardened and the external cylindrical surface ground to about 1.25 inches diameter. These rollers are shown in Fig. 2. Six were provided for these experiments.

4. ROLLER RACES

Two flat plates were used in the roller tests to represent bearings having a large diameter. These are shown in Fig. 2. In order to obtain data also upon bearings such as might be used—for example, for gun trunnions—two segmental bearings having inner diameters of 7 and 20 inches were made. The outer diameter was, of course, greater than the inner diameter by twice the diameter of the rollers. The smaller bearing is shown in Fig. 5. The larger bearing is shown in Figs. 6 and 7. Each of these bearings consisted of the inner race, two portions of the outer race, with apparatus for holding these parts in their proper relative position in a hydraulic testing machine having a capacity of 230 000 pounds. The smaller bearing is shown in the machine in Fig. 8. Side plates furnished bearings for a shaft through the inner race (see Fig. 7) constraining it to rotate about the axis of the bearing. Two rollers, diametrically opposite each other, were used in each of these bearings. As it was found that the rollers tended to become displaced, so that their axes were not parallel to the axis of the bearing, retainers or "cages" were made which rotated about the same shaft as the inner race. One of these cages is shown in Fig. 7. A lever attached to the shaft through the inner race allowed the torque required to rotate the inner race to be measured as shown in Fig. 8.

The bearing surfaces of all flat plates and bearings were hardened and ground.

5. HARDNESS AND DIMENSIONS

The hardness of all bearing parts was measured by the scleroscope, using the universal diamond pointed hammer. The dimensions of the bearing surfaces were also measured. These data are given in Table 2. In the case of the ball races it was found that the ends of grooves were harder than the middle portion of the groove. As the latter portion was used in the experimental work its hardness is given for the average value.

TABLE 2.—Dimensions and Hardness of Balls, Rollers, and Races

Specimen No.	Diameter or radius of curvature	Scleroscope hardness: Extreme variations of readings	Scleroscope hardness: Average
	Inches		
Diameter of balls	1.0003	62-69	66
Do	1.2503	57-63	60
Do	1.5000	64-70	68
Radius of ball races:			At middle of groove
31	.515	62-92	62
32	.515	65-95	65
33	.550	65-86	65
34	.550	60-93	60
35	.650	69-89	69
36	.650	63-90	63
37	.700	70-92	70
38	.700	61-91	61
27	.779	64-67	64
28	.778	60-91	60
29	.839	71-93	71
30	.843	71-91	71
Diameter of rollers:	Inches		
1	1.249	71-73	72
2	1.249	67-73	70
3	1.249	69-71	70
4	1.249	67-72	70
5	1.250	68-70	69
6	1.250	68-73	70
Radius of inner roller races:			
43	3.499	80-92	86
44	10.000	97-102	99
Radius of outer roller races:			
39	11.252	70-93	81
40	11.252	95-97	96
41	4.750	62-70	66
42	4.750	71-74	73
Plates:			
25	Flat within	93-101	97
26	.0002	94-100	97

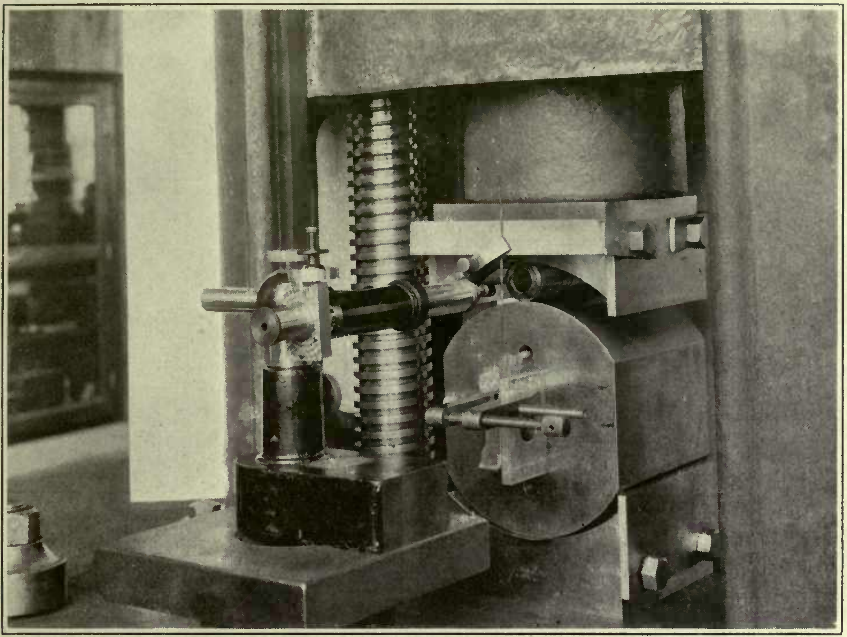

FIG. 5.—*Apparatus for measuring the deformation of a roller in a race having an inner diameter of 7 inches*

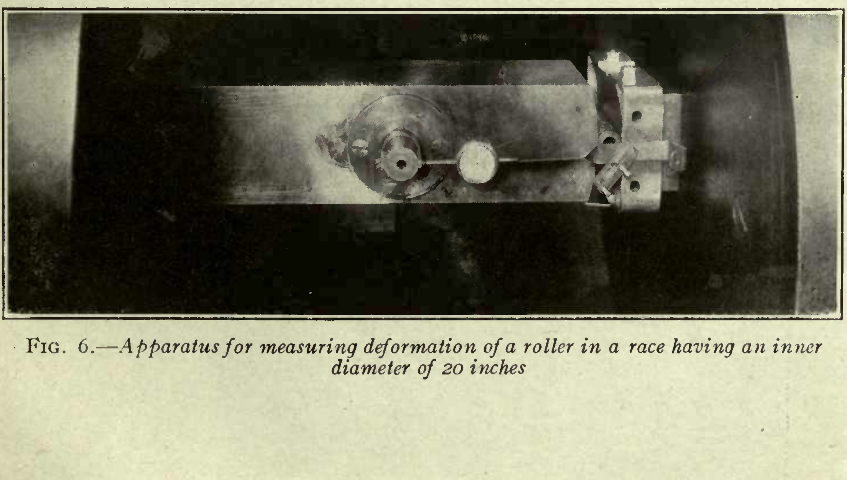

FIG. 6.—*Apparatus for measuring deformation of a roller in a race having an inner diameter of 20 inches*

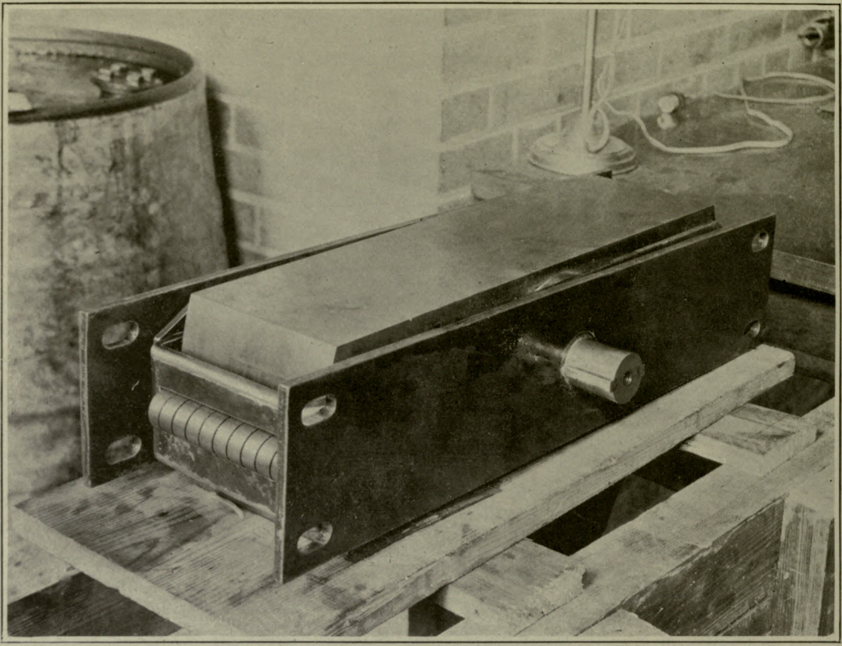

FIG. 7.—*Retainer for roller with inner race*

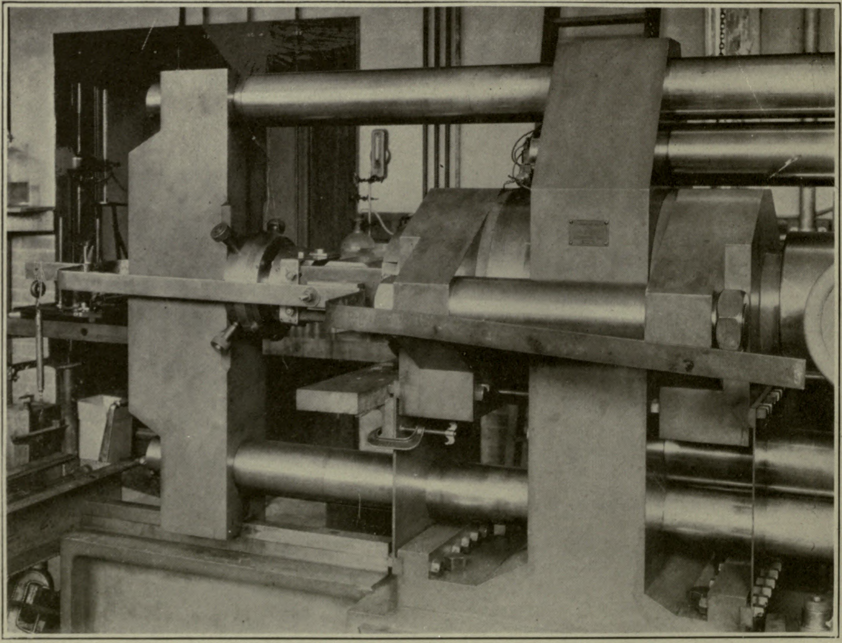

FIG. 8.—*Apparatus for making static friction test of roller and races having an inner diameter of 7 inches*

TABLE 3.—Static Friction of Ball (1 Inch Diameter)

Load on ball, pounds	Radius of races 0.515 inch				Radius of races 0.550 inch			
	Friction, pounds	Ratio friction to load		Coefficient of rolling friction	Friction, pounds	Ratio friction to load		Coefficient of rolling friction
		Observed value	Graph value			Observed value	Graph value	
250	0.11	0.00044	0.00044	0.00022	0.12	0.00048	0.00048	0.00024
500	.23	.00046	.00046	.00023	.31	.00062	.00052	.00026
750	.37	.00049	.00049	.00025	.42	.00056	.00056	.00028
1000	.51	.00051	.00054	.00027	.62	.00062	.00062	.00031
1250	.79	.00063	.00063	.00032	.84	.00067	.00068	.00034
1500	1.19	.00079	.00075	.00038	1.15	.00077	.00078	.00039
1750	1.53	.00087	.00089	.00045	1.63	.00093	.00092	.00046
2000	2.13	.00107	.00105	.00053	2.13	.00107	.00106	.00053
2250	2.81	.00125	.00123	.00062	2.71	.00120	.00120	.00060
2500	3.49	.00140	.00140	.00070	3.40	.00136	.00135	.00068

TABLE 4.—Static Friction of Ball (1.25 Inches Diameter)

Load on ball, pounds	Radius of races 0.650 inch				Radius of races 0.700 inch			
	Friction, pounds	Ratio friction to load		Coefficient of rolling friction	Friction, pounds	Ratio friction to load		Coefficient of rolling friction
		Observed value	Graph value			Observed value	Graph value	
250	0.12	0.00048	0.00044	0.00028	0.14	0.00056	0.00052	0.00032
500	.19	.00038	.00046	.00029	.26	.00052	.00053	.00033
750	.35	.00047	.00049	.00031	.43	.00057	.00054	.00034
1000	.50	.00050	.00051	.00032	.52	.00052	.00055	.00034
1250	.69	.00055	.00053	.00033	.66	.00053	.00057	.00036
1500	.87	.00058	.00057	.00036	.81	.00054	.00061	.00038
1750	1.00	.00057	.00064	.00040	1.16	.00066	.00071	.00044
2000	1.44	.00072	.00076	.00048	1.69	.00084	.00086	.00054
2250	2.10	.00093	.00090	.00056	2.49	.00111	.00106	.00066
2500	2.61	.00104	.00106	.00061	3.21	.00128	.00128	.00080

TABLE 5.—Static Friction of Ball (1.50 Inches Diameter)

Load on ball, pounds	Radius of races 0.779 inch				Radius of races 0.841 inch			
	Friction, pounds	Ratio friction to load		Coefficient of rolling friction	Friction, pounds	Ratio friction to load		Coefficient of rolling friction
		Observed value	Graph value			Observed value	Graph value	
250	0.15	0.00060	0.00055	0.00041	0.13	0.00052	0.00054	0.00041
500	.29	.00058	.00056	.00042	.27	.00054	.00054	.00041
750	.42	.00056	.00056	.00042	.43	.00057	.00055	.00041
1000	.54	.00054	.00056	.00042	.52	.00052	.00055	.00041
1250	.69	.00055	.00057	.00043	.70	.00056	.00056	.00042
1500	.88	.00059	.00057	.00043	.90	.00060	.00056	.00042
1750	1.02	.00058	.00059	.00044	1.02	.00058	.00058	.00044
2000	1.19	.00060	.00064	.00048	1.17	.00059	.00062	.00047
2250	1.65	.00073	.00072	.00054	1.60	.00071	.00069	.00053
2500	2.19	.00088	.00086	.00065	1.95	.00078	.00080	.00060

III. TESTS

1. STATIC FRICTION TEST ON BALL BEARING

(*a*) *Method of Test.*—The arrangements of the apparatus for these tests is shown in Fig. 1. Two balls were used with each pair of races in order to secure stability in the loaded condition. The lower ball race rests upon a plate mounted on two rollers. The upper ball race is loaded by a universal three-screw testing machine having a capacity of 50 000 pounds. A spherical bearing was used between the movable head of the testing machine and the upper ball race. After the desired load had been applied the lower ball race was drawn forward by a force exerted through the spring balance shown which rested on an antifriction roller. The smallest division on the spring balance represented 1 ounce.

The friction of the rollers was found by the method shown in Fig. 2 for each of the loads used for the balls. One-half of the friction for the four rollers was subtracted from the spring balance reading for the ball tests which gave the frictional resistance of the two balls.

In every case the bearings were started from rest. No attempt was made to measure the friction of the bearing after motion occurred, due to the fluctuations in the force and the short distance the bearing could be moved. The starting or static friction is always greater than the moving friction, so that the values given here are in any case the maximum. Care was taken to secure the following conditions during these tests:

1. All bearing surfaces were parallel to each other and also perpendicular to the action line of the load.
2. The balls and rollers were placed symmetrically with relation to the action line of the load.
3. The axes of the rollers were perpendicular to the axis of the ball groove.
4. The action line of the moving force was parallel to the axis of the ball groove.
5. The load was applied equally to the balls and rollers by a spherical bearing block.

It was found that the magnitude of the starting force varied considerably. The load exerted by the testing machine also fluctuated at the instant of starting but rarely more than 50 pounds. These fluctuations may have been due to the following causes:

1. Slight variations in the diameter of the balls and the rollers and variations in the surfaces of the races from the true cylinder or plane.

2. Nonuniform hardness of the bearing surfaces of the races. (The balls and rollers were much more uniform in hardness than the races.)

The conditions under which these tests were made represent ideal rolling friction along a straight line. They are never obtained in practice, so that values in practice may be much larger,

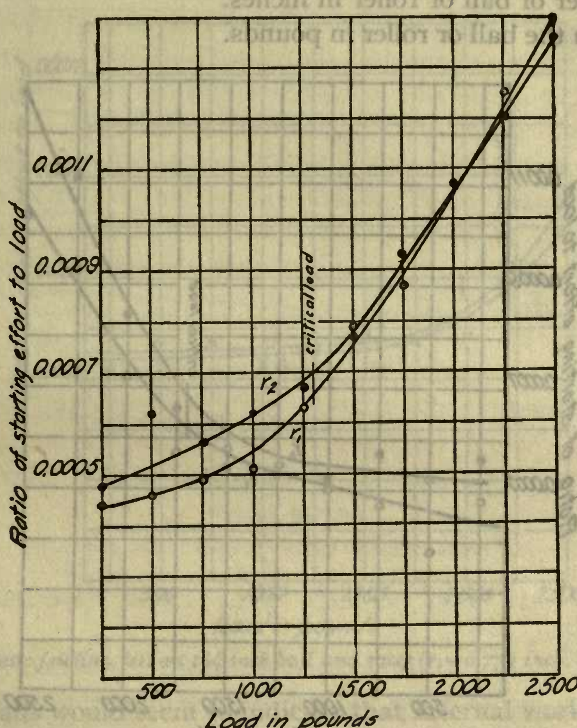

FIG. 9.—*Static friction test on 1-inch ball and races* (r_1=0.515 *inch*, r_2=0.550 *inch*)

due to the sliding friction which occurs. Even in these experiments there was some sliding friction, due to the fact that the area of contact between ball and race, although small, was appreciable. It was also impossible to secure exact arrangement of the parts of the apparatus.

(*b*) *Results*.—The results are given in Tables 3, 4, and 5 and in Figs. 9, 10, and 11. The values given in the tables for the friction are the averages of several trials for slightly different positions of the balls, rollers, and races. The graph values are obtained from the smooth curve drawn to represent the most probable values.

The coefficients of rolling friction were computed from the graph values by the following formula:[1]

$$\text{Coefficient of rolling friction} = \frac{Pd}{2Q}$$

in which:

$\frac{P}{2}$ = starting friction on one side for one ball or roller, in pounds.
d = diameter of ball or roller in inches.
Q = load on the ball or roller in pounds.

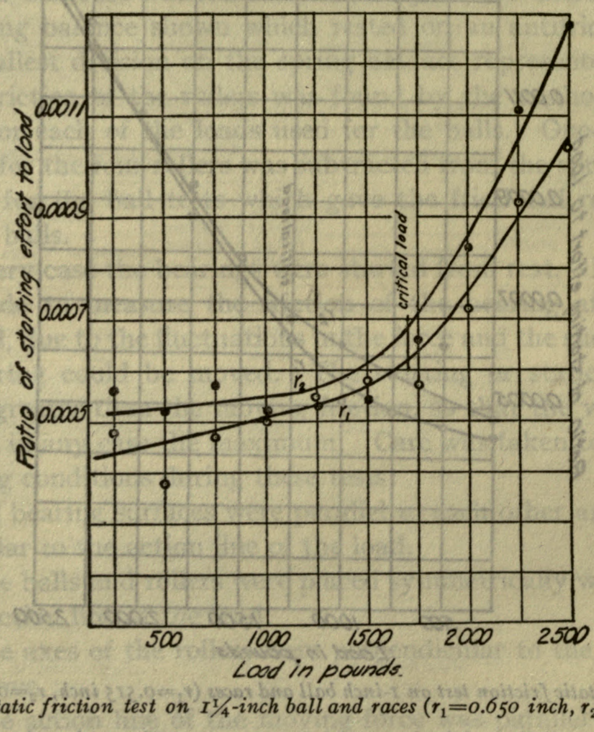

FIG. 10.—*Static friction test on 1¼-inch ball and races (r_1=0.650 inch, r_2=0.700 inch)*

For some of these tests the balls, rollers, and races were well coated with a good mineral lubricating oil. The observed values of the friction, when this was done, appeared to be the same as those obtained when no oil was used.

(*c*) *Conclusions.*—1. The starting friction is nearly the same for both sizes of groove. The groove having the larger radius gave the lowest value for the friction.

[1] R. Thurston, A Treatise on Friction and Lost Work, p. 82, 1885.

2. The ratio of starting friction to the load increases slowly as the load increases, then much more rapidly. The critical loads are approximately as follows:

Ball diameter in inches	Critical load in pounds
1.00	1300
1.25	1700
1.50	2200

If the frictional resistance is to be kept low, these critical loads should not be exceeded. The very rapid rise in the friction at

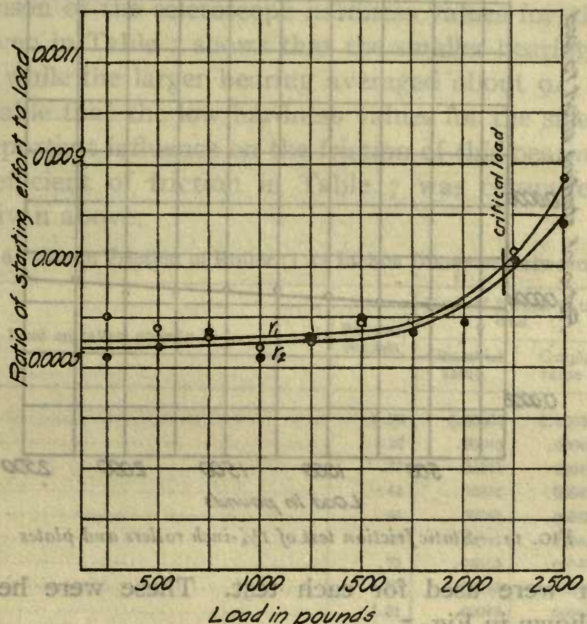

FIG. 11.—*Static friction test on $1\frac{1}{2}$-inch ball and races* ($r_1=0.779$ *inch*, $r_2=0.841$ *inch*)

greater loads would seem to indicate that internal work was being performed on the material of either the balls or races which might cause heating and their destruction if the bearings were operated continuously under loads greater than the critical loads.

3. The ratio of frictional resistance to load is practically the same for balls of all diameters up to the critical load and may be taken as 0.00055. For this reason the coefficient of rolling friction as found from the above equation was of little use in these tests.

4. Oil is of little, if any, use upon ball bearings in reducing the static frictional resistance.

2. STATIC FRICTION TEST ON ROLLER BEARING

(*a*) *Method of Test.*—The static friction of the rollers loaded between two *steel plates* was measured as for balls. The arrangement of apparatus is shown in Fig. 2.

The tests of static friction for the two segmental bearings were made in a hydraulic testing machine having a capacity of 230 000 pounds.

The arrangement of the apparatus for the smaller of these bearings is shown in Fig. 8. Two rollers diametrically opposite

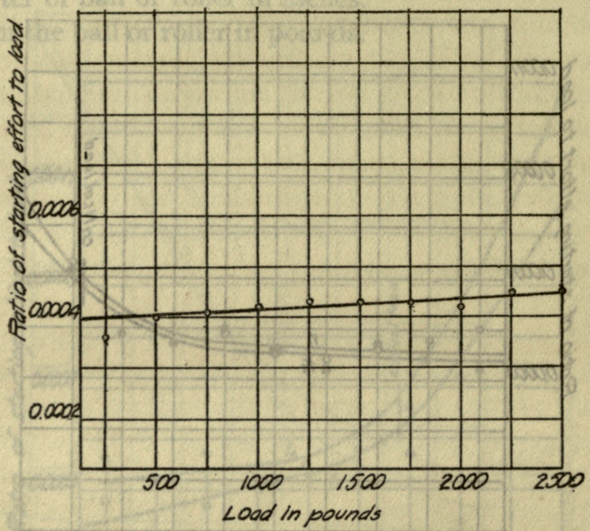

FIG. 12.—*Static friction test of 1¼-inch rollers and plates*

each other were used for each test. These were held in the retainers shown in Fig. 7.

The lever shown in Fig. 8, used for rotating the bearings under load, was 41 inches from the center of rotation to the point of application of the force. This lever was counterbalanced by one of equal length extending in the opposite direction. The force was applied through a spring balance, the smallest graduation of which represented 0.5 pound. Care was taken that the action line of the force was perpendicular to the lever arm. The observed force was used to compute the equivalent frictional force required to cause rotation if applied at the surface of the inner race.

(*b*) *Results.*—The results for these tests are given in Tables 6 and 7 and in Figs. 12 and 30. The values given for the friction are the averages of at least five determinations for each load, as it was found that the friction fluctuated considerably, depending on the position of the rollers with respect to the plane through the axis of the bearing. This was particularly true with the smaller bearing for which it was very difficult to secure satisfactory readings. This was due probably to the condition of unstable equilibrium of the whole system which existed during these tests and which was beyond the control of the experimenter.

This is the only explanation of the unexpected character of the curve for the smaller bearing in Fig. 30. Several other conditions such as inaccuracies in or nonuniform hardness of the bearing surfaces also affected the friction.

Comparison of the scleroscope hardness values for these bearings as given in Table 2 shows that the smaller bearing averaged about 78, while the larger bearing averaged about 94. It seems very probable that the low hardness values for the small bearing had an important influence on the friction of this bearing.

The coefficient of friction in Table 7 was computed by the formula given above.

TABLE 6.—Static Friction of Roller (1.25 Inches Diameter Between Plates)

Load on roller, pounds	Friction, pounds	Ratio of friction to the load		Coefficient of rolling friction
		Observed value	Graph value	
250	0.09	0.00036	0.000400	0.000250
500	.20	.00040	.000405	.000253
750	.31	.00041	.000410	.000256
1000	.42	.00042	.000415	.000259
1250	.54	.00043	.000420	.000262
1500	.65	.00043	.000425	.000265
1750	.75	.00043	.000430	.000268
2000	.85	.00042	.000435	.000271
2250	1.01	.00045	.000440	.000274
2500	1.14	.00046	.000445	.000277

TABLE 7.—Static Friction of Roller (1.25 Inches Diameter)

Load on roller, pounds	Radius of inner race 3.5 inches			Radius of inner race 10.0 inches		
	Friction, pounds	Coefficient of rolling friction		Friction, pounds	Coefficient of rolling friction	
		Observed value	Graph value		Observed value	Graph value
5 000	8.8	0.00110	0.00110	6.1	0.00076	0.00075
10 000	29.3	.00183	.00190	12.3	.00077	.00077
15 000	58.5	.00244	.00245	19.5	.00081	.00080
20 000	87.8	.00275	.00275	26.6	.00083	.00083
25 000	120.0	.00300	.00305	34.8	.00087	.00091
30 000	170.0	.00354	.00330	53.3	.00110	.00105
40 000	234.4	.00366	.00378	92.3	.00144	.00145
50 000	316.5	.00396	.00420	153.7	.00192	.00192

(*c*) *Conclusions.*—Consideration of the values for the coefficient of rolling friction for the bearing having a radius of 10 inches shows that the static friction is nearly constant up to a load of 25 000 pounds. For greater loads the friction increases rapidly. This is similar to the behavior of the balls, and it is believed that this critical load should be considered the allowable load on the roller.

Due to the unexpected character of the curve the critical load for the bearing having an inner diameter of 7 inches could not be determined.

The critical loads as obtained from the load friction diagram (Fig. 30) are approximately as follows:

Radius of inner races in inches	Critical load in pounds
10.0	25 000
3.5	...

3. COMPRESSION TEST ON BALL BEARING

(*a*) *Method of Test.*—The allowable load on a bearing may be determined by noting the greatest load which it will sustain without permanent deformation. (See Tables 8, 9, and 10.) The apparatus for this test was that used for the friction tests but arranged as shown in Fig. 3. A single ball was placed between the races and the load applied by the testing machine previously used.

(*b*) *Compression and Set.*—As it was impossible to measure the deformation of the ball under load, special apparatus was designed to measure the relative motion of the two races; that is, the deformation of balls and races combined. This apparatus is shown in Figs. 3 and 4. At each corner of the races is a steel rod secured to one race. Opposite it is a short steel lever carried by a horizontal shaft which is held in any position in which it may be placed by caps for the bearing loaded by long helical springs. Experience with this apparatus showed that the best results were obtained when the shaft rested in a triangular groove in the supports. The caps for the bearings were also grooved but were later turned over to present a plane surface to the shaft which was, therefore, held in a three-line bearing.

The end of the shaft which projects from the bearing carries a curved pointer, the end of which opposes the end of the pointer on the other side of the races. In Fig. 4, the rod secured to the upper race is seen at the left and the one secured to the lower race at the right. The levers are not visible but the pointers are clearly shown.

In use, the pointers are turned away from each other, the desired load is applied to the bearing, then the pointers are turned toward each other by hand so that each lever comes in contact with the corresponding rod. The distance between the two pointers is then measured by the micrometer microscope shown in Fig. 3. The arrangement of this apparatus is such as to give correct values, even if the races are slightly tilted during the test. The total deformation of ball and race combined under load may be obtained as well as the permanent deformation after removing the load. The pointers multiplied the movement of the levers 10 times. The arrangement of the pointers, in pairs, made the change in distance between pointers 20 times the change in the distance between the races.

TABLE 8.—Compression Test of Ball (1 Inch Diameter)

Load in pounds	Radius of races	Total deformation of ball and races			Permanent set of ball and races		Contact area			
		Observed value	Graph value	Hertz value	Observed value	Graph value	2a	2b	2b (Hertz value)	Area
	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch^3
500	0.515	0.00079	0.00088	0.00112	0.00003	0.00002	....	....	....	....
	.550	.00097	.00099	.00112	.00003	.00003	....	....	....	....
	.779	....	....	....	....	....	....	....	....	....
1000	.515	.00140	.00156	.00179	.00006	.00006	0.292	0.038	0.040	0.0087
	.550	.00170	.00172	.00177	.00007	.00007	.188	.043	.040	.0058
	.779	....	....	....	....	....	.098	.051	....	.0039
	∞	....	....	....	....	....	.056	.056	.056	.0025
1500	.515	.00212	.00213	.00234	.00013	.00012	....	....	....	....
	.550	.00231	.00233	.00232	.00014	.00014	....	....	....	....
	.779	....	....	....	....	....	....	....	....	....
2000	.515	.00265	.00264	.00283	.00021	00021	.360	.050	.050	.0141
	.550	.00283	.00287	.00282	.00023	.00024	.245	.058	.052	.0112
	.779	....	....	....	....	....	.122	.070	....	.0067
	∞	....	....	....	....	....	.073	.073	.072	.0042
2500	.515	.00314	.00310	.00329	.00033	.00032	....	....	....	....
	.550	.00336	.00336	.00327	.00038	.00037	....	....	....	....
	.779	....	....	....	....	....	....	....	....	....
3000	.515	.00355	.00352	.00371	.00048	.00047	.397	.057	.058	.0178
	.550	.00382	.00382	.00369	.00054	.00053	.273	.067	.058	.0143
	.779	....	....	....	....	....	.134	.080	....	.0084
	∞	....	....	....	....	....	.083	.083	.082	.0054
3500	.515	.00397	.00393	.00412	.00067	.00067	....	....	....	....
	.550	.00425	.00425	.00409	.00075	.00077	....	....	....	....
	.779	....	....	....	....	....	....	....	....	....
4000	.515	....	....	....	....	....	.423	.062	.064	.0206
	.550	....	....	....	....	....	.288	.073	.064	.0165
	.779	....	....	....	....	....	.141	.084	....	.0093
	∞	....	....	....	....	....	.089	.089	.090	.0062

Two microscopes, one at each end of the race, were used by which a difference in the distance between the pointers of 0.00004 inch could be observed by estimation. The displacement of either end of the ball race could therefore be measured within 0.000004 inch.

TABLE 9.—Compression Test of Ball (1.25 Inches Diameter)

Load in pounds	Radius of races	Total deformation of ball and races			Permanent set of ball and races		Contact area			
		Observed value	Graph value	Hertz value	Observed value	Graph value	2a	2b	2b (Hertz value)	Area
	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch^2
500	0.650	0.00081	0.00085	0.00104	0.00002	0.00002	....	....	....	....
	.700	.00063	.00087	.00104	.00002	.00003	....	....	....	....
	.779	....	....	....	....	....	....	....	....	....
1000	.650	.00154	.00150	.00166	.00005	.00005	0.283	0.044	0.044	0.0098
	.700	.00142	.00157	.00165	.00005	.00007	.183	.048	.044	.0069
	.779	....	....	....	....	....	.144	.053	....	.0060
	∞	....	....	....	....	....	.058	.058	.062	.0027
1500	.650	.00207	.00207	.00217	.00009	.00008	....	....	....	....
	.700	.00207	.00217	.00217	.00009	.00011	....	....	....	....
	.779	....	....	....	....	....	....	....	....	....
2000	.650	.00254	.00253	.00263	.00014	.00014	.367	.056	.054	.0161
	.700	.00265	.00268	.00261	.00015	.00017	.237	.063	.056	.0117
	.779	....	....	....	....	....	.182	.070	....	.0100
	∞	....	....	....	....	....	.075	.075	.076	.0044
2500	.650	.00299	.00299	.00303	.00021	.00021	....	....	....	....
	.700	.00314	.00317	.00303	.00023	.00026	....	....	....	....
	.779	....	....	....	....	....	....	....	....	....
3000	.650	.00339	.00340	.00342	.00030	.00030	.409	.063	.062	.0202
	.700	.00362	.00363	.00342	.00035	.00037	.265	.072	.064	.0150
	.779	....	....	....	....	....	.205	.079	....	.0126
	∞	....	....	....	....	....	.086	.086	.088	.0058
3500	.650	.00381	.00378	.00382	.00042	.00043	....	....	....	....
	.700	.00409	.00407	.00379	.00050	.00050	....	....	....	....
	.779	....	....	....	....	....	....	....	....	....
4000	.650	....	....	....	....	....	.432	.069	.070	.0234
	.700	....	....	....	....	....	.282	.078	.070	.0172
	.779	....	....	....	....	....	.220	.085	....	.0146
	∞	....	....	....	....	....	.094	.094	.096	.0069

(*c*) *Contact Area.*—Several different methods were tried of making visible the area of contact between the ball and the race. The one which was best suited for the purpose and was, therefore, used in this work was a thin film of lubricating oil on the surface of the race. This film applied with the fingers, which were used to wipe the surface almost dry, was extremely thin. The ball was well cleaned.

TABLE 10.—Compression Test of Ball (1.5 Inches Diameter)

Load in pounds	Radius of races	Total deformation of ball and races			Permanent set of ball and races		Contact area			
		Observed value	Graph value	Hertz value	Observed value	Graph value	2a	2b	2b (Hertz value)	Area
	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch²
500	0.779	0.00071	0.00072	0.00098	0.00002	0.00002	....	....	....	....
	.841	.00078	.00082	.00098	.00003	.00003	....	....	....	....
1000	.779	.00126	.00131	.00156	.00005	.00005	0.270	0.045	0.046	0.0096
	.841	.00149	.00147	.00155	.00005	.00006	.189	.049	....	.0075
	∞	....	....	....	....	....	.059	.059	.064	.0027
1500	.779	.00177	.00182	.00204	.00007	.00007	....	....	....	....
	.841	.00203	.00205	.00203	.00009	.00009	....	....	....	....
2000	.779	.00225	.00228	.00247	.00011	.00012	.373	.058	.058	.0170
	.841	.00256	.00257	.00246	.00014	.00014	.249	.065	....	.0127
	∞	....	....	....	....	....	.080	.080	.082	.0050
2500	.779	.00272	.00271	.00287	.00016	.00016	....	....	....	....
	.841	.00305	00303	.00285	.00021	.00020	....	....	....	....
3000	.779	.00311	.00311	.00324	.00022	.00022	.427	.065	.066	.0218
	.841	.00348	.00349	.00322	.00028	.00028	.285	.075	....	.0168
	∞	....	....	....	....	....	.094	.094	.094	.0069
3500	.779	.00349	.00348	.00359	.00030	.00029	....	....	....	....
	.841	.00395	.00393	.00357	.00038	.00038	....	....	....	....
4000	.779	....	....	....	....	....	.458	.072	.074	.0258
	.841	....	....	....	....	....	.309	.083	....	.0201
	∞	....	....	....	....	....	.103	.103	.104	.0083

TABLE 11.—Compression Test of Balls

Load on ball, in pounds	Total deformation of ball and races					
	Ball, 1 inch diameter; radius of races, 0.550 inch		Ball, 1.25 inch diameter; radius of races, 0.700 inch		Ball, 1.5 inch diameter; radius of races, 0.841 inch	
	Observed value	Graph value	Observed value	Graph value	Observed value	Graph value
	Inch	Inch	Inch	Inch	Inch	Inch
500	....	0.00080	....	0.00080	....	0.00080
4 000	0.00376	.00445	0.00389	.00430	0.00348	.00420
8 000	.00671	.00753	.00679	.00740	.00618	.00690
12 000	.00950	.01040	.00932	.01010	.00844	.00930
16 000	.01231	.01320	.01167	.01250	.01066	.01150
20 000	.01528	.01600	.01367	.01475	.01277	.01360
24 000	.01790	.01870	.01598	.01695	.01482	.01570
28 000	.02087	.02145	.01834	.01915	.01698	.01775
32 000	....	....	.02070	.02125	.01806	.01875

The area of contact between the race and the ball was distinctly visible, as it appeared darker than the surrounding surface.

The edges of this area were sharply defined. The thickness of the oil film was estimated by drawing a ball lightly across an oiled plate and measuring the width of the dark band. Knowing the diameter of the ball, the angle subtended by the band at the center of the ball was easily computed, and from this the versine of half this angle. This, multiplied by the radius of the ball, was assumed to be the thickness of the oil film. The area of contact was measured by means of a microscope reading (by estimation) to 0.0004 inch.

After applying the load to the ball resting on the oiled surface, the ball was removed and the total area of contact was computed.

(*d*) *Results*.—The results of these tests are given in Tables 8, 9, 10, and 11. In the tables are also given the values of the deformations and of the areas of contact calculated by Hertz's theory.[3] Hertz's results may be written:

$$a=\mu\sqrt[3]{\frac{P}{H}\delta}$$

$$b=\nu\sqrt[3]{\frac{P}{H}\delta}$$

$$\alpha=\xi\sqrt[3]{\left(\frac{P}{H}\right)^2\frac{1}{\delta}}$$

$$2(A+B)=\frac{4}{\delta}=\zeta_{11}+\zeta_{12}+\zeta_{21}+\zeta_{22}$$

$$2(A-B)=+\sqrt{(\zeta_{11}-\zeta_{12})^2+(\zeta_{21}-\zeta_{22})^2+2(\zeta_{11}-\zeta_{12})(\zeta_{21}-\zeta_{22})\cos 2\omega}$$

$$\cos\tau=\frac{A-B}{A+B},\ H=\frac{4}{3}\frac{E}{1-\delta^2}$$

where:

α = total deformation of ball and races combined

$\left.\begin{matrix}2a\\2b\end{matrix}\right\}$ = diameters of area of contact

P = load

E = Young's modulus = 30 000 000 lbs./in.[2]

δ = Poisson's ratio = 3/10

H = 44 000 000 lbs./in.[2]

$\zeta_{11}, \zeta_{12}; \zeta_{21}, \zeta_{22}$ are the reciprocals of the principal radii of curvature of the two bodies; ω, the angle between their principal planes

[3] Heinrich Hertz, Gesammelte Werke, Leipzig 1895, 1, pp. 155 to 173; and F. Heerwagen, Zeitschrift des Vereins deutscher Ingenieure, 45, pp. 1701 to 1705; 1901.

of curvature and μ, ν and ξ, transcendental functions of the auxiliary angle τ, expressed in terms of elliptic integrals. μ, ν and ξ have been taken from the tables of Hertz and Heerwagen and are given below in Table 12 which was prepared by Dr. L. B. Tuckerman.

TABLE 12.—Coefficients for Hertz's Theory

τ	μ	ν	ξ	τ	μ	ν	ξ
30 degrees	2.731	0.493	1.453	70 degrees	1.284	0.802	1.944
35 degrees	2.397	.530	1.550	75 degrees	1.202	.846	1.967
40 degrees	2.136	.567	1.637	80 degrees	1.128	.893	1.985
45 degrees	1.926	.604	1.709	85 degrees	1.061	.944	1.996
50 degrees	1.754	.641	1.772	90 degrees	1.000	1.000	2.000
55 degrees	1.611	.678	1.828	95 degrees	.944	1.061	1.996
60 degrees	1.486	.717	1.875	100 degrees	.893	1.128	1.985
65 degrees	1.378	.759	1.912				

The values of total deformation approach closely those given by theory as shown in Figs. 20, 21, and 22. The existing differences may be explained by the nonuniform hardness, the difference between the actual and the assumed elastic properties of the material, and in addition by the fact that the major diameter of the area of contact is not as assumed by the theory, very small in comparison with the diameter of the ball. The same is true for the area of contact.

These tests show that the radii of the races influence the amount of the total deformation and of the permanent set more than the theory would indicate and in the opposite direction, that is, the larger the radii of races, the greater the deformation.

The total deformation of the ball was not measured separately but the direct measurements of the set of the races and the ball showed that the permanent set of a ball even for a load of 30 000 pounds does not exceed 0.00020 inch for 1½-inch diameter ball nor 0.00015 inch for a 1-inch diameter ball. Thus the permanent set observed is due almost exclusively to the races. The carrying capacity of balls with races given in Tables 13 and 14 are therefore limited by the deformation of the races. If the races had been harder, the values would have been higher. The theoretical value of 2a (the major diameter of contact area) is not given in the tables since it is so large that even approximate agreement could not be expected.

The values for the area of contact are plotted in Figs. 13, 14, 15, 16, and 17. Those for the deformation are shown in Figs. 20, 21, 22, and 23. The tests showed that even up to very high loads, far beyond those actually used in practice, the law of strains does not undergo any sharp change. The total deformation of ball and races follows pretty closely the law of a straight line with only a slight tendency to decrease gradually with an increase of load. The permanent set follows, also, the law of a

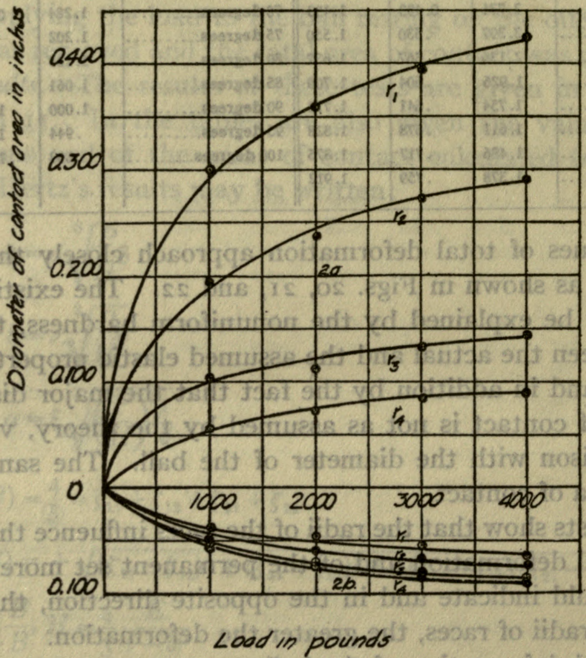

FIG. 13.—*Area of contact of 1-inch ball and races* ($r_1=0.515$ *inch*, $r_2=0.550$ *inch*, $r_3=0.779$ *inch*, $r_4=\infty$)

straight line but tends to increase gradually with an increase of load.

(*e*) *Conclusions*.—The allowable load on balls, as far as the permanent set is concerned, is limited to the load, which if increased, will produce a permanent set of either the balls or races, which would cause the bearing to fail to function properly. The permanent set will, in practice, first occur, probably, in the races. As the permanent set of the races grows very gradually, there is no definite indication of this load limit so that any limit selected is more or less arbitrary.

If we select 0.0001 inch [3] as the allowable permanent set of a race, we have from these tests the values of Table 13 for the carrying capacities of balls.

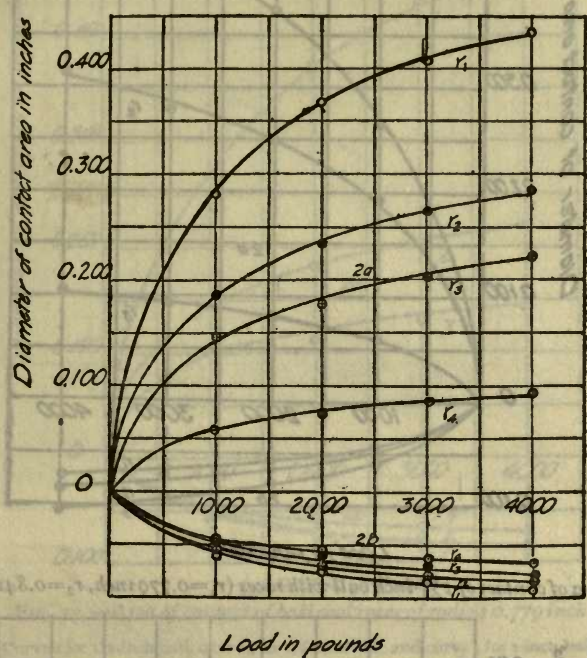

FIG. 14.—*Area of contact of 1¼-inch ball and races (r_1=0.650 inch, r_2=0.700 inch, r_3= 0.779 inch, $r_4=\infty$)*

TABLE 13.—Carrying Capacities of Balls with Races

Diameter of ball	Radius of race		Allowable load	
	r_1	r_2	r_1	r_2
	Inch	Inch	Pounds	Pounds
1.00 inch	0.515	0.550	2000	1800
1.25 inches	.650	.700	2500	2300
1.50 inches	.779	.841	2800	2500

A comparison of these values, with those given by the static friction test, shows that they are about 30 per cent larger. The allowable load on a ball may also be computed from the formula, $P=cd^2$, derived by Prof. Stribeck,[4] in which P is the load on the ball in kilograms; d is the diameter of the ball in centimeters,

[3] This value is often used as the allowable variation in the diameter of balls for bearings.

[4] Zeitschrift des Vereines deutscher Ingenieure, 45, p. 79; 1901.

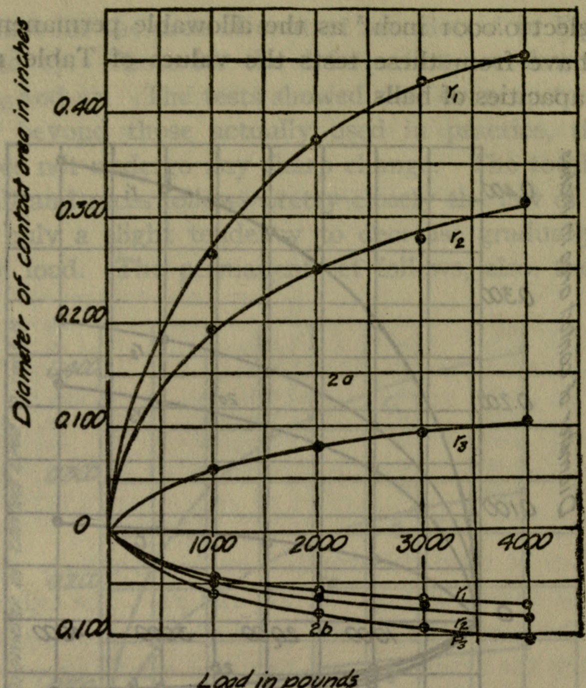

FIG. 15.—*Area of contact of 1½-inch ball with races* (r_1=0.779 *inch*, r_2=0.841 *inch*, $r_3=\infty$)

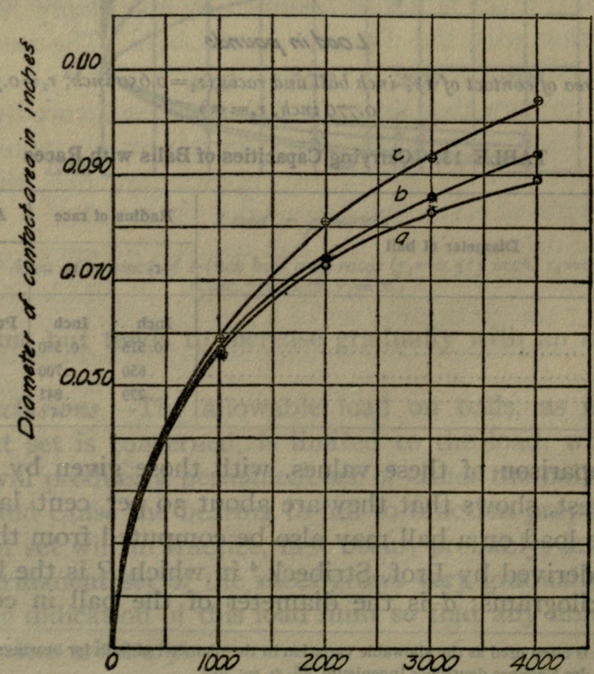

FIG. 16.—*Area of contact of ball and plates* (*a*=1-*inch*, *b*=1¼-*inch*, *c*=1½-*inch diameter*)

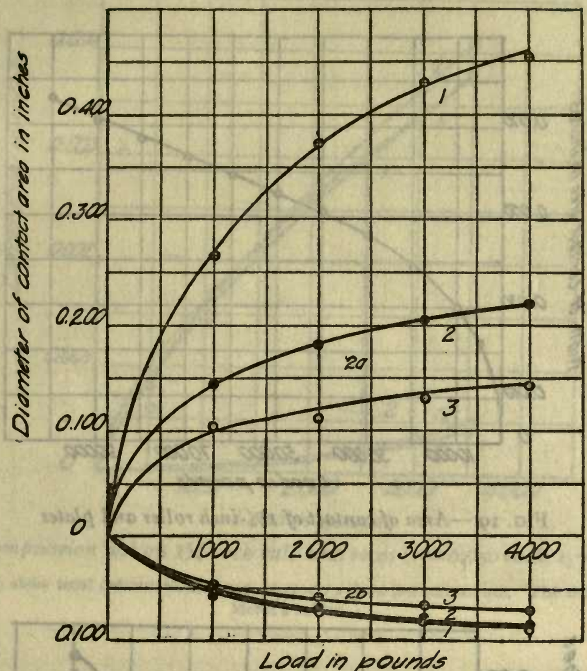

FIG. 17.—*Area of contact of ball and races of radius 0.779 inch*

Curve 1 for 1½-inch ball, curve 2 for 1¼-inch ball, and curve 3 for 1-inch ball

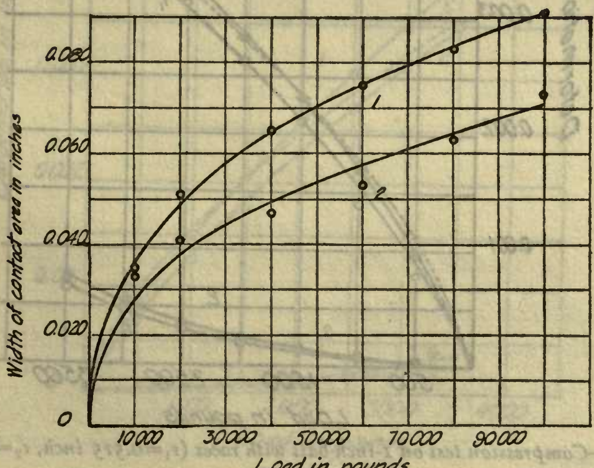

FIG. 18.—*Area of contact of a 1¼-inch roller between races of 3.5 inches and of 4.75 inches*

Curve 1, outer race; curve 2, inner race

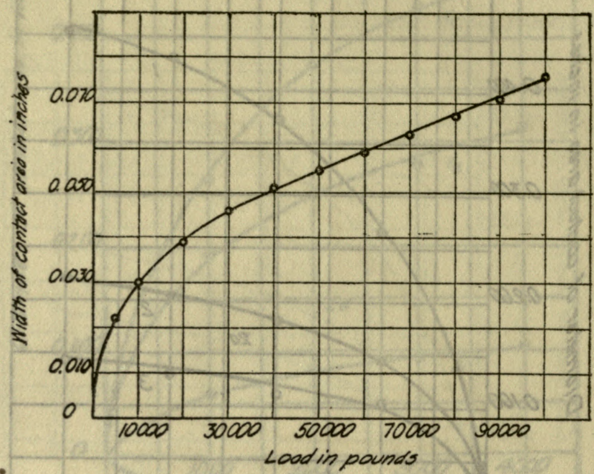

Fig. 19.—*Area of contact of 1¼-inch roller and plates*

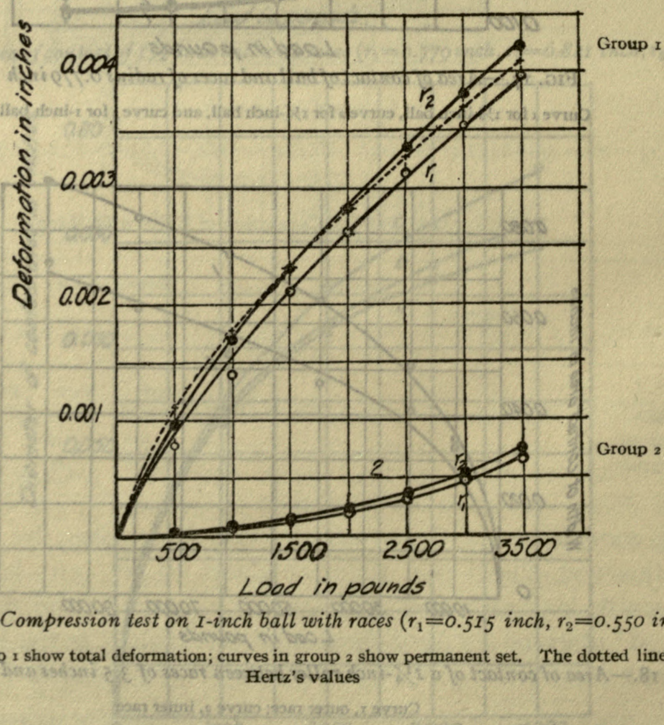

Fig. 20.—*Compression test on 1-inch ball with races (r_1=0.515 inch, r_2=0.550 inch)*

Curves in group 1 show total deformation; curves in group 2 show permanent set. The dotted line shows Hertz's values

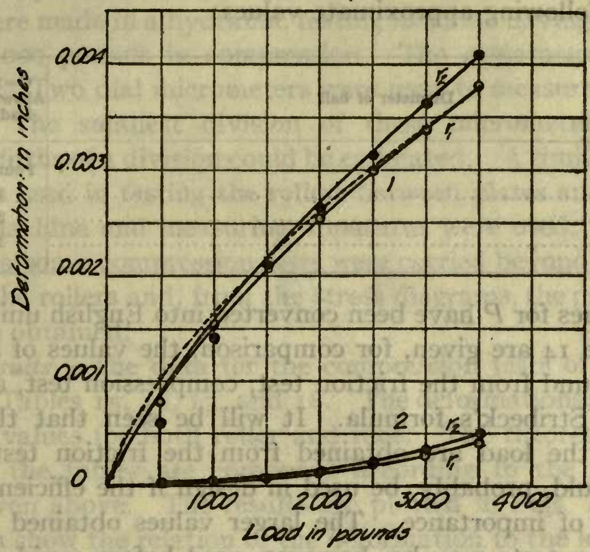

FIG. 21.—*Compression test on 1¼-inch ball with races (r_1=0.650 inch, r_2=0.700 inch)*

Curves in group 1 show total deformation; curves in group 2 show permanent set. The dotted line shows Hertz's values

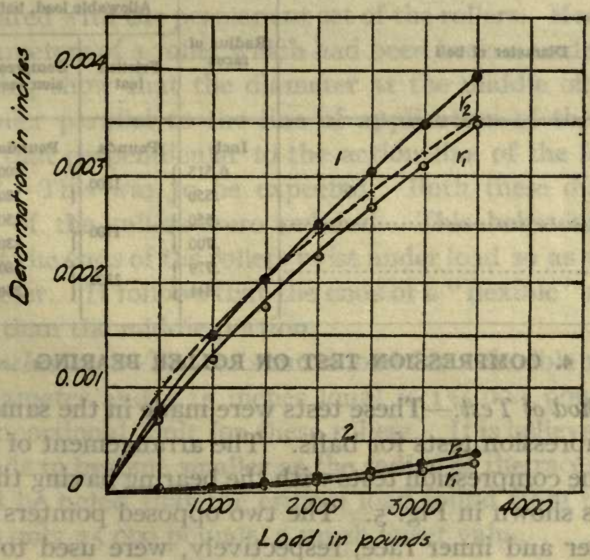

FIG. 22.—*Compression test on 1½-inch ball and races (r_1=0.779 inch, r_2=0.841 inch)*

Curves in group 1 show total deformation; curves in group 2 show permanent set. The dotted line shows Hertz's values

and *c* is a constant depending on the material. This formula gives the following approximate values:

Diameter of ball	C=100 Allowable load P	C=150 Allowable load P
	Pounds	Pounds
1.00 inch	1400	2100
1.25 inches	2200	3300
1.50 inches	3200	4800

The values for *P* have been converted into English units.

In Table 14 are given, for comparison, the values of allowable load, as found from the friction test, compression test, and those found by Stribeck's formula. It will be seen that the lowest values of the load are obtained from the friction test. These values should, probably, be used in design if the efficiency of the bearing is of importance. The larger values obtained from the compression tests may be, however, used before rapid deterioration of the bearings will result.

TABLE 14.—Carrying Capacities of Ball Bearings

Diameter of ball	Radius of races	Allowable load, ball with races		
		Friction test	Compression test	Stribeck formula (c=100)
	Inch	Pounds	Pounds	Pounds
1 inch	0.515	1300	2000	1400
	.550		1800	
1.25 inches	.650	1700	2500	2200
	.700		2300	
1.5 inches	.779	2200	2800	3200
	.841		2500	

4. COMPRESSION TEST ON ROLLER BEARING

(*a*) *Method of Test.*—These tests were made in the same manner as the compression tests for balls. The arrangement of the apparatus for the compression tests with the bearing having the smaller diameter is shown in Fig. 5. The two opposed pointers attached to the outer and inner race, respectively, were used to measure the deformation of the roller and races combined. A micrometer microscope was used at both ends of the roller to measure the distance between the ends of the pointers. The load was applied with a testing machine having a capacity of 100 000 pounds.

The compression tests with the bearings having the larger radius were made in a hydraulic testing machine having a capacity of 2 300 000 pounds in compression. The apparatus is shown in Fig. 6. Two dial micrometers were used to measure the deformation. The smallest division of these micrometers is 0.001 inch and fifths of a division could be estimated. A similar arrangement was used in testing the rollers between plates and the same testing machine and measuring apparatus were used. With this apparatus some compression tests were carried beyond the elastic limit of the rollers and, from the stress diagrams, the proportional limit was obtained.

(*b*) *Results.*—The data for the compression tests of rollers are given in Tables 15, 16, 17, and 18. The deformations are in each case the values for both roller and race. The theoretical values given in the tables are computed according to the formula of Hertz given above. The results are plotted in Figs. 24, 25, and 26, which show the relation of the deformation to the load. Figs. 18, 19, and 27 show the relation of area of contact to the load. The stress diagrams are shown in Figs. 28 and 29.

Inspection of the rollers and races showed that unlike the results with ball bearings the permanent set of the races was quite negligible compared with the permanent set of the rollers. Measurements of the diameters of a roller which had been broken under compressive loading show that the diameter at the middle of the length of the roller parallel to the line of application of the force was reduced, that perpendicular to the action line of the force it was increased. This was to be expected. Both these diameters at the ends of the rollers were reduced. This behavior seems to show that the ends of the rollers twist under load so as to decrease the diameter. It follows that the ends of a "flexible" roller carry less load than the middle portion.

(*c*) *Conclusions.*—The maximum load for a flexible roller (1.25 inches diameter and 5.25 inches long) is 135 000 pounds. This is the proportional limit for these rollers. It is believed that this value tends to become smaller as the radius of the races increases. It should be noted that the critical load found from the friction tests was only 25 000 pounds, a much lower value.

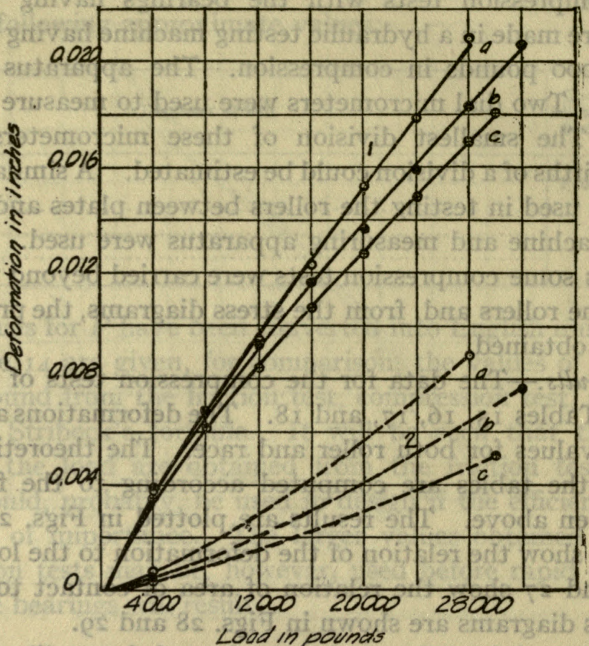

FIG. 23.—*Heavy compression test on 1-inch, 1¼-inch, and 1½-inch balls and races of 0.550-inch, 0.700-inch, and 0.841-inch radius*

Curves in group 1 show total deformation; curves in group 2 show permanent set; curve *a* is the test on the 1-inch ball, curve *b* is on the 1¼-inch ball, curve *c* on the 1½-inch ball

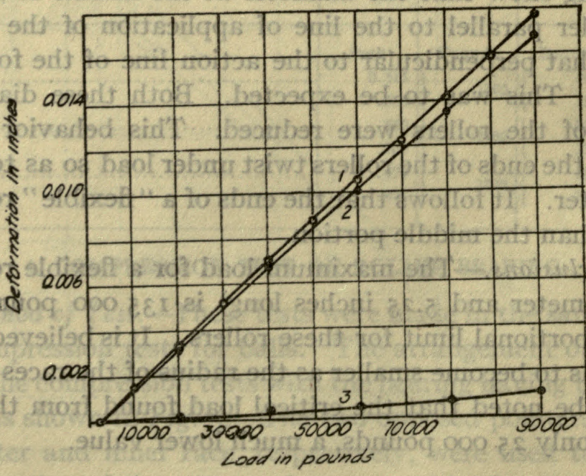

FIG. 24.—*Compression test on 1¼-inch roller between races of 3.5 inch and 4.75 inch radius*

Curve 1 shows total deformation, curve 2 shows elastic deformation, and curve 3 shows the permanent set

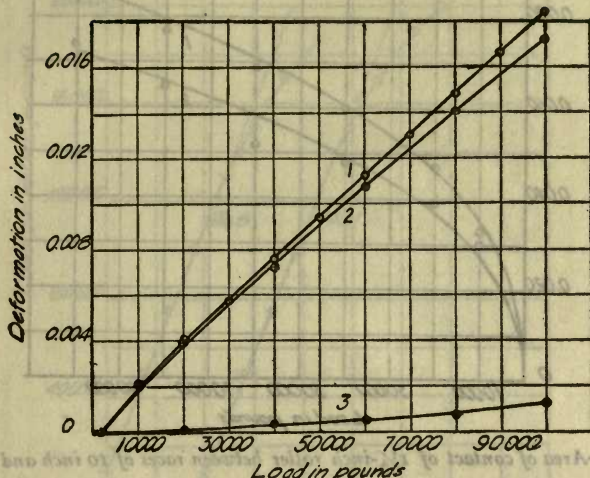

FIG. 25.—*Compression test on 1¼-inch roller between plates*

Curve 1 shows total deformation, curve 2 shows elastic deformation, and curve 3 shows permanent set

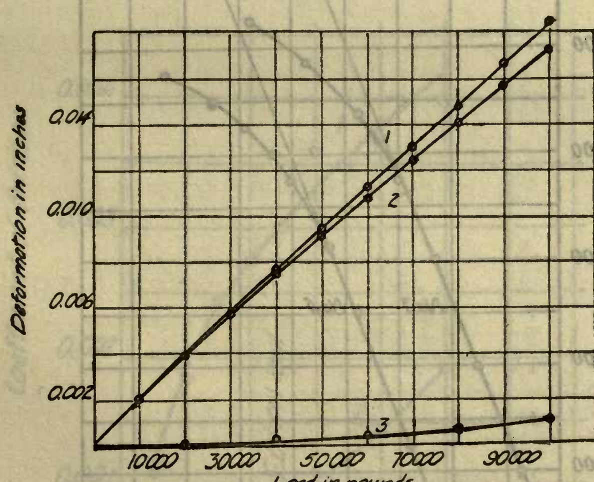

FIG. 26.—*Compression test on 1¼-inch roller between races of 10-inch and 11.25-inch radius*

Curve 1 shows total deformation, curve 2 shows elastic deformation, and curve 3 permanent set

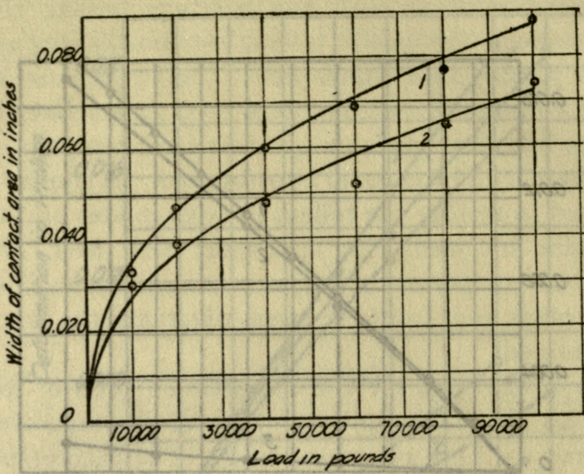

FIG. 27.—*Area of contact of 1¼-inch roller between races of 10 inch and 11.25 inch radius*

Curve 1, outer race; curve 2, inner race

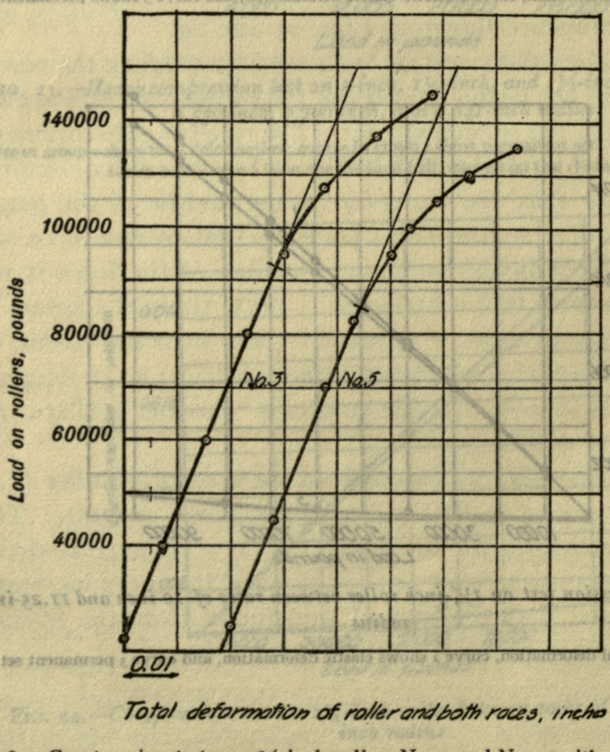

FIG. 28.—*Compression test on 1¼-inch rollers No. 3 and No. 5 with radius of inner race 3.5 inches*

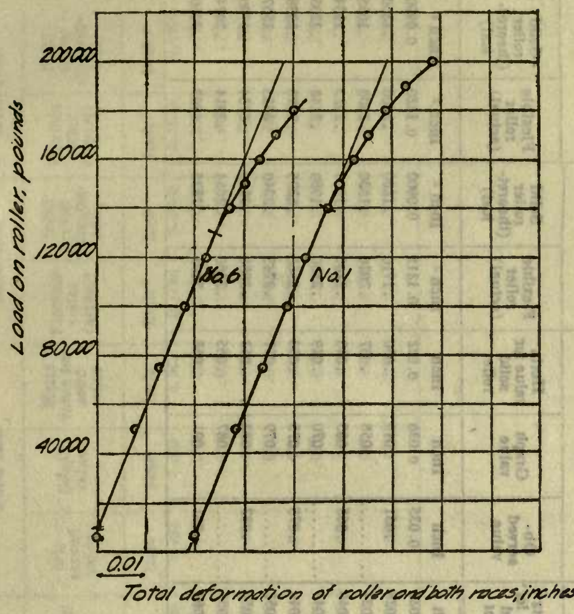

FIG. 29.—*Compression tests on 1¼-inch rollers No. 6 and No. 1 with radius of inner race 10 inches*

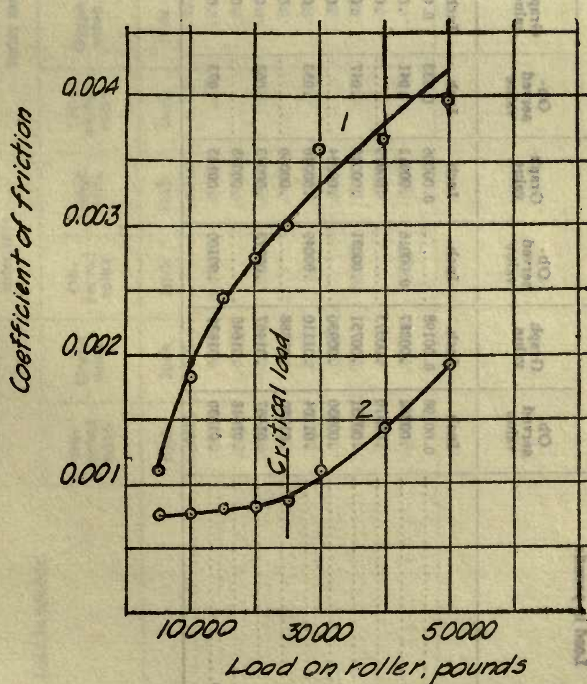

FIG. 30.—*Static friction test on 1¼-inch roller and races*

Curve 1, $r=3.5$ inches; curve 2, $r=10.0$ inches

TABLE 15.—Compression Test of Roller (1.25 Inches Diameter), Radius of Inner Race=3.5 Inches

Load in pounds	Total deformation		Permanent set of roller inner and outer race		Width of contact area						Contact area			
					Inner race			Outer race			Inner race		Outer race	
	Ob-served value	Graph value	Ob-served value	Graph value	Ob-served value	Graph value	Hertz value for solid roller	Ob-served value	Graph value	Hertz value for solid roller	Flexible roller (actual)	Solid roller (theoret-ical)	Flexible roller (actual)	Solid roller (theoret-ical)
	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch²	Inch²	Inch²	Inch²
10 000	0.00198	0.00198	....	0.00006	0.033	0.027	0.020	0.035	0.036	0.022	0.1215	0.0900	0.1620	0.0990
20 000	.00382	.00387	0.00016	.00012	.041	.038	.028	.051	.049	.030	.1710	.1260	.2206	.1350
30 000	.00570	.00573	....	.00019	....	.045	.034	....	.058	.037	.2026	.1530	.2608	.1665
40 000	.00751	.00751	.00031	.00026	.047	.049	.040	.065	.065	.043	.2206	.1800	.2923	.1934
50 000	.00930	.00930	....	.00034	....	.054	.044	....	.070	.049	.2430	.1980	.3148	.2205
60 000	.01104	.01110	.00049	.00046	.053	.057	.049	.075	.075	.053	.2566	.2205	.3373	.2386
70 000	.01280	.01288	....	.00060	....	.061	.052	....	.079	.057	.2745	.2340	.3555	.2567
80 000	.01467	.01467	.00073	.00073	.063	.064	.056	.083	.083	.062	.2880	.2520	.3734	.2790
90 000	.01648	.01648	....	.00088	....	.068	.059	....	.087	.065	.3060	.2654	.3914	.2924
100 000	.01830	.01830	.00100	.00105	.073	.071	.063	.091	.091	.068	.3193	.2835	.4093	.3060

TABLE 16.—Compression Test of Roller (1.25 Inches Diameter), Radius of Race=10 Inches

Load in pounds	Total deformation		Permanent set of roller inner and outer race		Width of contact area						Contact area			
					Inner race			Outer race			Inner race		Outer race	
	Observed value	Graph value	Observed value	Graph value	Observed value	Graph value	Hertz value for solid roller	Observed value	Graph value	Hertz value for solid roller	Flexible roller (actual)	Solid roller (theoretical)	Flexible roller (actual)	Solid roller (theoretical)
	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch²	Inch²	Inch²	Inch²
10 000	0.00208	0.00210	....	0.00003	0.030	0.027	0.020	0.033	0.034	0.022	0.1216	0.0900	0.1530	0.0990
20 000	.00394	.00403	0.00010	.00008	.038	.037	.028	.047	.046	.030	.1666	.1262	.2070	.1350
30 000	.00588	.00590	....	.00013	....	.044	.034	....	.054	.036	.1980	.1530	.2430	.1620
40 000	.00770	.00770	.00034	.00020	.048	.049	.040	.060	.060	.042	.2207	.1800	.2700	.1890
50 000	.00949	.00948	....	.00030	....	.054	.045	....	.066	.048	.2430	.2027	.2970	.2160
60 000	.01128	.01125	.00047	.00043	.052	.058	.050	.069	.071	.052	.2610	.2250	.3195	.2340
70 000	.01301	.01303	....	.00057	....	.062	.053	....	.075	.057	.2790	.2385	.3375	.2566
80 000	.01482	.01484	.00075	.00072	.065	.066	.057	.077	.080	.061	.2970	.2566	.3600	.2747
90 000	.01670	.01665	....	.00088	....	.069	.060	....	.083	.063	.3105	.2700	.3734	.2835
100 000	.01851	.01840	.00110	.00110	.074	.072	.064	.088	.087	.067	.3240	.2880	.3915	.3015

TABLE 17.—Compression Test of Roller (1.25 Inches Diameter), Between Two Plates

Load in pounds	Total deformation		Permanent set of roller and plates		Width of contact area			Contact area	
	Observed value	Graph value	Observed value	Graph value	Observed value	Graph value	Hertz value for solid roller	Flexible roller (actual)	Solid roller (theoretical)
	Inch	Inch	Inch	Inch	Inch	Inch	Inch	Inch [2]	Inch [2]
5 000					0.022	0.021	0.015	0.0945	0.0675
10 000	0.00243	0.00246		0.00007	.030	.029	.021	.1305	.0945
20 000	.00444	.00434	0.00008	.00013	.039	.039	.029	.1755	.1305
30 000	.00617	.00620		.00020	.046	.046	.036	.2070	.1620
40 000	.00800	.00803	.00035	.00029	.051	.051	.041	.2296	.1845
50 000	.00979	.00983		.00040	.055	.055	.046	.2474	.2070
60 000	.01162	.01164	.00049	.00053	.059	.059	.051	.2655	.2295
70 000	.01345	.01345		.00068	.063	.063	.055	.2835	.2474
80 000	.01522	.01523	.00079	.00085	.067	.067	.059	.3015	.2653
90 000	.01701	.01702		.00103	.071	.071	.062	.3193	.2787
100 000	.01892	.01880	.00128	.00125	.076	.075	.065	.3372	.2924

TABLE 18.—Compression Test of Roller (1.25 Inches Diameter)

Specimen (roller) No.	Radius of inner race 3.5 inches		Radius of inner race 10.0 inches	
	Proportional limit	Ultimate strength	Proportional limit	Ultimate strength
	Pounds	Pounds	Pounds	Pounds
3	143 000	211 000		
5	130 000	199 000		
6			130 000	184 000
1			140 000	206 000

WASHINGTON, March 21, 1921.