THE BRINELLING OF SMALL BALL BEARINGS.

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Thesis by

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INTRODUCTION.

The brinelling of ball bearings is a subject which has had comparitively little study. Since ball bearings have found most of their applications in such things as rotating shafts, there has been little reason to study their action under static loading conditions. This research was suggested by the Fafnir Ball Bearing Company because of the increasing use of ball bearings for static loads in aircraft. In the aircraft business the designers, because of strict weight requrirements, have used ball bearings up to their rated loads. Not infrequently these bearings brinelled. (A brinelled bearing is one in which a dent has been formed in one of the races by a ball) It is therfore evident that more should be known about the brinelling of bearings.

The original research on ball bearings was done about 1900 by Professor Stribeck in Germany. This research today forms the foundation for most bearing design. Stribeck proved the validity of Hertz's original theory of the contact stressed in bodies having the general shape of balls and races. (Ref.1)

In 1921 the Bureau of Standards concluded some tests on the brinelling of ball bearings. (Ref. 2) These tests were on single balls and races, and were made on balls of fairly large diameter. (1, $1\frac{1}{4}$, $1\frac{1}{2}$ inches) Two methods were used in these tests. The first method was to measure the starting friction of a ball-race combination under load. The second method was to measure permanent deformations in the combination. It was shown that these deformations were in the races and that the balls deformed but slightly. The first of these methods proved the most satisfactory.

Modern factory tests consist of loading a series of ball bearings to given loads and then disassembling them and measuring the races for brinells.

All research on ball bearings, since that of Stribeck, has tended to show the validity of his derived equations. For Maximum Static Non-Brinell rating Stribeck derived the now widely used formula,

$$P = knd^2$$

where,

P = rated load

n = number of balls in the bearing

d = diameter of the balls

k = a constant determined from tests

This formula has proven very satisfactory in the past. For it to yield good results, however, it is essential that the value for "k" be determined accurately. The purpose of this research then becomes, after a method of testing has been developed, to determine accurately when bearings fail and to see if these critical loads combine to give a constant for the above formula. It seems entirely possible that the above formula is not entirely complete, and that it might be modified to include other bearing dimensions. Hertz's theory (Ref. 2, 3, and 4) indicates that all race radii have a definite effect upon the stresses in the bearing.

SUGGESTED TEST METHODS

The modern industrial method of testing, which has been previously mentioned, seems to give quite satisfactory results, but is slow and subject to numerous inaccuracies. For these reasons one of the chief aims of this research was to develop a new rapid method which would give good accuracy and be applicable to industrial use. Several methods were suggested.

The first method that is usually thought of for this sort of work is the method of measuring deformations for various loads. Difficulty was anticipated for this method because of the small dimensions of the bearings and the very small deformations that were expected.

As has been mentioned, a method in which friction of the bearing underrload is measurd has been used successfully. In this case, however, this offered very severe experimental difficulties and so was considerd impractical. An alternative method is to measure the starting friction after a load has been applied and removed.

A very ellegant test method of an entirely different nature was also suggested. The suggestion was that a contact microphone be used to listen to a bearing after it had been subjected to a brinelling load. It seemed logical that the output of the microphone, after suitable amplifiaation, would give voltmeter readings that would be proportional to the brinell and that this data could be extropolated to a zero brinell load. This method seemed to offer definite admantages for industrial work.

TESTING EQUIPMENT.

For testing several testing machines were available in various sizes, so that the actual loading of the bearings caused little worry. To firmly hold the bearing in the testing machine a test device was designed and built. (figure 2) This was designed with several thoughts in mind and proved very satisfactory. In the first place the device must not deform to any measurable extent itself. Secondly it must be adaptable to various sizes of bearings. Thirdly it should be so designed that the bearing shaft could be rotated under load.

The first condition was met by machining the device from $1\frac{1}{2}$ x 5 inch cold rolled steel. The second desire was fulfilled by the use of sliding uprights which could be locked in position with a clamping arrangement. A system of sliding blocks was included so that the shaft could be raised from its Vee grooves onto small ball bearings which acted as rollers.

The housings for the bearings were designed so as to have an excess of material, since it was considered undesirable to have the bearings insufficiently supported. A typical housing is shown in figure 4. All housings were machined from mild steel. They were bored to tightly fit the bearings and were made with a slot that could be closed by tightening a bolt that was threaded into the housing. The width of the housings were slightly larger than the width of the bearings.

So that slight manufacturing differences would not cause differences in the tightness of the bearing fit, the shafts were ground with a slight taper. This taper was such that the average bearing fit tightly at the center of the shaft.

STARTING FRICTION TESTS.

Figure 5 shows a set-up that was used in an attempt to measure starting friction in a brinelled bearing as a measure of the depth of the brinell. By passing a measured current through the solenoid, the beam could be deflected and the deflection measured on the scale at the end of the boom.

The test procedure was to test a bearing by measuring changes in deflection for a given current. It was hoped that a brinell in the race would decrease the deflection for the same current on a good bearing.

The results of this test did not justify the hopes above mentioned. No difference was shown between good or bad bearings. In this test the bearing was loaded to about ten pounds by the use of boom weights as shown in the sketch. With the apparatus as shown it was not practical to use more load. If this had of been done, however, some results might have been obtained since the friction is proportionate to the load. The only other alternative is to increase the sensitivity of the apparatus.

TESTS USING A CONTACT MICROPHONE.

A great deal of time and effort was expended in attempting to develop a method using a contact microphone. For these tests I used a commercial contact microphone, or vibration pickup, of the crystal type. Figure 6 showa the general arrangement of the apparatus. The bearing was placed in the Vee grooves of the test device and raised onto the rollers. The bearing shaft was connected to a small motor with a length of rubber tubing. The probe of the pickup was threaded into the bearing housing, and the electrical leads from it were led to a filter built for this purpose. From the filter the leads ran to either a voltmeter of a commercial oscillograph.

Tests were made using good and badly Brinelled bearings. Various loads and shaft speeds were tried. Loads ranged up to 50 pounds and speeds from 10 to 200 r.p.m. The filter was deigned to elliminate all frequencies above 50 and was removed from time to time to note its effect. It was thought that the ordinary race noise would be removed by the filter, leaving only the clicks from the brinells.

This method gave no results. The trouble seemed to be that the ordinary noise from the ball bearing is much stronger than the noise due to failure in the race. Another difficulty with this system is that the bearings must by absolutely clean. The slightest trace of dust has much more effect than any other element entering. This effect of dirt seems to elliminate this method from the industrial class of test.

DEFORMATION METHOD.

The method which was finally accepted as the most satisfactory was the method of measuring deformations, total and permanent, under load. The apparatus for this method was finally developed to a system that seemed completely accurate.

Figure 2a shows the final form of the deformation measuring device. It, as can be noted, consisted of a set of multiplying levers on each side of the bearing housing and fastened to the housing. These levers were actuated by the relative upward motion of the bearing shaft refered to the housing. These arms moved the plunger of a standard Ames gauge. (1/100mm) The multiplication amounted to 12:1 and was considered satisfactory. Greater magnification may be used, but it was found that the accuracy so obtained was not warrented by the testing machine. The arms contacted the Ames gauge through a cross bar that was designed to average the deflections of the two arms. It was found that it was necessary to measure the deformations on each side of the bearing because of the tendency of the bearing to twist in the machine.

CHECKS ON THE SYSTEM. The testing equipment has been checked in several ways and there is no indication that it is faulty in design.

Figure 7 is the graphical result of tests made on a 203K ball bearing and on a solid ring of the same dimensions. The ring was machined from Utica,,a high grade tool steel. It may be noted from this graph that there is practically no definite relatioh between the solid ring and the beaing.

The other check made was to test the same bearing three times with various arrangements of lever mounting. The results of these tests are plotted in figure 8 and show no marked differences. The sketch indicates the various mounting methods tried. Sketch (a) is the actual test set-up and (b) and (c) are the check arrangements. Method (b) was designed to remove any angular housing deformation that might affect the results. Method (c) was dimiliar to (b), but was designed to remove the chance of motion at the pivot point. It may be noted that the deviations in these tests are of the same order as the usual experimental deviations for any one bearing and that no change can be blamed on any of the lever mountings.

It is believed that these checks clearly indicate that no measurable errors were introduced by the testing equipment.

HERTZ'S THEORY AS RELATED TO THIS PROBLEM.

Table I contains the tabular calculations from which the theoretical stresses in the ball bearings tested in this research can be obtained for any load. These calculations make use of the Hertz theory and are calculated according to a method given in reference 3 with some variations. Actually this theory was developed for materials which did not exceed the elastic limit. Stribeck, however, has shown that the theory is very good for normal ball bearing stresses. The use of the theory in this paper is only to obtain an approximation of the actual results.

Since the exact method used in these calculations differs slightly from those given in the references, and since the general method is not widely used, I include here a brief discussion of it. Another reason for discussing it here is that available English translations do not include numerical values that apply to small bearings of this type. In order to make the calculations for small bearings I have, in figure 16, plotted the necessary numerical values as taken from reference 2. It willbe found that these values cover all ball bearing sizes.

METHOD OF CALCULATING STRESSES IN BALL BEARINGS
(Reference 3 with variations)

The theory assumes that the contact area of the ball on the race is an ellipse of the following dimensions,

$$a = \mu \sqrt[3]{\frac{P_o}{H}\, m}$$

$$b = \nu \sqrt[3]{\frac{P_o}{H}\, m}$$

where,

P_o = the load on the uppermost ball

$H = E/12(1 - \delta^2)$

δ = Poissons ratio

and where m, μ, and ν, are obtained from the following formulae and from figure 16,

$$\cos\tau = \frac{A - B}{A + B}$$

where,

$$4/m = 2(A + B) = 1/r_{11} + 1/r_{12} + 1/r_{21} + 1/r_{22}$$

$$2(A - B) = 1/r_{21} - 1/r_{22}$$ (note, this is true only for a ball bearing where $r_{11} = r_{12}$)

where,

$r_{11} = r_{12}$ = radius of ball

r_{21} = groove radius

r_{22} = race radius

The maximum stress in the contact area is given by,

$$\sigma = 1.5 \frac{P}{\pi a \cdot b}$$

It should be noted that the stress will be largest in the inner race since the contact area is here the smallest.

In my calculations, however, it was convenient to separate P_o from the final stress expression. If this is done we get an expression,

$$\sigma = Q\, P_o^{1/3}$$

where,

$$Q = \frac{1.5 \quad H}{\pi (\mu \nu) \quad m^{2/3}} = \frac{5.94 \times 10^4}{(\mu \nu)\ m^{2/3}}$$

Stribeck has shown, reference 1, that the load on the uppermost ball is given by the formula,

$$P_o = (5/n)\ P$$

where,

P = total radial load on ball bearing

n = number of balls in bearing

Hence, for the final expression of stress in the inner race, we have,

$$\sigma = Q \left(\frac{5\ P}{n} \right)^{1/3}$$

RESULTS..

As has been mentioned, the greatest problem in this study was to find a good method of testing the bearings. It is believed that such a method has been found. It was desirable, after the test method was developed, to spend as much time as remained in testing a series of bearings with the purpose of checking the present bearing rating system. Because of limited time I was only able to test the following bearings.

BEARINGS TESTED

Quantity	Type
2	201K
2	202K
2	203F
2	204H
2	204W

I was informed that the basic design of the K, H, and F bearings were the same. This type is in general listed as the K or Conrad type and is the Non-Filling Slot Type. The W bearing is known as the Maximum type. The difference between a K and a W bearing of the same number is in the number of balls and in a 1% difference in groove radius. The W bearing has the most balls and the largest groove radius. The characteristics of the Various bearings are listed in Table I. It can be noted from a study of this Table that the bearings tested formed a fairly definite series in their design dimensions, and were chosen with this in mind.

Figures (9 to 13 are the results, in graphical form, of the tests. Two curves were obtained in each case and in general these curves are quite close to each other for any one type of bearing. Manufacturing differences probably explain most of the differences between two bearings of the same type.

In Table II are listed all of the permanent sets obtained in the tests. These values are plotted in Figure 14 in which the scale has been increased so as to better bring out the differences in the various curves. The values in Table II and Figure 14 are the values which are of interest in this problem because they are the Brinell measurments.

It is apparent from figure 14 that there is no definite load at which brinelling actually begins. In stead the bearings actually start to brinell at very small loads and it is only because of lack of sensitivity in the measuring device that this brinelling is not found earlier in the test. It may be noted here that the curves of figure 14 are similiar in form to those recorded in Reference 3.

In these tests the bearings were always loaded with one ball uppermost which is, of course, the critical condition. Permanent sets were measured at loads of from 300 to 500 pounds in these tests as this was more satisfactory from a physical standpoint than making the measurments at zero load. Tests were made which showed that there was no strange phenomena observed between zero and several hundred pounds, so that a single measured deformation caused by the initial load was all that was necessary to extropolate the data to zero.

DISCUSSION OF RESULTS.

Since there does not seem to be any definite load for any bearing at which brinelling starts,it becomes necessary to define some load that will serve in an analysis. Since in the manufacture of ball bearings a tollerance of 0.0001" is practice it seems logical that we define the Maximum Static Non-Brinell Load as the load that will cause a dent of 0.0001 inch to form in the race. If we then use this definition, we can obtain the critical brinell loads for the bearings tested from Figure 14. This has been done and the loads are listed in Table III and in Figure 17.

As has been mentioned previously in this paper Stribeck has shown that,for average bearings, the Maximum Static Non-Brinell Load bears a constant relation to the quantity nd^2. To see how well this held in our case I calculated values for Sribecks K (which I call K_1) and have recorded them in Table III. The average value, 2680, was then put in Stribecks formula and the expected loads calculated. These are listed in Table III and plotted in Figure 17.

The loads calculated by Stribeck's equation present two serious errors. For the 201K and the 202K the Knd^2 load is the same, but actually we see that the 202K bearing has a higher critical load. The test results show the 204H and 204W to have about the same non brinell-load, but the Stribeck formula indicates that the 204W should brinell at a much higher load. These difficulties seem definite even though the test data is not too extensive.

In order to try and explain the differences encountered between Stribeck's formula and experiment, I calculated the theoretical stresses in the bearing by Hertz's method. The stresses obtained were exceedingly high but seemed to be fairly constant at about 600,000 lbs. per. sq. in. Since the stress shows indications of being constant, I examined my calculations and was able to reduce the stress calculations to a form containing a constant and the bearing dimensions. As has been mentioned, the stress in a given bearing is given by, in my notation,

$$\sigma = Q\,P_o^{1/3}$$

where,

$$P_o = (5/n)P$$

where,

$P =$ the radial load on the bearing

if we assume that,

$$\sigma = \text{const.}$$

then,

$$P = K_2(\,n/Q^3\,)$$

In table III I have listed values calculated for this K_2, and have calculated rated load from the average of these values. These loads are plotted in figure 17. Actually the values obtained in this manner do not seem to be any more accurate than the values obtained by Stribeck's method, but they do tend to correct the deviations mentioned before. It can be seen that this method predicts a higher load for the 202K than for the 201K, and that it indicates a slightly lower load for the 204W than for the 204H bearings. Since the test data is so limited, very little can be said about either of these methods. It seems, however, that Stribecks equation may not be too accurate for the bearings tested.

It seems well at this point to attempt an explanation of the deviation noted in these tests. In the past Stribeck's equation has given very good results. This may be do in part to the fact that bearings used were not loaded to the rated values but were used with considerable margins. Another explanation can be given, however. If we reduce the Hertz equations previously given to a convenient form, we get.

$$r_{22} = \frac{-\,ar_1\ (1/\cos\tau + 1\ )}{\left[(\ 2a + 1\ ) - 1/\cos\tau\right]}$$

$$m = \frac{4\ r_1}{\frac{2a + 1}{a} + \frac{r_1}{r_{22}}}$$

where,

$a = r_{21}/r_1$ $\qquad$ $a = 2$ (51%) for W series bearings

From these equations it can be seen that for constant ball radius r_1, $\cos\tau$ gets larger for a decreasing value of r_{22}, and $\cos\tau$ gets smaller for an increasing r_{22}. Then for large values of the race radius r_{22}, we will find our values of μ and ν on the right side of Figure 16. In this region it can be seen that the curves for these values are not rapidly increasing. It then seems possible that ~~xx~~ an expression such as Stribeck's equati on would hold in this range, which corresponds to the range of large bearings, but would not hold at the other side of the graph which is the region of small bearings.

The method of rating which I have introduced here is quite complicated to use and so will probably never be employed. It is put in however to indicate a line of attack if this work is ever continued. In the data, incomplete as it may be, it is easy to see that the race radius r_{22} has a very marked effect upon the critical load. The radius of the groove, on the other hand, bears a constant relation to the ball radius and so does not seem to be important. I have attempted to obtain formulas that would express the results as presented here, but the test data is so incomplete and the actual mathematics so complex that no sucess has been enjoyed.

CONCLUSIONS...

The test method developed in this research seems to be accurate and trustworthy. It is a rapid method which should lend itself to industrial use. The method is inexpensive and requires little other than an Ames Gauge and an ordinary compression machine. Only a short time is required to master the experimental technic. It seems possible to obtain measurments to 0.00001 inches with experimental errors of less than 5 per cent.

The test data reported in this report is very incomplete and as such can not definitely indicate the validity of any rating system. What indications there are seem to be that Stribeck's equation is not accurate for bearings of this size, but that Hertz's theoretical work is good. It seems possible to calculate the Maximum Static Non-Brinell Loads from a formula based upon Hertz's work. This method is long and not as simple as could be desired. Since the test method is fast and easy to apply, it seems as though the best method of obtaining load ratings is from actual tests.

Future research work could be devoted to improving the finer features of the testing equipment, and to making extensive tests on complete series of bearings. Only when the results of such tests are available will useful formulae be derived.

ACKNOWLEDGMENT

The research reported in this paper has been greatly aided by several parties. I wish at this time to express my thanks for the cooperation given me by Mr. W. N. Fairfield of the Fafnir Bearing Company, and to the Fafnir Bearing Company for the ball bearings furnished. I am also grateful for the suggestions proffered by Dr. E. E. Sechler and Dr. A. L. Klein of the Guggenheim Aeronautical Laboratory of the California Institute of Technology, under whose supervision I worked.

REFERENCES.

(1) H. Hess. "A. S. M. E. Transactions", vol. 29
(A translation of Stribeck's original paper on ball bearings.)

(2) H. Hertz. "Gesammelte Werke." vol. I. pp 155 - 173

(3) H.L. Whittemore and S. N. Petrenko. "Technical Papers of the Bureau of Standards" No. 201

(4) Timoshenko. " Strength of Materials" vol. II

PHOTOGRAPHS

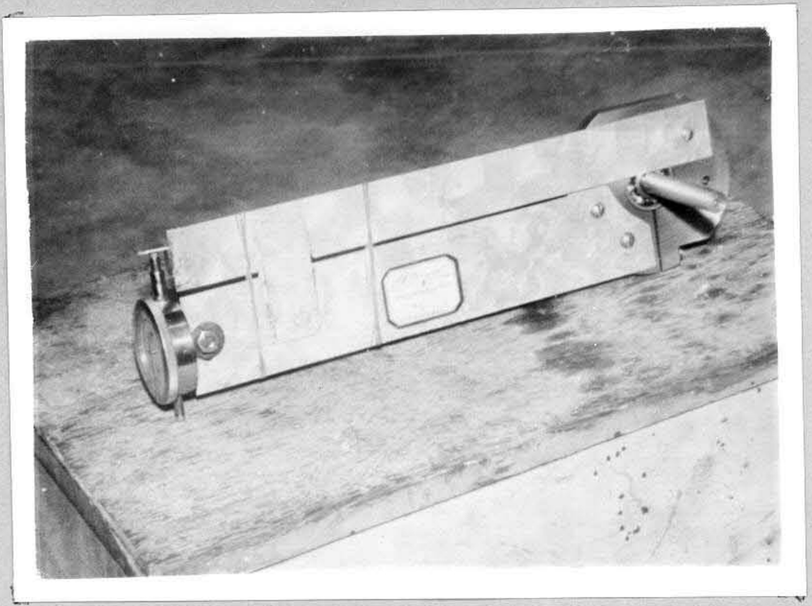


Fig. 1
An early form of the measuring device, showing housing, shaft, multiplying arms, and Ames gauge.

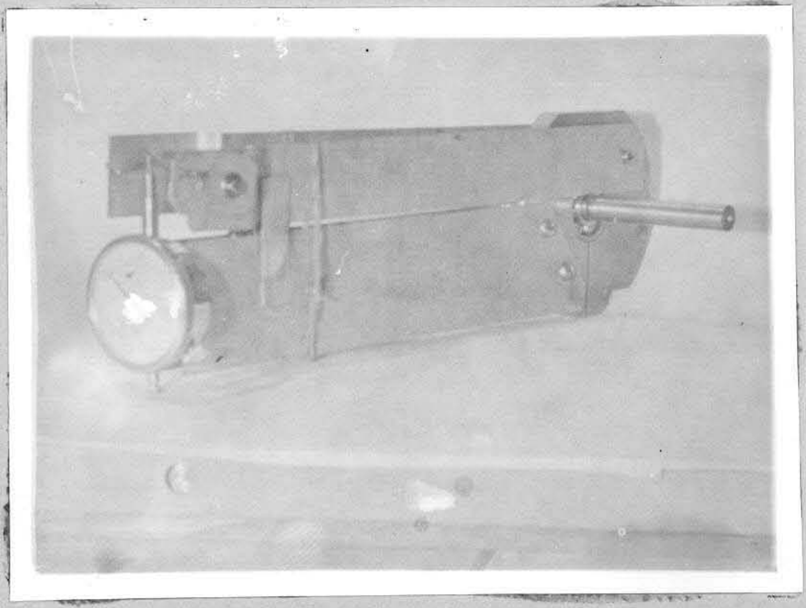


Fig. 1a
The final measuring device. Showing two arms and the connecting bar for averaging the two deflections.

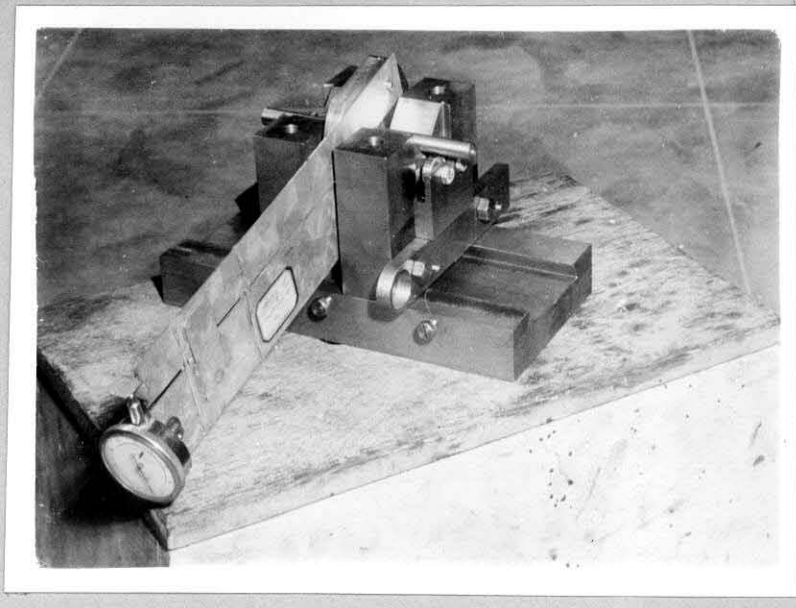


Fig. 2
Showing the measuring device in the support, ready for testing.

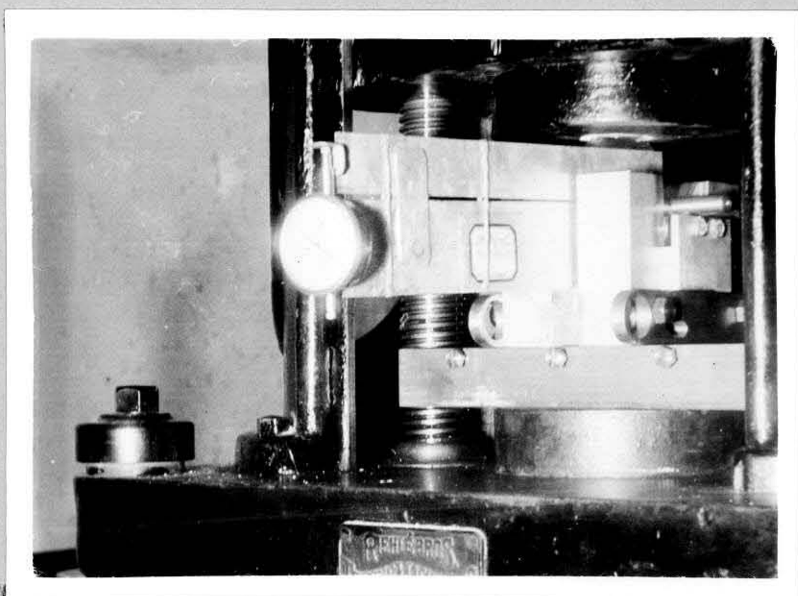


Fig. 3
The testing equipment in a standard testing machine.

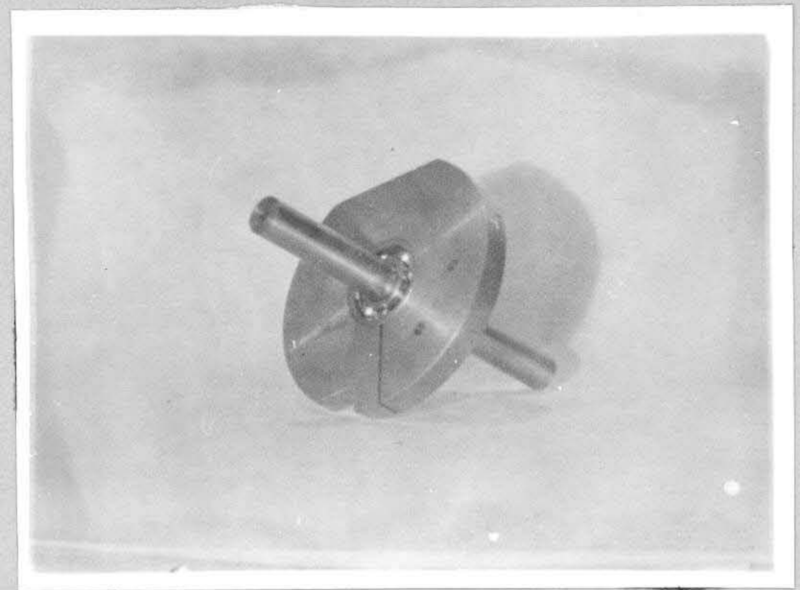


Fig. 4
A Ball Bearing in its housing and on its shaft.

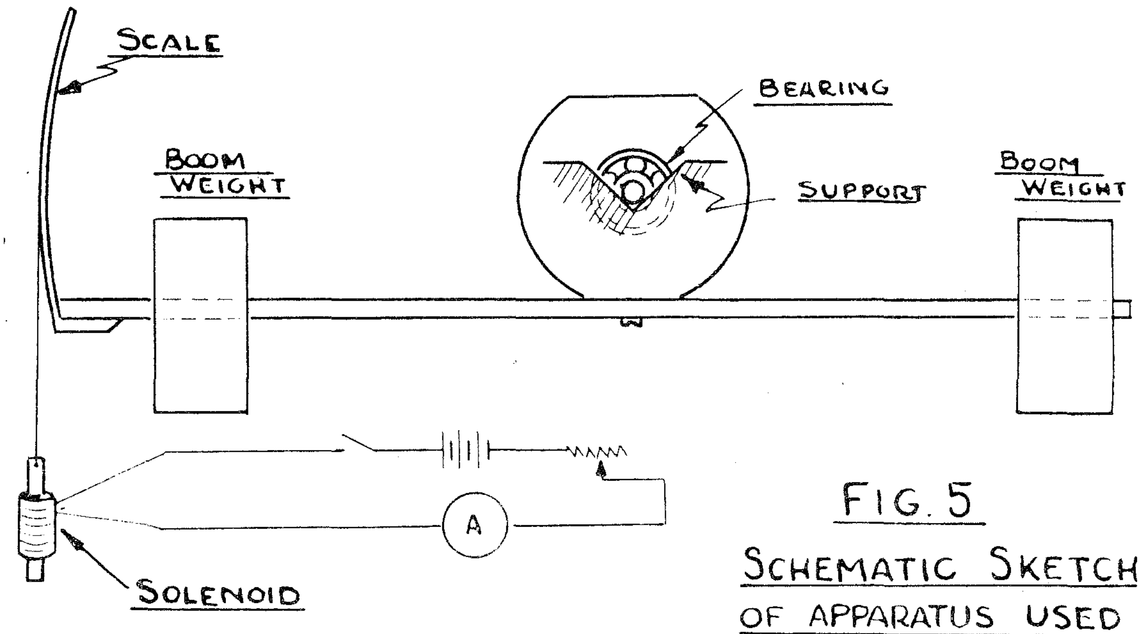
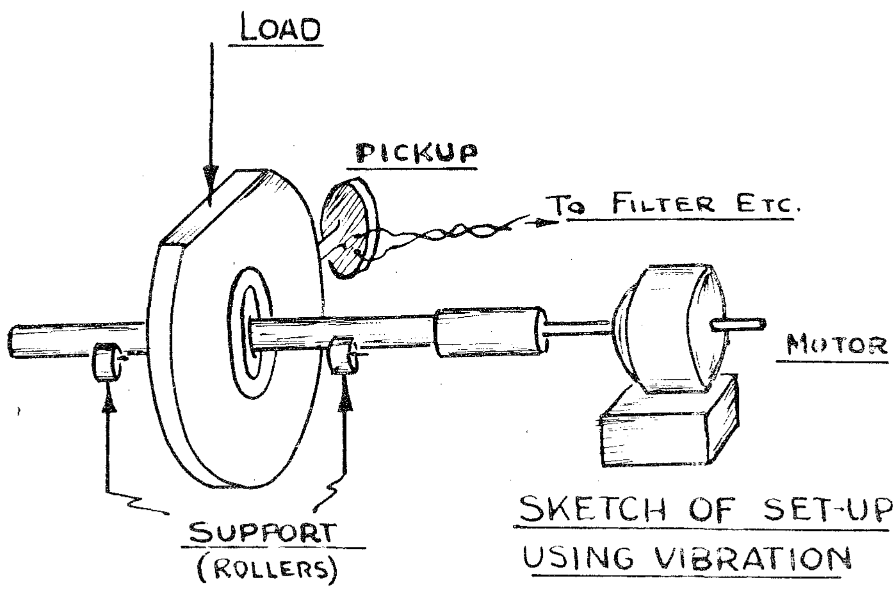


FIG. 5

SCHEMATIC SKETCH OF APPARATUS USED FOR STARTING FRICTION TESTS.



SKETCH OF SET-UP USING VIBRATION PICKUP.

FIG. 6

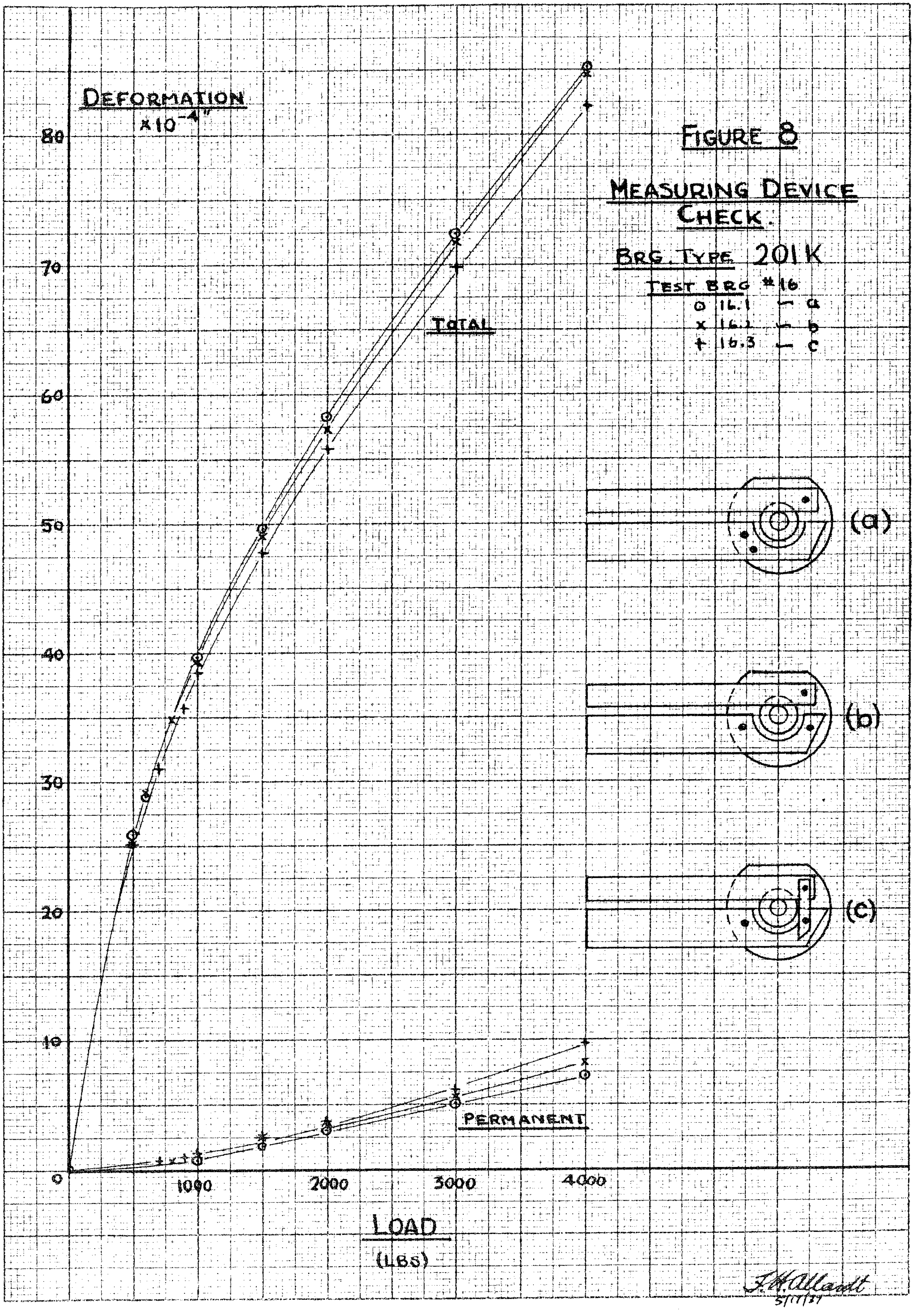
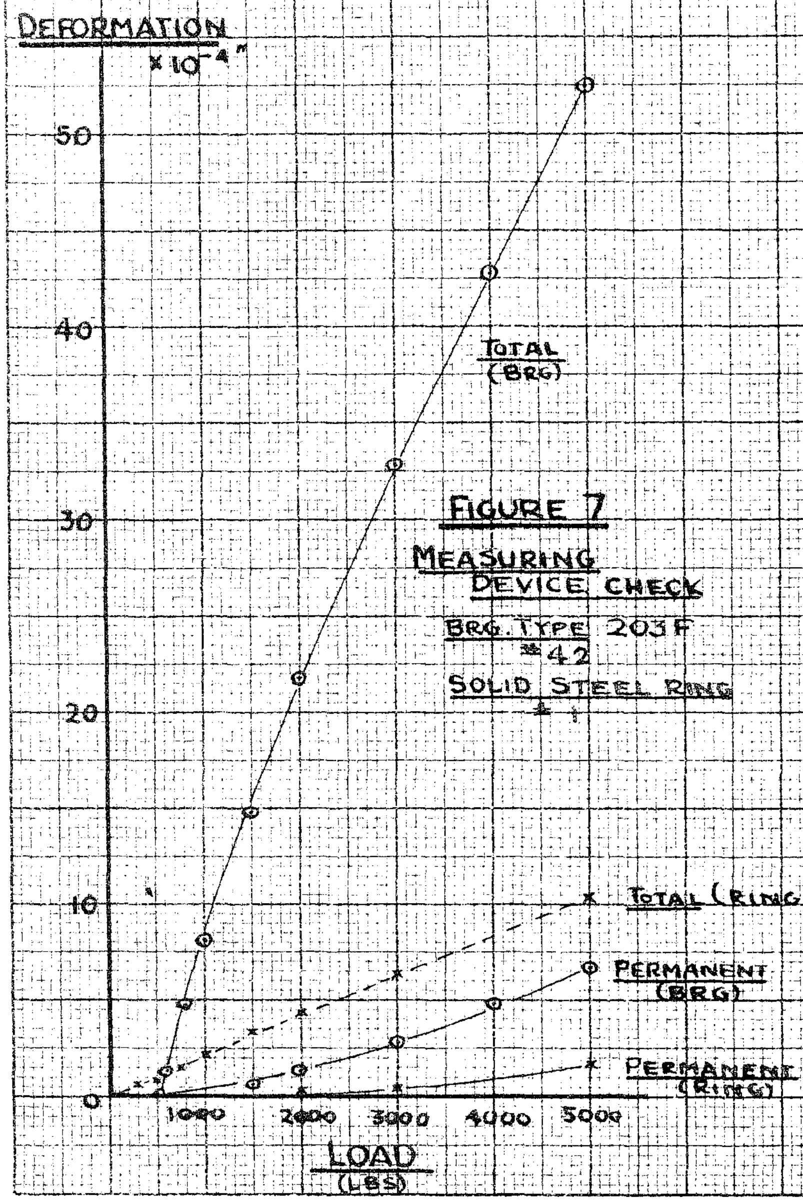
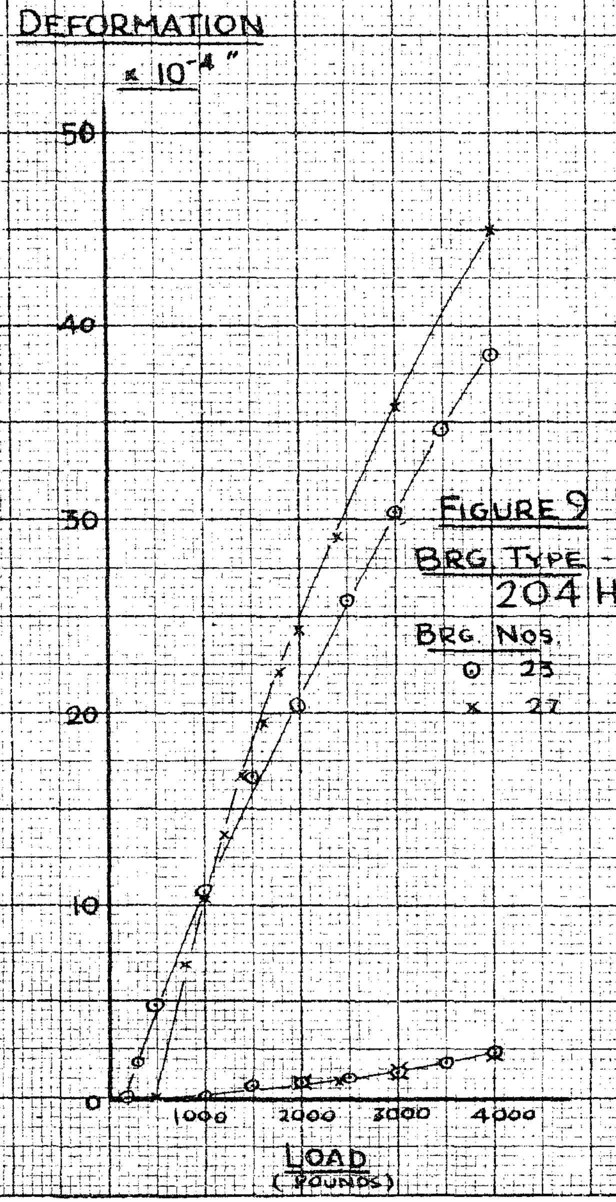
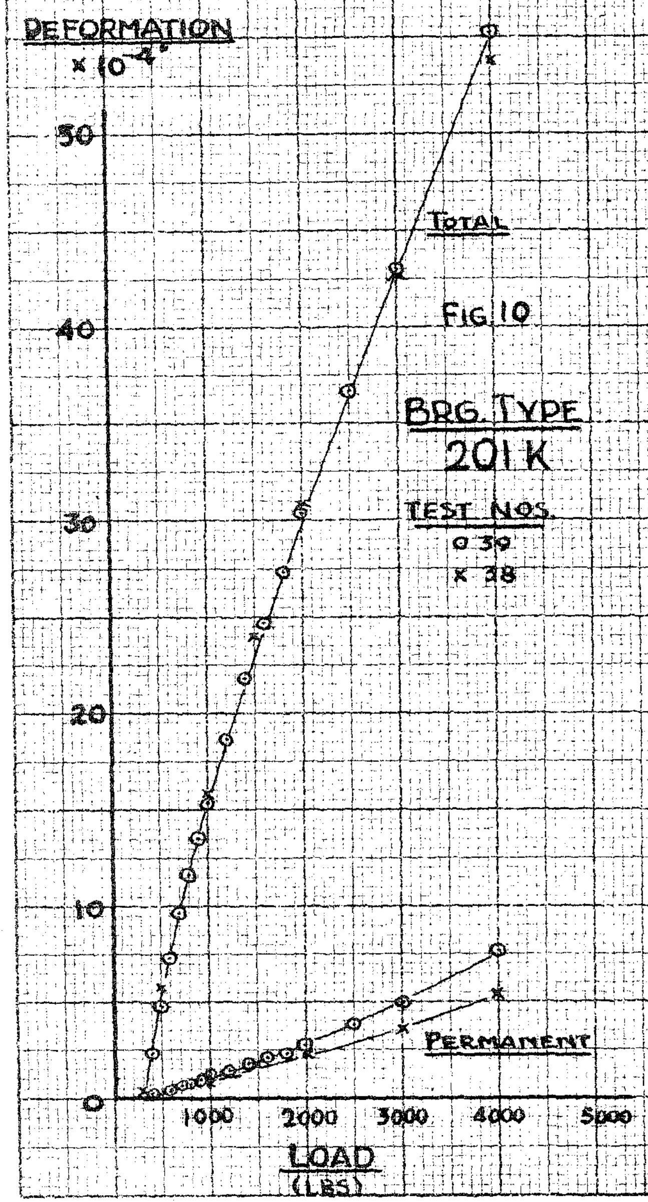
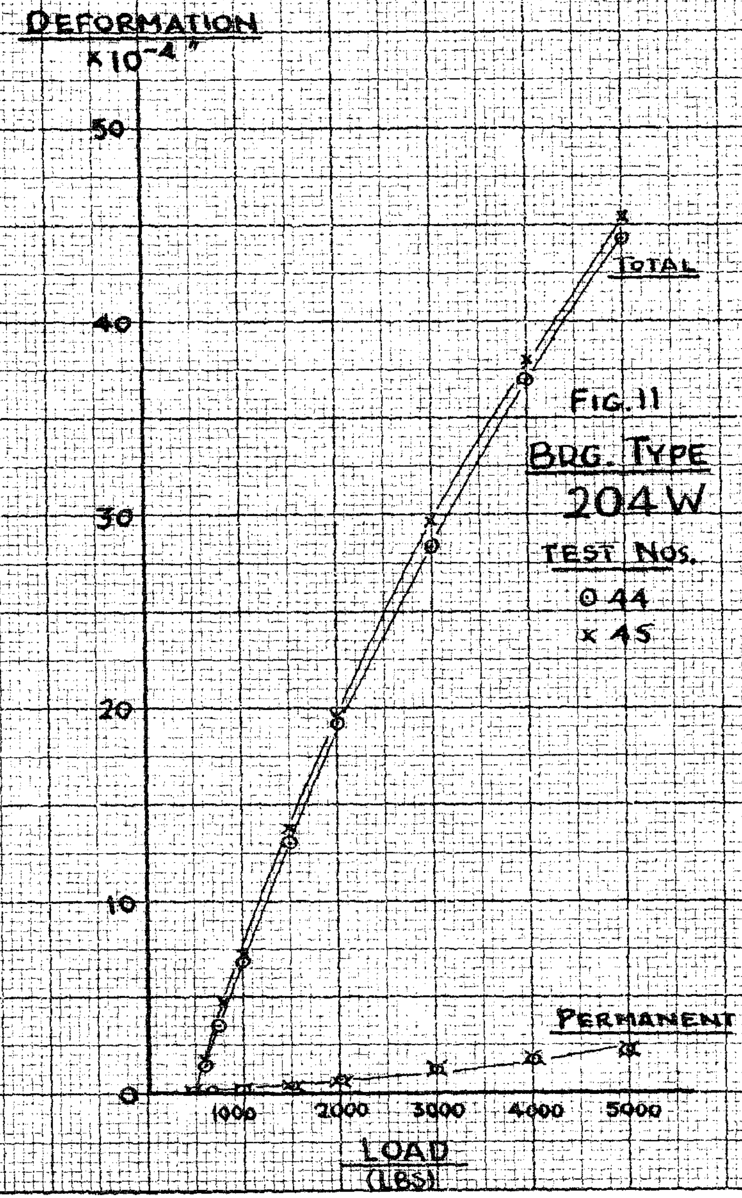


FIGURE 8
MEASURING DEVICE CHECK.
BRG. TYPE 201K
TEST BRG #16
o 16.1 — a
x 16.2 — b
+ 16.3 — c
DEFORMATION
x10^-4"
80
70
60
50
40
30
20
10
0
1000
2000
3000
4000
LOAD
(LBS)
TOTAL
PERMANENT
(a)
(b)
(c)


DEFORMATION
× 10⁻⁴"
50
40
30
20
10
0
TOTAL (BRG)
FIGURE 7
MEASURING DEVICE CHECK
BRG. TYPE 203F #42
SOLID STEEL RING #1
TOTAL (RING)
PERMANENT (BRG)
PERMANENT (RING)
1000
2000
3000
4000
5000
LOAD (LBS)


DEFORMATION
× 10⁻⁴"
50
40
30
20
10
0
FIGURE 9
BRG. TYPE - 204 H
BRG. NOS.
⊙ 23
× 27
1000
2000
3000
4000
LOAD (POUNDS)





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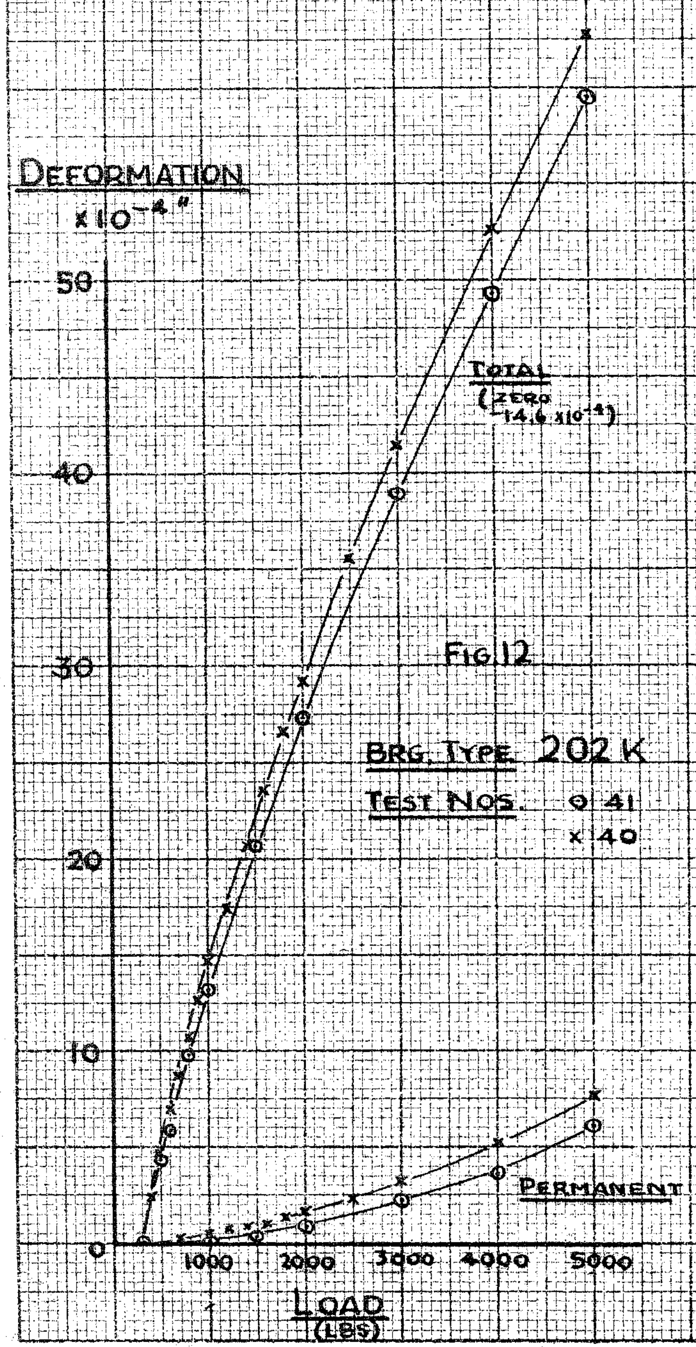
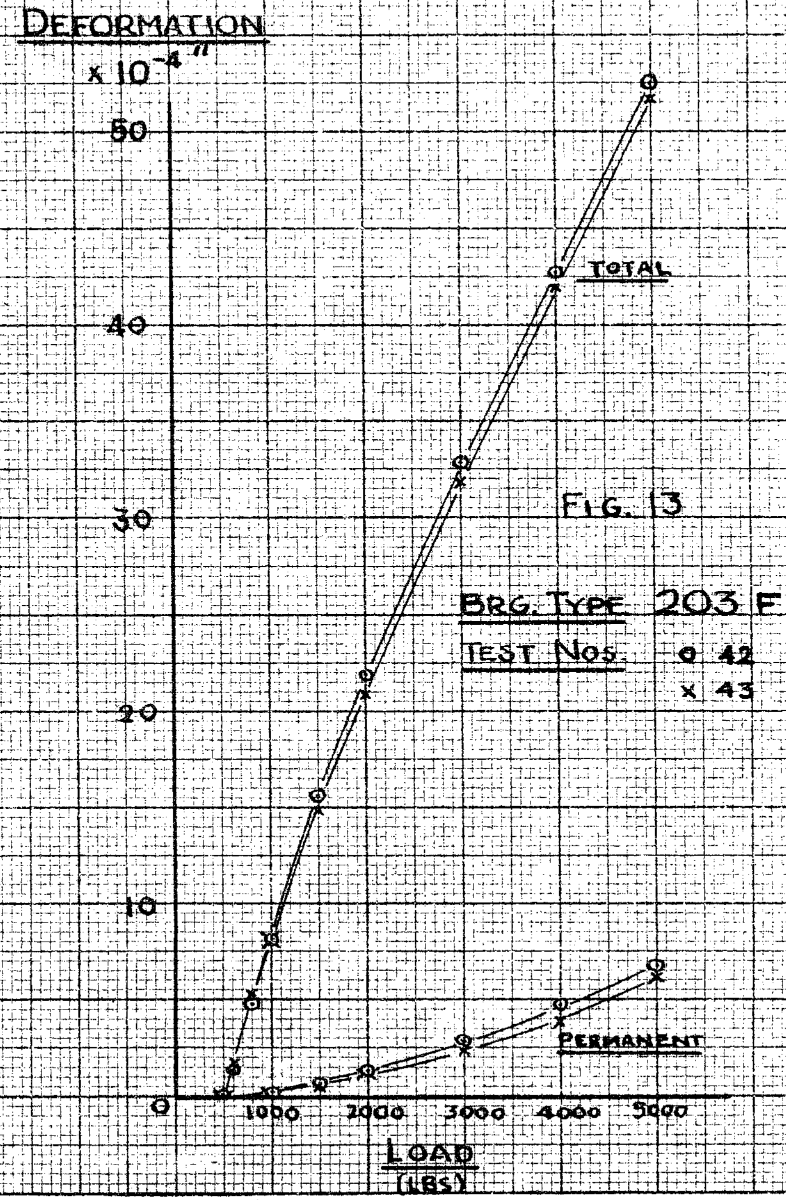
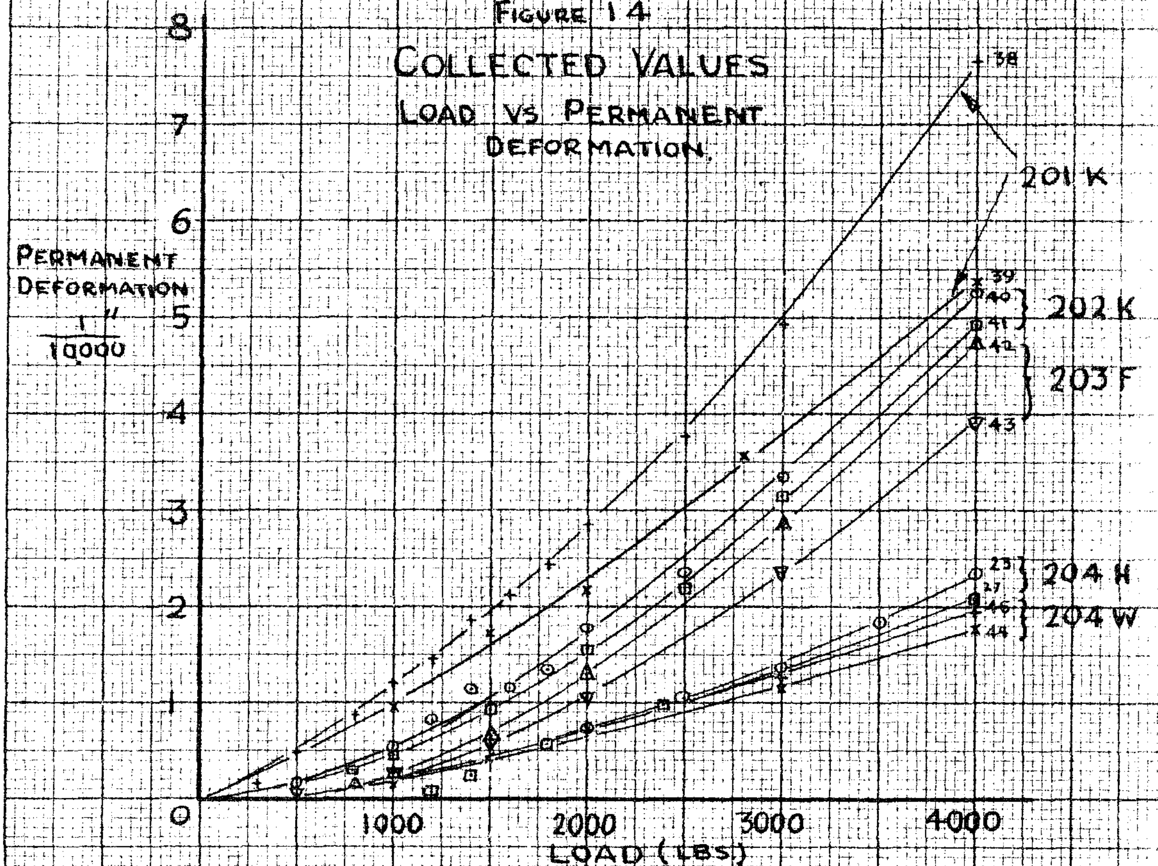

DEFORMATION
x 10⁻⁴"
50
40
30
20
10
0
TOTAL
(ZERO -14.6 x10⁻⁴)
FIG 12
BRG. TYPE 202 K
TEST NOS. ⊙ 41
x 40
PERMANENT
1000
2000
3000
4000
5000
LOAD
(LBS)


DEFORMATION
x 10⁻⁴"
50
40
30
20
10
0
TOTAL
FIG. 13
BRG. TYPE 203 F
TEST NOS. ⊙ 42
x 43
PERMANENT
1000
2000
3000
4000
5000
LOAD
(LBS)

5/11/37



(Permanent set in 1/10000 inch)

	201 K		202 K		203 F		204 W		204 H	
Load	#38	#39	#40	#41	#42	#43	#44	#45	#23	#27
300		.16								
500		.49	.16			.08				
800		.86		.30	.16			.16		
1000	.97	1.22	.57	.45	.28	.25	.16	.16	.08	
1200		1.46	.81							.08
1400		1.87	1.14							.24
1500	1.71			.91	.65	.59	.41	.41	.65	
1600		2.11	1.14							
1800		2.44	1.37							.57
2000	2.19	2.84	1.79	1.57	1.30	1.06	.73	.65	.73	.73
2400										98
2500		3.74	2.36	2.20					1.06	
2800	3.58									
3000		4.95	3.33	3.15	2.84	2.36	1.14	1.30	1.38	1.38
3500									1.87	
4000	5.36	7.64	5.29	4.91	4.71	3.90	1.79	1.95	2.35	2.03

TABLE II

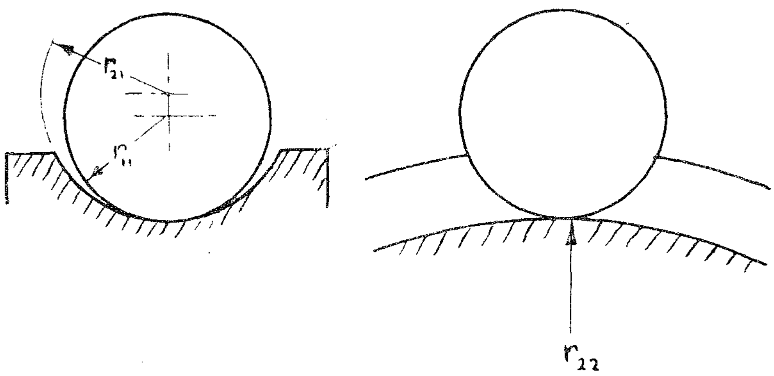
Test Data – Permanent Deformation vs Load

TABLE I

Brg. Type	Ball Size	r_{11}	r_{21}	r_{22}	m	nd^2	Q	n/Q^3
201K	1/4	.1250	.1275	.3080	.351	.437	6.64×10^4	23.9×10^{-15}
202K	1/4	.1250	.1275	.3671	.368	.437	6.42	26.4
203F	5/16	.1562	.1593	.4245	.450	.682	5.60	40.0
204H	11/32	.1718	.1753	.4925	.503	.826	5.21	49.3
204W	11/32	.1718	.1787	.4925	.496	1.18	5.93	48.1

TABLE III

Brg. Type	Load	K_1 P/nd^2	K_2 $\frac{P}{n/Q^3}$	$K_1 nd^2$	$K_2 \frac{n}{Q^3}$	P_o $5P/n$	Stress
201K	925	2110	387×10^{15}	1170	1170	660	579,000
202K	1500	3430	568	1170	1290	1070	665,000
203F	1850	2710	462	1830	1960	1320	645,000
204H	2450	2970	497	2215	2410	1750	628,000
204W	2575	2180	535	3160	2360	1288	674,000
Total		13400	2449×10^{15}				
Average		2680	489×10^{15}				



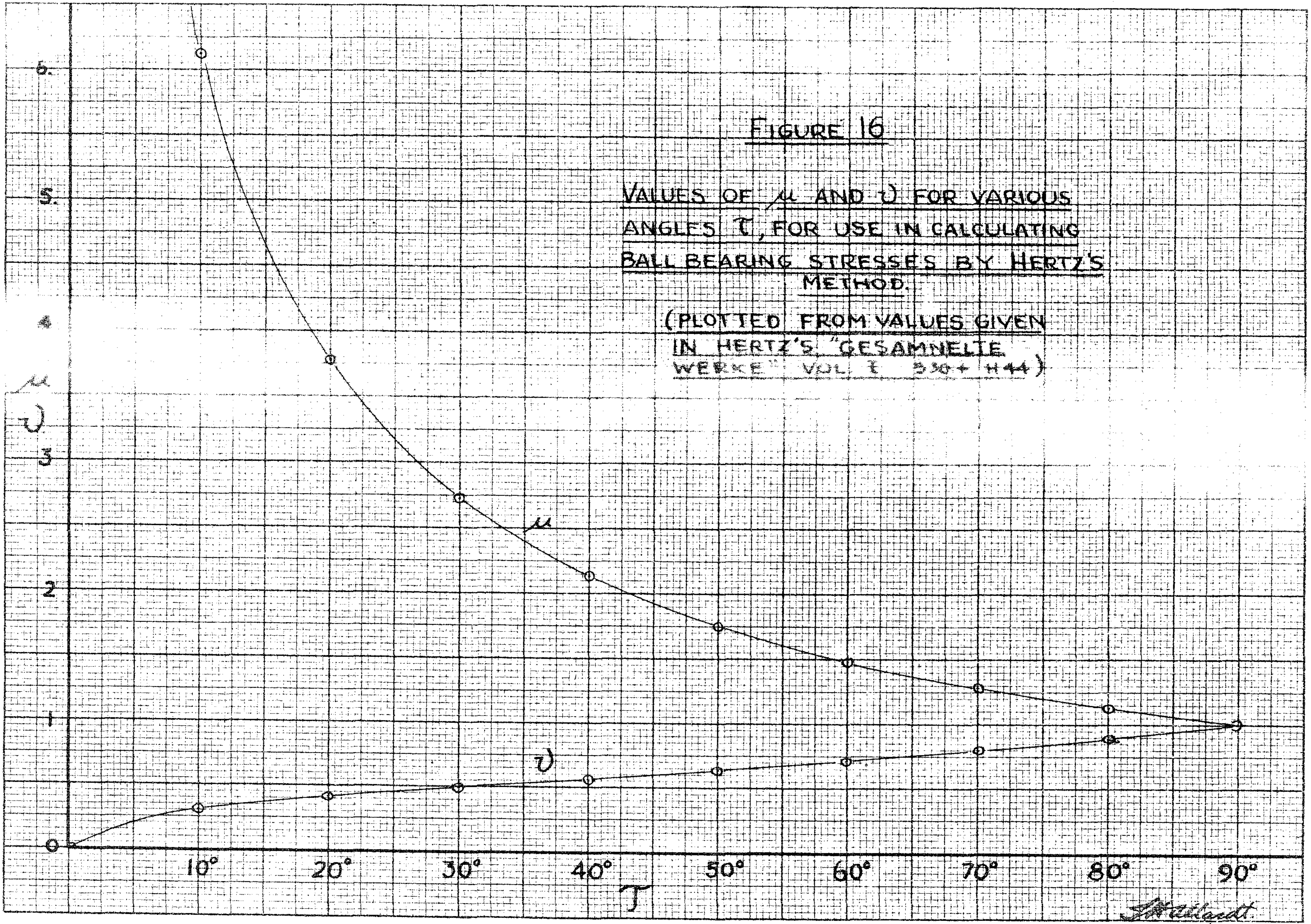
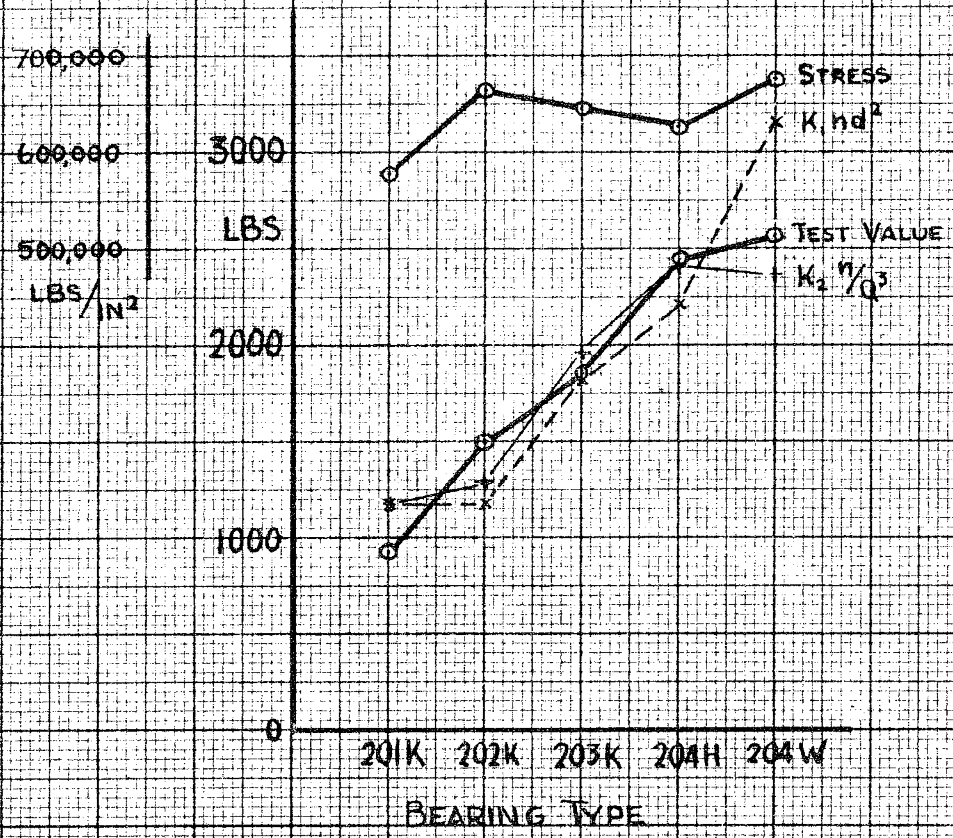


FIGURE 16
VALUES OF μ AND ν FOR VARIOUS ANGLES τ, FOR USE IN CALCULATING BALL BEARING STRESSES BY HERTZ'S METHOD.
(PLOTTED FROM VALUES GIVEN IN HERTZ'S "GESAMNELTE WERKE" VOL I 530 + H44)
μ
ν
0
1
2
3
4
5.
6.
10°
20°
30°
40°
50°
60°
70°
80°
90°
τ
5/20/37



700,000
600,000
500,000
LBS/IN²
3000
LBS
2000
1000
0
STRESS
K₁ nd²
TEST VALUE
K₂ n/Q³
201K
202K
203K
204H
204W
BEARING TYPE

FIGURE 17