COMPONENT HARDNESS DIFFERENCES AND THEIR EFFECT ON BEARING FATIGUE

by E. V. Zaretsky, R. J. Parker, and W. J. Anderson

Lewis Research Center
Cleveland, Ohio

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Lewis Research Center
National Aeronautics and Space Administration
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ABSTRACT 27083

The five-ball fatigue tester and full-scale rolling-element bearings were used to determine the effect of component hardness differences of SAE 52100 steel on bearing fatigue and load capacity. Maximum fatigue life and load capacity are achieved when the rolling elements of a bearing are one to two points (Rockwell C) harder than the races. There appears to be an interrelation among compressive residual stresses induced in the races during operation, differences in component hardness, and fatigue life. Differences in contact temperature and plastically deformed profile radii could not account for differences in fatigue life.

author

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NOMENCLATURE

a	deformation and wear area from surface trace, sq. in. $(a = D + W)$
C	load capacity, the load at which 90 percent of a group of bearings can endure 1,000,000 inner race revolutions, or for ball specimens, 1,000,000 stress cycles, lb $(C = P\sqrt[3]{L})$
D	deformation area from surface trace, sq in.
f_i	inner race conformity, percent
f_o	outer race conformity, percent
h	depth of running track from surface trace, in.
ΔH	hardness of rolling element minus race hardness for full-scale bearings or for five-ball test system, the hardness of the lower test balls minus the hardness of the upper test ball, Rockwell C
L	10-percent fatigue life, millions of revolutions or stress cycles
P	bearing or test system radial or thrust load, lb
P_n	normal ball load, lb
R	radius of ball, in.
R_p	effective radius of ball profile after plastic deformation and wear, in.
S_{max}	maximum Hertz stress, psi
S_{ry}	residual stress along axis of rolling, psi
W	wear area from surface trace, sq in.
β	contact angle, deg
$(\tau_{max})_r$	maximum shearing stress with residual stresses present, psi

INTRODUCTION

Much research has been directed toward increasing the fatigue life of ball and roller bearings and gears and, hence, their reliability. These efforts have led to increased operational reliability in engine and other aerospace equipment and components [1]. Fatigue testing has been conducted with bench-type component testers as well as with full-scale bearings and gears. A general conclusion drawn from much of these data is that material hardness plays an important role in determining rolling-contact fatigue life.

Several investigators using bench-type component testers [2 to 4] and full-scale bearings [5] have reported that rolling-element fatigue life increased with increasing hardness for several common bearing steels. Since deformation and wear tests indicate that resistance to permanent plastic deformation increases with increasing hardness, it was thus concluded that a qualitative correlation exists between fatigue life and resistance to plastic deformation [3 and 4].

Where plastic deformation does occur under rolling contact, the Hertz stress as calculated may be approximate only [6 and 7]. Material hardness or resistance to plastic deformation thus has a twofold effect on fatigue life; as hardness is decreased, fatigue life decreases because of an inherent decrease in material strength, but at the same time resistance to plastic deformation, and, thus, the contact stress decrease. This latter effect would increase fatigue life. Therefore, the two effects are acting in opposition to each other.

It has been shown that residual compressive stresses are developed below rolling-contact surfaces, the magnitude of which appeared to be a function of time [8 and 9]. Additionally, it is speculated that these residual stresses may be a function of material hardness. Residual compressive stresses induced by mechanical processing operations were found to increase the fatigue life of balls and complete bearings [10]. Thus, an additional variable that can be related to rolling-element fatigue is induced residual stress due to bearing operation.

The objectives of the research described in this paper, which is based on the work reported initially in references [11 and 12] were: (1) to determine if a maximum bearing fatigue life does exist at some optimum component hardness combination, (2) to determine if a relation exists among plastic deformation, relative hardness of bearing components, and fatigue life, and (3) to determine if residual stresses induced in the subsurface zone of resolved maximum shearing stress correlate with component hardness combinations and fatigue life. All experimental results were obtained with the same heat of material and lubricant batch except where indicated.

APPARATUS

The five-ball fatigue tester used in this investigation is shown schematically in Figs. 1(a) and (b) and was previously described in reference [3]. Essentially this fatigue apparatus consists of a 1/2-inch-diameter test ball pyramided upon four 1/2-inch-diameter lower test balls that are positioned by a separator and are free to rotate in an angular contact raceway (see Fig. 1(b)).

The upper test ball is analogous in operation to the inner race of a ball bearing, while the lower test balls and the angular contact raceway are analogous to the balls and the outer race of a ball bearing, respectively. For every revolution of the drive shaft, the upper test specimen receives 3 stress cycles. Instrumentation provides for automatic failure detection and shutdown. Lubrication is provided by means of a once through mist lubrication system.

The five-ball tester was modified in order to measure the surface temperature near the contact area of a modified upper test ball specimen during operation. Fig. 1(c) illustrates the test specimen and the mounting assembly, which is inserted into the drive spindle of the five-ball fatigue tester (see Fig. 1(a)). The specimen has a thermocouple attached, the tip of which is at one edge of the specimen running track. An axial hole was drilled through the drive spindle to insert the thermocouple wire. The thermocouple EMF was taken out through a slipring-brush assembly mounted at the top end of the drive spindle.

SPECIMENS AND PROCEDURE

SAE 52100 1/2-inch-diameter ball specimens were tested in the five-ball fatigue tester. The ball specimens were divided into 11 lots. Nine of the eleven lots were from the same heat of material according to the manufacturer. The two lots from the separate heat of material had a Rockwell C hardness greater than 66. A range of hardness was obtained for the remaining nine lots (table 1, lots A to I), which originally had a Rockwell C hardness of approximately 66, by varying the tempering temperature and the tempering time for each lot. A schedule of the heat treatment used for each of these lots is shown in table 1.

The lots were divided into five groups having nominal Rockwell C hardnesses of 60, 62, 63, 65, and 66. Balls from each hardness group were used as lower test balls with three lots of upper test balls having average Rockwell C hardnesses of 60.5, 63.2, and 65.2.

Retained austenite and prior austenitic grain size are also given in table 1. Since only the tempering temperature was varied between lots, cleanliness and prior austenitic grain size were held relatively constant.

Plastic deformation and wear data were obtained for upper test balls of the five nominal hardnesses run on lower test balls having average Rockwell C hardnesses of 60.5, 63.2, and 65.2. For each upper ball hardness, eight tests were run, each for 30,000 stress cycles. Six profile traces of each upper ball running track were made in a contour tracer at different locations around the ball perpendicular to the running track.

Fatigue and deformation and wear tests were conducted at a maximum Hertz stress of 800,000 psi, a drive shaft speed of 10,000 rpm, and a contact angle of 30° (indicated by β in Fig. 1(b)) with a highly purified naphthenic mineral oil. The race temperature in the fatigue tests stabilized at 150° to 165° F with no heat added. The five-ball system was considered failed when a fatigue spall developed on either the upper or lower test ball specimens. Because of the low stress developed on the outer race-ball contact, no failure occurred on the outer race.

Residual stress measurements were made on five upper test ball specimens having a Rockwell C hardness of 63.2 run against lower test balls of Rockwell C hardness 60 to 66 under the aforedescribed test conditions. The results were compared with fatigue lives obtained with the same hardness combinations.

Fatigue tests were also conducted with 207-size radial ball bearings. The dimensions of the bearings are as follows:

Inner race diameter, in.	1.6648
Outer race diameter, in.	2.5411
Ball diameter, in.	0.4375
Inner race conformity, f_i, percent	51
Outer race conformity, f_o, percent	52
Specification	ABEC 5

The inner and outer races (all from the same heat) were tempered to a nominal hardness of Rockwell C 63. The balls were divided into four groups and tempered according to the tempering schedules given in table 1 to produce nominal Rockwell C hardnesses of 60, 63, 65, and 66. Four lots of bearings were assembled, each containing balls of a specific hardness. These bearings were run at a radial load of 1320 pounds (producing maximum Hertz stresses of 352,000 and 336,000 psi at the inner and outer races, respectively), and a speed of 2750 rpm, with the highly purified napthenic mineral oil lubricant and no heat added.

RESULTS AND DISCUSSION

Fatigue Life Results

For applications where high reliability is of paramount importance, early failure of bearings is of primary interest. Hence, the significant life on a Weibull plot is the 10-percent life. The fatigue data was analyzed according to the statistical methods of reference [13]. The 10-percent lives are tabulated in table 2. The load capacity C, where $C = P\sqrt[3]{L}$, is also summarized in table 2 for each hardness combination tested, and is plotted in Fig. 2 as a function of ΔH, the hardness of the lower test balls minus that of the upper test ball.

System fatigue life and load capacity are found to be maximum where the lower test ball hardness was one to two points (Rockwell C) greater than that of the upper test ball for varying hardnesses of both components. These results indicate that a maximum bearing life can be achieved if the rolling elements of a bearing are one to two points (Rockwell C) harder than the races.

Failure Location

From probability theory and reference [14], it was determined that, for upper and lower test balls of equal fatigue strength, the probability of a failure occurring in either one or the other is approximately equal. It was shown in references [3] and [4] that the life of a given rolling element increased with increased hardness of that element. Consequently, it would be expected that where the hardness of a component was increased, the probability of failure occurring in the component would decrease. Therefore, for the series of tests reported herein it was expected that, as the hardness of the lower balls was increased with a given upper ball hardness, there would be a greater probability of the upper test ball failing. Table 2 tabulates the number of upper ball failures, lower ball failures, and the number of tests where both upper and lower test ball failures occurred. The failure index indicates the number of fatigue failures relative to the number of tests in each series. Where there was an upper and lower ball failure, it was assumed from the nature of the testing and the method of detecting a failure that the lower ball failed prior to the upper ball. The percent of total failures occurring in the upper ball is also given in table 2 and is plotted against ΔH in Fig. 3. As was expected, increased ΔH resulted in more upper test ball failures. Where the upper and lower test balls were of the same hardness (i.e., $\Delta H = 0$), approximately half the failures in each series occurred in the upper test ball.

Examination of table 2 and Fig. 2 shows that the data obtained with the upper test balls having an average Rockwell C hardness of 63.2 are generally representative of all other data obtained. From the Rockwell C 63.2 upper test ball data, the 10-percent lives of the upper and lower test balls were determined. These data are plotted separately as a function of ΔH together with the system 10-percent lives in Fig. 4. It can be seen from this figure that the upper test ball appears to control the trend in system life.

Deformation and Wear

Deformation and wear and thus contact stress can be affected by material hardness [3 and 15]. The alteration produced on the rolling-contact surfaces takes three basic forms: (a) elastic deformation, (b) plastic deformation, and (c) wear. The latter two forms result in permanent alteration of the ball surface contour that can be measured after testing. Figure 5 is a schematic diagram of the transverse section of a ball surface showing this permanent alteration.

Deformation and wear data were obtained for upper test balls having nominal hardnesses of 60, 62, 63, 65, and 66 run against lower test balls having average Rockwell C hardnesses of 60.5, 63.2, and 65.2, under the conditions previously described. Average values for the deformation and wear areas are given in table 3. By the use of trigonometric relations, an effective ball radius R_p at the point of contact can be calculated [16] in terms of a, h, and R

$$R_p = \frac{(a/h)^2}{2[R - \sqrt{R^2 - (a/h)^2} - h]}$$

On the basis of this equation for R_p, an effective maximum Hertz stress for 30,000 stress cycles of operation can be calculated for each component hardness combination. (Based upon previous experience, the value R_p obtained after 30,000 stress cycles approximates the value that would be obtained after an indefinite number of stress cycles). In order to account for the plastic deformation that may or may not be accumulative in the lower test balls (because of their unknown degree of randomness of rotation), three calculations were made for each hardness combination. Deformation of a lower test ball was assumed to be one of the following: (a) none, (b) equal to that of the upper ball, or (c) equal to the value obtained with the reverse hardness combination. Contact stresses recalculated on the basis of the assumptions are given in table 3. Based on these recalculated stresses, theoretical relative 10-percent lives were calculated based on the relationship $L \propto 1/S_{max}^9$. These values are also presented in table 3 and are compared with the relative experimental 10-percent lives. It is apparent from these values that the differences in effective Hertz stress with varying component hardness combinations cannot account for the actual differences in fatigue life.

Contact Temperature

A possible cause of differences in fatigue life with different hardness combinations may be the contact temperature induced by sliding with the contact zone [6]. Temperature gradients in the contact zone can induce thermal stresses and alter the calculated maximum shearing stress. Temperature measurements were taken at the edge of the running track of a series of upper test balls having a Rockwell C hardness of 63.2 run against five series of lower test balls having average hardnesses from 60.5 to 66.4. Table 4 contains a tabulation of these data. These temperatures are a better approximation of the actual temperature in the contact zone of a ball specimen than temperatures measured at the outer diameter of the race. It will be noted from these data that there are no significant differences in the measured near contact temperature of the hardness combinations measured. These results tend to indicate that thermal stresses due to temperature gradients in the contact zone of two rolling bodies of different hardnesses cannot account for the difference in fatigue life discussed herein.

Residual Stress

It was shown in references [17] and [9] that residual compressive stresses are developed in bodies in rolling-contact; the magnitude of which appeared to be a function of time. Additionally, residual compressive stresses induced by mechanical processing operations were found to increase the fatigue life of balls and complete bearings [10]. A unit volume on the upper test ball running track in the five-ball system is stressed many more times than a point on any of the lower test balls, so that it would be likely that the upper test ball would absorb more energy and build up a proportionally greater amount of subsurface residual stress than would each of the lower test balls.

Induced residual stress can either increase or decrease the maximum shearing stress [12], according to the following equation:

$$(\tau_{max})_r = -3.22\times10^6\left(\frac{P_n}{R^2 S_{max}}\right)^{1/2} - \frac{1}{2}\,(\pm S_{ry})$$

where the positive or negative sign of S_{ry} indicates a tensile or a compressive residual stress, respectively. A compressive residual stress would reduce the maximum shearing stress and increase fatigue life [12] according to the following equation:

$$L \propto \left(\frac{1}{(\tau_{max})_r}\right)^9$$

Five upper test balls having a Rockwell C hardness of 63.2 that were run for approximately the same number of stress cycles against lower balls having Rockwell C hardnesses of 59.7, 61.8, 63.4, 65.0 and 66.2 were selected for residual stress measurements. Standard x-ray diffraction techniques (private communication from R. Lindgren and W. E. Littmann of the Timken Roller Bearing Company, Canton, Ohio) were used to measure residual stresses at a depth of 0.005 inch beneath the running track.

The residual stresses below the track (table 5) were found to be compressive and varied between 178,000 and 294,000 psi. Their values were considerably higher than those anticipated (based on the data from [17 and 9]). Even so, it should be noted that these measured stresses are less than the true values because the x-ray beam could not be focused entirely in the stressed zone.

Background residual stresses (measured outside the stressed zone) were zero in samples 2, 3, and 4 and 59,000 and 20,000 psi compressive in specimens 1 and 5. These background stresses would have a tendency to increase the value of the measured residual stresses in the stressed zone. Consequently the values for specimens 1 and 5 might be high relative to the other samples.

The measured compressive residual stresses are plotted as a function of ΔH in Fig. 6. From this figure, it is noted that the measured stress increases with increasing lower test ball hardness to an intermediate hardness where a peak was obtained. On the basis of these limited data, the apparent maximum residual stress occurs at a value of ΔH slightly greater than zero.

The measured values of compressive residual stress were used to calculate theoretical 10-percent lives of the upper test ball using the aforementioned relationships. These calculated 10-percent lives, shown in table 5, predict a peak life at the maximum compressive residual stress which occurs at a ΔH slightly greater than zero. Although these results which are based on limited residual stress measurements, do not show the predicted peak life at a ΔH of one to two points (Rockwell C hardness) such as was experimentally determined, it is apparent that an interrelation exists among differences in component hardness, induced compressive residual stress, and fatigue life.

Full-Scale Bearing Tests

In order to illustrate the effect of ΔH on bearing fatigue and load capacity, four lots of SAE 52100 207-size deep groove ball bearings, each with balls of a specific hardness and races of Rockwell C hardness 63, were fatigue tested at a radial load of 1320 pounds, a speed of 2750 rpm with the mineral lubricant and no heat added. The results of these tests are shown in Fig. 7 and tabulated in table 6. The relative bearing load capacity is shown as a function of ΔH in Fig. 8. For comparison purposes, the range of predicted relative capacity based on the five-ball fatigue data is also presented. As with the five-ball system, it can be concluded that maximum bearing fatigue life and load capacity can be achieved where the rolling elements of the bearing are one to two points (Rockwell C hardness) greater than the races.

SUMMARY

System fatigue lives were determined in five-ball fatigue tester with components having various hardness combinations. Upper test balls of Rockwell C hardnesses of 60.5, 63.2, and 65.2 were run against lower test balls of nominal Rockwell C hardnesses 60, 62, 63, 65, and 66. These tests were run with no heat added at an initial maximum Hertz stress of 800,000 psi, 10,000 rpm, and a 30° contact angle using a highly purified napthenic mineral oil lubricant. Residual stress measurements were made on upper test ball specimens of Rockwell C hardness 63.2 run against lower test balls of varying hardnesses. Plastic deformation and wear, and near contact temperatures of five-ball tester specimens were also studied. Four lots of 207-size deep groove ball bearings each with balls of a specific hardness were fatigue tested at a radial load of 1320 pounds, a

speed of 2750 rpm with the mineral oil lubricant and no heat added. The following results were obtained:

1. Bearing fatigue life and load capacity were found to be maximum where the rolling elements of the bearing are one to two points (Rockwell C hardness) greater than the races for varying hardnesses of both components.

2. An interrelation is indicated among differences in component hardness, induced compressive residual stress, and fatigue life. The apparent maximum residual stress occurs where the rolling elements are of slightly greater hardness than the race.

3. Differences in plastic deformation and wear for different hardness combinations could not account for measured differences in fatigue life.

4. The measured near contact temperatures based on data obtained with five hardness combinations were not significantly different indicating that any thermal effect on fatigue life could not account for differences in life.

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TABLE 1. - SAE 52100 MATERIAL PROPERTIES AND HEAT TREATMENT

Designation Group	Lot	Average Rockwell C hardness	Retained austenite, percent by volume[a]	Heat treatment[b] First temper in oil	Second temper in oil
I	A	59.7	>2	60 minutes at 250° F	60 minutes at 450° F
	B	60.5	8.9		
II	C	61.8	12.8	60 minutes at 250° F	60 minutes at 350° F
	D	61.9	12.3		
III	E	63.2	12.5	60 minutes at 250° F	60 minutes at 320° F
	F	63.2	12.8		
	G	63.4	15.6		
IV	H	65.0	18.4	60 minutes at 250° F	90 minutes at 250° F
	I	65.2	—		
V	J	66.2	11.8	60 minutes at 250° F	None
	K	66.4	13.3		

[a] All groups had prior austenitic grain size (ASTM) of 12.

[b] All groups austenitized for 30 minutes at 1550° F to 1600° F and all quenched to 125° F prior to tempering.

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TABLE 2. - SYSTEM FATIGUE LIFE AND LOAD CAPACITY OBTAINED WITH VARYING

HARDNESS COMBINATIONS IN FIVE-BALL FATIGUE TESTER

[Initial maximum Hertz stress, 800,000 psi; system thrust load, 390 lb; contact angle, 30°; room temperature; material, SAE 52100 steel.]

Upper test ball Rockwell C hardness (and designation[a])	Lower test ball Rockwell C hardness (and designation[a])	Difference in Rockwell C hardness between lower and upper test balls, ΔH	10-Percent fatigue life, millions of stress cycles	Weibull slope	System load capacity based on experimental life, C^b, lb (c)	Confidence number[d] percent	Failure index (number of failures out of number of specimens tested)	Number of upper test ball failures	Number of lower test ball failures	Number of upper and lower test ball failures	Number of upper test ball failures (e)
60.5 (I-B)	60.5 (I-B)	0	6.2	0.869	624	88	20 out of 21	11	7	2	55
	61.9 (II-D)	1.4	17.3	1.428	876	--	22 out of 22	12	6	4	55
	63.2 (III-E)	2.7	8.1	.839	682	77	18 out of 22	12	4	2	67
	65.2 (IV-I)	4.7	9.7	.839	725	74	20 out of 20	16	1	3	80
	66.4 (V-K)	5.9	10.0	1.036	730	72	19 out of 20	13	3	3	69
63.2 (III-F)	59.7 (I-A)	-3.5	1.2	1.036	362	99	20 out of 21	0	20	0	0
	61.8 (II-C)	-1.4	4.3	1.111	553	92	22 Out of 22	6	13	3	27
	63.4 (III-G)	.2	7.7	1.111	670	76	22 out of 23	13	3	6	59
	65.0 (IV-H)	1.8	14.2	1.327	830	--	22 out of 22	12	4	6	55
	66.2 (V-J)	3.0	3.7	.839	525	91	22 out of 23	18	2	2	82
65.2 (IV-I)	60.5 (I-B)	-4.7	1.7	0.810	405	98	17 out of 21	5	12	0	29
	61.9 (II-D)	-3.3	3.5	1.235	513	98	17 out of 18	1	14	2	6
	63.2 (III-E)	-2.0	5.1	1.072	584	90	18 out of 18	6	5	7	33
	65.2 (IV-I)	0	11.6	1.150	770	64	18 out of 18	9	2	7	50
	66.4 (V-K)	1.2	17.2	1.279	876	--	20 out of 21	9	3	8	45

[a]See table I.

[b]$C = P\sqrt[3]{L}$ where P is load on test system and L is 10-percent life of system.

[c]See Fig. 2.

[d]Percentage of time that 10-percent life obtained with each hardness combination will have same relation to hardness combination in that series exhibiting highest 10-percent life.

[e]See Fig. 3.

TABLE 3. - DEFORMATION AND WEAR AND THEIR EFFECT ON MAXIMUM HERTZ STRESS FOR VARYING HARDNESS COMBINATIONS IN FIVE-BALL FATIGUE TESTER

[Initial maximum Hertz stress, 800,000 psi; contact angle, 30°; 30,000 stress cycles; room temperature.]

Upper test ball Rockwell C hardness (and designation[a])	Lower test ball Rockwell C hardness (and designation[a])	Upper test ball deformation area from surface trace, sq in.	Upper test ball wear area from surface trace, sq in.	Calculated radius of upper test ball, in.	Effective maximum Hertz stress, psi			Relative theoretical 10-percent life			Relative experimental 10-percent life
					(b)	(c)	(d)	(b)	(c)	(d)	
60.5 (I-B)	60.5 (I-B)	1.02×10^{-6}	0.47×10^{-6}	0.483	7.29×10^{5}	6.48×10^{5}	6.48×10^{5}	1.22	1.69	1.39	0.36
61.8 (II-C)		.53	.15	.398	7.45	6.87	[e]6.72	1.00	1.00	1.00	1.00
63.2 (III-E)		.42	.06	.311	7.71	7.43	6.90	.73	.50	.79	.47
65.2 (IV-I)		.27	0	.276	7.85	7.71	7.01	.62	.35	.69	.56
66.4 (V-K)		.28	0	.274	7.86	7.74	[e]7.05	.62	.37	.65	.58
60.5 (I-B)	63.2 (III-E)	1.22×10^{-6}	0.25×10^{-6}	0.553	7.19×10^{5}	6.23×10^{5}	6.90×10^{5}	2.08	6.20	1.87	0.08
61.8 (II-C)		.70	.01	.430	7.39	6.71	[e]7.13	1.62	3.18	1.38	.30
63.2 (III-E)		.45	.01	.323	7.67	7.33	7.33	1.16	1.36	1.08	.54
65.2 (IV-I)		.34	0	.286	7.80	7.63	7.39	1.00	1.00	1.00	1.00
66.4 (V-K)		.26	0	.280	7.83	7.68	[e]7.40	.96	.94	1.00	.27
60.5 (I-B)	65.2 (IV-I)	1.22×10^{-6}	0.28×10^{-6}	0.576	7.16×10^{5}	6.15×10^{5}	7.01×10^{5}	2.25	7.30	2.34	0.10
61.8 (II-C)		.74	.02	.458	7.33	6.58	[e]7.20	1.81	3.90	1.82	.20
63.2 (III-E)		.52	0	.356	7.56	7.11	7.39	1.32	1.97	1.45	.30
65.2 (IV-I)		.29	0	.290	7.79	7.59	7.59	1.05	1.10	1.14	.68
66.4 (V-K)		.23	0	.281	7.83	7.67	[e]7.70	1.00	1.00	1.00	1.00

[a]See table I.

[b]No deformation and wear of lower test ball assumed.

[c]Deformation and wear of lower test ball assumed equal to that of upper test ball.

[d]Deformation and wear of lower test ball assumed equal to that value obtained with reverse hardness combination.

[e]Estimated

TABLE 4. - TEMPERATURE AT EDGE OF CONTACT ZONE FOR MODIFIED FIVE-BALL FATIGUE TESTER WITH 1/2-INCH-DIAMETER SAE 52100 STEEL BALLS

[Initial maximum Hertz stress, 800,000 psi; shaft speed, 10,000 rpm; contact angle, 30°.]

Upper test ball Rockwell C hardness (and designation[a])	Lower test ball Rockwell C hardness (and designation[a])	Temperature (no heat added), °F	
		Race	Contact zone of upper test ball
63.2 (III-E)	60.5 (I-B)	135	184
	61.8 (II-C)	132	184
	63.2 (III-E)	130	187
	65.0 (IV-H)	130	186
	66.2 (V-J)	125	180

[a]See table I.

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TABLE 5. - RESIDUAL STRESS MEASUREMENTS OF UPPER TEST BALL SPECIMENS

HAVING ROCKWELL C HARDNESS OF 63.2

Specimen number	Lower test ball Rockwell C hardness	Difference in Rockwell C hardness between lower and upper test balls. ΔH	Specimen running time, millions of stress cycles	Measured residual stress at depth of 0.005 inch below ball surface removed by electropolishing, psi		Calculated 10-percent life of upper test ball based on residual stresses,[a] millions of stress cycles
				Under track	Outside track	
1	59.7	-3.5	36.1	[b]-178×10^3	[b]-59×10^3	0.391
2	61.8	-1.4	32.4	-198	0	.641
3	63.4	.2	37.9	-294	0	11.3
4	65.0	1.8	40.0	-223	0	1.278
5	66.2	3.0	39.1	-257	-20	3.23

[a]Based on experimental 10-percent life of 11.3×10^6 stress cycles at $\Delta H = 0.2$.

[b]Negative sign denotes compressive residual stress.

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TABLE 6. - BEARING FATIGUE LIFE AND LOAD CAPACITY WITH VARYING HARDNESS;

RADIAL LOAD, 1320 LB; SPEED, 2750 RPM; RACE ROCKWELL C HARDNESS, 63

Ball Rockwell C hardness	Difference Rockwell C between ball and races, ΔH	10-percent fatigue life, millions of inner race revolutions	Bearing radial capacity based on experimental life, C lb	Ratio of C to C of ΔH = 2	Confidence number, percent	Failure index (number of failures out of number of bearings tested)
60	-3	21	3640	0.58	89	14 out of 28
63	0	77	5620	.90	60	11 out of 25
65	2	106	6250	1.00	--	12 out of 28
66	3	74	5540	89	62	14 out of 27

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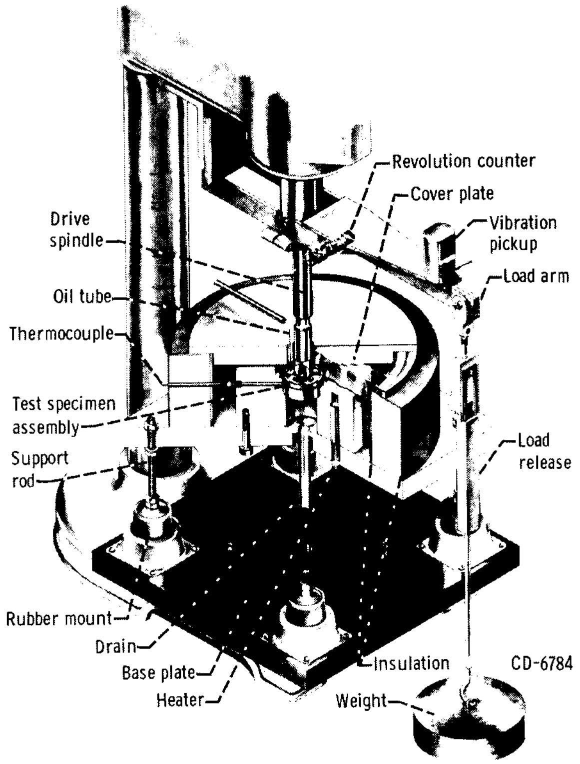
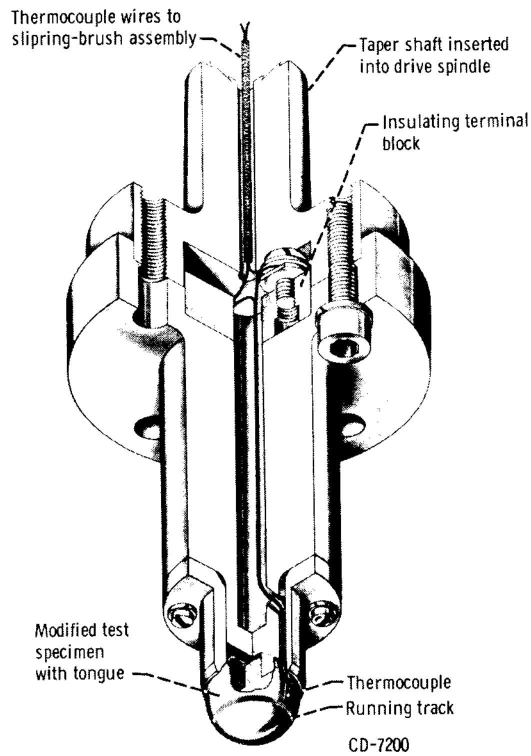
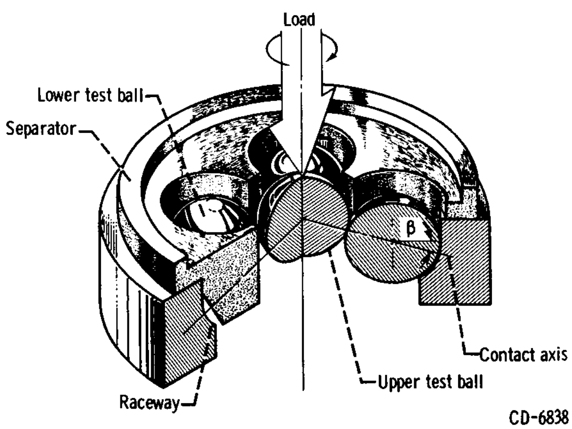


(a) Cutaway view of five-ball fatigue tester.



(c) Operating-temperature measuring device.



(b) Schematic of five-ball tester.

Fig. 1. - Test apparatus.

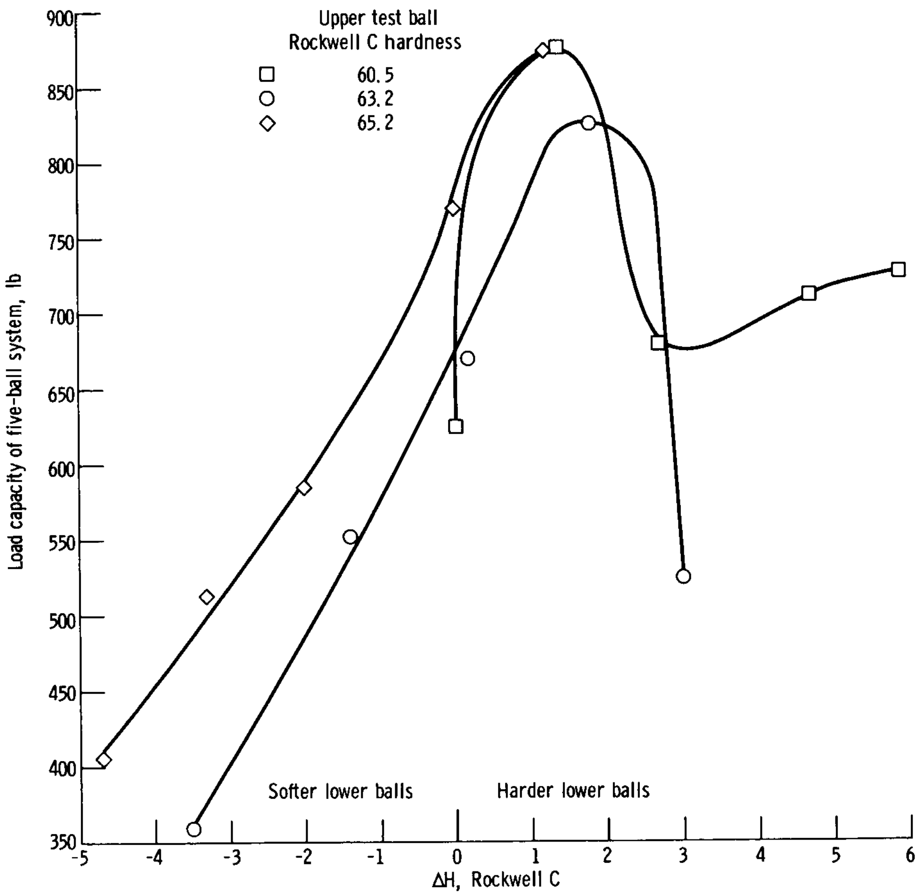


Fig. 2. - Load capacity of five-ball system as a function of difference in hardness between lower test balls and upper test ball.

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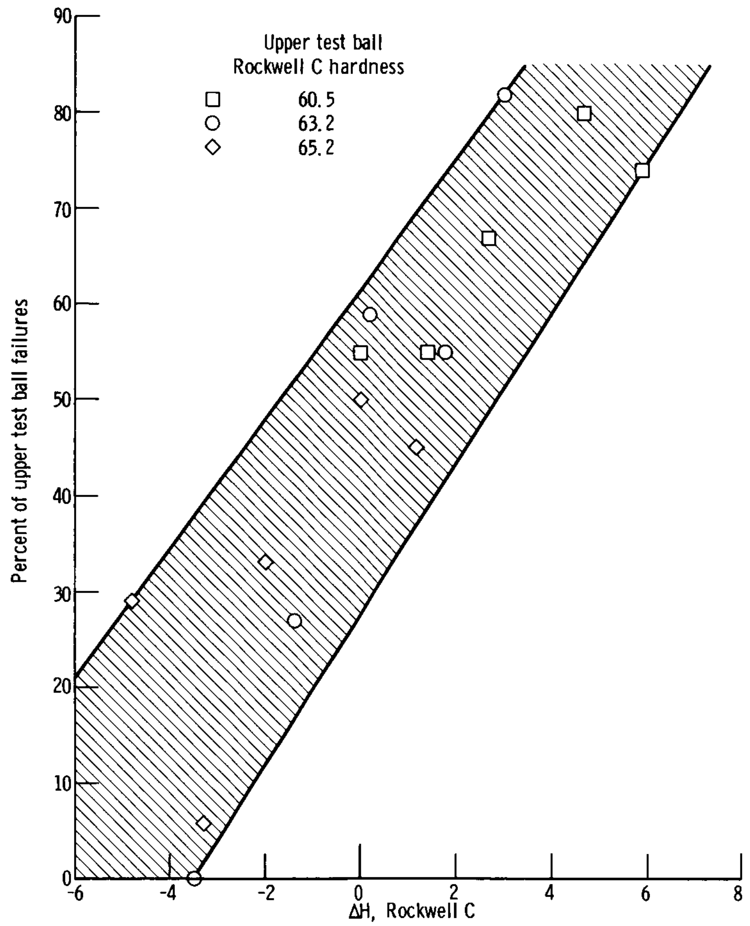

Fig. 3. - Percent of upper test ball failures for test group as a function of difference in hardness between lower test balls and upper test ball.

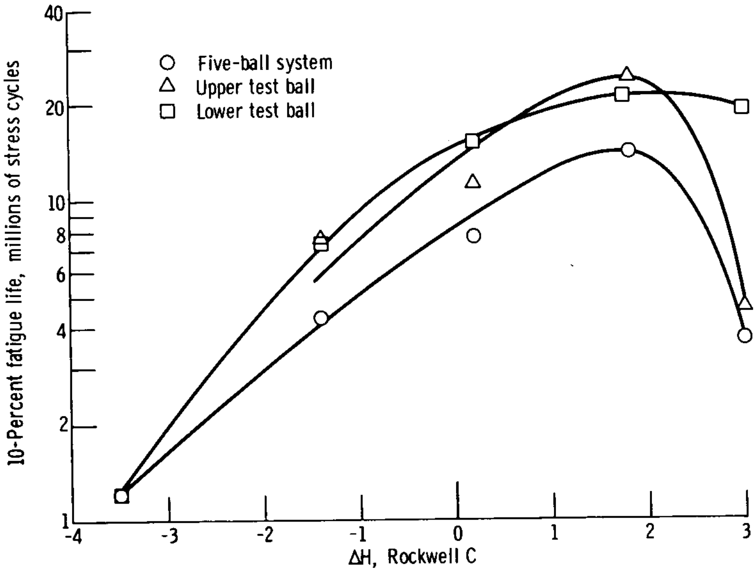

Fig. 4. - 10-Percent life of five-ball system and components as a function of difference in hardness between lower test balls and upper test ball having Rockwell C hardness of 63.2.

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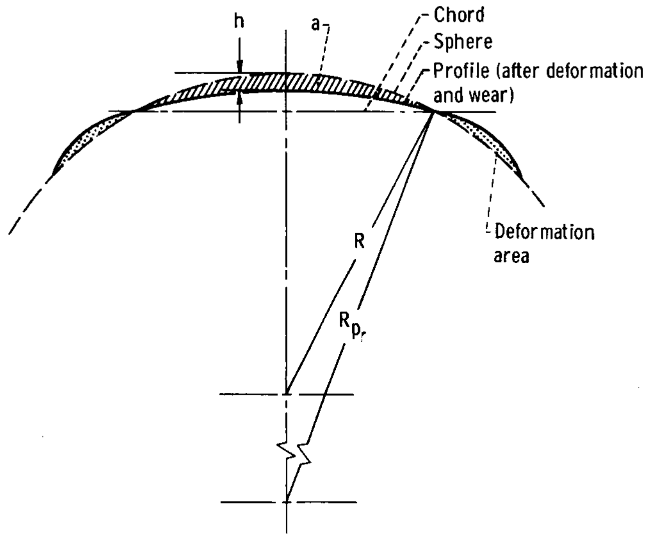

Fig. 5. - Surface profile after deformation and wear of cross section of stressed ball track. (Not to scale.)

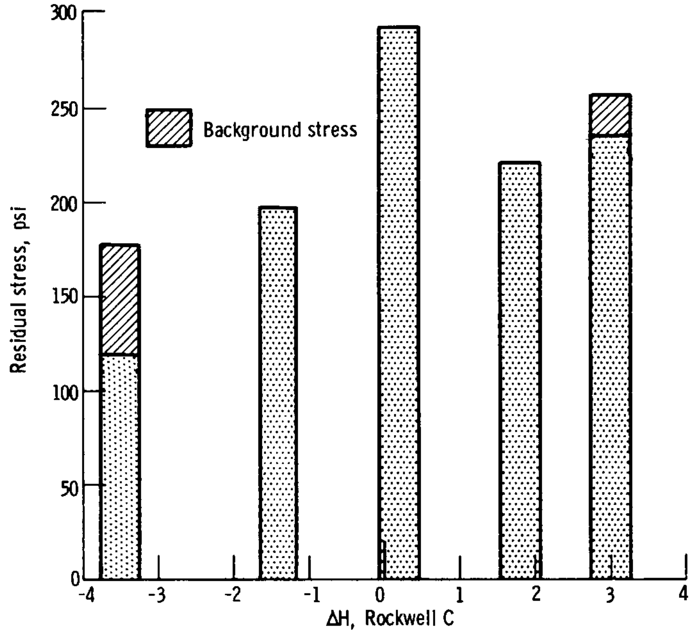

Fig. 6. - Measured compressive residual stress in track of upper test balls having Rockwell C hardness of 63.2 as a function of difference in hardness between lower test balls and upper test ball.

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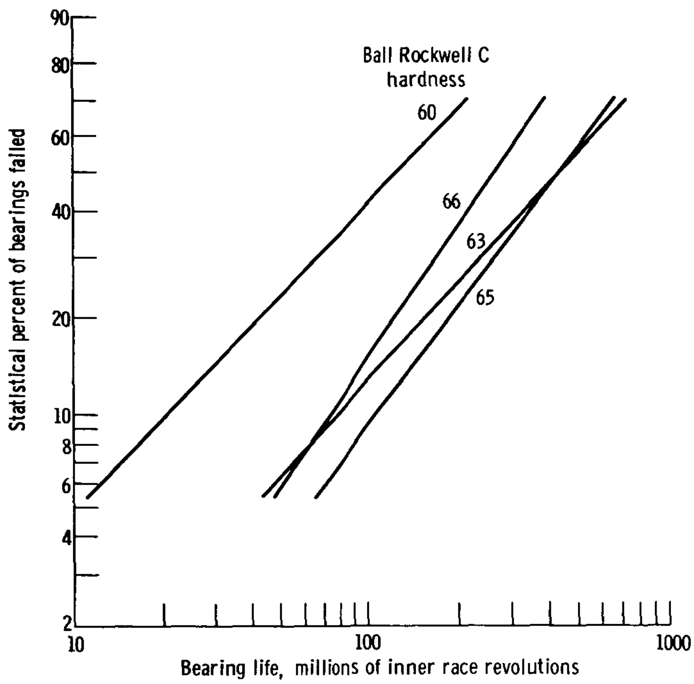

Fig. 7. - Rolling-contact fatigue life of 207-size deep groove ball bearings having SAE 52100 races of Rockwell C hardness 63 with SAE 52100 balls of varying hardness. Radial load, 1320 pounds; speed, 2750 rpm; no heat added.

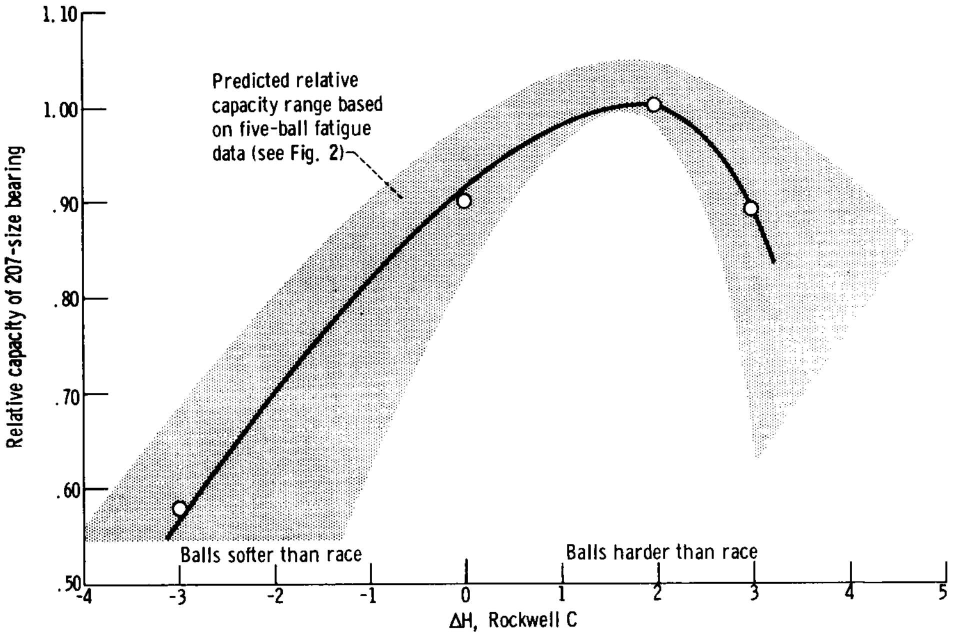

Fig. 8. - Relative radial load-carrying capacity of 207-size deep groove ball bearing as a function of difference in hardness between balls and races.