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EVALUATION OF LOAD-LIFE RELATION WITH BALL BEARINGS AT 500° F

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ABSTRACT

A survey of the literature suggests that a stress-life exponent of approximately 12 is typical of vacuum-processed steels for ball bearings rather than the exponent of 9 which has been generally accepted by the bearing industry and bearing users. Tests run with vacuum-degassed AISI 52100 balls in the five-ball fatigue tester at four maximum Hertz stress levels in the range from 650 000 to 875 000 psi showed good agreement with the literature. However, tests run with consumable-electrode vacuum melted AISI M-50 steel angular-contact ball bearings at 500° F at three thrust loads did not show significant deviation from the accepted ninth power stress-life relation.

INTRODUCTION

The generally accepted relation between load (or stress) and life in a rolling-element bearing results from early work of Lundberg and Palmgren [1 and 2][3]. It was determined that life varied with the inverse cubic power of load for point contact (ball bearings).

The inverse-cubic load-life relation has been included in the AFBMA Standards for ball bearings [3] and has been generally accepted by ball bearing manufacturers and users. Since stress is proportional to the cube root of load for ball bearings, the load-life relation can be

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[3]Numbers in brackets designate references at end of paper.

resolved into a stress-life relation where life L varies as the inverse ninth power of stress ($L \propto (1/\text{stress})^9$).

The value of the stress-life exponent has a significant effect on predicting life of ball bearings and the choice of a bearing for a particular application. Although data from bench type rolling-element fatigue testers have proven to be a useful substitute and have shown good correlation with full-scale bearing data, it is desirable to test full-scale bearings for some of the more severe applications. The mainshaft bearings of advanced airbreathing turbojet engines promise to be one of these severe applications. Increased power requirements in these engines have dictated that these bearings operate at higher temperatures, higher speeds, and higher loads. In current turbine engines, mainshaft bearings operate at bearing temperatures up to 425° F and maximum speeds approaching 2.2 million DN. (DN is the bearing bore in millimeters multiplied by shaft speed in rpm.) It is anticipated that more advanced engine designs may require that bearings operate at temperatures between 500° and 550° F and speeds exceeding 2.2 million DN. Projection of these trends to the 1990's would predict bearing temperatures to 600° F and bearing speeds of 3 to 4 million DN producing higher bearing operating stresses.

Fatigue tests at temperatures from 400° to 600° F with a 120-millimeter bore angular-contact ball bearings at 1.4 million DN and using a synthetic paraffinic oil have produced fatigue lives in excess of 13 times AFBMA predicted (catlog) life [4 and 5]. One conclusion from this work was that there was little difference among the bearing lives at 400°, 500°, and 600° F. Since the lubrication in these tests was predominatly in the elastohydrodynamic (EHD) regime, the failure mode was classical subsurface fatigue. Therefore, under adequate EHD lubricating conditions the effects of temperature on fatigue life in the range of 400° to 600° F are statistically insignificant.

The effect of load (or stress) on fatigue life at these more severe conditions must likewise be studied to determine if the stress-life relation for more nominal conditions is applicable.

There is some question whether the accepted inverse ninth power stress-life relation applies to the cleaner, double-vacuum-melted materials currently being used for the more advanced applications. Data in [6 to 9] indicate a stress-life exponent in the range of 11 to 13.

The objectives of the research reported herein were to (1) determine the stress-life relation for vacuum-degassed AISI 52100 balls in the five-ball fatigue tester, (2) clarify the discrepancies among published values of stress-life exponents, and (3) determine the stress-life relation for consumable-electrode vacuum melted AISI M-50 ball bearings at 500° F.

Tests were conducted with one-half-inch diameter balls in the five-ball fatigue tester at maximum Hertz stresses in the range of 650 000 to 875 000 psi and were initially reported in [10].

Fatigue tests at 500° F with consumable-electrode vacuum melted AISI M-50 angular-contact ball bearings (120-mm bore) were conducted by General Electric Company under NASA contract NAS 3-11148.

BACKGROUND

Fatigue tests with ball bearings [11 to 13] tended to verify the inverse cubic load-life relation. Styri [11] in 1951 presented data with two types of ball bearings. One group of 6207 deep-groove ball bearings under various radial loads from 775 to 3880 pounds determined life to be inversely proportional to load to the 3.3 power. Another group of 1207 double-row, self-alining bearings were tested such that the maximum Hertz stress at the outer-race-ball contact varied from 580 000 to 810 000 psi. Here it was found that life varied inversely with the ninth power of stress (or the third power of load).

Cordiano and others [12] reported in 1956 that for 217-size thrust-loaded ball bearings, the load-life exponent was 2.7, 3.2, and 4.2 for three lubricants (a water-glycol base, a phosphate ester base, and a phosphate ester, respectively). These, with the exception of the phosphate-ester lubricant, tend to verify the inverse cubic relation. This exception is not clearly understood but is attributed, by the authors of [12] to the "superior" lubrication qualities of this lubricant at lighter loads.

In 1963, McKelvey and Moyer [13] reported that, with four groups of SAE 4620 carburized-steel crowned rollers (elliptical contact), fatigue life varied inversely with maximum Hertz stress to the eighth to ninth power. Maximum Hertz stress in these tests varied from 262 000 to 478 000 psi. Again, the inverse cubic relation was generally confirmed.

Several publications [14 to 18] report data with bench type rolling-element fatigue testers, rather than full-scale bearings, that show good agreement with the inverse cubic relation between load and life. It is more common in these tests to report the inverse ninth power relation between stress and life. These data are for AISI 52100 and AISI M-50 steels, and at least two sets of data [15] are with air-melted material, whereas the other references do not state the melting process.

More recent data [6 to 8] indicate a greater effect of stress on life than the inverse ninth power relation. Schatzberg [6] reports a stress-life exponent of 12 for vacuum-degassed AISI 52100 balls in a four-ball fatigue tester. Data presented in [7 and 8] and reanalyzed in [9] also indicate a stress-life exponent in the range of 11 to 13 for consumable-electrode vacuum-melted AISI 52100 balls also in a four-ball fatigue tester. The 10-percent lives as functions of maximum Hertz stress for several of these data are shown in figure 1. These data are summarized in table I.

Also shown in figure 1 and table I are data from [19] with AISI 52100 toroidal rollers. The stress-life exponents for these data are in the range of from 15 to 19. These exponents are much higher than those from any other published data. The lack of correlation of these data [19] with other data remains unexplained.

It is interesting to note that the data from the bench-type fatigue testers [6, 14 to 18] were obtained at maximum Hertz stresses from 526 000 to 1 300 000 psi and that the different exponents observed do not appear to be influenced by the stress range. The bearing life data of [11], which cover the maximum Hertz stress range from approximately 300 000 to 810 000 psi, tend to confirm the lack of varying exponent in different stress ranges. However, [7 and 8] propose that the stress-life exponent varies in the stress range from approximately 500 000 to 800 000 psi because of plastic deformation.

FIVE-BALL FATIGUE TESTER

The five-ball fatigue tester used in the current study is shown in figure 2 and has been described in detail in [20]. The test assembly consists of an upper test ball pyramided upon four lower test balls that are positioned by a separator and are free to rotate in an angular-contact raceway. System loading and drive are supplied through a vertical drive shaft, which grips the upper test ball. For every revolution of the drive shaft, the upper test ball received three stress cycles from the lower test balls.

Lubrication is provided by a once-through, mist lubrication system. In these tests, lubricant was paraffinic mineral oil without additives with a viscosity of 28.4 centistokes at 100° F. Vibration instrumentation detects a fatigue failure on either the upper or a lower test ball and automatically shuts down the tester. This provision allows unmonitored operation and a consistent criterion for failure.

All one-half-inch diameter balls used in the five-ball tests were from one heat of vacuum-degassed AISI 52100 and had a nominal Rockwell C hardness of 61 at room temperature.

HIGH-TEMPERATURE BEARING FATIGUE TESTER

The 120-millimeter bore angular contact ball bearings were fatigue tested in the high-temperature fatigue tester shown in figure 3. This tester is described in detail in [21]. Essentially the tester consists of a shaft to which two test bearings are attached. Loading is supplied through a system of 10 springs which thrust load the bearings. Drive of the test rig is accomplished by a flat belt on a crowned spindle (not shown in the illustration).

Lubrication is provided to the test bearings through a jet-feed lubrication system by a pump immersed in a heated oil reservoir. The reservoir has approximately a 3-gallon capacity. The pump is capable of circulating the oil through the system at 3 gallons per minute at 600° F. Gravity drainage for the lubricant is provided by a single exit under each test bearing as well as from the bellows in the center of the bearing assembly.

The lubricant used for these bearing fatigue tests was a synthetic paraffinic oil with an antiwear additive with a viscosity of 447 centistokes at 100° F and 3.6 centistokes at 500° F. Additional lubricant properties are given in table II. Because of the relatively poor oxidation resistance of the paraffinic oil, all tests were conducted in an inert atmosphere, with nitrogen gas used as the inerting medium.

Instrumentation provided for automatic shutoff by monitoring and recording bearing temperatures, oil temperature, bearing vibration, nitrogen flow rate and pressure, as well as support bearing lubricant flow and pressure. A change in any of these parameters from those programmed for the test conditions terminated the test. Oxygen content within the bearing housing assembly was monitored frequently during operation. An infrared pyrometer was used to measure inner-race temperature, using a sapphire window sight tube aimed at the inner race of the first test bearing (fig. 3).

TEST BEARINGS

The test bearings were ABEC-5 grade, split inner-race 120-millimeter bore angular-contact ball bearings having a nominal contact angle of 20 degrees. The inner and outer races were manufactured from one heat of consumable-electrode vacuum-melted (CVM) AISI M-50 steel, and the balls were manufactured from a second heat. The nominal Rockwell C hardness of the balls and races was 63 at room temperature. Each bearing contained 15 balls of 13/16-in. diameter. The cage was a one-piece outer-land riding type made of a nickel-base alloy (AMS 4892) having a nominal Rockwell C hardness of 33. The retained austenite content of the ball and race material was measured at less than 3 percent. The inner- and outer-race curvatures were 54 and 52 percent, respectively. All components with the exception of the cage were matched within ± 0.5 point Rockwell C. This matching assured a nominal differential hardness in all bearings (i.e., the ball hardness minus the race hardness, commonly called ΔH) of zero. Chemical analysis and heat treatment of the bearing material are given in tables III and IV, respectively.

RESULTS AND DISCUSSION

Five-Ball Fatigue Results

Four groups of one-half inch diameter balls were tested in the five-ball fatigue tester at four levels of maximum Hertz stress ranging from 650 000 to 875 000 psi. Test conditions included a contact angle of 30° and a shaft speed of 10 000 rpm with a paraffinic mineral oil as the lubricant.

The results of the fatigue tests are shown in the Weibull plots of figure 4. Both upper- and lower-ball failures were considered in determining the five-ball system life in the Weibull analysis. Life of the five-ball system decreased with increasing stress as is seen in figure 4 and table V.

Also shown in table V are the 90-percent confidence limits on the 10-percent lives as calculated by methods of [22]. The interpretation of these limits is that the true 10-percent life at each condition will fall between these limits 90 percent of the time. It is seen that for adjacent stress levels the confidence limits overlap, which would indicate that the life differences are not statistically significant.

For example, in table V the experimental estimate of the 10-percent life of 439 million stress cycles at 650 000 psi falls within the 90-percent confidence limits of 40 to 560 million stress cycles for the 725 000 psi condition. However, the limits do not overlap for nonadjacent stress levels (e.g., 650 000 and 800 000 psi conditions). This observation coupled with the consistent trend of decreasing life with increasing stress indicates good statistical significance in the data.

The 10- and 50-percent lives at each stress condition are plotted in figure 5 against maximum Hertz stress. Straight lines fitted to these data by the method of least squares indicate that for these data, $L \propto (1/S_{max})^n$ where n equals 12 for both the 10- and 50-percent life levels. These results correlate well with the data in [6 to 8], which show values of n to be 11.5 and 12.4.

Effect of Ball Temperature

In the present five-ball tests the temperature of the test ball assembly was not controlled, but temperatures were measured with thermocouples at the outer-race and at the center of the upper ball. These data are shown in table VI. Temperatures at both locations increased with increasing stress, and ball temperature was consistently higher than race temperature, as expected.

Since lubricant viscosity decreases with increasing temperature, the lubricant in the upper-ball-lower ball contacts would have the lowest viscosity at the highest stress (see Table VI). Lubricant viscosity affects fatigue life according to the following relation $L \propto \nu^m$ where $m = 0.2$ or 0.3 from [23 and 24, respectively], and ν is the kinematic viscosity of the lubricant. The experimental fatigue results should reflect this effect, which would tend to decrease life at the higher stresses. The 10-percent lives are adjusted for this viscosity effect by the above relation with m taken as 0.25. As shown in table VI, the effect is small and can be considered negligible. With the viscosity effect considered, the stress-life exponent is only reduced to 11.5.

Effect of Vacuum Processing

The most recent published data [6 to 8] have been attained with vacuum-processed AISI 52100. At least two sets of earlier data [15] that show the exponent to be 9 are with air-melted materials. The other references [11 to 14] and [16 to 18] do not state whether the materials were air-melted or vacuum-processed. If it is assumed that these were air-melted materials, which is probable (because of the earlier dates), it then becomes apparent that cleaner, vacuum-processed materials may be a significant factor in raising the stress-life exponent to approximately 12.

In examining the literature and the conditions of the various tests, no factor other than the effect of vacuum processing is apparent that could explain the differences in the stress-life exponents. The published work covered a variety of lubricants, a wide load and stress range, and both thrust and radial loads with no consistent trends apparent.

The greater sensitivity to stress with vacuum-processed materials is not surprising if one considers the effect of vacuum processing on steel cleanliness. Vacuum-processed steels have fewer nonmetallic inclusions than air-melted steels. A hard, nonmetallic inclusion acts as a stress concentrator or stress raiser. The effect on fatigue life of increasing Hertz stress on a steel containing many inclusions may be overshadowed by the stress raising effect of the inclusions. In a cleaner vacuum-processed steel, the effect of increasing Hertz stress would become more apparent. The stress-life exponent would then be higher for a vacuum-processed steel than for an air-melted steel.

Evidence of similar effects of stress raisers on fatigue life of rotating-beam specimens is seen in [25 to 27]. In [25], rotating-beam specimens of a chrome-nickel steel show a greater effect of stress on life as inclusion count is decreased. Similarly, in [26 and 27], a greater effect of stress on life is seen with unnotched steel rotating-beam specimens than with specimens with a sharp notch. In these examples, the inclusions and the sharp notches are stress concentrators. These references also provide some assurance that the higher stress life exponent (12) for the vacuum-processed steels in the study reported herein can be attributed to the lack of inclusions and their lack of a stress raising effect.

Bearing Fatigue Tests

High reliability is a key requirement for advanced aircraft application. Current advanced aircraft engines dictate operating temperatures of approximately 425° F. With increased speed, however, and particularly in the Mach 3+ regime, the heat rejection capacity of the lubricant cooling system becomes marginal because of the increase in ram air temperature, the limits on temperature rise of the fuel, and the greater demands for cooling capacity elsewhere. As a result, bearing temperatures are expected to approach, if not exceed, 500° F. It is conceivable that at these temperatures the stress-life exponent may be different from the normally accepted value of 9 or 10 and the twelfth power relation indicated by the data reported herein. A proper stress-life exponent value then becomes more than of mere academic interest inasmuch as an engine designer requires a reliable analytic tool to predict bearing life and performance.

Fatigue tests were performed with 120-millimeter bore angular-contact ball bearings made of consumable-electrode vacuum melted (CVM) AISI M-50 steel at thrust loads of 4600, 5800, and 7310 pounds. The test conditions included a speed of 12 000 rpm (1.44×10^{6} DN) and an outer-race temperature of 500° F in a high-temperature fatigue tester. The maximum Hertz stresses shown in table VII include effects of approximately 200 pounds per ball centrifugal force.

The test results are given in figure 6 and are summarized in table VII. The failure index indicates the number of failed bearings out of those tested. The predicted life for the three thrust load conditions are shown in figure 6(d) and given in table VII. The predicted results were calculated using the method of Harris [28] which utilizes a stress-life exponent of 9. A material factor obtained by the method of Chevalier, et al [29] was used.

In most fatigue test experience, it can reasonably be expected that experimental lives will generally fall within plus or minus 50 percent of the predicted life value if all parameters such as material processing and elastohydrodynamic (EHD) effects are factored into the life equations. It is expected that the elastohydrodynamic (lubricant) life factor is somewhat lower than that calculated by methods of [28] which uses conventional EHD theory. Data of [30 to 32] indicate that, under the stress conditions of these bearing tests, the film thickness prediction of conventional EHD theory is very likely optimistic. Therefore, the EHD lubricant life factor is probably lower than that calculated by [28]. The resultant predicted lives may be slightly lower than shown in figure 6(d). However, current state of the art does not allow a more accurate prediction.

A statistical analysis of the experimental data utilizing the methods of Johnson [22], indicates that there are no statistical differences at the 10-percent life level among the fatigue lives obtained with the three loads. For the 4600 and 5800 pound life tests, the experimental results are more than half the predicted value. However, the data appear, within reason, to approximate a ninth-power stress-life relation.

The results for the 7310 pound test present an anomally. At the 10-percent life level, the life is over twice that which was predicted while at the 50-percent life level, the life is less than the predicted value. These fatigue results at the 7310 pound load comprise two sets of test data run under the same conditions with the same lubricant and bearing components from the same heats of material. Results from the two sets of data were nearly identical and are grouped together in figure 6(c). The high slope of the life distribution would suggest a high load for the lubrication condition present. However, this would not explain the apparent increase in life with increased load at the 10-percent life level.

The data for the three load conditions does not show significant deviations from the ninth power stress-life relation. These bearing life data do not support the results of the bench-type fatigue tester data showing a twelfth power stress-life relation. As a result, it cannot be recommended that this exponent be changed in the life equations until more full-scale bearing fatigue data run at varying load conditions are obtained at more moderate temperature conditions.

SUMMARY

One-half-inch diameter balls were tested in the five-ball fatigue tester at maximum Hertz stresses in the range of 650 000 to 875 000 psi. The ball material was one heat of vacuum degassed AISI 52100. The outer-race temperature was in the range of 120° to 162° F. Angular-contact ball bearings were fatigue tested at 4600, 5800, and 7310 pounds thrust load

at an outer-race temperature of 500° F. These 120-millimeter bore bearings were made from CVM AISI M-50. The following results were obtained:

1. With the vacuum-degassed AISI 52100 balls in the five-ball fatigue tester, rolling-element fatigue life decreased with increased maximum Hertz stress according to the relation $L \propto (1/S_{max})^{12}$. A survey of the literature suggests that a stress-life exponent of approximately 12 is more typical of vacuum-processed materials.

2. The 120-millimeter bore bearings fatigue tested at 500° F did not show significant deviation from the ninth power stress-life relation.

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TABLE I. - SUMMARY OF PUBLISHED BENCH-TYPE RIG ROLLING-ELEMENT FATIGUE DATA RELATED TO STRESS-LIFE EFFECTS

Reference	Test type	Publication date	Material: Type	Material: Melting process	Lubricant	Load range, kg	Maximum Hertz stress range, psi	Load-life exponent	Stress-life exponent
14	Four-ball	1956	EN-31 (AISI 52100)	(a)	Mineral oil	200 to 600	[b]6.6×10^5 to 9.5×10^5	2.8	[c]8.4
15	Spin-rig	1957	AISI 52100	Air melt	SAE 10 mineral oil	----------	6.0×10^5 to 7.5×10^5	---	10.4
15	Spin-rig	1957	MV-1 (AISI M-50)	Air melt	SAE 10 mineral oil	----------	6.0×10^5 to 7.5×10^5	---	9.7
16	R-C rig	1958	MV-1 (AISI M-50)	(a)	MIL-L-7808	----------	6.4×10^5 to 7.77×10^5	---	9.7
17	Four-ball	1958	EN-31 (AISI 52100)	(a)	Diester	400 to 600	[b]8.3×10^5 to 9.5×10^5	3.6	[c]10.8
18	Crowned-disk	1960	AISI 52100	(a)	#60 Spindle oil	----------	5.26×10^5 to 7.3×10^5	---	[d]8.5
19	Toroids	1962	AISI 52100	(a)	Navy symbol 2190 TEP	----------	8.4×10^5 to 8.9×10^5	---	[d]15 to 19
7	Four-ball	1965	AISI 52100	Consumable-electrode vacuum remelted	Mineral oil	----------	6.1×10^5 to 8.0×10^5	---	[e]12.4
6	Four-ball	1969	AISI 52100	Vacuum degassed	Squalene	----------	9.25×10^5 to 1.3×10^6	---	11.5
6	Four-ball	1969	AISI 52100	Vacuum degassed	Squalene + 100 ppm H_2O	----------	9.25×10^5 to 1.3×10^6	---	11.7

[a]Melting process not reported.
[b]Approximate stress range, not reported by authors of reference.
[c]Based on load-life exponent times 3.
[d]Estimated, not reported by authors of reference.
[e]Re-analyzed in [9].

TABLE II. - PROPERTIES OF SYNTHETIC PARAFFINIC OIL USED IN 500° F BEARING TESTS

Kinematic viscosity, cS, at	
100° F	447.0
210° F	40.4
400° F	6.0
500° F	3.6
Flash point, °F	520
Fire point, °F	595
Autoignition temperature, °F	760
Pour point, °F	-60
Volatility (6.5 hr at 500° F), weight percent	14.2
Specific heat at 500° F, Btu/(lb)(°F)	0.750
Thermal conductivity at 500° F, Btu/(hr)(ft)(°F)	0.06
Specific gravity at 500° F	0.714
Additives	Antiwear agent

TABLE III. - CHEMICAL ANALYSIS OF THE AISI M-50 BEARING MATERIAL

Element	Weight percent	
	Races	Balls
C	0.79	0.83
Mn	.21	.22
P	.11	.009
S	.003	.003
Si	.22	.22
Cr	3.89	3.99
Mo	4.29	4.23
V	1.00	1.00
Ni	.07	----
W	.04	----
Fe	Bal.	Bal.

TABLE IV. - HEAT TREATMENT CYCLE FOR AISI M-50 BEARINGS (DOES NOT INCLUDE STRESS RELIEF TREATMENTS DURING FINAL GRINDING)

Preheat	1550° F
Austenitize	$2025\pm15^{\circ}$ F
Quench	To 1000° F in 250° F oil; air cool to room temperature.
Tempering:	
First cycle	1025° F for 2 hr; air cool; -120° F for 3 hr.
Second cycle	1025° F for 2 hr; air cool; -120° F for 3 hr.
Third cycle	1025° F for 2 hr; air cool.

TABLE V. - FIVE-BALL FATIGUE TEST RESULTS

[1/2-in.-diam vacuum-degassed AISI 52100 balls; contact angle, 30°; shaft speed, 10 000 rpm.]

Maximum Hertz stress, psi	Life, millions of upper-ball stress cycles						Weibull slope	Failure index[a]
	L_{10}			L_{50}				
	Lower 90-percent confidence limit	L_{10} estimate	Upper 90-percent confidence limit	Lower 90-percent confidence limit	L_{50} estimate	Upper 90-percent confidence limit		
6.5×10^5	160	439	1200	1300	2040	3200	1.23	19 out of 77
7.25	40	150	560	900	1640	3000	.79	23 out of 98
8.0	9	34.8	140	120	214	400	1.04	15 out of 37
8.75	5	13.1	35	60	95.1	140	.95	33 out of 34

[a]Failure index indicates number of failures out of total number of tests.

TABLE VI. - LIFE ADJUSTMENT FOR DECREASED LUBRICANT VISCOSITY DUE TO HIGHER TEMPERATURE

AT HIGHER STRESSES

Maximum Hertz stress, psi	Outer-race temperature, °F	Ball temperature,[a] °F	Lubricant viscosity at ball temperature, cS	Ten-percent life, millions of upper-ball stress cycles	Ten-percent life adjusted for viscosity at 650 000 psi maximum Hertz stress
6.5×10^5	120	150	11.0	439	439
7.25	129	156	10.0	150	153
8.0	146	195	5.8	34.8	40.9
8.75	162	[b]210	4.9	13.9	16.4

[a]Measured with thermocouple at the center of the upper ball.
[b]Linear extrapolation from lower stress data.

TABLE VII. - FATIGUE LIFE RESULTS WITH 120-mm BORE ANGULAR-CONTACT BALL BEARINGS AT THREE THRUST LOADS

[Material, AISI M-50 steel; speed, 12 000 rpm; lubricant, synthetic paraffinic oil with antiwear additive.]

Load, lb	Maximum Hertz stress,[a] ksi		Life, millions of inner-race revolutions				Confidence number at L_{10} level
			Theoretical		Experimental		
	Inner race	Outer race	L_{10}	L_{50}	L_{10}	L_{50}	
4600	300	252	302	1510	166	502	--
5800	323	267	159	795	113	349	67
7310	347	284	83	415	194	342	78

[a]Includes effects of approximately 200 pounds per ball centrifugal force.

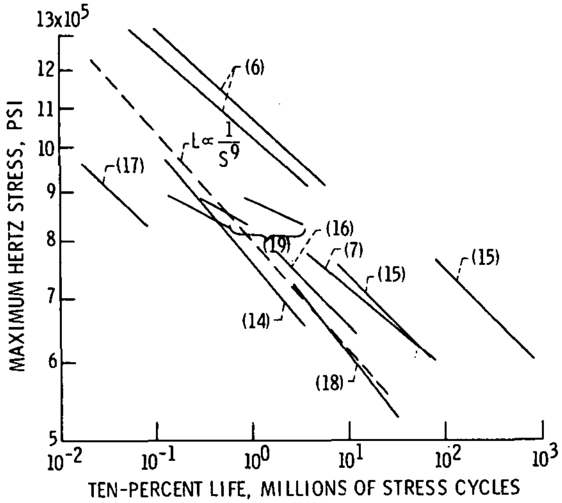
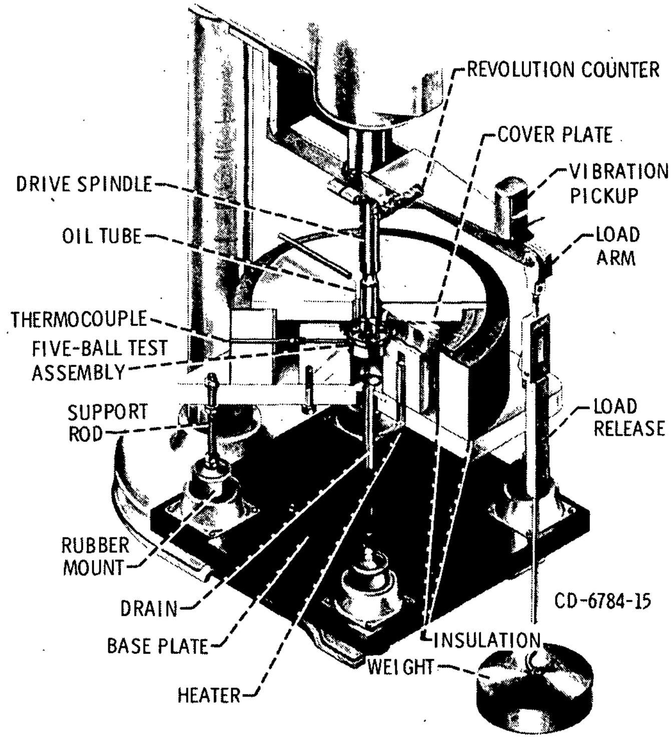
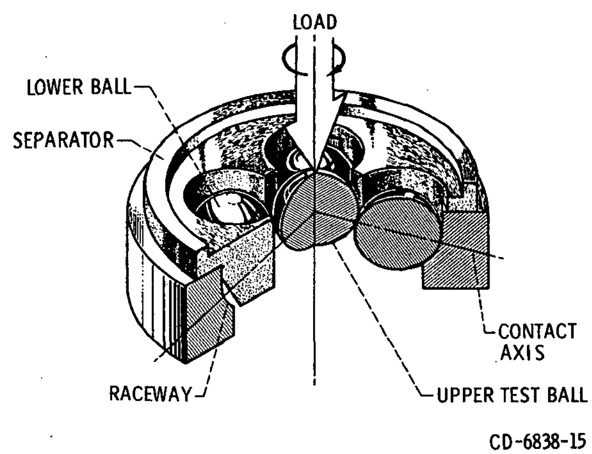


Figure 1. - Stress-life relation: summary of published data.



(a) CUTAWAY VIEW OF FIVE-BALL FATIGUE TESTER.



(b) FIVE-BALL TEST ASSEMBLY.

Figure 2. - Test apparatus.

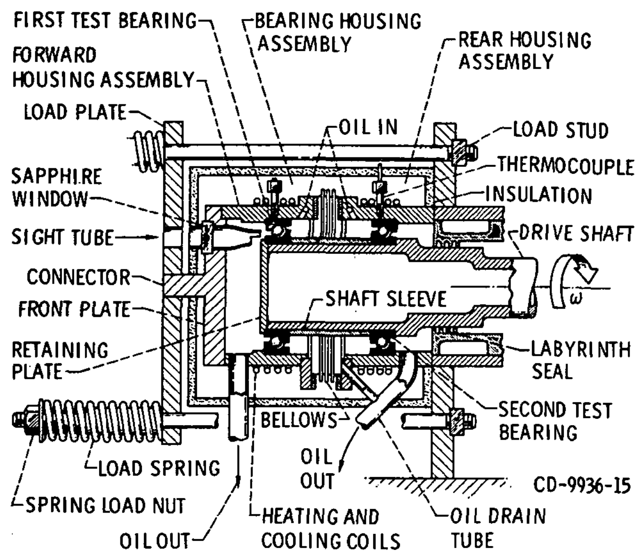


Figure 3. - High-temperature bearing fatigue test apparatus.

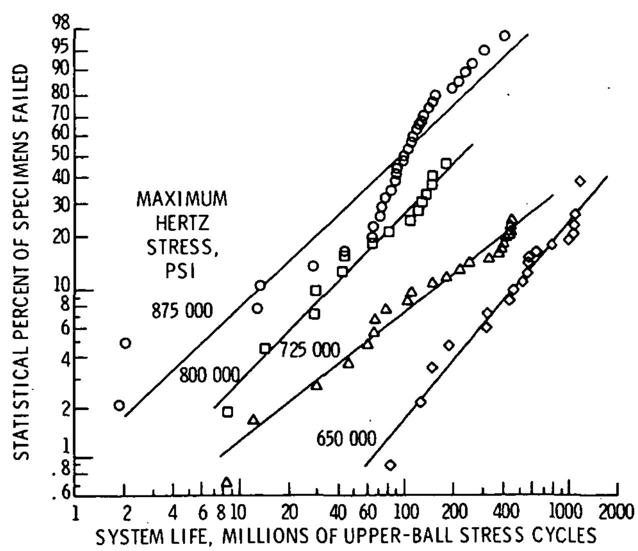


Figure 4. - Rolling-element fatigue life of 1/2-in. diameter AISI 52100 steel balls in the five-ball fatigue tester. (Shaft speed, 10 000 rpm; contact angle, 30°; lubricant, paraffinic mineral oil.)

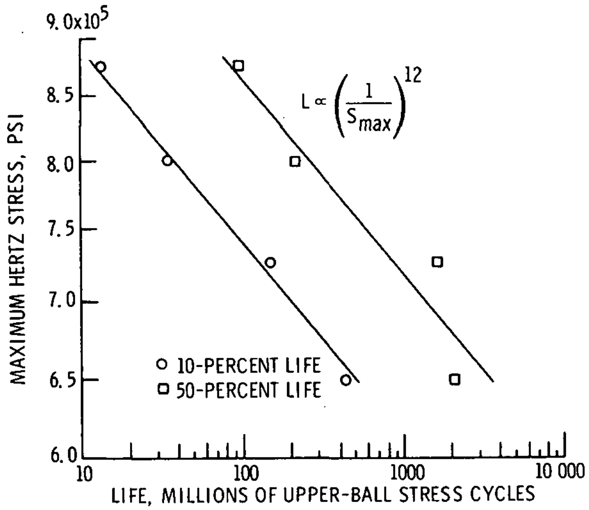


Figure 5. - Effect of maximum Hertz stress on life of vacuum-degassed AISI 52100 balls tested in five-ball fatigue tester.

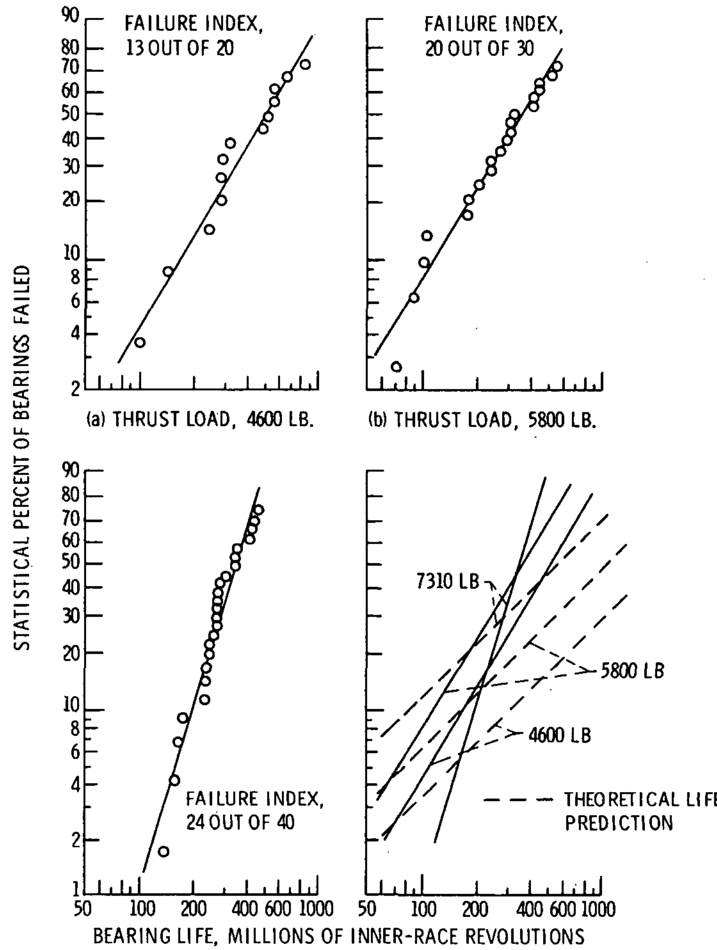


Figure 6. - Effect of thrust load on the life of 120-mm bore angular-contact ball bearings at 500° F and a speed of 12 000 rpm.