NASA TECHNICAL NOTE

NASA TN D-6745

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NASA TN D-6745

REEVALUATION OF THE STRESS-LIFE RELATION IN ROLLING-ELEMENT BEARINGS

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • APRIL 1972

1. Report No. NASA TN D-6745	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle REEVALUATION OF THE STRESS-LIFE RELATION IN ROLLING-ELEMENT BEARINGS		5. Report Date April 1972
		6. Performing Organization Code
7. Author(s) Richard J. Parker and Erwin V. Zaretsky		8. Performing Organization Report No. E-6768
		10. Work Unit No. 764-74
9. Performing Organization Name and Address Lewis Research Center National Aeronautics and Space Administration Cleveland, Ohio 44135		11. Contract or Grant No.
		13. Type of Report and Period Covered Technical Note
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D. C. 20546		14. Sponsoring Agency Code
15. Supplementary Notes		

16. Abstract

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17. Key Words (Suggested by Author(s)) Bearings; Rolling-element bearing; Bearing life; Bearing fatigue life	18. Distribution Statement Unclassified - unlimited

19. Security Classif. (of this report) Unclassified	20. Security Classif. (of this page) Unclassified	21. No. of Pages 17	22. Price* $3.00

*For sale by the National Technical Information Service, Springfield, Virginia 22151

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SUMMARY

Four groups of 12.7-millimeter- (1/2-in.-) diameter balls were tested, each at a level of maximum Hertz stress in the range of 4.5×10^9 to 6.0×10^9 N/m^2 (650 000 to 875 000 psi). Tests were run in the five-ball fatigue tester at a contact angle of 30° and a shaft speed of 10 000 rpm with a paraffinic mineral oil as the lubricant. All balls were from one heat of vacuum-degassed AISI 52100 material. The 10-percent fatigue lives at the four stress levels followed the relation that fatigue life is inversely proportional to maximum Hertz stress raised to the power of 12. This result agrees with a survey of the literature which suggests that a stress-life exponent of approximately 12 is typical of vacuum-processed bearing steels. The exponent of 9, which was initially determined and verified with air-melted materials, has been generally accepted by the bearing industry. Results of this investigation indicate that the load-life relation for vacuum-processed bearing steels may be a fourth power relation rather than the third power relation that has been generally accepted.

INTRODUCTION

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

Since stress is proportional to the cube root of load for ball bearings, the load-life relation for ball bearings can be resolved into a stress-life relation where life, $L \propto (1/\text{stress})^9$. This relation has been generally accepted by ball bearing manufacturers and users. There is at least one exception where a manufacturer has indicated that the life of ball bearings varies inversely with the fourth power of load (or twelfth power of stress) (ref. 3). Nevertheless, the inverse cubic load-life relation has been included in the AFBMA Standards for ball bearings (ref. 4).

Subsequent fatigue tests with ball bearings (refs. 5 to 7) tended to verify this inverse cubic relation. Styri (ref. 5) in 1951 presented data with two types of ball bearings. One group of 6207 deep groove ball bearings under various radial loads from 3.45×10^3 to 1.73×10^4 newtons (775 to 3880 lbf) determined life to be inversely proportional to load to the 3.3 power. Another group of 1207 double-row, self-alining bearings was tested such that the maximum Hertz stress at the outer-race-ball contact varied from 4.0×10^9 to 5.6×10^9 N/m^2 (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 (ref. 6) 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 reference 6, to the ''superior'' lubrication qualities of this lubricant at lighter loads.

In 1963, McKelvey and Moyer (ref. 7) 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 1.8×10^9 to 3.3×10^9 N/m^2 (262 000 to 478 000 psi). Again, the inverse cubic relation was generally confirmed.

Several publications (refs. 8 to 12) 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 (except for ref. 8, which does not state the type of steel). At least two sets of data (ref. 9) are with air-melted material, whereas the other references do not state the melting process.

More recent data (refs. 13 to 15) indicate a greater effect of stress on life than the inverse ninth power relation. Schatzberg (ref. 13) reports a stress-life exponent of 12 for vacuum-degassed AISI 52100 balls in a four-ball fatigue tester. Data presented in references 14 and 15 and reanalyzed in reference 16 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 reference 17 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

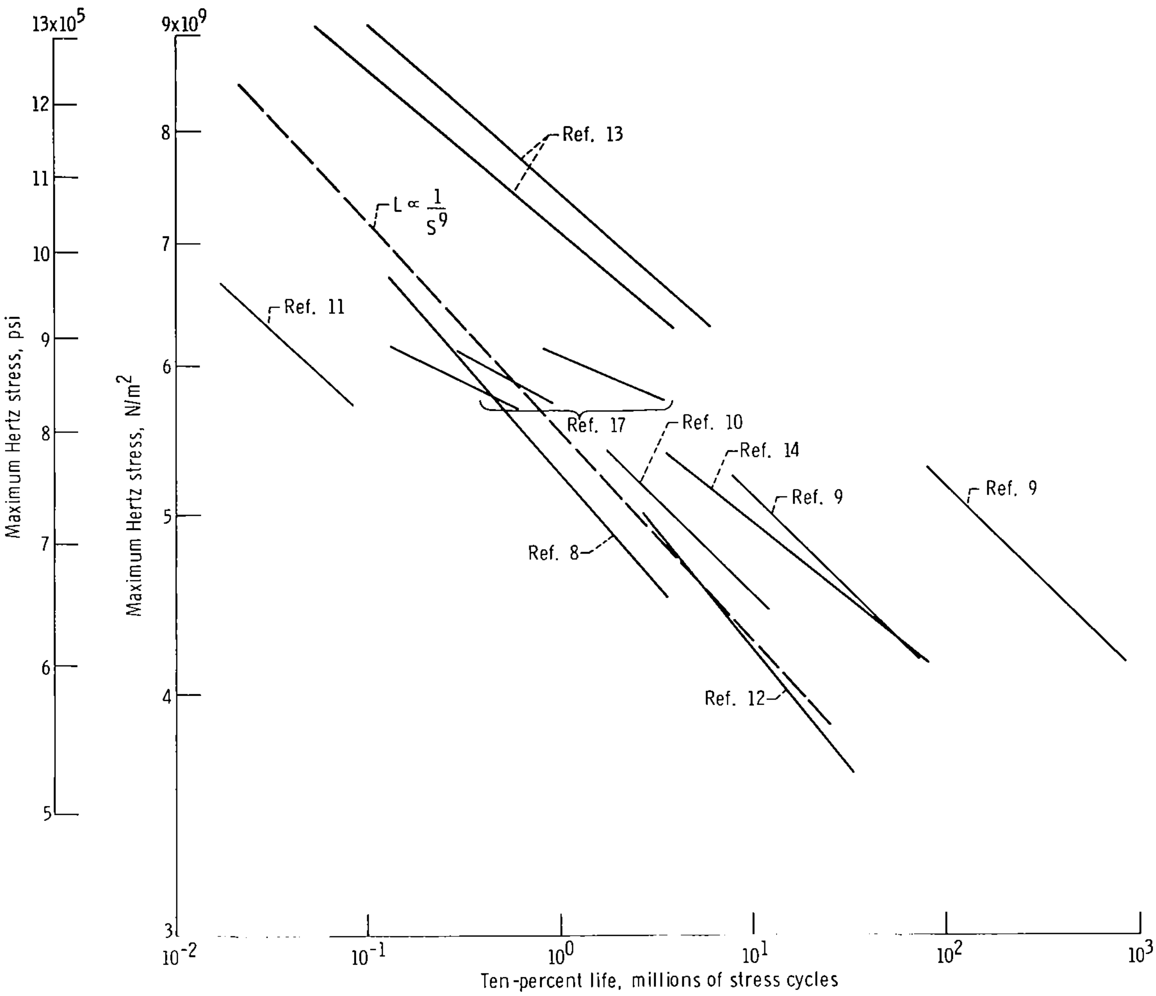


Figure 1. - Stress-life relation: summary of published data.

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

Reference	Test type	Publication date	Material		Lubricant	Load range, kg	Maximum Hertz stress range		Load-life exponent	Stress-life exponent
			Type	Melting process			N/m^2	psi		
8	Four-ball	1956	----------	(a)	Mineral oil	200 to 600	[b]4.5×10^9 to 6.6×10^9	[b]6.6×10^5 to 9.5×10^5	2.8	[c]8.4
9	Spin-rig	1957	AISI 52100	Air melt	SAE 10 mineral oil	---------	4.1×10^9 to 5.2×10^9	6.0×10^5 to 7.5×10^5	---	10.4
9	Spin-rig	1957	MV-1 (AISI M-50)	Air melt	SAE 10 mineral oil	---------	4.1×10^9 to 5.2×10^9	6.0×10^5 to 7.5×10^5	---	9.7
10	R-C rig	1958	MV-1 (AISI M-50)	(a)	MIL-L-7808	---------	4.4×10^9 to 5.4×10^9	6.4×10^5 to 7.77×10^5	---	9.7
11	Four-ball	1958	EN-31 (AISI 52100)	(a)	Diester	400 to 600	[b]5.7×10^9 to 6.6×10^9	[b]8.3×10^5 to 9.5×10^5	3.6	[c]10.8
12	Crowned-disk	1960	AISI 52100	(a)	#60 Spindle oil	---------	3.7×10^9 to 5.1×10^9	5.26×10^5 to 7.3×10^5	---	[d]8.5
17	Toroids	1962	AISI 52100	(a)	Navy symbol 2190 TEP lubricating oil	---------	5.8×10^9 to 6.1×10^9	8.4×10^5 to 8.9×10^5	---	[d]15 to 19
14	Four-ball	1965	AISI 52100	Consumable-electrode vacuum remelted	Mineral oil	---------	4.2×10^9 to 5.5×10^9	6.1×10^5 to 8.0×10^5	---	[e]12.4
13	Four-ball	1969	AISI 52100	Vacuum degassed	Squalene	---------	6.4×10^9 to 9.0×10^9	9.25×10^5 to 1.3×10^6	---	11.5
13	Four-ball	1969	AISI 52100	Vacuum degassed	Squalene + 100 ppm H_2O	---------	6.4×10^9 to 9.0×10^9	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 ref. 16.

lack of correlation of these data (ref. 17) with other data remains unexplained.

It is interesting to note that the data from the bench-type fatigue testers (refs. 8 to 13) are in the range from 3.6×10^9 to 9.0×10^9 N/m^2 (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 reference 5, which cover the maximum Hertz stress range from approximately 2.1×10^9 to 5.6×10^9 N/m^2 (300 000 to 810 000 psi), tend to confirm the lack of varying exponent in different stress ranges. However, references 14 and 15 propose that the stress-life exponent varies in the stress range from approximately 3.4×10^9 to 5.5×10^9 N/m^2 (500 000 to 800 000 psi) because of plastic deformation.

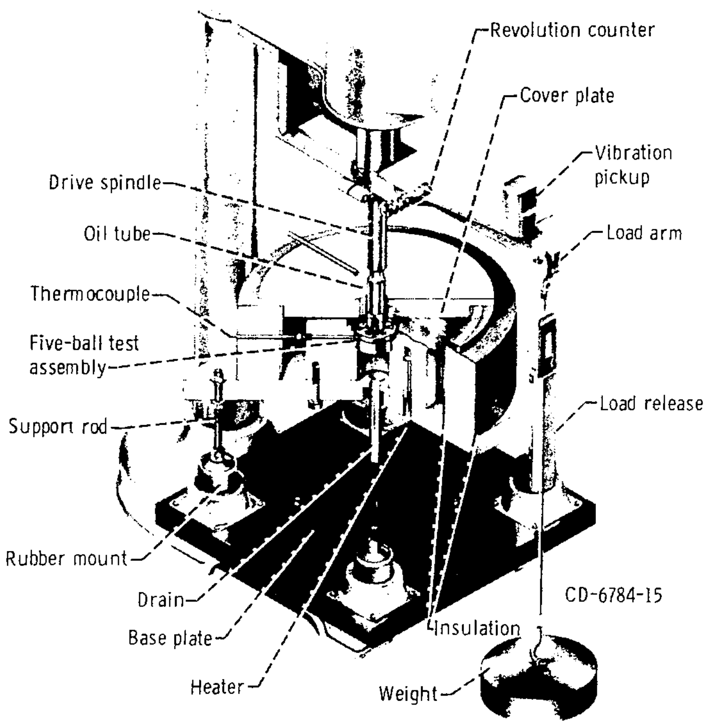
The objectives of the research reported herein were (1) to determine the stress-life relation for vacuum-degassed AISI 52100 balls in the five-ball fatigue tester and (2) to clarify the discrepancies among published values of stress-life exponents. Tests were conducted with 12.7-millimeter- (1/2-in.-) diameter balls in the maximum Hertz stress range from 4.5×10^9 to 6.0×10^9 N/m^2 (650 000 to 875 000 psi), at a shaft speed of 10 000 rpm, and at a contact angle of 30° with a paraffinic mineral oil as the lubricant. Four series of tests were run, each at one of four stresses, in the range given above. All fatigue results were obtained with a single batch of lubricant and a single material heat.

APPARATUS AND PROCEDURE

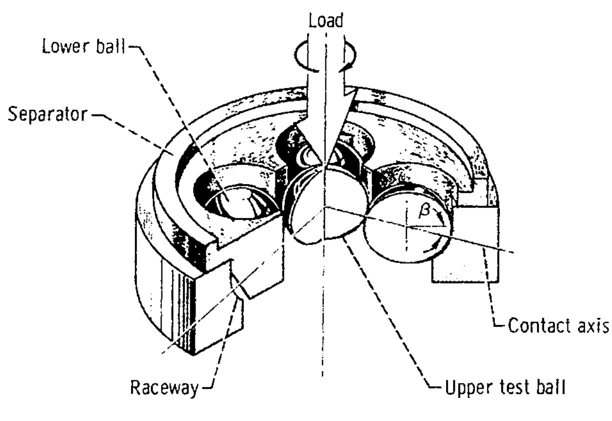
Five-Ball Fatigue Tester

The NASA five-ball fatigue tester was used for all tests conducted. The apparatus is shown schematically in figure 2 and is described in detail in reference 18. This fatigue tester consists essentially 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. The upper test ball and raceway are analogous in operation to the inner and outer races of a bearing, respectively. The separator and the lower balls function in a manner similar to the cage and the balls in a bearing.

Lubrication is provided by a once-through, mist lubrication system. The lubricant was a paraffinic mineral oil without additives with a viscosity of 28.4 centistokes (28.4×10^{-6} m^2/sec) at 311 K (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.

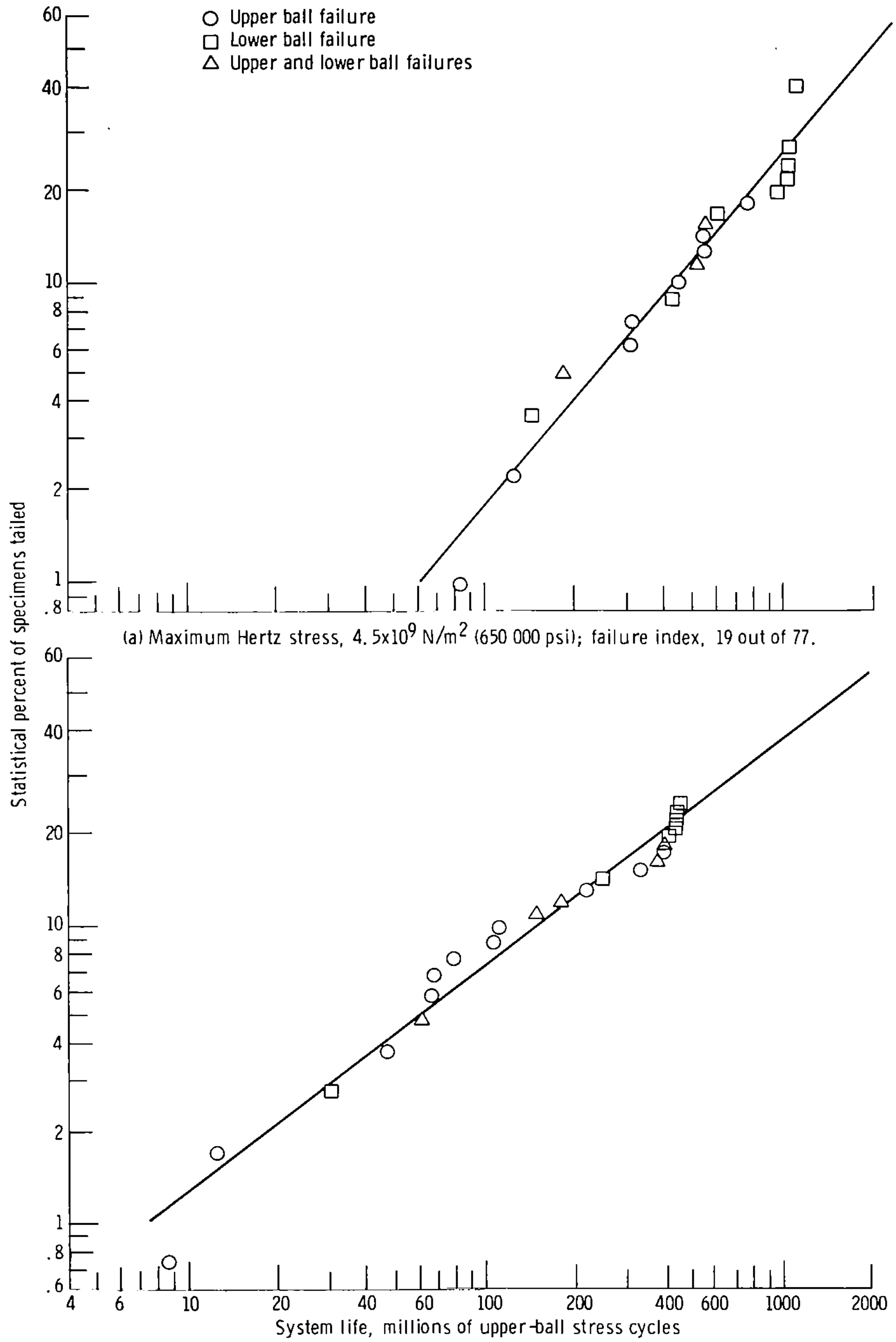


(a) Cutaway view of five-ball fatigue tester.



(b) Five-ball test assembly.

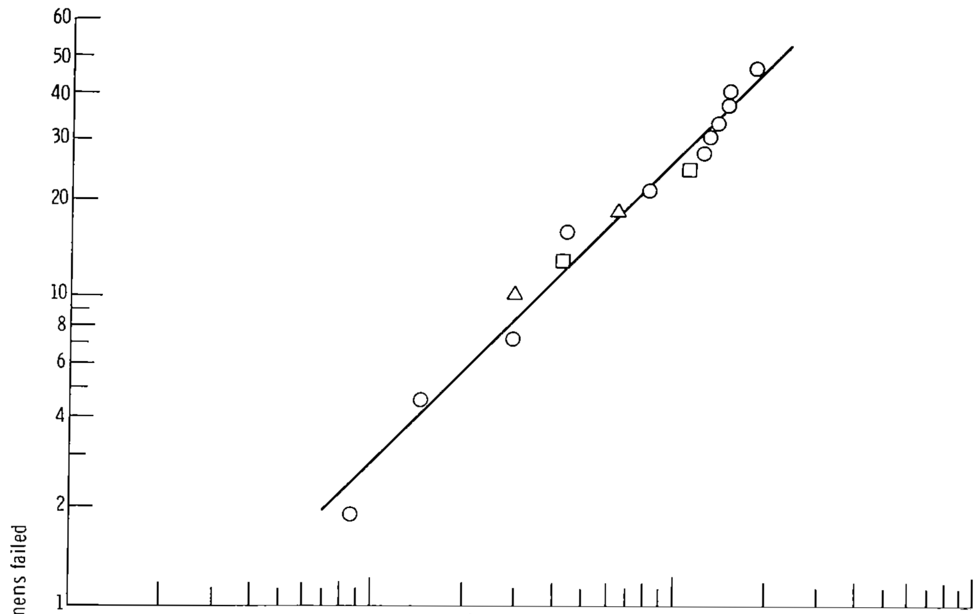
Figure 2. - Test apparatus.



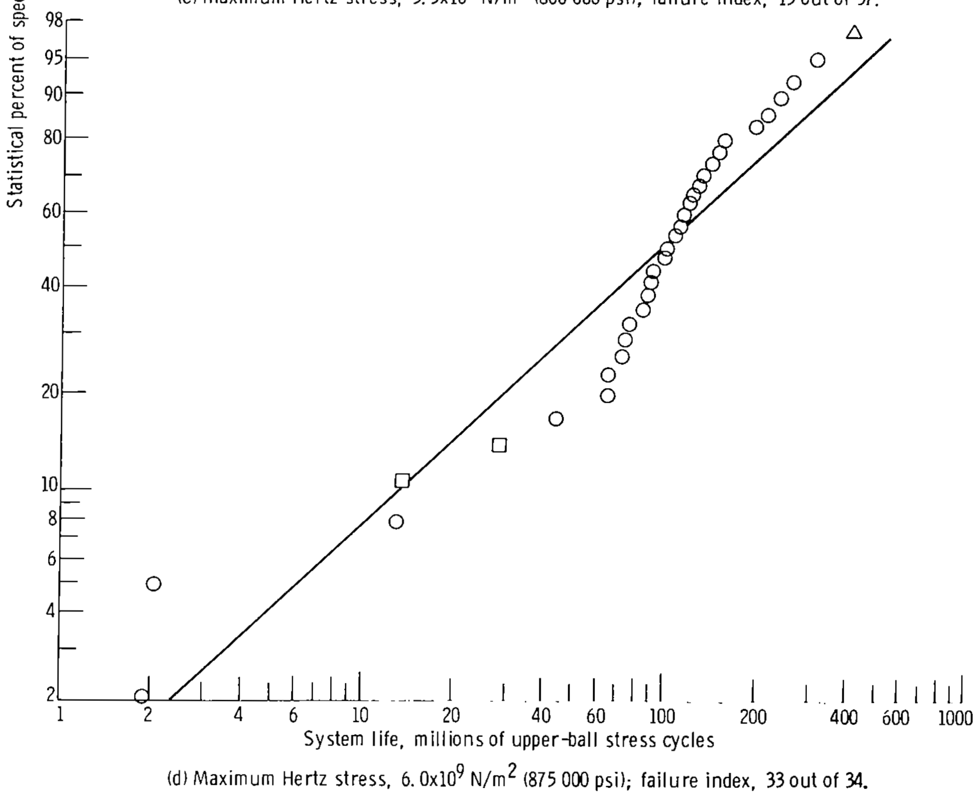
(a) Maximum Hertz stress, 4.5x10⁹ N/m² (650 000 psi); failure index, 19 out of 77.

(b) Maximum Hertz stress, 5.0x10⁹ N/m² (725 000 psi); failure index, 23 out of 98.

Figure 3. - Rolling-element fatigue life of 12.7-millimeter-(1/2-in.-) diameter AISI 52100 balls in the five-ball fatigue tester. Shaft speed, 10 000 rpm; contact angle, 30°, lubricant, paraffinic mineral oil.

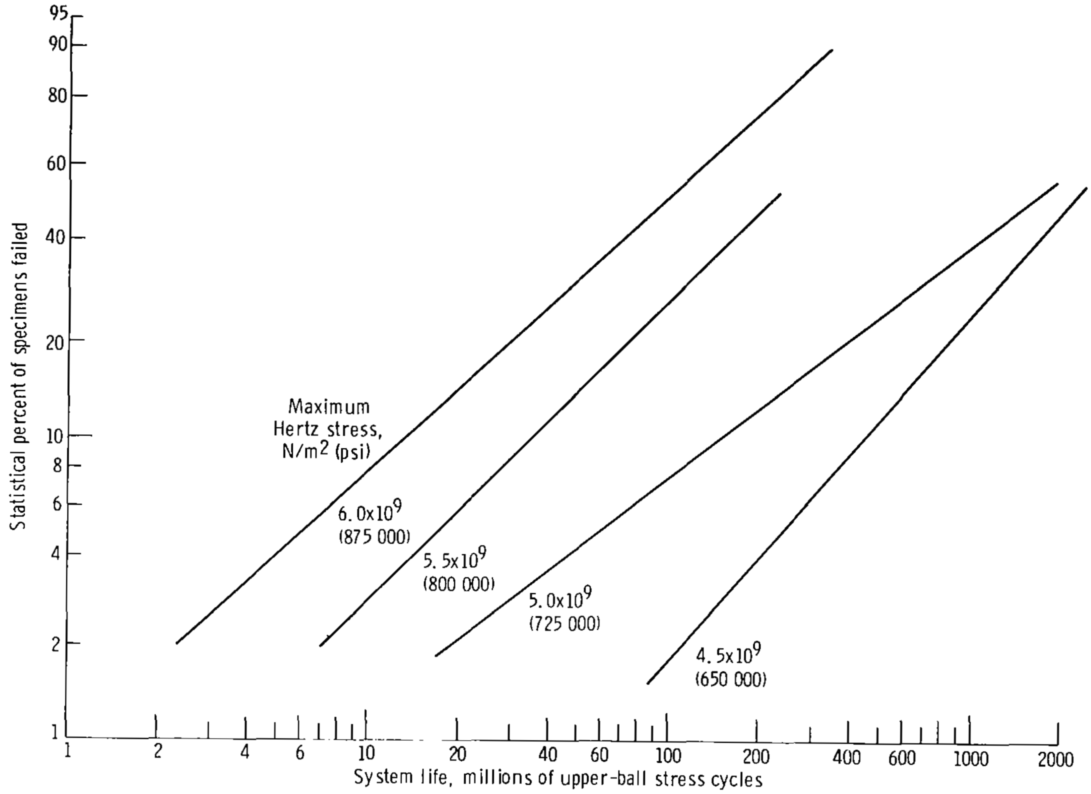


(c) Maximum Hertz stress, 5.5×10^9 N/m^2 (800 000 psi); failure index, 15 out of 37.



(d) Maximum Hertz stress, 6.0×10^9 N/m^2 (875 000 psi); failure index, 33 out of 34.

Figure 3. - Continued.



(e) Summary.

Figure 3. - Concluded.

Fatigue Testing

Before they were assembled in the five-ball fatigue tester, all test-section components were flushed and scrubbed with ethyl alcohol and wiped dry with clean cheesecloth. The test balls were examined for imperfections at a magnification of 15 diameters. After examination, all test balls were coated with test lubricant to prevent corrosion and wear at startup. A new set of five balls was used for each test. Each test was suspended when a fatigue failure occurred (on either an upper test ball or a lower test ball) or when a preset cutoff time was reached. The speed, outer-race temperature, and oil flow were monitored and recorded at regular intervals. After each test, the outer race of the five-ball system was examined visually for damage. If any damage was discovered, the race would be replaced before further testing. The stress that was developed in the contact area was calculated by using the Hertz formulas given in reference 19.

Method of Presenting Fatigue Results

The statistical methods of reference 20 for analyzing rolling-element fatigue data were used to obtain a plot of the log-log of the reciprocal of the probability of survival as a function of the log of upper-ball stress cycles to failure (Weibull coordinates). For convenience, the ordinate is graduated in statistical percent of specimens failed. From a plot such as this (see fig. 3), the number of upper-ball stress cycles necessary to fail any given portion of the specimen group may be determined.

For purposes of comparison, the 10-percent life on the Weibull plot was used. The 10-percent life is the number of upper-ball stress cycles within which 10 percent of the specimens can be expected to fail; this 10-percent life is equivalent to a 90-percent probability of survival. The failure index indicates the number of specimens that failed out of those tested.

RESULTS AND DISCUSSION

Five-Ball Fatigue Results

Four groups of 12.7-millimeter- (1/2-in.-) diameter balls were tested in the five-ball fatigue tester at four levels of maximum Hertz stress ranging from 4.5×10^9 to 6.0×10^9 N/m^2 (650 000 to 875 000 psi). Test conditions included a contact angle of 30^o and a shaft speed of 10 000 rpm with a paraffinic mineral oil as the lubricant. All balls were from one heat of vacuum-degassed AISI 52100 and had a nominal Rockwell C hardness of 61 at room temperature.

The results of the fatigue tests are shown in the Weibull plots of figure 3. 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 3(e) and table II(a). Also shown in table II(a) are the 90-percent confidence limits on the 10-percent lives as calculated by methods of reference 20.

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 adjacent stress levels.

For example, in table II(a) the experimental estimate of the 10-percent life of 439 million stress cycles at 4.5×10^9 N/m^2 (650 000 psi) falls within the 90-percent confidence limits of 40 to 560 million stress cycles for the 5.0×10^9 N/m^2 (725 000 psi) condition. However, the limits do not overlap for nonadjacent stress levels (e.g., 4.5×10^9 and 5.5×10^9 N/m^2) (650 000 and 800 000 psi conditions)). This observation coupled with

TABLE II. - FIVE-BALL FATIGUE TEST RESULTS

(a) Upper- and lower-ball failures

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

Maximum Hertz stress, N/m²	Maximum Hertz stress, psi	L_{10}: Lower 90-percent confidence limit	L_{10} estimate	L_{10}: Upper 90-percent confidence limit	L_{50}: Lower 90-percent confidence limit	L_{50} estimate	L_{50}: Upper 90-percent confidence limit	Weibull slope	Failure index[a]
		Life, millions of upper-ball stress cycles							
4.5×10^{9}	6.5×10^{5}	160	439	1200	1300	2040	3200	1.23	19 out of 77
5.0	7.25	40	150	560	900	1640	3000	.79	23 out of 98
5.5	8.0	9	34.8	140	120	214	400	1.04	15 out of 37
6.0	8.75	5	13.1	35	60	95.1	140	.95	33 out of 34
(b) Upper-ball failures, only									
4.5×10^{9}	6.5×10^{5}	140	519	1900	1350	2460	4400	1.21	11 out of 77
5.0	7.25	40	180	830	1000	2110	4400	.76	16 out of 98
5.5	8.0	9	37.7	160	140	256	490	.98	13 out of 37
6.0	8.75	5	14.5	40	70	106	170	.95	31 out of 34

[a]Indicates number of failures out of total number of tests.

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 4 against maximum Hertz stress. Straight lines fitted to these data by the method of least squares indicates 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 references 13 to 15, 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 at the outer race were measured. As is noted in table III, the outer-race temperature increased with increased stress. To obtain a better understanding of the temperature of the lubricant in the upper-ball - lower-ball contact, tests were run with a thermocouple in the center of the upper-ball. These data are also shown in table III. Ball temperature also increased with increased stress and was higher than the race temperature, as expected.

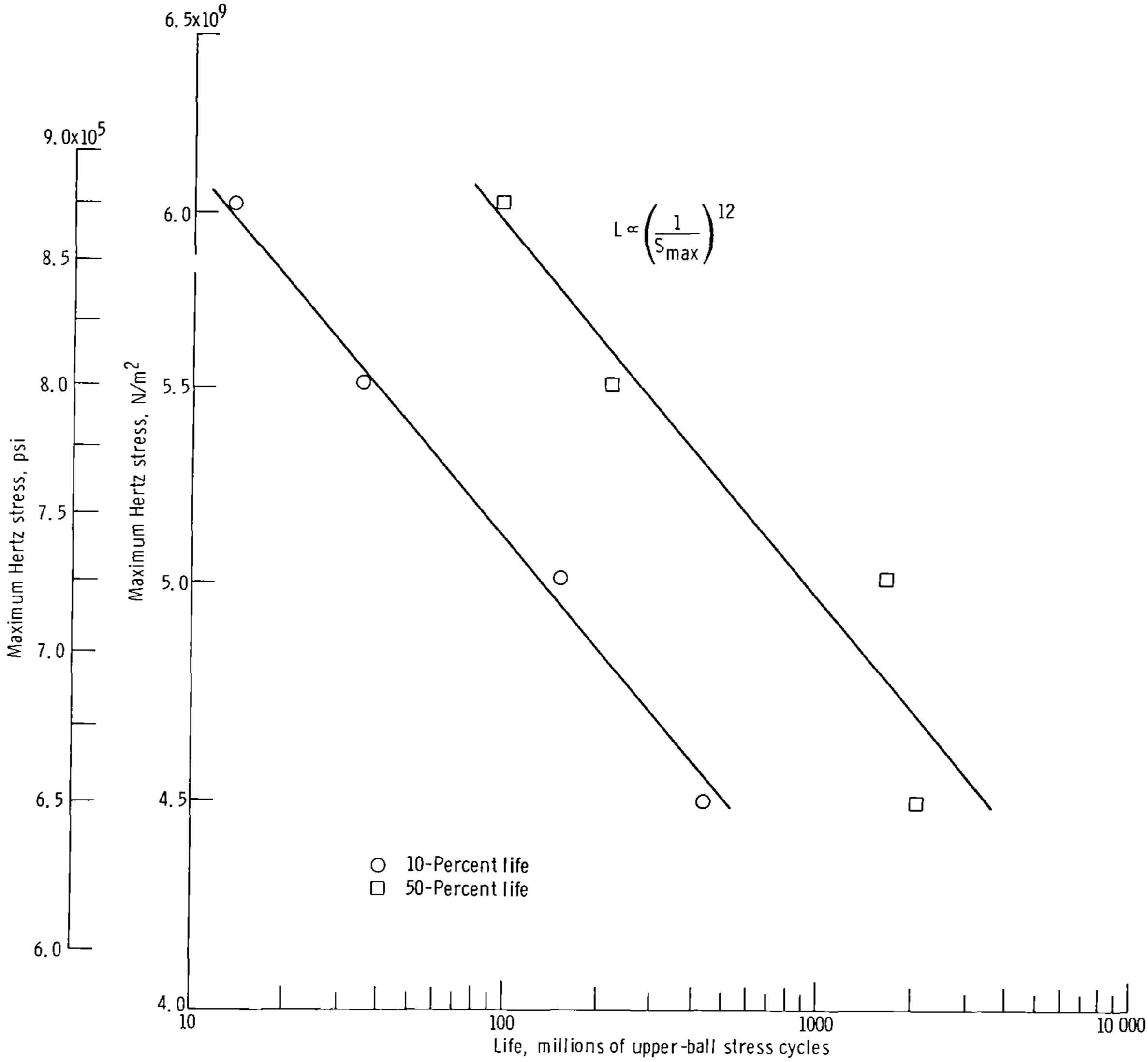


Figure 4. - Effect of maximum Hertz stress on life of vacuum-degassed AISI 52100 balls tested in five-ball fatigue tester.

At the higher stresses, the lubricant in the upper-ball - lower-ball contact would be at a lower viscosity since viscosity decreases with increased temperature. Lubricant viscosity affects fatigue life according to the following relation $L \propto \nu^m$ where $m = 0.2$ or 0.3 from references 21 and 22, respectively, and ν is the kinematic viscosity of the lubricant. The experimental fatigue results would then 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 is shown in table III, the effect is small and can be considered negligible. However, with the viscosity effect considered, the stress-life exponent is only reduced to 11.5, which also correlates well with the data from references 13 to 15.

TABLE III. - LIFE ADJUSTMENT FOR DECREASED LUBRICANT VISCOSITY DUE TO HIGHER TEMPERATURE AT HIGHER STRESSES

Maximum Hertz stress		Outer-race temperature		Ball temperature[a]		Lubricant viscosity at ball temperature		Ten-percent life, millions of upper-ball stress cycles	Ten-percent life adjusted for viscosity at 4.5×10^{9} N/m^2 (650 000 psi) maximum Hertz stress
N/m^2	psi	K	^{o}F	K	^{o}F	cS	m^2/sec		
4.5×10^{9}	6.5×10^{5}	322	120	339	150	11.0	11×10^{-6}	439	439
5.0	7.25	327	129	342	156	10.0	10	150	153
5.5	8.0	336	146	364	195	5.8	5.8	34.8	40.9
6.0	8.75	345	162	372	[b]210	[b]4.9	[b]4.9	13.1	16.4

[a]Measured with thermocouple at the center of the upper ball.
[b]Extrapolated.

Effect of Vacuum Processing

The most recent published data (refs. 13 to 15) have been attained with vacuum-processed AISI 52100. At least two sets of earlier data (ref. 9) that show the exponent to be 9 are with air-melted materials. The other references (refs. 5 to 8 and 10 to 12) 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 (becuase of the earlier dates), it then becomes apparent that the cleaner, vacuum-processed material may be more sensitive to stress changes, thus yielding a stress-life exponent of approximately 12. This would translate to a load-life exponent of 4, which is in agreement with at least one manufacturer's catalog (ref. 3).

Other Effects

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 and a wide load and stress range with no consistent trends apparent.

These data included tests with spinning (or sliding) and rolling and tests with nearly pure rolling as in references 9, 10, and 12. There is an indication of a trend toward a higher stress-life exponent when spinning is present in the rig tests. The data of reference 8 is an exception to this trend. However, when the full-scale bearing fatigue tests of references 5 to 7 are considered, this trend is not apparent. Both radial and thrust loaded bearings were tested, and similar exponents were observed even though thrust loaded ball bearings experience a much greater degree of spinning in the ball-race contact.

The five-ball fatigue test data reported herein were also analyzed considering upper-ball failures only (i.e., considering lower-ball failures as suspensions in the Weibull analysis). As is seen in table II(b), this analysis shows slightly greater lives in all cases, but the general trends are not altered.

General Comments

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 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 references 23 to 25. In reference 23, rotating-beam specimens of a chrome-nickel steel show a greater effect of stress on life as inclusion count is decreased. Similarly, in references 24 and 25, 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 can be considered as stress concentrations. 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.

SUMMARY OF RESULTS

Four groups of 12.7-millimeter- (1/2-in.-) diameter balls were tested, each at a level of maximum Hertz stress in the range of 4.5×10^{9} to 6.0×10^{9} N/m^{2} (650 000 to 875 000 psi). Tests were run in the five-ball fatigue tester at a contact angle of 30° and a shaft speed of 10 000 rpm with a paraffinic mineral oil as the lubricant. All balls were from one heat of vacuum-degassed AISI 52100 material.

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. The exponent of 9, which has been generally accepted by the bearing industry, was initially determined and verified with air-melted materials.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, February 1, 1972,
764-74.

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NASA-Langley, 1972 —— 15 E-6768