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NASA FIVE-BALL FATIGUE TESTER - OVER 20 YEARS OF RESEARCH

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ABSTRACT

The paper reviews, from both a technical and historic perspective, the results of research conducted using the NASA Five-Ball Fatigue Tester. The test rig was conceived by W. J. Anderson in late 1958. The first data was generated in March 1959. Since then a total of approximately 500,000 test hours have been accumulated on a group of eight test rigs which are capable of running 24 hours a day, 7 days a week. Studies have been conducted to determine the effect on rolling-element fatigue life of contact angle, material hardness, chemistry, heat treatment, and processing, lubricant type and chemistry, elastohydrodynamic film thickness, deformation and wear, vacuum, and temperature as well as Hertzian and residual stresses. Correlation was established between the results obtained using the five-ball tester and those obtained with full-scale rolling-element bearings.

INTRODUCTION

The determination of the rolling-element fatigue life of a bearing made from a particular steel or of the effect of a specific lubricant on fatigue is an expensive undertaking. The very high scatter or dispersion in fatigue lives makes it necessary to test a large number of specimens. The cost is high both in money and time. High-quality aircraft engine bearings can cost several hundred dollars each, and test duration can run from several hundred to several thousands of hours depending on the load conditions and speed.

Bearing companies and various research laboratories throughout the world have pioneered the use of bench type rolling-element fatigue testers which can simulate to various degrees the conditions found under full-scale bearing operation. These bench type testers accelerate the generation of rolling-element fatigue data because testing is done at higher stresses resulting in reduced cost. The results obtained with the bench type testers have been used to indicate trends and to rank materials and lubricants. In achieving this objective, bench type rigs have been very successful.

Early rolling-element fatigue testing was performed with what might be termed a two-disk machine [1]. However, up until 1956 most rolling-element fatigue testing was performed on full-scale bearings. In 1957, NACA (now NASA) published their first results [2] with what was termed the "NASA spin rig" [3]. About the same time that work was being performed with the spin rig, Pratt and Whitney Aircraft began utilizing what they termed a "one-ball rig" [4], General Electric Company devised what they termed the "R-C (rolling-contact) rig" [5].

In 1958 it became apparent to William J. Anderson, who then headed the rolling bearing research at the NASA Lewis Research Center, that the number of parameters that could be studied in the spin rig was limited. The reason was that the test load was a function of the orbital velocity of the balls in this rig. Hence, specimen loading could not be varied without varying other parameters. About this time, Professor F. Barwell in Great Britian modified a sliding four-ball wear tester by allowing the three lower balls, which were previously fixed, to rotate in their cylindrical cup [6]. Maintaining very high contact stresses, e.g., >6.9 GPa (1,000,000 psi), Barwell was able to induce a pit or spall in the upper ball running track which resembled those obtained in full-scale bearings. Anderson, capitalizing on the idea of a ball loaded against lower rotating balls, conceived of the five-ball fatigue tester. Anderson's concept incorporated an upper test ball pyramided upon four lower test balls which are positioned by a separator and are free to rotate in an angular contact raceway (see Fig. 1). The implementation of this concept was left to Thomas L. Carter and Erwin V. Zaretsky who in January 1959 designed and fabricated the first five-ball fatigue tester at the NASA Lewis Research Center. The tester produced its first data in March 1959. Since then a total of approximately 500,000 test hours have been accumulated on a group of eight test rigs which are capable of running 24 hours a day, 7 days a week. The objective of the work reported herein is to summarize over 20 years of test results obtained from the Five-Ball (Rolling-Element) Fatigue Tester and to assess the correlation of these data with full-scale bearing data and, thus, the influence on current bearing design and operation.

FIVE-BALL FATIGUE TESTER

The NASA five-ball fatigue tester simulates very closely the kinematics of a thrust-loaded, full-scale bearing. The tester, which was first described in [7,8], essentially comprises an upper test ball pyramided upon four lower test balls which are positioned by a separator and are free to rotate in an angular contact raceway. Loading and drive are supplied through a vertical shaft as illustrated in Fig. 1(a). By varying the pitch diameter of the four lower test balls, the contact angle β, Fig. 1(b), may be varied. The angular spin velocity in the upper ball-lower ball contact, ω_2, approximately equals $\omega_1 \sin \beta$ [7,8]. By utilization of a vibration type failure-detection and shut-down system, long-term unmonitored tests are made possible.

Lubrication in this rig is accomplished by introducing droplets of a fluid into a gas stream directed at the test balls. Lubricant flow rate can be controlled by adjusting the pressure upstream of a long capillary tube; pressure drop through the capillary tube is sufficient to give excellent control for the small flow rates required. The test ball assembly is loaded by dead-weight acting on the drive shaft through a load arm. Contact load is a function of the drive shaft load and the contact angle, β. For every revolution of the drive shaft, a point on the running track of the upper test ball is stressed three times [7,8].

The system housing assembly is supported by rods held in flexible rubber mounts as shown in Fig. 1(a). Positioning of the rods and rubber mounts provides for alignment of the raceway and the four lower test balls with respect to the upper test ball and the drive shaft. Minor misalignments are absorbed by the flexible rubber mounts, as are vertical oscillations of the drive shaft created by motor and drive belt vibration. Such vibrations, if not absorbed, will increase the stress level of the ball specimens thus influencing fatigue results. Modifications of this rig have been used for operation in a vacuum (low pressure) environment, and at temperatures from -300° to 2000° F (Figs. 1(c), (d) and (f)).

DISCUSSION OF RESULTS

The results of the various research programs conducted in the five-ball fatigue tester are discussed in terms of the effect on fatigue life of the particular parameter or variable being investigated. These include material effects (hardness, component hardnesses, residual stresses, alloying element, processing and non-ferrous materials), lubricant effects (type, viscosity, temperature and additives), geometry effects (contact angle and ball hollowness), stress effects, and wear debris analysis. Finally there is a comparison with full-scale bearing data.

Effect of Material Hardness

The fatigue spin rig and the five-ball tester were used to determine the rolling-contact fatigue life at room temperature of groups of AISI M-1, AISI M-50, Halmo, and WB-49 steel balls tempered to various hardness levels. Nominal test conditions included 5.52 GPa (800,000 psi) maximum theoretical (Hertz) compressive stress and a synthetic diester lubricant. The fatigue-life results were compared with: material hardness, resistance to plastic deformation in rolling contact, and previously published tensile and compressive strength data for these same heats of material. The results of these tests were reported in [8-11].

A summary of these tests is shown in Fig. 2. Rolling-element fatigue life or, analogously, load-carrying capacity of each of the four alloy compositions studied, increased continuously as material hardness increased. The improvement in load capacity was in the order of 30 to 100 percent over the hardness range tested. No maximum fatigue life or load capacity at intermediate hardness values was observed. In contrast, elastic-limit and yield strength measurements [8] made on bar specimens from the same heats of materials showed optimum values at an intermediate hardness level and did not correlate with fatigue life. Resistance to plastic deformation, however, measured with ball specimens in rolling contact, did correlate with fatigue life.

These results were substantiated with bearings made of AISI 52100 having Rockwell C hardnesses up to 63; for these bearings, fatigue life increased with increasing hardness [12]. These findings, however, were contrary to the conclusions reached in [13] wherein a maximum fatigue life was predicted at an intermediate hardness level for AISI M-50.

Differential Hardness

It was reported in [14,15] that plastic deformation can reduce the contact stress as much as 10 percent at theoretical maximum Hertz stress levels between 4.10 and 5.52 GPa (600,000 and 800,000 psi) in the five-ball fatigue tester. Therefore, where plastic deformation does occur under rolling contact, the calculated Hertz stress may only be approximate [14,16]. Material hardness or resistance to plastic deformation has a two-fold effect on fatigue life: as hardness is decreased, fatigue life decreases because of an inherent decrease in material strength, but at the same time plastic deformation increases and the contact stress decreases. This latter effect would increase fatigue life. The two effects are, therefore, acting in opposition to each other. Based upon this premise and the apparent contradiction between the results obtained in both the spin rig and five-ball fatigue tester and those reported in [13], a program was undertaken to determine the effect of hardness differences between upper and lower test balls on rolling-element fatigue life.

AISI 52100 steel balls from one heat of material were tempered to a range of Rockwell C hardnesses from 59.7 to 66.4. Groups of balls having average Rockwell C hardnesses of 60.5, 63.2, and 65.2 were used as upper test balls and run with lower test balls of nominal Rockwell C hardnesses of 60, 62, 63, 65, and 66. Nominal test conditions included an average race temperature of 339 K (150° F), 5.52 GPa (800,000-psi) maximum (Hertz) compressive stress, and a highly purified naphthenic mineral oil lubricant. The fatigue life results were compared with component hardness combinations, plastic deformation of the upper test ball, retained austenite, grain size, and contact temperature. The results of these tests are reported in [17,18] and summarized in Fig. 3.

In general, for a specific upper test ball hardness, the rolling-element fatigue life and the load-carrying capacity of the test system (considering both upper and lower ball failures) increased with increasing lower test ball hardness to an intermediate hardness where a peak life was attained. For further increases in hardness of the lower balls, system life and capacity decreased. The peak life hardness combination occurred for each of three lots of upper test balls where the hardness of the lower test balls was approximately one to two Rockwell C hardness points greater than that of the upper test ball. The improvement in load capacity at the intermediate hardness where a peak life was observed was as much as 130 percent when compared to the capacity at the lowest hardness.

There was no apparent correlation between fatigue life and material properties such as retained austenite and grain size. Additionally, no correlation existed between the resistance to plastic deformation between elements of different hardnesses and fatigue life. Further, no correlation existed between the rolling-element fatigue life and the temperatures measured at the edge of the contact zone (Fig. 1(e)) for each hardness combination.

To illustrate the effect of hardness differential, ΔH, on bearing fatigue and load capacity, four lots of AISI 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 5870 N (1320 pounds), a speed of 2750 rpm with a mineral oil lubricant, and with no heat added [18,19]. The relative bearing load capacity is shown as a function of ΔH in Fig. 4. For comparison purposes, the range of predicted relative capacity based on the five-ball fatigue data is also presented. The correlation with the five-ball data is excellent. It can be concluded that maximum bearing fatigue life and load capacity can be achieved where the hardness of the rolling elements of the bearing is one to two points Rockwell C greater than the races.

Based upon the above results, the data of [12], which were among the earliest data to show an effect of hardness on life, were reanalyzed considering hardness differences between the balls and race [18]. The reanalyzed data not only indicated the optimum ΔH relation, but also exhibited relative life values of approximately the same magnitude with varying ΔH as those shown for the 207-size bearings in [18].

From the data of [17] a relation which approximates the effect of bearing hardness on fatigue life was obtained where

$$\frac{L_2}{L_1} = e^{0.1(Rc_2 - Rc_1)}$$

L_1 and L_2 are the bearing ten-percent fatigue lives at bearing hardnesses Rc_1 and Rc_2, respectively, at a ΔH of 0.

Residual Stress

It was shown in [20,21] 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 [22]. 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 shear stress [23], according to the following equation:

$$(\tau_{max})_r = -4.67\times10^6 \left(\frac{P_n}{R^2 S_{max}}\right)^{1/2} - \frac{1}{2} (\pm S_{ry})$$

where

P_n normal load

R radius of curvature of sphere

S_{max} maximum Hertz stress

S_{ry} residual stress in the y (rolling direction)

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 shear stress and increase fatigue life [23] according to the following:

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

Five upper test balls having a Rockwell C hardness of 63.2 that were run in the five-ball tester [17] 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 were used to measure residual stresses at a depth of 0.13 mm (0.005 in.) beneath the running track, which was the calculated depth of the maximum shear stress. The residual stresses below the track were found to be compressive and varied between 1.23 and 2.03 GPa (178,000 and 294,000 psi).

The measured compressive residual stresses are plotted as a function of ΔH in Fig. 5. From this figure, note 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 ten-percent lives of the upper test ball using the aforementioned relationships. These calculated ten-percent lives 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.

Based upon the results obtained with the five-ball tests, subsurface residual stress measurements were made on AISI 52100 steel inner races from the 207-size deep-groove ball bearings in which ΔH ranged from -1.1 to 3.5 points Rockwell C [24]. These bearings had been run at an inner-race speed of 2750 rpm and a radial load of 5870 N (1320 pounds) producing maximum Hertz stresses of 2.43 and 2.32 GPa (352,000 and 336,000 psi) at the inner and outer races, respectively. The residual stress measurements were made in a circumferential direction at various depths below the inner-race running track surface on a total of 19 of the bearings that had been run for nominally 200, 600, and 1600 hours. The measurements indicated that the maximum compressive residual stresses occur in approximately the same ΔH range for which the maximum fatigue lives were observed. Again, good correlation with five-ball tester results was indicated. Additionally, no relation between running time and residual stress could be determined from these measurements.

Lubricant Effects

Bearing failure by fatigue is affected by the physical and chemical properties of the lubricant. Knowledge of how these chemical and physical properties affect rolling-element fatigue is a useful guide in both selecting existing lubricants for bearing applications and in developing new lubricant formulations. Specifically, for aircraft engine bearing applications, it is important to know the effect these lubricants have on the bearing life and reliability.

Effect of lubricant type. - The NASA fatigue spin rig and five-ball fatigue tester were used to determine the rolling-element fatigue lives at room temperature and at 422 K (300° F) of groups of AISI M-2 and AISI M-1 steel balls run with nine lubricants having varied chemical and physical characteristics [16]. These lubricants were classified as three basic types: esters, mineral oils, and silicones. Longer fatigue lives were obtained at room temperature with the silicone and the dioctyl sebacates than with other lubricants; at 422 K (300° F), however, the silicone and the mineral oils gave the longer lives. The relative order of fatigue results obtained in the five-ball fatigue tester compared favorably with that obtained with 7208-size bearings tested with the same lubricants (Fig. 6).

A correlation is suggested between the plastically deformed-profile radius of the upper test ball running track at ambient temperature after 30,000 stress cycles of operation and rolling-element fatigue life. The fatigue results obtained did not correlate with temperatures measured at the edge of the contact zone of the test specimens (Fig. 1(e)) during rolling contact in the five-ball fatigue tester at 422 K (300° F) outer-race temperature. The measured contact temperature differences among the lubricants were insignificant.

Elastohydrodynamic film thickness measurements using X-ray techniques were made with a rolling-contact disk machine under simulated five-ball test conditions [15] with several of the lubricants used in [16]. Under certain conditions [15], elastohydrodynamic lubrication was found to exist at initial maximum Hertz stress levels up to 5.52 GPa (800,000 psi).

Polyphenyl ethers. - In the early 1960s among the most often considered lubricants for bearings, gears, and hydraulic systems in applications where high thermal and oxidative stability and resistance to nuclear radiation were necessary were the polyphenyl ethers. Because the polyphenyl ethers have these desirable properties along with relatively low vapor pressures, they were being considered as a lubricant for a closed organic lubrication loop in space power generation systems such as SNAP-8. In such a system, the lubricant would be exposed to pressures less than atmospheric.

A modified five-ball fatigue tester (Fig. 1(c)) was used to determine the relative lubricating characteristics in rolling contact of polyphenyl ethers and mineral oils [25]. Test conditions were a race temperature of 422 K (300° F), a shaft speed of 4900 revolutions per minute, and a test duration of 6 hours with AISI M-50 balls. Measurements on the upper ball were made to determine the effects on wear of reduced pressure, lubricant degassing, contact angle, and contact stress.

Use of a four-ring polyphenyl ether (4P3E) resulted in several times more wear than a naphthenic mineral oil when the fluids were tested at pressures near their vapor pressures at 422 K (300° F). In tests at atmospheric pressure in an argon atmosphere, the 4P3E polyphenyl ether exhibited more wear than a paraffinic oil. Greater wear occurred when the 4P3E polyphenyl ether was tested at a pressure near its vapor pressure than at atmospheric pressure in argon with all other conditions equal. Increased wear at higher contact angles and higher contact stresses was accompanied by increased darkening of the 4P3E polyphenyl ether. The polyphenyl ethers appear to be inferior to the mineral oils in their ability to provide elastohydrodynamic lubrication. However, rolling-contact fatigue tests in the five-ball fatigue tester indicated that the fatigue life with a 5P4E polyphenyl ether at 422 K (300° F) and atmospheric pressure may be expected to be comparable to that with the mineral oils.

Reduced pressure environment. - Rolling-element fatigue life is affected by lubricant viscosity. In general, as lubricant viscosity increases, so does the fatigue life of a rolling-element bearing [26]. Decreases in apparent bulk viscosity of a super-refined mineral oil and an ester based oil were obtained by saturating the lubricating fluid with nitrogen gas [27]. Conversely, exposing the lubricating fluid to a reduced pressure environment can increase the bulk viscosity by degassing the lubricant.

A modified five-ball fatigue tester (Fig. 1(c)) was used to investigate the effect of a reduced-pressure environment on rolling-element fatigue life, deformation, and wear of AISI 52100 steel balls with a super-refined naphthenic mineral oil used as the lubricant. Tests were conducted at atmospheric pressure, at a 20° contact angle, with a thrust load of 267 N (60 pounds) to produce an initial maximum Hertz stress of 5.52 GPa (800,000 psi) at a shaft speed of 4900 rpm with no heat added [28,29]. The atmospheric pressure tests served as reference data for tests conducted at a reduced pressure, approximately the vapor pressure of the lubricant. For these reduced-pressure tests, the ball specimens were lubricated either by a quasi-mist method and or by immersion in the lubricant. In both cases, all other test conditions were the same as those in the atmospheric-pressure tests.

No significant difference in the fatigue lives was observed for tests conducted at atmospheric pressure and those conducted at reduced ambient pressure. The amounts of deformation and wear for the two pressure conditions differed little, regardless of the lubrication mode employed. This was an indication that a sufficient elastohydrodynamic lubricating film existed at the reduced-pressure levels.

Cryogenic environment. - In the late 1960s interest focused on the possible use of liquid lubrication in place of a dry lubricant film for bearings operating in a cryogenic environment. A lubricant capable of forming an elastohydrodynamic film in cryogenic applications must be liquid in the cryogenic temperature range. It must also be able to operate at the maximum system temperature without evaporation. The fluid should be chemically inert and not be susceptible to water absorption, which may cause corrosion of the bearing components. In addition, good heat-transfer properties are desirable.

A class of fluids that exhibits many of the properties required for cryogenic applications is the fluorinated polyether. While some of the fluid properties such as viscosity and heat-transfer characteristics are clearly defined, the ability of the fluids to provide adequate lubrication and life needed to be determined.

Rolling-element lubrication tests were conducted with 12.7-mm- (1/2-inch-) diameter AISI 52100 steel balls in a five-ball fatigue tester modified for cryogenic temperature testing (Fig. 1(d)) [30]. Test conditions included a drive shaft speed of 4750 rpm, a maximum Hertz stress range of 3.45 to 5.52 GPa (500,000 to 800,000 psi), outer-race temperatures of 89 to 227 K (160° to 410° R), and contact angles of 10°, 20°, 30°, and 40°. Four fluorinated polyether fluids with different viscosities were used as the lubricants.

The deformation and wear of the test specimens lubricated with the fluorinated polyether fluids compared favorably with those of test specimens run at room temperature with conventional lubricants. This result indicates that the fluorinated polyether fluids have the ability to form adequate elastohydrodynamic films at outer-race temperatures from 89 to 227 K (160° to 410° R).

The fatigue lives with the four fluorinated polyether fluids in the cryogenic temperature range were compared with that of a super-refined mineral oil at room temperature [31]. There was no statistical difference between the fatigue lives obtained with the fluorinated polyether fluid and the super-refined mineral oil at 169 K (305° R) and 328 K (590° R), respectively. This result indicates that at lower temperatures with the polyether, no fatigue life derating of AISI 52100 is necessary. The operating characteristics of the fluorinated polyether fluids at cryogenic temperatures compared favorably with those of the super-refined mineral oil at room temperature.

Traction fluids. - In recent years there has been renewed interest in mechanical transmissions which use the principle of traction for power transfer. In these devices, torque is transmitted between rolling elements across a thin elastohydrodynamic lubricant film under high contact pressure. Consequently, the magnitude of the contact pressure required to transmit a given amount of torque is principally dependent on the tractive properties of the lubricant. Lubricants formulated for traction drive applications typically have high coefficients of traction. These oils are generically called traction fluids.

Rolling-element fatigue tests were run in the five-ball fatigue tester with two traction fluids, a tetraester with and without additives, a synthetic paraffinic oil and a paraffinic mineral oil [32,33]. Standard test conditions for all tests consisted of a maximum Hertz stress of 5.52 GPa (800,000 psi), a contact angle of 30°, and an upper-ball speed of 10,700 rpm. All tests were run at room temperature (i.e., no heat added). Outer-race temperatures varied from 339 to 353 K (150° to 175° F) for the synthetic lubricants and averaged approximately 336 K (146° F) for the mineral oil.

The results of the rolling-element fatigue tests are summarized in Table 1. The paraffinic mineral oil gave longer lives than any of the synthetic lubricants tested. The ten-percent life with the mineral oil was 34.8 million stress cycles whereas the lives with the synthetic lubricants ranged from 9.5 million cycles to 17.9 million cycles. The statistical significance of the differences in these lives from that of the mineral oil is reflected by the confidence numbers given in Table 1. They indicate the percent of time that components lubricated with the mineral oil would show fatigue lives superior to the fatigue lives of identical components lubricated with each of the synthetic lubricants. A confidence number greater than 95 percent, which is equivalent to a 2σ confidence level, indicates a high degree of certainty. Although all of the synthetic lubricants show lives less than the mineral oil, only the lives with the tetraester base and the synthetic paraffinic oil approach statistically significant differences at the ten-percent life level. At the fifty-percent life level, all experimental lives with the synthetic lubricants are significantly less than the mineral oil.

The rolling-element fatigue lives obtained with the traction fluids were not significantly different on a statistical basis from those obtained with the tetraester oil or the paraffinic mineral oil.

Erratic test behavior was observed in those tests using the traction fluid that contained an antiwear additive. Rolling-element surface distress and overheating occasionally caused premature termination of tests with this lubricant.

Lubricant Additive Effects

Lubricant additives can prevent or minimize wear and surface damage to bearings and gears whose components are in contact under very thin film or boundary lubrication conditions. These antiwear or extreme pressure (EP) additives either adsorb onto the surfaces or react with the surfaces to form protective coatings or surface films. The value of additives for this protection is well known, and their use is commonplace.

To determine the effects of antiwear and EP additives on rolling-element fatigue life, it is necessary to run fatigue tests at EHD film conditions where classical subsurface rolling-element fatigue is the sole mode of failure. This objective was accomplished by testing in the NASA five-ball fatigue tester at test conditions where classical subsurface rolling-element fatigue could be expected [34] with three different steel ball materials.

Rolling-element fatigue tests were conducted in the five-ball fatigue tester using a base oil with and without surface active additives. The 12.7-mm- (0.500-in.-) diameter test balls were either AISI 52100, AISI M-50, or AISI 1018 steel. The test lubricant was an acid-treated white oil containing either 2.5 percent sulfurized terpene, 1 percent didodecyl phosphite, or 5 percent chlorinated wax. Nine combinations of materials and lubricants (six containing additives) were tested at conditions including a maximum Hertz stress of 5.52 GPa (800,000 psi), a shaft speed of 10,700 rpm, and a race temperature of 339 K (150° F). The results of these tests are summarized in Table 2.

Based oil plus 2.5 percent sulfurized terpene. - For both the AISI 52100 and AISI 1018 steels the 2.5 percent sulfurized-terpene additive apparently reduced fatigue life approximately 50 percent. For the AISI M-50, however, the fatigue life was essentially unchanged from that obtained with the base oil. The combined confidence number for the three successive tests is 92 percent. Combining the data for the three materials assumes that the reduction in life is unrelated to the bearing steel chemistry. With this assumption, it can be concluded that the 2.5 percent sulfurized terpene can reduce rolling-element fatigue life by as much as 50 percent under the test conditions reported.

Base oil plus 1 percent didodecyl phosphite. - The 1 percent didodecyl-phosphite additive was run with the AISI 52100 and AISI 1018. The combined confidence number for these two series of tests was 68 percent or approximately equivalent to a 1 sigma confidence and was not considered statistically significant. The results with this additive showed essentially no statistical difference between the lives using the base oil with and without the additive.

Base oil plus 5 percent chlorinated wax. - Only in the tests with the chlorinated-wax additive with AISI 52100 balls was surface distress observed. In this case, eight tests were not included in the analysis because of multiple spalling and considerable surface distress in the running track. Even with these early failures deleted from the analysis, the life with this additive was significantly less than that without the additive.

It was evident that with this lubricant-additive-material-test condition, the EHD film was marginal. This result suggests the possibility that the lubricant rheology of the base oil (viscosity and/or pressure-viscosity effects) has been altered by the chlorinated-wax additive. Further, the influence of this additive can be detrimental to rolling-element fatigue life.

The additives used with the base oil did not change the life ranking of bearing steels in these tests where rolling-element fatigue was of subsurface origin. That, is regardless of additive content of the lubricant, the lives with the three materials ranked in descending order as follows: AISI 52100, AISI M-50, AISI 1018.

Effect of Contact Angle

Aside from the material and lubricant criteria required for improving bearing life and reliability, design considerations are also important. The bearing contact angle is one of these design factors. Bearing contact angle can be defined as "the angle made by a line passing through the points of contact of the ball and both raceways with a plane perpendicular to the axis of the bearing when both races are centered with respect to each other and one race is axially displaced with respect to the other without the application of measurable force." The actual bearing contact angle depends on various operating variables, including bearing load and speed, and may be different at both races.

As contact angle is increased, spin velocity ω_s (and therefore relative sliding between the ball and race) is increased. The sliding is a function of the angular speed of the bearing inner race, the geometry of the bearing, and the applied thrust load. The ball spin affects surface traction forces and heat generation. These in turn affect the elastohydrodynamic film thickness and, possibly, subsurface shear stress. All these factors in turn can affect rolling-element fatigue life.

It became necessary in view of the aforementioned to determine the effect on bearing fatigue and load capacity of the changes in various factors caused by a change in contact angle. The five-ball fatigue tester was used to determine the rolling-element fatigue lives of 12.7 mm (1/2-in.) diameter M-1 steel balls at four contact angles: 10°, 20°, 30°, and 40°. These tests were run at an initial maximum Hertz stress of 5.52 GPa (800,000 psi) using a synthetic diester lubricant meeting the MIL-L-7808C specification [14]. The contact angle for the five-ball fatigue test assembly is as shown in Fig. 1(b).

Figure 7 shows the fatigue results for these test conditions. These results indicate that fatigue life decreases with increasing contact angle. The data are in agreement with other data obtained in a Pratt and Whitney modified one-ball fatigue tester [14] for 30°, 45° and 55° contact angles. These data are also shown in Fig. 7. (Since both sets of data had a common contact angle of 30°, the fatigue lives were compared to the fatigue lives at this angle.

To obtain an estimate of the variation of contact temperature with contact angle β, temperature measurements were made in a modified five-ball fatigue tester (Fig. 1(e)). Each upper ball had a thermocouple attached, the tip of which was approximately 0.25 mm (0.01 in.) away from the edge of the running track. Several measurements were made at running conditions of 5.52 GPa (800,000 psi) a shaft speed of 10,000 rpm, and contact angles of 10°, 20°, 30°, and 40°. The data obtained with these balls are summarized in Fig. 8. These temperatures are believed to represent a much better approximation of actual contact temperatures than the temperatures measured at the outside diameter of the race. A linear increase in temperature is observed with increasing contact angle. The near-contact temperature at the

40° contact angle is approximately 33 K (60 F) higher than that at the 10° contact angle.

Rolling-element fatigue tests were conducted with 12.7-mm (1/2-in.) diameter AISI 52100 steel balls in the five-ball fatigue tester modified for cryogenic temperature testing [35]. Test conditions included a drive shaft speed of 4750 rpm, a maximum Hertz stress of 5.52 GPa (800,000 psi), an outer-race temperature of 130 K (235° R); a lubricant temperature of 170 K (305° R); and contact angles of 20°, 30°, and 40°. A fluorinated ether fluid was used as the lubricant.

There was a decrease in fatigue life with increased contact angle. However, the differences among the fatigue lives obtained at 20°, 30°, and 40° contact angles were not statistically significant. This trend of decreasing fatigue life with increased contact angle in fluorinated ether fluid at 170 K (305° R) is identical to the trend with the diester fluid at contact angles of 20°, 30°, and 40° at 328 K (590° R). Analysis indicates that the decrease in fatigue life is caused by decreased elastohydrodynamic (EHD) film thickness and decreased ball hardness caused by increased contact temperature.

Effect of Bearing Steel

Alloying elements. - A considerable number of studies has been performed to determine the rolling-element fatigue lives of various bearing materials. (A list of references related to these studies can be found in [36].) However, none of these previous studies maintained the required close control of operating and processing variables such as material hardness, melting technique, and lubricant type and batch for a completely unbiased material comparison. It was necessary to compare these materials in rolling-element fatigue tests or actual bearing tests. The more standard mechanical tests such as tension and compression tests or rotating beam tests were not correlatable with rolling-element fatigue results [8].

Eight steels (AISI 52100, M-1, M-2, M-10, M-50, M-42, T-1, and Halmo) were tested in the five-ball fatigue tester [36-38]. Groups of 12.7-mm- (1/2-in.-) diameter balls of each of these materials were tested at a maximum Hertz stress of 5.52 GPa (800,000 psi), a contact angle of 30°, and a shaft speed of 10,000 rpm. Tests were run at a race temperature of 339 K (150° F) with a super-refined naphthenic mineral oil as the lubricant.

In each of the tests, all five balls of the five-ball system were from the particular material lot being tested. From 25 to 30 five-ball tests were run for each material lot. Each test was suspended when either an upper or a lower ball failed or when a cutoff time of 100 hours was reached.

The ten-percent lives for each material lot are shown in Table 3. A 2 to 1 ratio in the ten-percent lives of two lots of the same material is observed with both AISI 52100 and AISI M-10. The only difference between lots of the same material is that they were heat treated separately. The different values of fatigue life among the material lots cannot be attributed to the slight variations in the material properties such as hardness, grain size, retained austenite, and cleanliness since no clear trends were apparent. These slight material property differences may be a result of slight variations in execution of the heat treatment. The variation in ten-percent fatigue lives between material lots may also be normal scatter in rolling-element fatigue data. Based upon previous experience, a 2 to 1 ratio in fatigue lives among lots of the same material is not unusual.

The tests with all three heat treatment lots of each material were grouped together to compare the fatigue lives of the various materials. A Weibull analysis was performed on the combined results for each material.

The materials containing the greatest weight percentage of alloying elements were those with the lowest fatigue lives. This effect is shown in Fig. 9, where relative ten-percent lives are plotted against total weight percent of the alloying elements tungsten, chromium, vanadium, molybdenum, and cobalt. Individually, each element shows no consistent effect on fatigue life. The possible exception is tungsten which is present in significant quantities in the four lowest lived materials, AISI M-1, AISI M-2, AISI T-1, and AISI M-42. The effect, therefore, seems to be a cumulative one. When present in high percentages, these alloying elements appear to be detrimental to rolling-element fatigue life.

There was a strong indication that a relation existed among these alloying elements and the size and distribution of the metal carbides, which vary with alloy content and heat treatment. In [39] a carbide parameter was derived based upon a statistical analysis which related rolling-element fatigue life to the total number of residual carbide particles per unit area, median residual carbide size, and percent residual carbide area. A comparison of the data contained in Table 3 and the carbide parameter C' are shown in Fig. 10.

<u>Processing variables</u>. - Rolling-element bearings for main-shaft applications in aircraft turbine engines are made primarily of either AISI M-50 or 18-4-1 steels. AISI M-50 has been used in such applications since about 1957 and is today the standard material used by American jet-engine manufacturers [40]. The use of 18-4-1 for these applications is primarily by the European jet engine manufacturers. Both materials contain alloying elements to promote high hardness and good hardness retention at the high temperatures experienced by main shaft bearings. Both contain significant chromium and vanadium, but AISI M-50 contains molybdenum, whereas 18-4-1 contains tungsten.

Current aircraft turbine engine manufacturers' material specifications demand a double vacuum melted (VIM-VAR, for vacuum induction melt, vacuum arc remelt) AISI M-50 steel for mainshaft bearings. With this material, ball bearing fatigue lives of nearly 100 times AFBMA predicted life have been obtained [40]. Reduction in inclusion content, trace elements, and interstitial gas content due to double vacuum melting is considered responsible for a major portion of this life advancement.

Rolling-element fatigue tests were conducted in the five-ball fatigue tester with VIM-VAR AISI M-50, electro-flux remelt (EFR) 18-4-1, and VAR 18-4-1 [41]. Two groups of each material were subjected to slightly different heat treatments. Results and test conditions are shown in Fig. 11. Individual data points for each Weibull plot are given in [41].

A statistical comparison of the test results indicates that the rolling-element fatigue life of VIM-VAR AISI M-50 is not significantly different from EFR or VAR 18-4-1. The slight variations in heat-treatment procedures for each material resulted in statistically insignificant differences in rolling-element fatigue life. These results are contrary to what would have been predicted based the total percent of alloying elements as previously discussed. Since the 18-4-1 material had a higher alloy content it would have been expected to have a lower fatigue life. It is suggested by these data that the melting process may have a far stronger influence on rolling-element fatigue life than the material chemistry.

AMS 5749 steel combines the tempering, hot hardness, and hardness retention characteristics of AISI M-50 steel with the corrosion and oxidation resistance of AISI 440C stainless steel. The effects of double vacuum melting and retained austenite on the rolling-element fatigue life of AMS 5749 steel were determined in tests run in the five-ball fatigue tester [42,43]. Results and test conditions are shown in Fig. 12.

Double vacuum melting (VIM-VAR) produced an AMS 5749 material with a rolling-element fatigue life at least 14 times that for the same material with vacuum induction melting alone. The VIM-VAR AMS 5749 steel balls gave lives from 6 to 12 times greater than VIM-VAR AISI M-50 steel balls. Similar tests in the rolling-contact (RC) fatigue tester [43] showed no significant difference in the lives of the two materials. The difference between 5-ball results and RC rig results are not yet understood.

A material processing method commonly called ausforming has been studied in the five-ball fatigue tester. The ausforming process consists of an isothermal "warm working" operation performed while the material is in a metastable austenitic condition. The austenite is subsequently transformed in either the lower bainite or martensite transformation region of the time-temperature-transformation (TTT) curve. To apply the ausforming method, the steel must have a sluggish transformation behavior in the temperature range where "warm working" is to take place. AISI M-50 is such a steel.

The five-ball fatigue tester was used to determine the effect of ausforming on the rolling-element fatigue life of 11.1-mm (7/16-in.) diameter consumable-electrode vacuum melted (CVM) AISI M-50 steel balls [44]. One group of ausformed balls and two groups of CVM processed AISI M-50 steel balls were tested at a maximum Hertz stress of 5.52 GPa (800,000 psi), a shaft speed of 10,000 rpm, and a contact angle of 23° with no heat added. A paraffinic mineral oil was used as the lubricant.

The ausformed balls were fabricated from AISI M-50 bar material which was extruded to a cross-sectional area 20 percent of the original area (80 percent reduction in cross-sectional area) while the material was in a metastable austenitic condition. The ten-percent fatigue life of this group of ausformed balls was three and four times that of two groups of CVM processed AISI M-50 balls.

Ausforming results in a reduction in size and a more uniform distribution of carbide particles compared to conventionally processed AISI M-50. Thus, the smaller more uniformly distributed carbide particles could have accounted for the longer fatigue life by lessening the severity of dislocation pileups which cause stress concentrations and accelerate crack initiation and propagation. It is believed that the "warm work" imparted to the material during the ausforming process provided more numerous nucleation sites which accounted for the relatively uniform carbide precipitation. The added strain energy also speeded up the time-dependent precipitation process which was apparently completed before cooling to room temperature.

Unpublished data from NASA tests performed with VIM-VAR AISI M-50 120-mm bore angular-contact ball bearings made by ausforming showed that forging laps were induced in the raceways of the bearing because of the relatively low forging temperature. These forging laps acted as nuclei for fatigue spalls in a rather short period of time. The conclusion reached is that ausforming can result in improved rolling-element fatigue life, but problems with forging, particularly large, massive parts, and the costs thereof far exceeded its benefits.

Nonferrous Materials

In the late 1950s and early 1960s it was anticipated that aerospace technology would require bearings to operate reliably at temperatures between 644 and 1366 K (700° and 2000° F). Since this temperature range is beyond the range in which ferrous bearing materials are capable of operating for any significant length of time, refractory materials such as ceramics and cermets were considered.

The first nonferrous material studied in the five-ball fatigue tester was a crystallized glass ceramic (Pyroceram 9608) [7,10]. Rolling-element fatigue spalls similar to those observed for bearing steel balls were produced but the room-temperature load carrying capacity was found to be approximately one-fifteenth that of a group of previously tested AISI M-1 bearing steel balls. Tests at maximum Hertz stresses of 1.83, 2.03, and 2.28 GPa (265,000, 295,000, and 330,000 psi) and a speed of 2430 rpm showed fatigue life of the ceramic varied inversely with approximately the 11th to 13th power of stress.

The rate of fatigue spall development was much slower than that which is normal for steel balls. An intentional overrun for a period several times that which produced the initial fatigue spalling did not produce a general breakdown of the test surface. The scatter between short- and long-lived specimens was much lower for the crystallized glass ceramic than is normal for bearing steels (higher Weibull slope). Since the ceramic does not have appreciable foreign matter comparable to nonmetallic inclusions in steel, this low scatter indicates that the degree of structural homogeneity may be an important factor in life scatter of bearing materials.

Variation in contact angle, which produces variations in relative sliding of the contacting surfaces, did not produce any significant differences in fatigue life of specimens run at the same contact load (Hertz stress). The fatigue life at 644 K (700° F) was approximately one-third of that observed at room temperature. This may not indicate a thermal effect in the material itself since variation in lubricant viscosity over this temperature range could produce this difference in life. These tests indicated that the crystallized glass ceramic may be useful in low-load, short-duration applications where an operating temperature above the limits of steels is the paramount design consideration.

Subsequent to the tests with the crystallized glass ceramic, tests were conducted in the five-ball fatigue tester with four other refractory materials: hot-pressed alumina, cold-pressed-and-sintered alumina, self-bonded silicon carbide, and nickel-bonded titanium carbide cermet [45-47]. Results and test conditions are shown in Fig. 13. Support balls were AISI 52100 and AISI M-50 steel in the 300 and 644 K (80 and the 700 F) tests, respectively. Preliminary tests were also performed with each of the four refractory materials at higher temperatures from 866 to 1366 K (1100 to 2000 F). For these tests, hot-pressed alumina support balls were used with molybdenum disulfide lubrication in a modified five-ball tester (Fig. 1(f)).

The failure pits in all four materials were shallow, eroded areas, apparently of surface origin and were unlike fatigue pits found in bearing steels. Progression of an incipient failure for all four materials was a slow process that frequently consumed one half of the total running time of the specimen.

The lives of hot-pressed alumina and nickel-bonded titanium carbide cermet varied inversely with stress to an average power of approximately 10. However, the cold-pressed-and-sintered alumina and the self-bonded silicon carbide were less sensitive and exhibited an average stress-life exponent ranging from 7 to 8.

The capacity at 300 K (80 F) of hot-pressed alumina was about 7 percent that of a typical bearing steel but was the highest of the four materials studied. Preliminary tests at elevated temperatures indicated that hot-pressed alumina is capable of rolling-contact operation at temperatures up to 1366 K (2000 F) without gross wear or plastic deformation.

It was concluded in [47] that the ability of a ceramic or cermet material to be functional in rolling-contact applications appears to be related to its physical properties and surface condition. Consequently, surface finish appears to be an important criterion for long life. Since surface finish was related to the amount of porosity or to the presence of a weak second phase, such conditions should be avoided for proposed high-temperature bearing materials.

Hot-pressed silicon nitride has a low specific gravity (41 percent that of bearing steel) and has a potential application as low mass balls for very high-speed ball bearings. Hot-pressed silicon nitride balls were tested under rolling-contact conditions in the five-ball fatigue tester [48-50]. Test conditions were maximum Hertz stresses of 4.27 and 5.52 GPa (620,000 and 800,000 psi), a race temperature of 328 K (130° F), a speed of 9400 rpm, and a super-refined naphthenic mineral oil as the lubricant. Fatigue lives were compared with those for typical bearing steels AISI 52100 and AISI M-50. A analytical program was used to predict the dynamic performance characteristics and fatigue life of high-speed ball bearings with silicon nitride balls relative to that of bearings containing steel balls.

Extrapolation of the experimental results to contact loads which result in stress levels typical of those in rolling-element bearing applications indicated that hot-pressed silicon nitride running against steel may be expected to yield fatigue lives comparable to or greater than those of bearing quality steel running against steel. The fatigue spalls on the silicon nitride balls were similar in appearance to those observed in tests with typical bearing steels. The fatigue life with the hot-pressed silicon nitride is considerably greater than that of any other ceramic or cermet tested.

A digital computer analysis indicates that there is no improvement in the lives of 120-millimeter-bore angular-contact ball bearings of the same geometry operating at DN values from 2 to 4 million where hot-pressed silicon nitride balls are used in place of steel balls. The higher modulus of elasticity of silicon nitride tends to offset the benefits of its lower density. The principal value of silicon nitride may lie in bearings in marginally lubricated corrosive environments rather than in extensive speed applications.

Hollow Rolling Elements

In the mid 1960's analyses indicated that bearings operating at DN values (bearing bore in mm times shaft speed in rpm) above 1.5 million, would experience reduced life because of high centrifugal forces and increased Hertz stresses at the outer-race/ball contacts. At that time engine designers anticipated turbine bearing speeds to increase to the range of 2.5 to 3 million DN after 1980, with consequent significant decreases in fatigue life.

A possible solution to the problem caused by ball centrifugal force was the use of hollow balls in the bearing. Obviously, removal of a large percentage of the mass from the ball will reduce the centrifugal force.

The five-ball fatigue tester was used to determine the rolling-element fatigue lives of solid 12.7-mm (1/2-in.) diameter balls [51,52] and hollow balls with 2.54 mm (0.100 in.) wall thickness (21.7 percent from that of a solid ball) [51,52]. The upper test balls fabricated from consumable-electrode vacuum melt (CVM) AISI 52100 steel were run against solid AISI 52100 steel lower support balls. Results and test conditions are shown in Fig. 14.

The hollow balls were fabricated by a technique that included rough-forming hemispherical shells, joining them together by electron-beam welding, heat treating, and finishing to an Anti-Friction Bearing Manufacturers Association (AFBMA) 10 specification, comparable to that of the solid balls were fatigue tested to obtain comparative data. The results of Fig. 14 indicate that the differences in fatigue life between the two groups of balls are not statistically significant.

The probability of a fatigue spall occurring in the weld area was calculated as 6.5 percent for those tests in which the running track crossed the weld. One of the ten fatigue spalls that occurred on the hollow balls was in the weld. This spall was a classical subsurface type similar in appearance to those occurring in the area outside of the weld and in the conventional solid balls.

Only minor differences in hardness were measured on a section of the hollow ball wall cut through a diameter normal to the weld. Similar metallographic structures were observed on the weld zone and the parent material. These results indicated extremely good control during the manufacturing process, and that the presence of a weld zone should not be detrimental to the rolling-element fatigue life of the balls.

To explore the effects of higher percentages of weight reduction additional testing was conducted with CVM AISI M-50 17.5-mm (11/16-in.) diameter balls having a wall thickness of 1.52 mm (0.060 in.) [52,53]. This represents a weight reduction of 56.5 percent from that of a solid ball. These results are shown in Fig. 15. The ten-percent life obtained with these hollow balls was approximately one-fifth that for a comparable solid ball.

Seven of these hollow balls were electropolished to determine the orientation of the weld. On each of six failed balls, the running track crossed the weld. The fatigue spall on five of these six balls was adjacent to or directly on the weld. The spall on the other ball was clearly away from the weld area. The running track on the seventh ball did not cross the weld. This ball had no fatigue spall after 300 hours (530 million stress cycles) running time.

Four of the failed balls with the spall adjacent to the weld were sectioned parallel to and through the running track. In each ball, a crack was found that passed through the weld area and connected the outside surface spall to the inside surface.

From the width, shape, and pattern of these cracks, it was concluded that they initiated at the inner surface of the ball and propagated to the outer surface, resulting in a spall that resembles those of classical subsurface rolling-element fatigue. It was also concluded that these hollow balls from the fatigue tests had probably failed in flexure due to a stress concentration in the region of the weld. The stress concentration appeared to be at the notch formed at the edge of the weld.

The ball with the spall away from the weld area was also sectioned through the spall parallel to the running track. No cracks could be seen connecting the spall and the inner surface of the ball. This failure was apparently not a result of flexure of the wall and most probably was classical subsurface fatigue.

The 12.7-mm (1/2-in.) diameter AISI 52100 steel hollow balls, with a 21.7 percent weight reduction compared satisfactorily with solid balls in fatigue tests. However, the 17.5-mm (11/16-in.) diameter AISI M-50 steel hollow balls with a 56.5 percent weight reduction did not compare very well with solid balls in fatigue tests. The fact that the balls were of different material and size should not have made any significant difference in the fatigue comparisons. It is probable that a weight reduction of 56.5 percent results in a wall that is merely too thin for use in a bearing ball.

Additional experiments were conducted [54] to compare the rolling-element fatigue life of electron-beam-welded hollow balls with 1.78 mm (0.070-in.) wall thickness with previously run solid balls and 1.52-mm

(0.060-in.) wall thickness hollow balls. These balls, also made from AISI M-50 steel, had a weight reduction of 50 percent.

The fatigue life of these hollow balls, determined in the five-ball fatigue tester at a maximum Hertz stress of 4.83 GPa (700,000 psi) was not significantly less than that of the previously run solid balls of the same material. The ball failures on the hollow balls with the 50 percent weight reduction resulted from classical subsurface fatigue and did not occur in the weld area. However, subsequent tests [53,54] with 215-series bearings with hollow balls with both the 56.5 and the 50 percent weight reductions had ball failures attributed to flexure fatigue. These bearings were not heavily loaded (thrust load of 2240 N (500 lb)), but the normal ball load was more than twice that of the five-ball test. It is concluded that hollow balls with a weight reduction of 50 percent or greater have a wall that is too thin for use in bearings for aircraft turbine engines.

A number of additional problems exist with hollow balls for use in bearings that are inherent in the manufacture of the ball. Two signficant problems are: (a) difficulty in maintaining uniform wall thickness and (b) controlling the weld penetration around the periphery of the ball. Unless these problems are overcome, hollow balls will have an unbalance and/or a slightly different stiffness at the weld. Under dynamic conditions, these factors could adversely affect the life of the ball.

Stress-Life Relation

Four groups of 12.7-mm-(1/2-in.-) diameter balls were tested, each at a level of maximum Hertz stress in the range of 4.48 to 6.03 GPa (650,000 to 875,000 psi) [55,56]. 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 results of these tests are shown in Fig. 16. The 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 [55,56] 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. 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 [56].

Ferrographic analysis

A Ferrographic analysis was used to determine the types and quantities of wear debris generated during accelerated rolling-element fatigue tests in the five-ball fatigue tester [57,58]. Lubricant samples were taken from tests previously reported in [42,43]. Ball specimens were made of AMS 5749, a corrosion resistant, high-temperature bearing steel. The lubricant was a super-refined naphthenic mineral oil. Conditions included a maximum Hertz stress of 5.52 GPa (800,000 psi) and a shaft speed of 10,000 rpm. Four types of debris were observed: normal rubbing wear particles, fatigue microspall particles, spheres, and friction polymer deposits. For about half of the tests, the fatigue spall particle rating and the number of spherical particles reached a maximum during the last one-third of the test durations. The number of spheres observed in these accelerated fatigue tests was much less than others have reported during long duration testing under lower loads. Laminar particles, sometimes associated with rolling contact fatigue, were not observed in this study. The characterization of wear debris as a function of time was of limited use in predicting fatigue failures in these accelerated tests.

Correlation with Full-Scale Bearings

A prime purpose of accelerated rolling-element fatigue testing is to study phenomena which occur in rolling-element bearings and in a relative short time period at reduced cost. Ultimately, however, the various trends and discoveries which are recognized from these tests must be incorporated in and validated with full-scale bearings.

While it may be difficult, although not impossible, to predict actual bearing life [59] from them, or based on them, bench-type fatigue tests, such as those run in the NASA five-ball fatigue tester, can rank material, lubricant, or operating variables on a relative life basis [5,14-18,36]. This is illustrated in Fig. 6 which is a comparsion of the relative load capacity in the five-ball tester to that obtained with 7208-size bearings where lubricant type is the variable [16]. In Fig. 17, there is shown a comparison of relative load capacity obtained in the five-ball system to that obtained with the 207-size bearings where ball hardness is the variable [18]. As can be seen for both the material hardness and lubricant variables, the correlation of the five-ball tester results with full scale bearings on a relative basis is excellent. For both sizes of bearings, the maximum stress level was approximately 2.41 GPa (350,000 psi) at the inner race-ball contact as compared to 5.52 GPa (800,000 psi) for the five-ball tester.

Additional work was performed with 120-mm bore ball bearings made from AISI M-1, AISI M-50 and WB-49 [36,60]. The bearings were tested at an outer-race temperature of 589 K (600° F). The ten-percent lives of the AISI M-50 and AISI M-1 bearings exceeded the calculated AFBMA life by factors of 13 and 6, respectively. The bearings with WB-49 races showed lives less than AFBMA life.

The magnitudes of the life differences seen in these bearing tests at 589 K (600° F) correlate well with the five-ball fatigue test results. Relative 10-percent fatigue lives are shown in Fig. 18. Using the AISI M-50 ten-percent life as a comparison, the AISI M-1 data for the five-ball fatigue tests and the bearing tests agree remarkably well. WB-49 and AISI M-42, both alloys containing relatively high percentages of cobalt and similar in microstructure, show reasonably good agreement between the five-ball fatigue tests and the bearing tests.

These examples of excellent correlation of five-ball fatigue tester data with full-scale bearing fatigue data show that this tester can reliably identify qualitative effects of many variables on rolling-element fatigue life. As an accelerated test, it does so in a timely and relatively inexpensive manner. As such it has had a broad and significant effect on full-scale bearing research and, thus, on present bearing state-of-the-art.

SUMMARY OF RESULTS

The NASA five-ball fatigue tester was conceived by W. J. Anderson in late 1958. The first data were generated in March 1959. Since then a total of approximately 500,000 test hours have been accumulated on a group of eight test rigs which are capable of running 24 hours a day, 7 days a week. Studies have been conducted on the effect on rolling-element fatigue life of contact angle, material hardness, chemistry, heat treatment and processing, lubricant type and chemistry, elastohydrodynamic film thickness, deformation and wear, vacuum, and temperature as well as Hertzian and residual stresses. From a survey of all of the reports of tests using the five-ball fatigue tester the following results were obtained.

1. A relationship was established between component hardness and fatigue life. Maximum bearing fatigue life can be achieved where the rolling elements of the bearing are one to two points (Rockwell C hardness) (ΔH) greater than the races.

2. A relation was established between subsurface compressive residual stress and rolling-element fatigue life. The value of the maximum shear stress was effectively reduced by the presence of the compressive residual stress thus increasing the life.

3. An interrelation was established between lubricant type, elastohydrodynamic film thickness and rolling-element fatigue life.

4. There was no apparent effect of a reduced pressure environment on rolling-element fatigue.

5. The operating characteristics and rolling-element fatigue life of fluorinated polyether fluids at cryogenic temperatures compared favorably with those of a super-refined mineral oil at room temperature.

6. Rolling-element fatigue lives obtained with traction fluids were not significantly different on a statistical basis from those obtained with a tetraester oil or a paraffinic mineral oil.

7. Lubricant additives can reduce rolling-element fatigue life below that obtained with the lubricant base stock without additives. However, the additives used with the base oil did not change the life ranking of bearing steels in those tests where rolling-element fatigue was of subsurface origin.

8. There was a decrease in fatigue life with increased contact angle (ball spin). The decrease in fatigue life was attributed primarily to decreased elastohydrodynamic film thickness caused by increased contact temperature.

9. A relation was established between material alloying element and rolling-element fatigue life. This related to a carbide factor which was statistically derived which in turn related rolling-element fatigue life to the total number of residual carbide particles per unit area, median residual carbide size, and percent residual carbide area.

10. Double vacuum melting had a far greater effect on improving rolling-element fatigue life than material chemistry.

11. Ausforming improved the fatigue life of AISI M-50 steel balls over conventionally forged balls of the same heat of material. However, for full-scale bearings, the cost of ausforming exceeded its benefits.

12. The use of nonferrous materials at temperatures to 1366 K (2000° F) was established. However, the lives obtained were significantly less than those obtained at room temperature with conventional bearing steels.

13. The rolling-element fatigue life of hot-pressed silicon nitride material was found to be considerably greater than that of any other ceramic or cermet material tested.

14. The primary mode of failure for hollow rolling elements was by flexure fatigue. Where failure was by classical subsurface rolling-element fatigue, there was no statistical difference in life between the hollow and solid balls.

15. The fatigue lives at four stress levels followed the relation that fatigue life is inversely proportional to maximum Hertz stress to the power of 12. This would suggest that 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.

16. The characterization of wear debris as a function of time using ferrographic analysis was of limited use in predicting fatigue failures in accelerated rolling-element fatigue tests.

17. A qualitative correlation was established between the rolling-element fatigue results in the five-ball fatigue tester and full-scale bearing tests.

18. The use of the five-ball fatigue tester has been established as a reliable, relatively inexpensive and timely method of determining qualitative effects on rolling-element fatigue life.

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TABLE 1. - ROLLING-ELEMENT FATIGUE LIFE OF AISI 52100 BALLS LUBRICATED WITH ONE OF SEVERAL SYNTHETIC LUBRICANTS OR A PARAFFINIC MINERAL OIL IN THE FIVE-BALL FATIGUE TESTER [33]

[Maximum Hertz stress, 5.52 GPa (800,000 psi); shaft speed, 10 700 rpm; contact angle, 30°.]

Lubricant	Rolling-element fatigue life, millions of upper-ball stress cycles		Weibull slope	Failure index[a]	Confidence number,[b] percent	
	10-percent life	50-percent life			10-percent life	50-percent life
Paraffinic mineral oil	34.8	214	1.04	15 out of 37	--	---
Formulated tetraester	14.3	71.1	1.17	35 out of 39	83	>99
Tetraester base	10.5	46.2	1.27	38 out of 40	92	>99
Traction fluid 1	17.9	115.9	1.01	21 out of 32	69	94
Traction fluid 2	17.3	72.7	1.31	15 out of 28	72	>99
Synthetic paraffinic oil	9.5	91.9	.83	26 out of 36	87	>99

[a]Number of failure out of total number of tests.
[b]Percent of time that the fatigue life with the mineral oil will be greater than that with the specific synthetic lubricant.

TABLE 2. - ROLLING-ELEMENT FATIGUE LIFE OF STEEL BALLS WITH AN ACID-TREATED WHITE OIL WITH AND WITHOUT SURFACE-ACTIVE ADDITIVES [34]

[Maximum Hertz stress, 5.52 GPa (800,000 psi); shaft speed, 10 700 rpm; contact angle, 30°; race temperature, 339 K (150° F).]

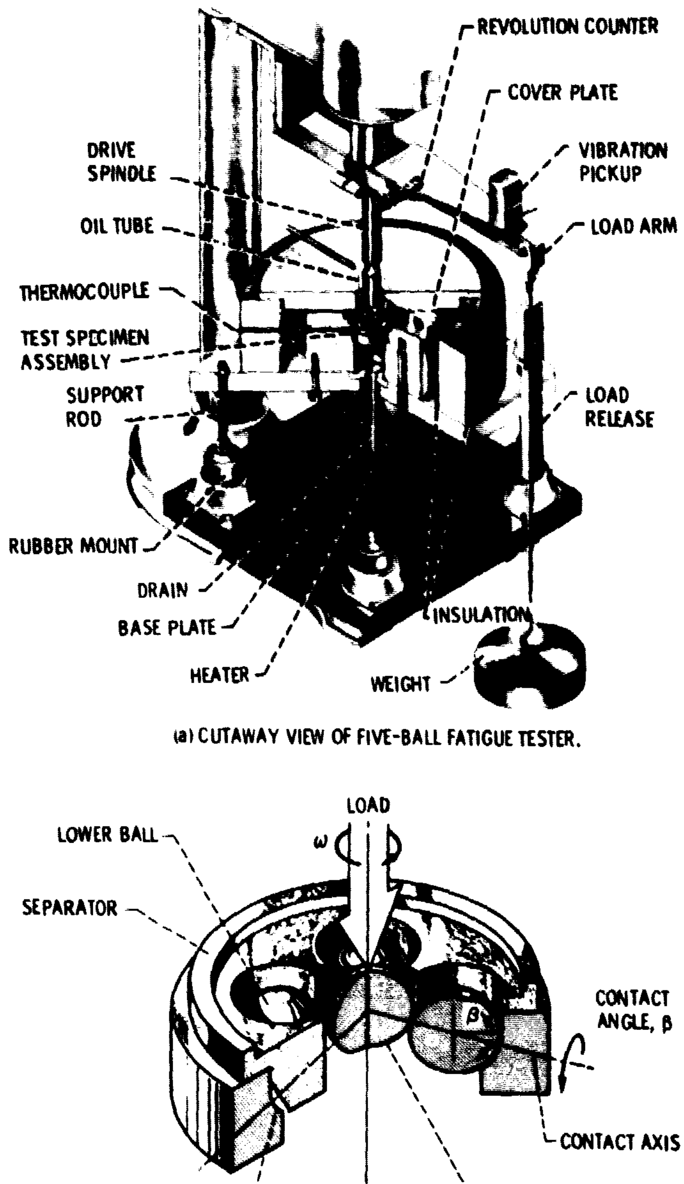
Material	Lubricant	Rolling-element fatigue life, millions of upper-ball stress cycles		Weibull slope	Failure index[a]	Confidence number,[b] percent
		L_{10}	L_{50}			
AISI 52100	Base oil	37.0	202	1.11	10 out of 30	--
	Base oil plus 2.5 percent sulfurized terpene	17.8	39.2	2.39	24 out of 30	81
	Base oil plus 1 percent didodecyl phosphite	33.9	92.8	1.87	25 out of 30	54
	Base oil plus 5 percent chlorinated wax	4.8	15.1	1.63	30 out of 30	99
AISI M-50	Base oil	14.6	44.5	1.69	28 out of 30	--
	Base oil plus 2.5 percent sulfurized terpene	15.9	34.1	2.47	30 out of 31	59
AISI 4118	Base oil	7.8	25.0	1.61	30 out of 30	--
	Base oil plus 2.5 percent sulfurized terpene	4.2	26.7	1.02	29 out of 30	83
	Base oil plus 1 percent didodecyl phosphite	6.4	24.4	1.40	30 out of 30	65

[a]Indicates number of failures out of total number of tests.
[b]Percentage of time that the 10-percent life with the base oil will be greater than (or less than, as the case may be) the 10-percent life with the additive.

TABLE 3. - FATIGUE RESULTS WITH GROUPS OF ONE-HALF INCH DIAM BALLS RUN IN FIVE-BALL FATIGUE TESTERS. MAXIMUM HERTZ STRESS, 5.52 GPa (800,000 psi), CONTACT ANGLE, 30 deg, SHAFT SPEED, 10,300 rpm; RACE TEMPERATURE, 339 K (150° F) [39]

Material[f]	Heat treatment lot	Average hardness, R_c	Retained austenite, volume percent[a]	Austenitic grain size[b]	Cleanliness rating[c]		Life, millions of upper ball stress cycles		Slope	Failure index[e]	Carbide parameter, C'
					Class[d]	Type	B_{10}	B_{50}			
52100	A	62.5	4.90	13	B1	Heavy	18.5	114	1.04	22 out of 29	1.22
	B	62.0	4.10	13	D1	Thin	30.1	130	1.29	22 out of 29	1.17
	C	62.5	.80	13	D1	Thin	12.9	84	1.00	19 out of 25	.76
	Combined	----	----	--	--	-----	21.2	109	1.15	63 out of 83	1.00
Halmo	A	60.8	0.60	8	D2	Heavy	22.0	74	1.56	25 out of 30	0.58
	B	60.8	1.00	8	D1	Heavy	12.9	57	1.26	28 out of 30	.64
	C	61.1	1.70	8	D2	Heavy	12.9	68	1.14	26 out of 30	.69
	Combined	----	----	--	--	-----	16.4	66	1.35	79 out of 90	.63
M-50	A	62.6	1.90	10	B1	Heavy	15.3	35	2.29	29 out of 29	0.60
	B	62.2	2.90	9	D2	Heavy	12.3	36	1.73	24 out of 30	.54
	C	62.3	1.50	10	D1	Heavy	14.2	48	1.55	26 out of 29	.50
	Combined	----	----	--	--	-----	14.4	33	1.89	79 out of 88	.56
M-10	A	62.2	1.10	9	D3	Heavy	19.4	65	1.56	26 out of 30	0.36
	B	62.0	2.40	6	D2	Heavy	8.3	46	1.11	30 out of 30	.39
	C	61.6	1.60	6	D1	Thin	13.1	42	1.62	[illegible] out of 30	.35
	Combined	----	----	--	--	-----	13.7	50	1.40	85 out of 90	.37
T-1	A	61.4	7.30	11	B1	Heavy	6.6	50	0.92	26 out of 30	0.26
	B	61.4	5.20	9	D1	Thin	8.4	59	.97	26 out of 30	.33
	C	61.0	9.50	10	D1	Heavy	8.5	74	.87	23 out of 30	.30
	Combined	----	----	--	--	-----	8.6	59	.98	75 out of 90	.29
M-1	A	63.3	[illegible]	10	B2	Heavy	8.2	43	1.13	29 out of 30	0.32
	B	63.4	3.30	9	A1	Heavy	5.9	37	1.02	29 out of 30	.34
	C	63.5	1.00	8	A2	Heavy	6.7	33	1.18	29 out of 29	.38
	Combined	----	----	--	--	-----	7.6	38	1.18	87 out of 89	.33
M-2	A	63.4	1.70	6	B1	Heavy	5.8	35	1.05	28 out of 30	0.36
	B	63.4	2.40	10	D1	Thin	5.5	26	1.23	28 out of 29	.36
	C	63.4	2.30	9	D1	Heavy	4.2	31	.95	29 out of 30	.35
	Combined	----	----	--	--	-----	5.7	30	1.13	85 out of 89	.36
M-42	A	61.8	1.00	9	A1	Thin	1.0	6.6	0.97	30 out of 30	0.07
	B	61.3	4.40	10	D1	Heavy	1.7	8.9	1.12	27 out of 30	.07
	C	61.3	4.90	6	D1	Heavy	1.4	5.8	1.33	30 out of 30	.07
	Combined	----	----	--	--	-----	1.4	7.0	1.18	87 out of 90	.07

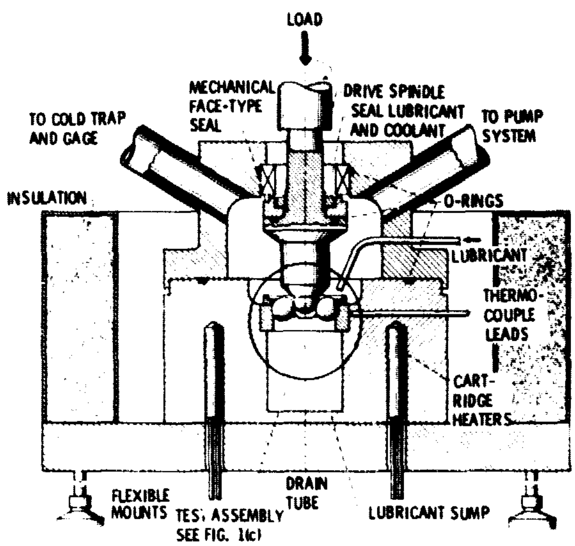
[a]Determined from the integrated peak intensities of (220)γ and (200)α planes.
[b]ASTM E 112-63.
[c]ASTM E45 63, Method A (table shows predominate inclusion class and type).
[d]Inclusion classes: A-Sulfides, B-Alumina, C-Silicates, D-Globular oxides.
[e]Indicates number of failures out of total number of tests.
[f]Defined in [39].



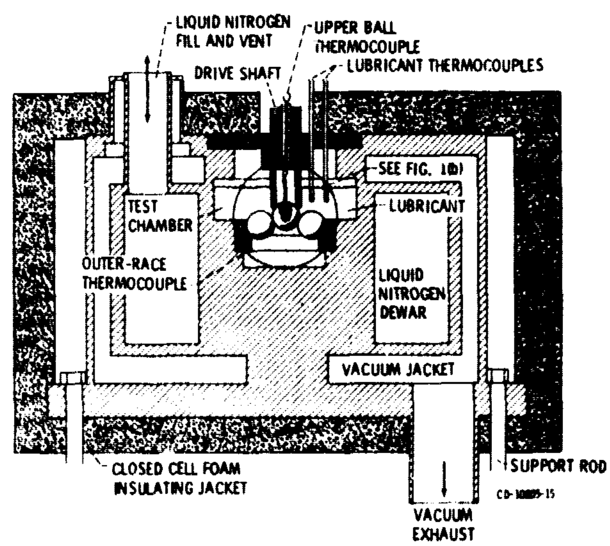
(a) CUTAWAY VIEW OF FIVE-BALL FATIGUE TESTER.

(b) FIVE-BALL ASSEMBLY.

Figure 1. - Five-ball fatigue tester.

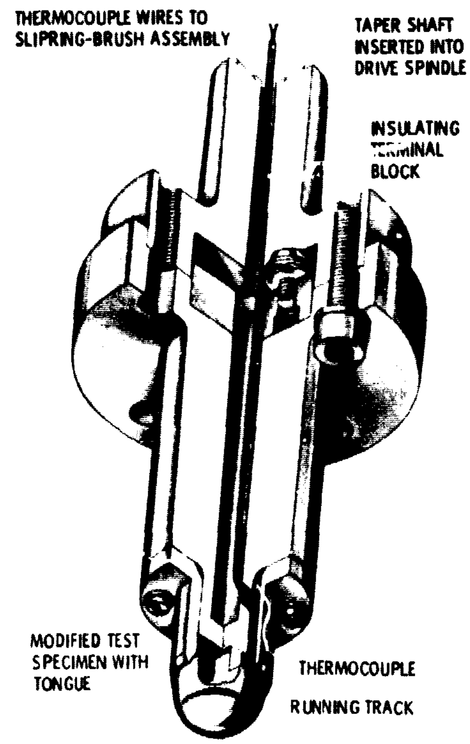


(c) Modification for tests at reduced pressures.

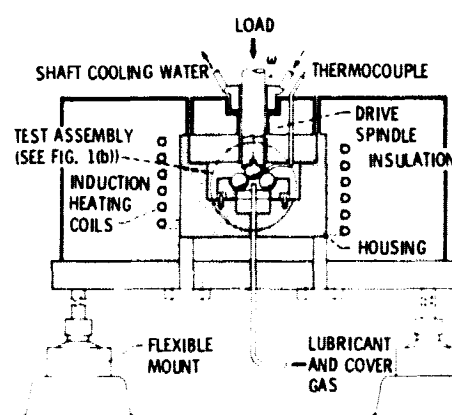


(d) Modification for low temperature and cryogenic testing.

Figure 1. - Continued.



(e) Operating-temperature measuring device.



(f) Modification for tests at 1366 K (2000° F).

Figure 1. - Concluded.

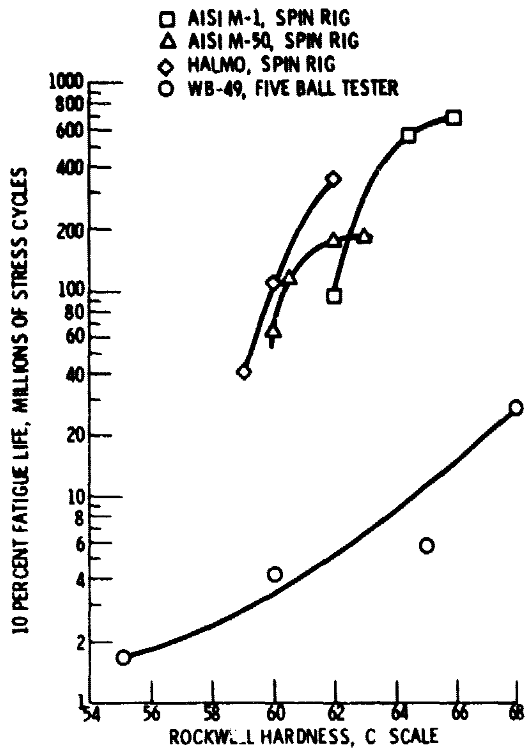


Figure 2. - Ten percent fatigue life as a function of hardness for four tool steels [9].

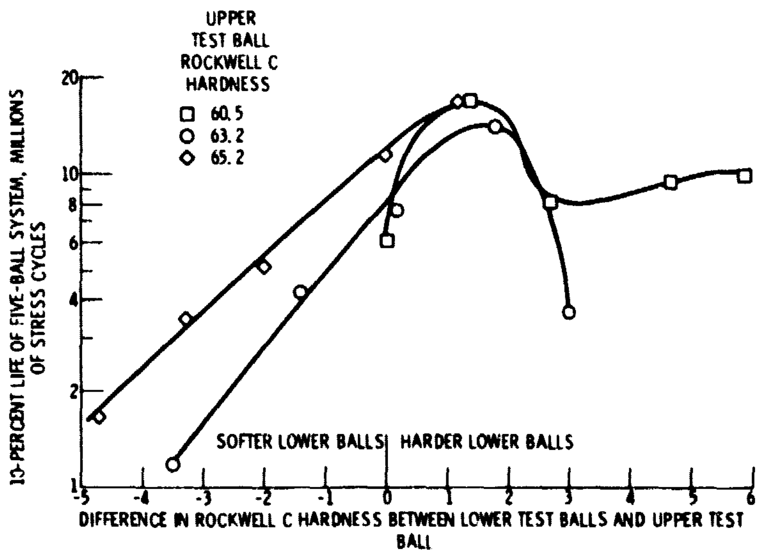


Figure 3. - Ten-percent life of five-ball system as a function of difference in hardness between lower test balls and upper test ball with lower and upper balls of AISI 52100 steel [17].

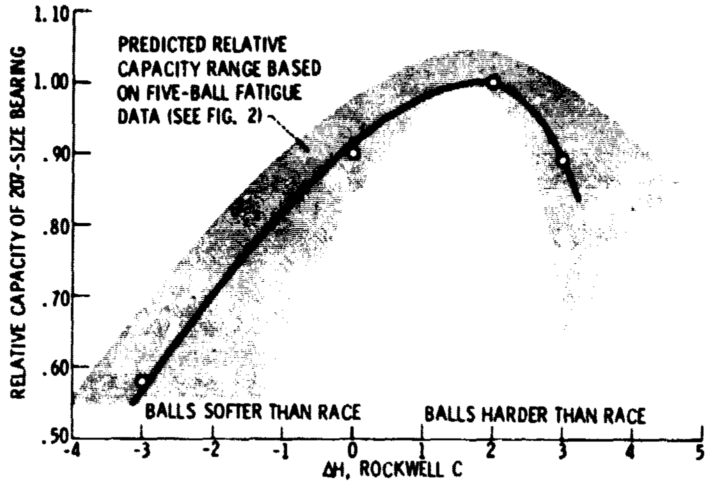


Figure 4. - Relative radial load-carrying capacity of 207-size deep groove ball bearing as a function of difference in hardness between balls and races [18].

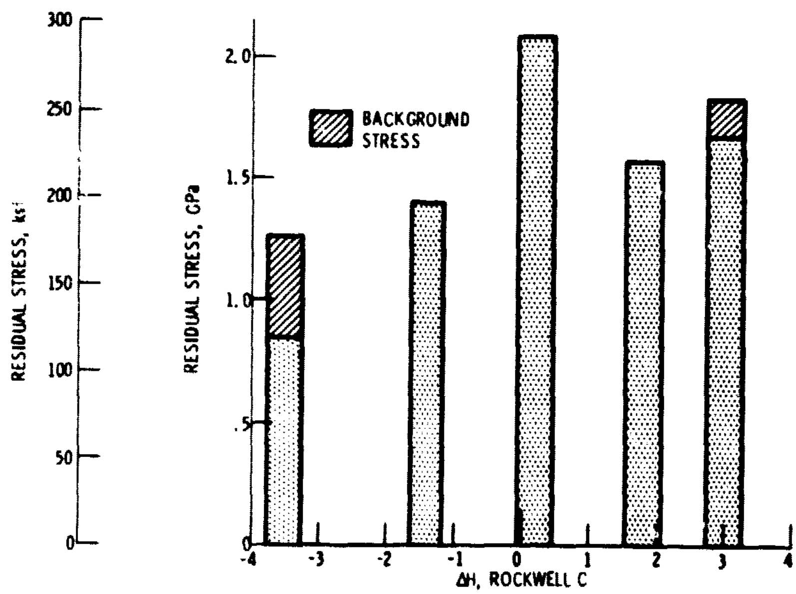


Figure 5. - Measured compressive residual stress in track of upper test balls having Rockwell C hardness of 63.2 as a function of difference in hardness between lower test balls and upper test ball [18].

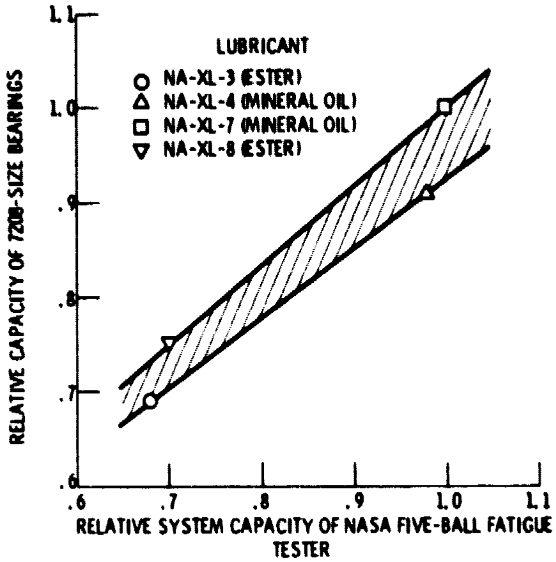


Figure 6. - Relative load-carrying capacity of 7208-size bearings as function of relative capacity of five-ball fatigue tester for various lubricants at 422 K (300° F) [16].

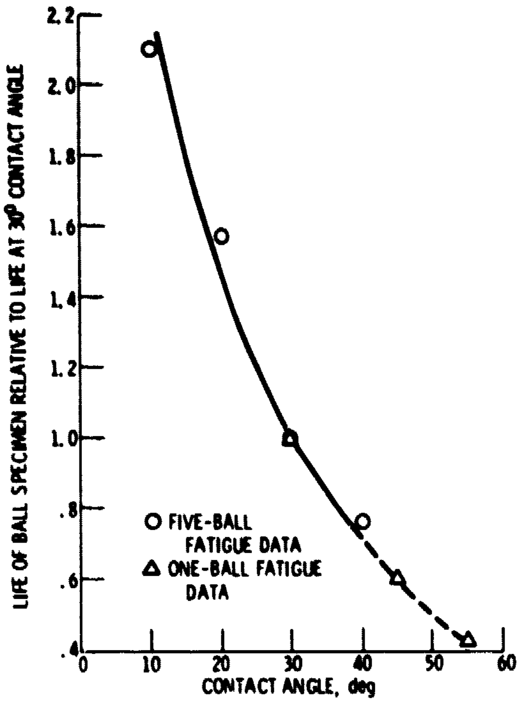


Figure 7. - Effect of contact angle on rolling-element fatigue for a constant Hertz stress [14].

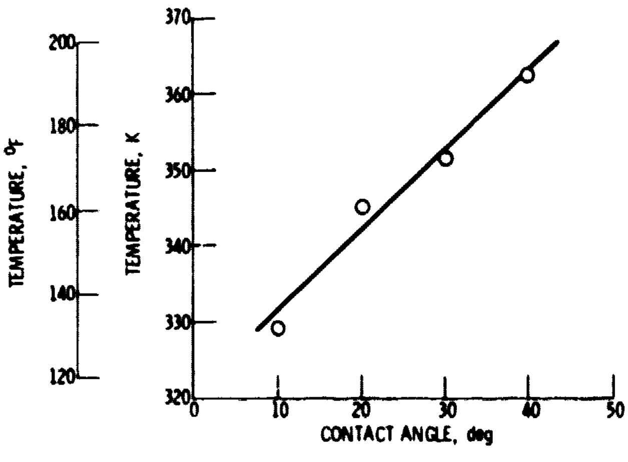


Figure 8. - Temperature at edge of contact area as a function of contact angle. Maximum Hertz stress, 5.52 GPa (800 000 psi); 10 000 rpm; five-ball fatigue tester [14].

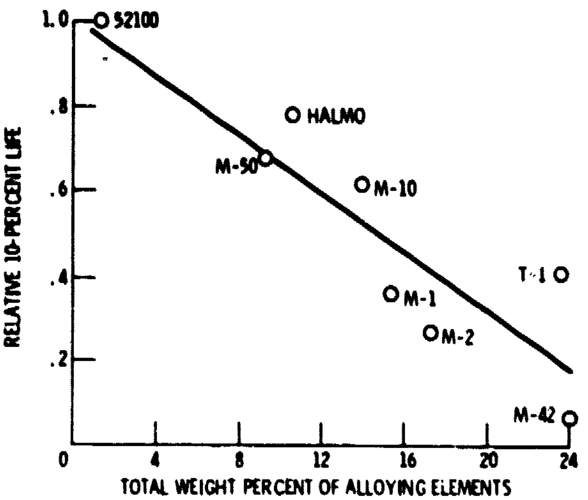


Figure 9. - Effect of total weight percent of alloying elements tungsten, chromium, vanadium, molybdenum, and cobalt on rolling-element fatigue life at 339 K (150° F) [36].

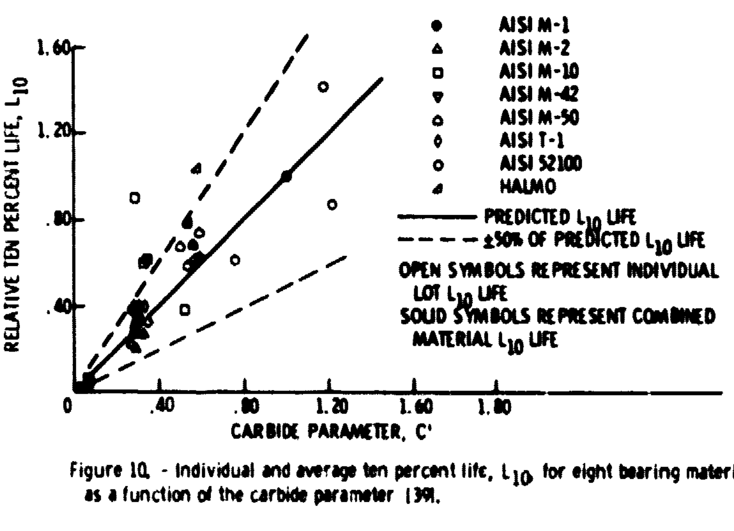


Figure 10. - Individual and average ten percent life, L_{10}, for eight bearing materials as a function of the carbide parameter [39].

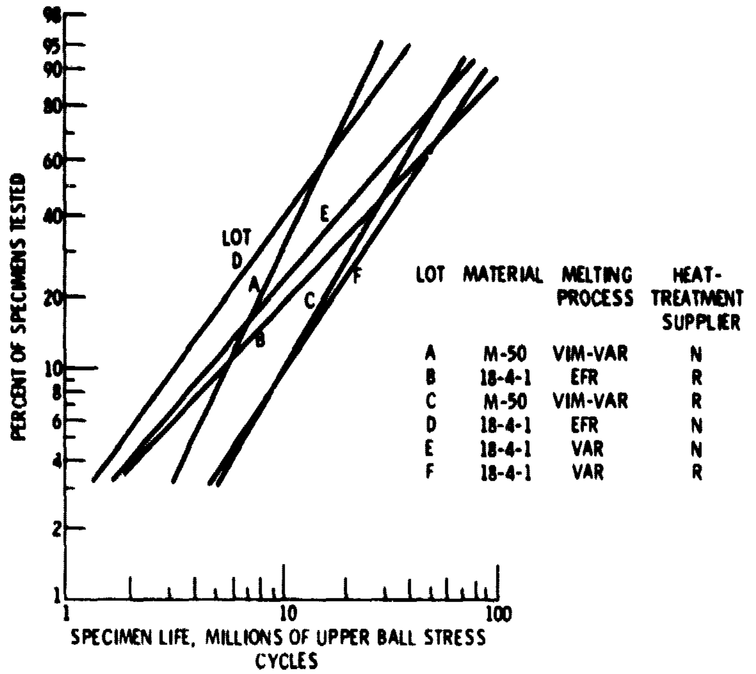


Figure 11. - Rolling-element fatigue life of VIM-VAR AISI M-50, EFR 18-4-1, and VAR 18-4-1 steel balls in five-ball fatigue tester. Maximum Hertz stress, 5520 megapascals (800 000 psi); shaft speed, 10 700 rpm; contact angle, 30°; temperature, 339 K (150° F). Type II ester (MIL-L-23699) lubricant [41].

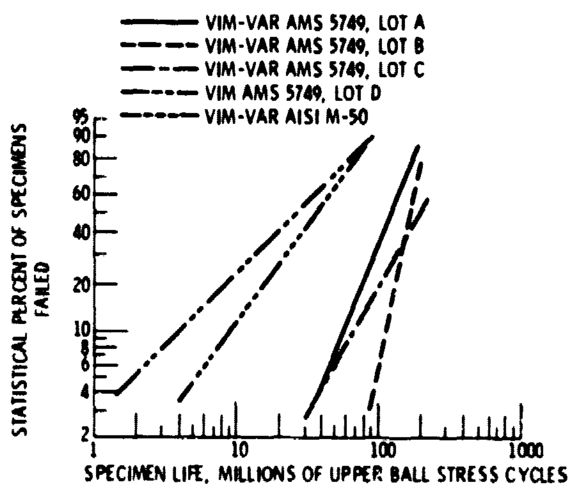


Figure 12. - Summary of rolling-element fatigue data with AMS 5749 and AISI M-50 balls in five-ball fatigue tester. Maximum Hertz stress 5.52 GPa (800 000 psi); shaft speed, 10 700 rpm; contact angle, 30°; temperature, 339 K (150° F). Super refined naphthenic mineral oil lubricant [42].

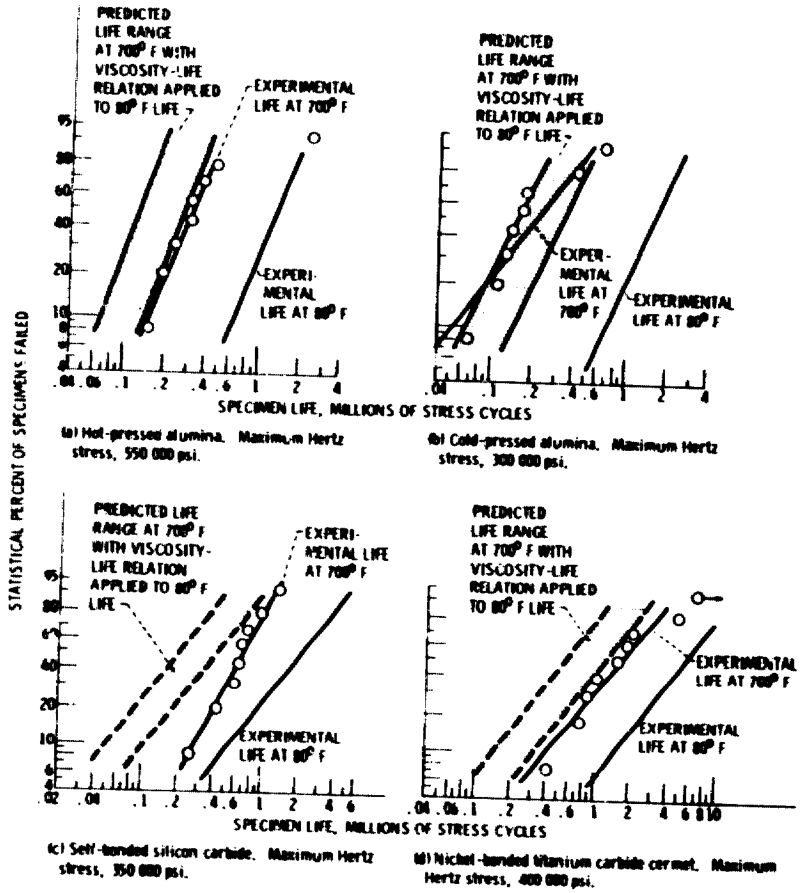


Figure 13. - Effect of 700° F race temperature on life of 12.7-mm, (1/2-in.) diameter ball specimens of four refractory materials in five-ball fatigue tester. Shaft speed, 950 rpm; contact angle, 20°; lubricant, highly refined naphthenic mineral oil; viscosity at 80° F, 150 cs; viscosity at 700° F, 0.6 cs; maximum Hertz stress, 1.72 to 4.48 GPa (250 000 to 650 000 psi) [47].

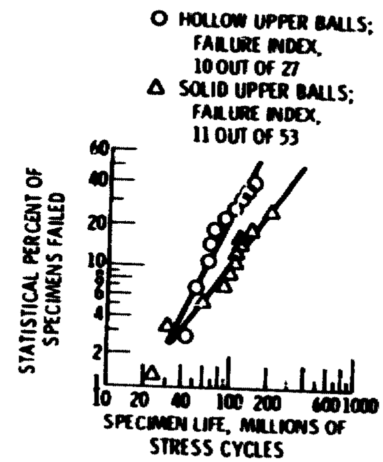


Figure 14. - Rolling element fatigue life of CVM AISI 52100 steel hollow and solid balls in the five-ball fatigue tester. Maximum Hertz stress, 5.52 GPa (800 000 psi), speed 10 600 rpm; no heat added; lubricant, super-refined naphthenic mineral oil. Contact angle, 20°.

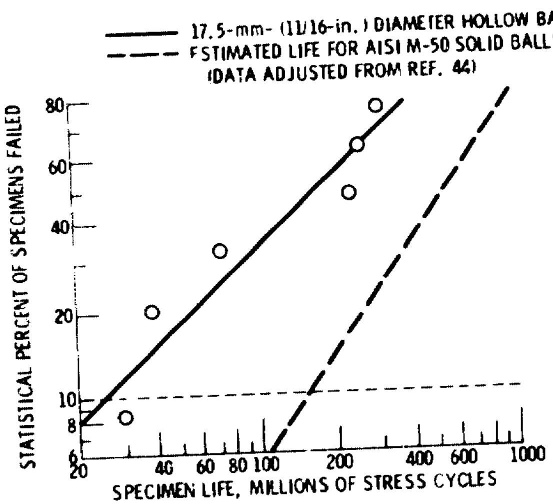


Figure 15. - Rolling-element fatigue life of CVM AISI M-50 steel balls in the five-ball fatigue tester. Maximum Hertz stress, 4.83 GPa (700 000 psi); speed, 10 600 rpm; no heat added; lubricant super-refined naphthenic mineral oil. Contact angle, 34.5°.

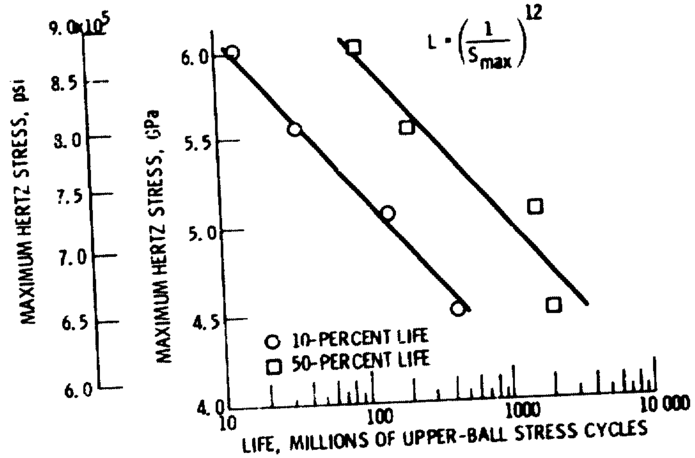


Figure 16. - Effect of maximum Hertz stress on life of vacuum-degassed AISI 52100 balls tested in five-ball fatigue tester [55].

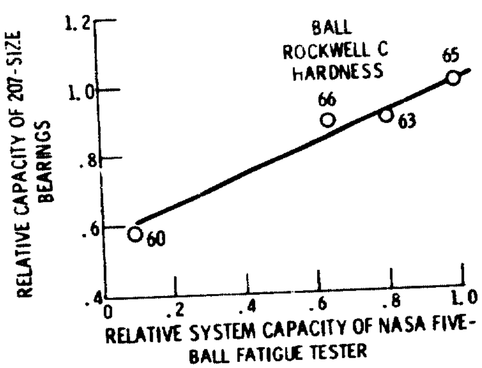


Figure 17. - Relative capacity of 207-size bearings as a function of relative capacity of five-ball fatigue tester for various ball hardness. Nominal hardness of upper ball (for five-ball tests) and inner race (for bearings), 63 Rockwell C [18].

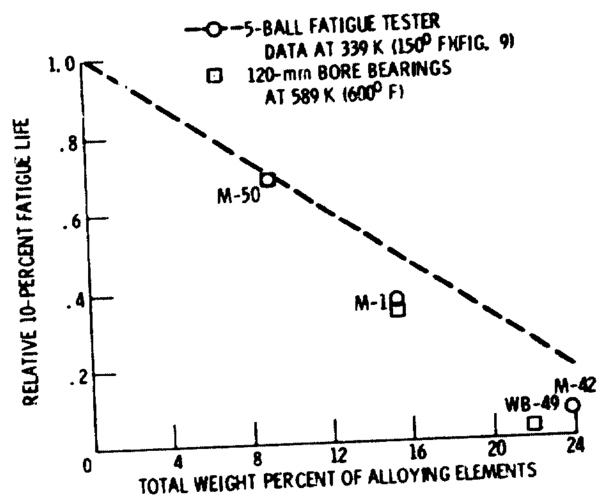


Figure 18. - Effect of total weight percent of alloying elements on fatigue life of 120-mm bore bearings at 589 K (600° F) [36].