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DEVELOPMENT OF HIGH-SPEED ROLLING-ELEMENT BEARINGS - A HISTORICAL AND TECHNICAL PERSPECTIVE

by

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ABSTRACT

Projections by the major United States aircraft turbine engine manufacturers in the late 1950's indicated that the trends in airbreathing turbojet engines would dictate ball and roller bearings operating at speeds to 3 million DN and temperatures to 589 K (600° F). Helicopter transmission development in the 1960's suggested the need for tapered-roller bearings to operate at speeds to 2 million DN. In 1959 NASA began a research program to encompass these projected temperature and speed requirements for large bore ball and roller bearings. In the 1970's NASA expanded its high-speed bearing program to encompass tapered-roller bearings as well as small-bore ball and roller bearings. Based on this NASA work, temperature capabilities of rolling-element bearings for aircraft engines have moved from 450 to 589 K (350° to 600° F) with increased reliability. High bearing speeds to 3 million DN can be achieved with a reliability exceeding that which was common in commercial aircraft. Capabilities of available bearing steels and lubricants were defined and established. Computer programs for the analysis and design of rolling-element bearings were developed and experimentally verified. The reported work is a summary of NASA contributions to high-performance engine and transmission bearing capabilities.

INTRODUCTION

Projections by the major United States aircraft turbine engine manufacturers in the late 1950's indicated that the trends in airbreathing turbojet engines would dictate ball and roller bearings operating at speeds to 3 million DN (where DN equals the bearing bore in millimeters multiplied by the shaft speed in revolutions per minute) and temperatures to 589K (600° F) for the 1970's and 1980's. These were operating speeds and temperatures for which engineering technology and lubricants did not exist. In February 1959 engineers of the NASA Lewis Research Center planned a bearing technology program to encompass these temperature and speed requirements.

It became of first order importance to determine which lubricants were capable of operating at elevated temperatures. The NASA engineers instituted a test program using the NASA-designed five-ball fatigue tester for this purpose. More than 30 potential high-temperature lubricants were evaluated, starting in February 1959.

Based on the results of this early NASA investigation, a program was instituted in the fall of 1962, with SKF Industries to test groups of 25-mm-bore angular-contact ball bearings using the best 11 lubricants from the previous five-ball tests. These bearings were tested at temperatures from 478 to 589 K (400 to 600° F) and at a speed of 43 000 rpm.

Concurrent with the lubricant test program at SKF Industries, an in-house program conducted by the NASA engineers was undertaken to evaluate the rolling-element fatigue life of bearing steels suitable for high-temperature operation. The materials tested were the potential high-temperature steels AISI M-50, M-1, M-2, M-10, T-1, Halmo, and WB-49, and the conventional bearing material AISI 52100, which was used as a reference.

NASA engineers also conducted an in-house program to determine the hot-hardness characteristics of the high-temperature bearing steels of interest. It is known that the rolling-element fatigue life of a bearing is closely related to the hardness of the bearing components.

In March 1965 a contract was entered into between the NASA Lewis Research Center and General Electric's Aircraft Engine Group to design and build a high-temperature, high-speed bearing test rig capable of testing large-diameter bearings under conditions of load, speed, and temperature typical of those expected in advanced high-performance turbine engines. NASA and General Electric engineers designed an advanced angular-contact ball bearing for testing similar to that used on the J-79 turbojet engines. The bearing test program included tests with both an advanced ester lubricant and a synthetic paraffinic oil in an air environment at a temperature of 492 K of (425° F) and a shaft speed of 12 000 rpm or 1.44 million DN.

Bearing tests were also conducted at outer-race temperatures of 478, 533, and 589 K (400°, 500°, and 600° F) with the synthetic paraffinic oil under a low-oxygen environment of less than 0.1 percent oxygen, by volume. Tests were also conducted with a polyphenyl ether lubricant at an outer-race temperature of 589 K (600° F) in both an air and a low-oxygen environment. A series of bearing tests was performed with a perfluorinated ether fluid which had not been investigated with the 25-mm-bore bearings at SKF Industries.

Two additional steels were evaluated at 589 K (600° F) with the 120-mm-bore bearing design. These were AISI M-1 and WB-49. Both of these materials were produced by a consumable-electrode vacuum-melting process.

Having established the high-temperature technology, attention was focused on the high-speed aspect that engine designers considered imperative for advanced aircraft engine designs for the 1980's. The then present working knowledge of bearing technology was limited to 2 million DN. Using a High-Speed Bearing Dynamics Computer Program the design of a 120-mm-bore, angular-contact ball bearing was analyzed and modified for operation at speeds to 3 million DN or 25 000 rpm. The bearings were manufactured from vacuum-induction melted, consumable-electrode vacuum-remelted (VIM-VAR) AISI M-50 material. The cage was a silver-plated, one-piece inner land riding type made of an iron-base alloy.

On June 29, 1973 tests were begun with two of these specially designed ball bearings. Test conditions included a speed of 3 million DN and a temperature of 492 K (425° F). Under-race lubrication was provided as well as outer-race cooling. Twenty-five hundred test hours later the bearings were removed undamaged from the test rig. The longest lived, high-speed, rolling-element bearings in the world had survived. Twenty-eight more of these bearings were tested at these conditions. A 10-percent life over 20 times expected design life was achieved.

Helicopter transmission development in the 1960's suggested the need for tapered-roller bearings to operate at speeds to 2 million DN. In the 1970's NASA expanded its high-speed bearing program to encompass tapered roller bearings as well as small-bore ball and roller bearings. The NASA program is both experimental and analytical with the analytic results verified by experiment. The work reported is a summary of NASA contributions to high-performance engine and transmission bearing capabilities.

HIGH-TEMPERATURE LUBRICANT SELECTION AND LIFE

Lubricant Selection

In a lubricant screening program 25-mm-bore angular-contact ball bearings, made from consumable-electrode vacuum-melted (CVM) AISI M-1 steel, were tested with 11 lubricants in a low-oxygen environment [1][1]. These lubricants, their base stock, and kinematic viscosities at three temperatures are listed in table 1. Bearing test conditions included speeds from 20 000 to 45 000 rpm, maximum Hertz stresses at the inner race ranging from 1.3×10^9 to 2.4×10^9 N/m^2 (189 000 to 347 000 psi), and outer-race temperatures from 478 to 589 K (400° to 600° F).

High-temperature failure modes can be catagorized as fatigue pitting, surface glazing and pitting, and surface smearing or deformation. The fatigue failures reported in this study were generally more extensive and were associated with more surface distress (glazing and superficial pitting) than classical fatigue spalls normally experienced under more conventional temperature conditions.

It is interesting to note that with the synthetic paraffinic oil with the antiwear additive, no glazing, superficial pitting, fatigue spalling, or wear occurred for any of the 10 bearings tested. Each bearing was run for at least 180 hr. For the same lubricant without the additive under the same conditions (i.e., temperatures between 561 to 589 K (550° to 600° F)), some glazing was observed.

Bearing Lives at 492 K (425° F) in Air

Tests were conducted with ABEC-5 grade, split-inner-race 120-mm-bore angular-contact ball bearings having a nominal contact angle of 20°. The balls and races were manufactured from consumable-electrode vacuum-melted (CVM) AISI M-50 steel with a nominal Rockwell C hardness of 63 at room temperature [2].

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[1]Refers to references at the end of the text.

Bearing test conditions were a thrust load of 25 800 N (5800 lb) which produced a maximum Hertz stress on the inner race of 2.23×10^9 N/m^2 (323 000 psi), an outer-race temperature of 492 K (425° F), and a shaft speed of 12 000 rpm. Two lubricants were used, an advanced ester oil and a low viscosity synthetic paraffinic oil; both were run in an air environment. Properties of these lubricants are given in table 2. Ten percent bearing fatigue lives from these tests are shown in figure 1. The ester fluid produced a life approximately 6 times that predicted by the Anti-Friction Bearing Manufacturers Association (AFBMA) method, while the synthetic paraffinic oil produced a life more than 10 times the AFBMA life.

At temperatures above 492 K (425° F) the ester fluid's viscosity is such that it is questionable whether it can produce an adequate EHD film [2]. On the other hand, the synthetic paraffinic lubricant's viscosity is adequate to support an EHD film, but the lubricant oxidizes rapidly above 492 K (425° F). As a result, at temperatures much above 492 K (425° F), a relatively inert environment must be provided with less than 0.1 volume percent oxygen.

Bearing Lives at 589 K (600° F)

Groups of 120-mm-bore angular-contact ball bearings of similar design and operating at the same test conditions as those bearings at 492 K (425° F) in air were fatigue tested with three lubricants at 589 K (600° F) [3,4]. The lubricants were a high-viscosity synthetic paraffinic oil with antiwear and antifoam additives, a fluorocarbon fluid, and a polyphenyl ether with an oxidation inhibitor and antifoam additive. Properties of these three lubricants are given in table 2. The rolling-element fatigue lives of the 120-mm ball bearings with the three lubricants are summarized in figure 2.

Synthetic paraffinic oil. - The synthetic paraffinic oil gave a 10-percent fatigue life more than 13 times the AFBMA-predicted life. There were no statistical differences in fatigue lives between bearings run in a low-oxygen environment (less than 0.1 volume percent oxygen) at 589 K (600° F) [5] and bearings run in air at 492 K (425° F). Metallurgical examination of the bearings indicated that failure was by classical rolling-element fatigue. The fatigue spalls were subsurface in origin, initiating in the zone of resolved maximum shearing stress. There was no apparent or measurable wear on the bearing surfaces at the 589 K (600° F) operating temperature; however, there were some signs of surface glazing. This suggests that at 589 K (600° F) some asperity contact of the mating surfaces occurred [4], although this phenomenon had no significant effect on the fatigue results.

Polyphenyl ether. - Tests were conducted with 120-mm-ball bearings of the same design with the 5P4E polyphenyl ether at an outer-race temperature of 589 K (600° F) in an air environment [3,4]. Preliminary tests at a maximum Hertz stress of 2.23×10^9 N/M^2 (323 000 psi) in a 0.1 volume percent oxygen environment showed severe bearing wear and damage after only a few hours operation because of the inability of the bearings to stabilize at a temperature of 589 K (600° F). As a result of these tests, the maximum Hertz stress on the inner race was lowered to 2.03×10^9 N/m^2 (295 000 psi), and a series of 26 bearings was tested in an air environment. Of these bearings, only two failed by fatigue. This is an insufficient number of failures to permit an accurate life estimate. However, a rough estimate of 0.75 of the

AFBMA-predicted life was made for the polyphenyl ether on the basis of the fatigue data.

Despite this apparently less hostile environment and the reduced load, most of the tests with the polyphenyl ether had to be suspended because of a large amount of ball wear. In tests running from 2 to 65 hr the average reduction in ball diameter was approximately 0.0254 mm (0.001 in.). After a relatively short running time, a stable suspension of wear particles in the polyphenyl ether fluid exists. These particles can act as an abrasive, which may accelerate the wear processes. However, 10 bearings ran for over 450 hr with minimal wear. The ball diameters on these bearings were reduced approximately 0.00762 mm (0.0003 in.). On all the bearings tested glazing was present on the contacting surfaces. However, with the long-lived bearings, no micropitting of the surface accompanied the glazing.

Fluorocarbon fluid. - Preliminary tests with the fluorocarbon fluid at 589 K (600° F) and at a maximum Hertz stress of 2.23×10^9 N/m^2 (323 000 psi) under a low-oxygen environment (less than 0.1 volume percent) produced considerable ball wear and/or surface distress [3,4]. As with the polyphenyl ether, the maximum Hertz stress on the inner race was lowered to 2.03×10^9 N/m^2 (295 000 psi). The 10-percent life was approximately three times AFBMA-predicted life.

The performance of the fluorocarbon fluid was not consistent. The predominant lubrication mode ranged from boundary to elastohydrodynamic. In some tests ball wear was extensive as evidenced by a decrease in ball diameter of 0.0127 to 0.0508 (0.0005 to 0.002 in.) within 500 hr of operation. Conversely, there were some tests, which were terminated at 500 hr, exhibiting extremely good surfaces with no evidence of incipient fatigue failure or measurable wear. Endurance tests with this fluid at 533 K (500° F) in an air environment resulted in extremely poor bearing life.

BEARING MATERIAL SELECTION AND LIFE

Ball and Race Materials

The bearing industry has used AISI 52100 steel as a standard material since 1920. This is a high-carbon chromium steel which also contains small amounts of manganese, silicon, nickel, copper, and molybdenum. For bearings it is generally vacuum processed and has a high degree of cleanliness from rigorous control of the melting process.

A commonly accepted minimum tolerable hardness for bearing components is 58 Rockwell C. At a hardness below this value brinelling of the bearing races can occur. Since hardness decreases with temperature, conventional bearing materials such as AISI 52100, can be used only to temperatures of about 450 K (350° F). Much effort has gone into developing steel alloys suitable for higher temperatures.

There has been a considerable number of studies performed to determine the rolling-element fatigue lives of various bearing materials [6-16]. However, none of these 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. The more standard mechanical tests such as tension and compression tests or rotating beam tests are not correlatable with rolling-element fatigue results [11].

Rolling-element fatigue tests were run with eight through-hardened bearing materials at 339 K (150° F) [17-19]. These materials were AISI 52100, M-1, M-2, M-10, M-42 (similar to WB-49), M-50, T-1 and Halmo. Balls of each material having diameters of 12.7 mm (1/2 in.) were run in the five-ball fatigue testers. Care was taken to maintain constant all variables known to affect rolling-element fatigue life. The longest lives at 339 K (150° F) were obtained with AISI 52100. Ten-percent lives of the other materials ranged from 7 to 78 percent of that obtained with AISI 52100. A trend is indicated toward decreased rolling-element fatigue life with increased total weight percent of alloying elements (fig. 3) [19].

Three bearing materials were investigated in rolling-element fatigue tests with 120-mm-bore ball bearings at 589 K (600° F) using a synthetic paraffinic oil in a low-oxygen environment. These materials were AISI M-50, WB-49, and AISI M-1 [20]. AISI M-50 is a martensitic, high-speed tool steel, which has been used in critical bearing applications for the past two decades. The steel has good through hardenability and was developed primarily for use as a high-strength, high-wear-resistant tool steel. The material for these bearings was produced by the consumable-electrode, vacuum melting (CVM) process. The fatigue life with the 120-mm-bore ball bearings made from AISI M-50 is shown in figure 4. The material hardness was controlled at room temperature to Rockwell C 63±1 for the rings and Rockwell C 63±0.5 for the balls.

AISI M-1 is also a high-speed tool steel which has been under investigation as a potential high-temperature bearing material. The material hardness for the AISI M-1 test bearings was controlled to Rockwell C 63±1 for the rings and Rockwell C 63±0.5 for the balls. The fatigue life obtained with the 120-mm-bore ball bearings made from CVM AISI M-1 is compared in figure 4 with the AISI M-50 bearings under the same operating conditions.

The WB-49 material was developed specially for high-temperature bearing applications [21]. It contains considerably more alloying elements than either the AISI M-50 or the AISI M-1 material. The CVM WB-49 rings were heat treated to a room temperature hardness of Rockwell C 64±0.5. The WB-49 bearings utilized AISI M-1 steel balls from the same heat as those for the AISI M-1 bearings. Previous experience has shown that WB-49 balls could not be manufactured without producing incipient microcracking [22]. As a result, balls made from WB-49 had extremely short fatigue lives. Fatigue results with the WB-49 bearings are summarized in table 3 and are compared in figure 4 with the AISI M-50 and AISI M-1 bearings.

From these data the 10-percent fatigue-life difference between the AISI M-50 and M-1 steels can be considered statistically insignificant at 589 K (600° F). However, the differences between the WB-49 and both the AISI M-50 and M-1 materials is statistically significant. For the M-50 and M-1 bearings run with the synthetic paraffinic oil, the experimental bearing 10 percent life exceeds the AFBMA-predicted (catalog) life by a factor in excess of 13 and 6, respectively. As a result, no derating of bearing life is required for these two materials. However, fatigue life with the WB-49 was less than half the AFBMA-predicted (catalog) life; hence, this material would have to be derated. The results of the bearing tests 589 K (600° F) correlated well with the results of the five-ball fatigue data at 339 K (150° F) (fig. 5) [19].

Material Hardness Effects

Heat treatment can significantly influence several rolling-element bearing material properties. Most bearing procurement specifications do not designate heat treatment but rather call for certain material characteristics such as grain size and hardness, which are controlled by the heat-treat cycle. Hardness is the most influential heat-treat-induced variable in rolling-element fatigue [11,15,23].

Hot-hardness measurements were made for groups of AISI 52100, Super Nitralloy, AISI M-1, AISI M-50, Halmo, WB-49, Matrix II, WD-65, and modified AISI 440 C [24-26]. The results of these hardness measurements are shown in figure 6. These normalized data show that regardless of the initial hardness, the hot hardness of the individual materials shows the same functional dependence; that is, the charges in hardness with increasing temperature are all independent of initial hardness.

EFFECT OF LOAD AND STRESS

It is commonly accepted that the life of rolling-element bearings is inversely proportional to stress to the 9th or 10th power. However, data which are summarized and presented in [27] suggest 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. A proper stress-life exponent value then becomes more than of mere academic interest inasmuch as an engine designer requires a reliable analytic tool to predict bearing life and performance.

Fatigue tests were performed with the 120-mm-bore angular-contact ball bearings made of consumable-electrode vacuum-melted (CVM) AISI M-50 steel at thrust loads of 20 462, 25 800 and 32 517 N (4600, 5800, and 7310 lb). The test conditions included a speed of 12 000 rpm (1.44 million DN) and an outer-race temperature of 533 K (500° F) in a high-temperature fatigue tester. The maximum Hertz stresses shown in table 3 include effects of approximately 890 N (200 lb) per ball centrifugal force [28].

The test results are summarized in table 3. The predicted life for the three thrust load conditions are given in table 3. The predicted results were calculated using the method of Harris [29] which utilizes a stress-life exponent of 9. A material factor obtained by the method of Chevalier, et al., [30], was used. The data for the three load conditions did not show significant deviation from the ninth power stress-life relation. This result would suggest that even though there is evidence [27-28] that the ninth power relation may not be rigorous, there is no justification to charge its use in determining bearing life.

LUBRICATION MODES FOR HIGH SPEED

Jet Lubrication

A question which was required to be answered with definite certainty was "What is the speed limitation for reliable operation of jet lubricated, large-bore, angular-contact ball bearings?" Tests were conducted at a shaft speed of 20 800 rpm (2.5 million DN) with an inner-race land riding cage using jet lubrication (fig. 7) [31]. The criteria for successful operation comprised the ability of the bearing to operate over the spectrum of bearing thrust loads which may be reasonably expected in actual turbojet engine applications. The results of these tests indicated that successful operation at a thrust load of 22 240 N (5000 lb) and a speed of 16 700 rpm (2 million DN) could only be obtained at a flow rate of 3.8×10^{-3} cm^3 (1.0 gal) per min, or higher. At 20 800 rpm (2.5 million DN) the bearing inner-race temperature could be stabilized at 478 K (400° F) at a thrust load 6672 N (1500 lb) and flow rate of 8.3×10^{-3} m^3 (2.2 gal) per min. At the lower flow rates the bulk bearing temperature began to exceed the estimated operating temperature limitations of the lubricant (495 K (431° F)) [32]. This temperature limitation is based on the lubricants thermal and oxidative stability and its ability to form an adequate elastohydrodynamic film at the bearing operating temperature. From these test results it was concluded that the limiting speed of their bearings at low load at the optimal lubricant conditions with an inner-race land riding cage is less than 2.5 million DN. However, high oil jet velocities might extend the maximum permissible operating bearing speeds to values greater than 2.5 million DN because of better oil penetration and more efficient cooling. Practical limits of jet size and supply pressure would have to be determined.

Under-Race Lubrication

Parametric tests were conducted with the thrust loaded 120-mm-bore ball bearings to 3 million DN using under-race lubrication [33-35]. In this method of lubrication, lubricant is centrifugally injected through the split inner race and shoulders of an angular-contact ball bearing by means of a plurality of radial holes (fig. 8). As a result, both the cooling and lubricant functions are accomplished at the inner race. The results previously discussed show that bearings operating with jet lubrication under certain conditions are limited to speeds less than 2.5 million DN. It was therefore necessary to determine if there is a crossover where it becomes necessary or desirous to use one lubrication system over another.

Data for both under-race lubricated and jet lubricated bearings are presented in figure 9. The under-race lubricated bearings were provided with outer-race cooling. However, outer-race cooling generally had an insignificant effect on the inner-race temperature [34].

The results shown in figure 9(a) indicated that all operating conditions the under-race lubricated bearings had lower temperatures than the dual-orifice jet lubricated bearings. At 12 000 rpm (1.44 million DN) the temperature difference was approximately 22 K (40° F) and at 16 700 rpm (2 million DN), the temperature difference is approximately 44 K (80° F). Beyond 2 million DN the bearing temperature with under-race lurbication increases only nominally while the temperature of the jet lubricated bearings increases at an accelerated rate. Hence, proper thermal management using jet lubrication is not achievable at the higher speeds. From the above it is easily concluded that under-race lubrication results in lower operating temperatures.

The data of figure 9(b) compare power loss for the two different lubrication systems. As was reported in [33,34], power loss is a function of the amount of lubricant penetrating the bearing cavity. This is due to viscous drag and lubricant churning [36]. From figure 9(b) the power loss with under-race lubricated bearings is higher than with the jet lubricated bearings. At 12 000 rpm (1.44 million DN) the under-race lubricated bearing power loss was approximately 1-kW (1.3-hp) greater than the jet lubricated bearings. At 16 700 rpm (2 million DN), the difference was approximately 2.3 kW (3.1 hp). The power loss with the under-race lubricated bearing with a flow rate of 4.9×10^{-3} cm^3 (1.3 gal) per min was equivalent to a jet lubricated flow rate of approximately 6.8×10^{-3} cm^3 (1.8 gal) per min. If bearing power loss is a function of lubricant flowing through or in the bearing cavity, then it can be reasonably concluded that, for a given jet lubricant flow, approximately 70 percent of the lubricant penetrates the bearing cavity at speeds to at least 2 million DN. At higher speeds this percentage probably decreases due to centrifugal force and windage effects [29].

COMPUTER PERFORMANCE PREDICTIONS

The bearings run at 3 million DN were designed using the results of the calculations made by the computer program first described in [37] and subsequently updated by [38]. This program is referred to as COMB. Later comparisons were made using the different and somewhat more comprehensive bearing-shaft computer program described in [39]. This program is referred to as SHABERTH. To effect a direct comparison of predicted and experimental bearing performance, the computer programs were run at the stated operating conditions of the bearings tested in [31]. The first calculations were done with the COMB program. Then the comparisons using SHABERTH were made. The effect of operating conditions on inner- and outer-race temperatures were made as well as determining power loss [40].

Representative calculations compared with experimental results are shown in figure 10. In general, the COMB program predicted bearing race temperatures reasonably well at low speeds. However, this program underestimated bearing power loss by a factor of 2.

The SHABERTH program which had two different traction models for the lubricant, one designated NASA and the other SKF, predicted race temperatures and bearing power losses reasonably well. The program predicted slightly higher bearing power losses using the NASA version than when using the SKF version.

BEARING LIFE AT 3 MILLION DN

Test Bearing

The test bearings were ABEC-5 grade, split inner-race 120-mm-bore ball bearings. These bearings were nearly identical to those used in the previous investigations. The inner and outer races, as well as the balls, were manufactured from one heat of vacuum-induction melted, vacuum arc-remelted (VIM-VAR) AISI M-50 steel. The previous bearings tested were made from CVM AISI M-50. The nominal hardness of the balls and races was Rockwell C-63 at room temperature. Each bearing contained 15, 2.0638-cm (13/16-in.) diameter balls. The retained austenite content of the ball and race material was less than 3 percent. The cage was a one-piece inner-land riding type, made out of an iron-base alloy (AMS 6415) heat treated to a Rockwell C hardness range of 28 to 35 and having a 0.005-cm (0.002-in.) maximum thickness of silver plate (AMS 2410). The cage was balanced within 3 g-cm (0.042 oz-in.).

The inner- and outer-race curvatures of the bearing were 54 and 52 percent, respectively. The nominal contact angle was 24°. The previous bearings tested had a nominal contact angle of 20°.

All components with the exception of the cage were matched within ±1 Rockwell C point. This matching insured a nominal differential hardness in all bearings (i.e., the ball hardness minus the race hardness, commonly called ΔH) of zero [9]. Surface finish of the balls was 2.5 μcm (1 μin.) AA, and the inner and outer raceways were held to a 5-μcm (2-μin.) AA maximum surface finish.

The bearing design permitted under-race lubrication by virtue of radial holes machined into the halves of the split inner races. Provision was also made for inner-race land-to-cage lubrication, by the incorporation of several small diameter holes radiating from the bore of the inner race to the center of the inner-race shoulder.

Fatigue-Life Results

Two groups of 120-mm-bore angular-contact ball bearings were fatigue tested with a tetraester lubricant at a bearing temperature of 492 K (425° F). Test conditions were a shaft speed of 12 000 or 25 000 rpm (1.44 or 3.0 million DN) and a bearing thrust load of 66 721 N (5000 lb). The fatigue life results of these tests are shown in figure 11. At 1.44 and 3.0 million DN, 84 483 and 74 800 bearing test hours were accumulated, respectively [41].

Only one bearing failed at 1.44 million DN (12 000 rpm). As a result the bearing life distribution presented in figure 11 for this speed is based on the results at 3.0 million DN (25 000 rpm) and is only an estimate.

The experimental life results were compared with the predicted life obtained by the methods of Lundberg and Palmgren [42,43] considering the centrifugal effects at 25 000 rpm but not elastohydrodynamic effects. The predicted 10-percent life was 21×10^6 inner-race revolutions at 3.0 million DN. The 10-percent life, based on the experimental life previously discussed (fig. 2) [2], adjusted for differences in contact angle, thrust load, and speed, is approximately 105×10^6 inner-race revolutions. This life coincides with current design practice which would use a material factor of five multiplied by the predicted life. The major difference, besides contact angle, between the bearings tested at 3 million DN and those bearings tested previously [2] at the lower speeds was the use of the double-vacuum-melted (VIM-VAR) AISI M-50 steel as opposed to consumable-electrode vacuum-melted (CVM) AISI M-50 steel.

For the 12 000 rpm (1.44 million DN) tests, the single failure which occurred was on one ball. For the 25 000 rpm (3.0 million DN) tests, of the six failures encountered, three occurred on the inner race, and three occurred on single balls. Metallurgical analysis of the failed bearings established that the failures were initiated by classical subsurface rolling-element fatigue. In this failure mode a spall of subsurface origin is formed. The spall acts as a stress raiser which in the presence of higher hoop stresses, at higher speeds such as 3 million DN, can cause the inner race to fracture [41]. Hence, race fracture at very high speeds can be a serious problem. Its solution must incorporate both fracture mechanics methodology and materials development, aimed at improving the fracture strength of the high-speed bearing steels [41].

CONTINUING RESEARCH

The approach and techniques which led to the solution for designing and operating at high-speed, angular-contact ball bearings, were successfully applied by NASA engineers to cylindrical-roller bearings [44,45], small-bore ball bearings [46-48], and large-bore tapered-roller bearings [49-52]. Research is also being conducted on small-bore tapered-roller bearings and small-bore cylindrical-roller bearings, and spherical roller bearings. While a detailed discussion of these NASA rolling-element bearing research programs are beyond the scope of this paper, a brief description is provided.

Cylindrical-Roller Bearings

An ABEC-5 grade 118-mm-bore roller bearing was studied parametrically at speeds from 10 000 to 25 000 rpm (1.2 to 3.0 million DN). The bearing had a round outer ring (not preloaded), and provisions were made for lubrication and cooling through the inner ring in a manner similar to the 3 million DN 120-mm-bore angular-contact ball bearings previously discussed. In some tests the outer ring was also cooled. The bearing material was CVM AISI M-50.

The bearing ran successfully at 3 million DN with very small evidence of cage slip. Load, which was varied from 2200 to 8900 N (500 to 2000 lb), had no significant effect on bearing temperature or cage slip over the speed range tested. Bearing temperature varied inversely with cage slip for all test conditions. Cooling the outer ring decreased its temperature but increased the inner-ring temperature. Heat rejected to the lubricant (power loss within the bearing) increased with both shaft speed and total oil flow rate to the inner race [44].

The operating characteristics of the bearing were calculated using the computer program CYBEAN [53-55]. The predicted results of inner- and outer-race temperatures and heat transferred to the lubricant generally compared well with experimental data to 3 million DN, radial loads to 8900 N (2000 lb) and total lubricant flow rates to 0.0102 m^3/min (2.7 gal/min).

Small-Bore Ball Bearings

Small advanced aircraft turbojet engines require bearings that can operate at 2.5 million DN. Parametric tests were conducted with 35-mm-bore angular-contact ball bearings lubricated by either jet or by flowing oil through axial grooves and radial holes machined in the inner ring of the bearing [48]. Test conditions were a thrust load of 667 N (150 lb), speeds from 48 000 to 72 000 rpm (1.68 to 2.5 million DN), and an oil-inlet temperature of 394 K (250° F). Outer-race cooling was also used in some tests. Successful tests were achieved with both jet lubrication and inner-race lubrication. The jet-lubricated bearings had lower outer-race and higher inner-race temperatures than the under-race lubricated bearing. Maximum power loss of 2.8 kW (3.7 hp) was experienced at 2.5 million DN. Maximum cage slip of 7 percent occurred at this speed. Additional test results with different cage designs and using combined radial and thrust loads are published in [46,47].

Tapered-Roller Bearings

Tapered-roller bearings are being used in some helicopter transmissions to carry combined radial, thrust and moment loads and in particular, those loads from bevel gears such as high-speed input pinions. The performance of 120.65-mm-bore high-speed design tapered-roller bearings was investigated at shaft speeds of 20 000 rpm (2.4 million DN) under combined thrust and radial load [51]. Using a tapered-roller bearing computer analysis [56], the test bearing design was optimized for high-speed operation. Temperature distribution and bearing heat generation were determined as a function of shaft speed, radial and thrust loads, lubricant flow rates, and lubricant inlet temperature. The high-speed design tapered-roller bearing operated successfully at shaft speeds up to 20 000 rpm under heavy thrust and radial loads. Bearing temperatures and heat generation with the high-speed design bearing were significantly less than those of the modified standard bearing tested [49]. Cup (outer ring) cooling was effective in decreasing the high cup temperatures to levels equal to the cone (inner ring) temperature. These results were similar to those obtained with the high-speed ball and roller bearings previously discussed.

Endurance life tests were run with the standard design and the optimized high-speed design bearings at speeds of 12 500 and 18 500 rpm (1.5 and 2.2 million DN), respectively [52]. Standard design bearings of vacuum melted (CVM) AISI 4320 and CBS-1000M and high-speed design bearings of CVM CBS-1000M and through-hardened VIM-VAR AISI M-50 were run under heavy combined radial and thrust load until fatigue failure or until a preset cutoff time of 1100 hr was reached. The standard design bearings made from CBS-1000M ran to approximately six times rated catalog life. Twelve identical bearings of AISI 4320 material ran to 10 times the rated catalog life without failure. Cracking and fracture of the cones of AISI M-50 high-speed design bearings occurred at 18 500 rpm due to high tensile hoop stresses. Four CVM CBS-1000M high-speed design bearings ran to 24 times rated catalog life without any spalling, cracking or fracture failure.

SUMMARY

Projections by the major United States aircraft turbine-engine manufacturers in the late 1950's indicated that the trends in airbreathing turbojet engines would dictate ball and roller bearings operating at speeds to 3 million DN and temperatures to 589 K (600° F). Helicopter transmission development in the 1960's suggested the need for tapered-roller bearings to operate at speeds to 2 million DN. In 1959 NASA began a research program to encompass these projected temperature and speed requirements for large-bore ball and roller bearings. In the 1970's NASA expanded its program to encompass tapered roller bearings as well as small-bore ball and roller bearings. The following results were obtained:

1. High bearing speeds to 3 million DN can be achieved with a fatigue life exceeding that which was common in commercial aircraft.

2. Temperature capabilities of rolling-element bearings for aircraft engines have moved from 450 to 589 K (350° to 600° F) with increased reliability.

3. Capabilities of available bearing steels, lubricants, and bearing designs were defined and established.

4. Computer programs for the analysis and design of rolling-element bearings were developed and experimently verified.

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TABLE 1. - KINEMATIC VISCOSITIES OF SCREENING LUBRICANTS

Lubricant type	Base stock	Kinematic viscosity, cs[a]		
		311 K (100° F)	372 K (210° F)	533 K [b](500° F)
Modified polyphenyl	Blend of 3-ring and 4-ring components	26	4.3	0.82
	Blend of 3-ring and 4-ring components	56	5.9	85
Polyphenyl ether	5P4E	365	13.1	1.07
	5P4E[c]	365	13.1	1.07
	6P5E	1831	24.7	1.20
Ester	Mixed polyester-diester	40	8.4	1.70
	Diester[c,d,e]	37	7.8	1.50
Hydrocarbon	Synthetic paraffinic	314	32	2.9
	Synthetic paraffinic[d]	314	32	2.9
	Super-refined naphthenic mineral oil[c,d,e]	79	8.4	1.1
	Super-refined paraffinic mineral oil	480	28	2.1

[a]Manufacturers' data.
[b]Estimated values.
[c]Oxidation inhibitor additive.
[d]Anti-wear additive.
[e]Anti-foam agent additive.

TABLE 2. - TEST LUBRICANT PROPERTIES

Property	Lubricant description				
	Advanced ester	Synthetic paraffinic oil		Fluorocarbon fluid	5P4E Polyphenyl ether
		Low viscosity	High viscosity		
Additives	Oxidation and corrosion inhibitors, antiwear agent	Antiwear and antifoam agents oxidation inhibitor	Antiwear and antifoam agents	None	Oxidation inhibitor, Antifoam agent
Kinematic viscosity, cs, at - 311 (100° F) 372 (210° F) 477 (400° F) 589 (600° F)	 29.0 5.4 1.5 ---	 61.0 8.9 1.9 ---	 443.3 39.7 5.8 2.2[a]	 298.3 29.8 4.6 1.8[a]	 358.0 13.0 2.1 0.9[a]
Flash point, K (° F)	533 (500)	539 (510)	542 (515)	None	556 (540)
Fire point, K (° F)	---	575 (575)	589 (600)	None	622 (660)
Autoignition temperature, K (° F)	717 (830)	644 (700)	703 (805)	None	886 (1135)
Pour point, K (° F)	233 (-40)	225 (-55)	236 (-35)	239 (-30)	279 (40)
Volatility 6.5 hr at 533 K (500° F) wt. %	2.0[b]	---	14.2	18.0	8.5
Specific heat at 533 K (500° F), J/(kg)(K) (Btu/(hr)(ft)(°F))	2557 (0.59)[b]	---	3033 (0.70)	1387 (0.32)	2297 (0.53)
Thermal conductivity at 533 K (500° F), J/(m)(s)(K) (Btu/(hr)(ft)(°F))	---	---	0.12 (70×10^{-3})	0.009 (52×10^{-3})	0.135 (78×10^{-3})
Specific gravity at 533 K (500 °F)	0.84	0.71	0.71	1.51	1.01

[a]Extrapolated value
[b]At 478 K (400° F)

TABLE 3. - FATIGUE LIFE RESULTS WITH 120-mm BASE ANGULAR-CONTACT BALL BEARINGS AT THREE THRUST LOADS

[Material, AISI-M-50 steel; speed, 12 000 rpm; temperature, 533 K (500° F); lubricant, synthetic paraffinic oil with antiwear additive.]

Load, lb	Maximum Hertz stress,[a] ksi		Life, millions of inner-race revolutions				Confidence number at L_{10} level
	Inner race	Outer race	Theoretical		Experimental		
			L_{10}	L_{50}	L_{10}	L_{50}	
4600	300	252	302	1510	166	502	--
5800	323	267	158	795	113	349	67
7310	347	284	83	415	194	342	78

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

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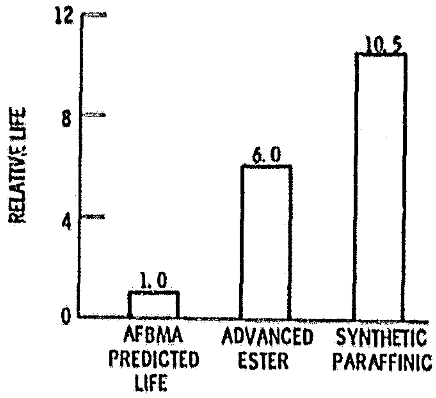

Figure 1. - Bearing fatigue life at 492 K (425° F) in air.

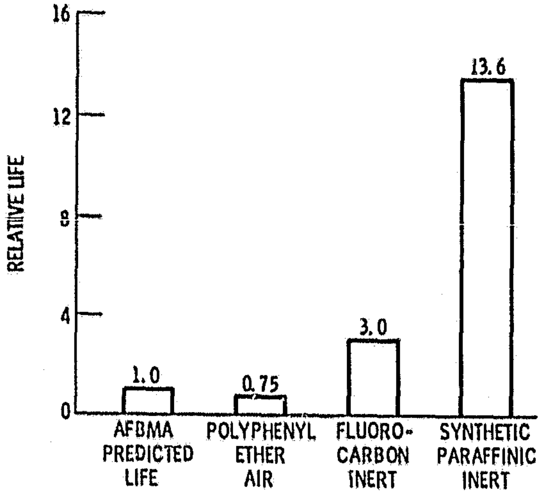

Figure 2. - Bearing fatigue life at 589 K (600° F).

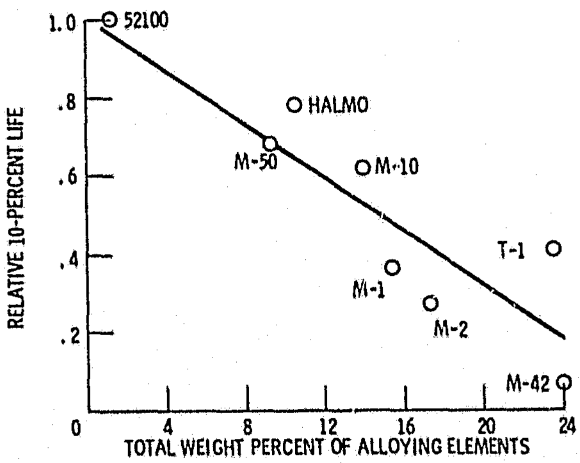

Figure 3. - Effect of total weight percent of alloying elements tungsten, chromium, vanadium, molybdenum, and cobalt on rolling-element fatigue life at 339 K (150° F).

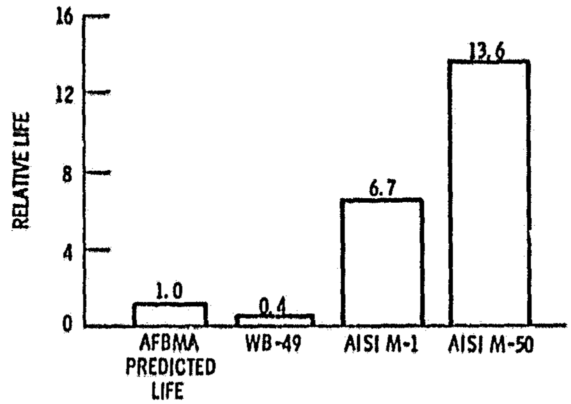

Figure 4. - Bearing life with three steels at 589 K (600° F) with synthetic paraffinic lubricant in low-oxygen environment.

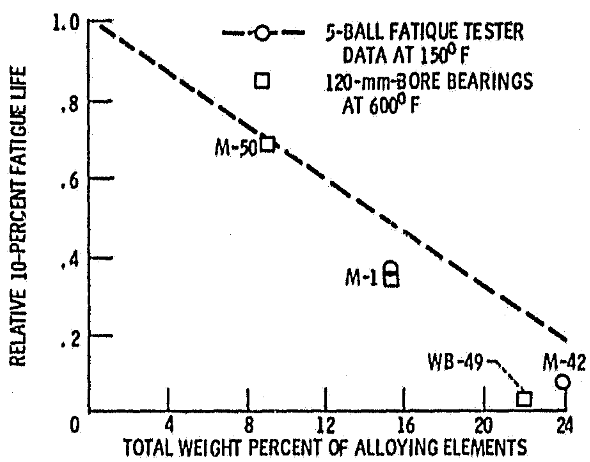

Figure 5. - Effect of total weight percent of alloying elements on fatigue life of 120-mm-bore bearings at 589 K (600° F).

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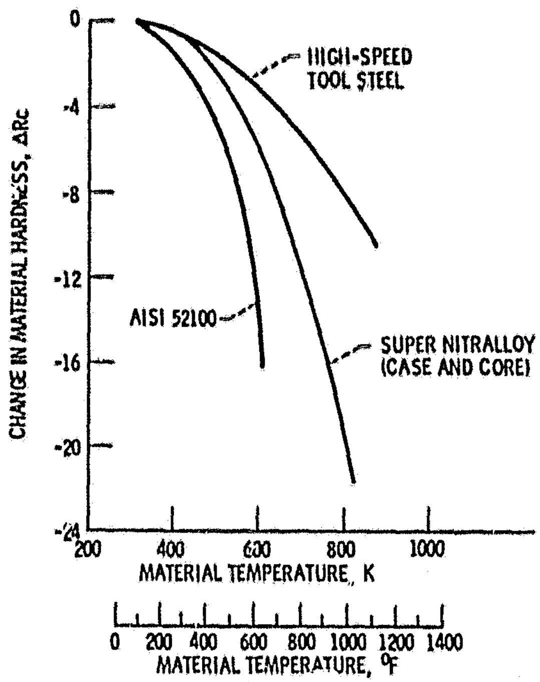
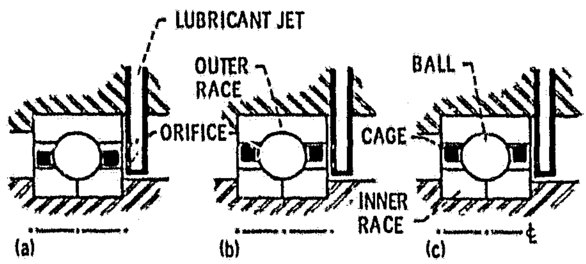


Figure 6. - Normalized short-term hot hardness as a function of temperature.



(a) Dual orifice. Inner-land riding cage.

(b) Dual orifice. Outer-land riding cage.

(c) Single orifice. Outer-land riding cage.

Figure 7. - Bearing lubrication for inner- and outer-race land riding cages. Number of jets, 2 per bearing.

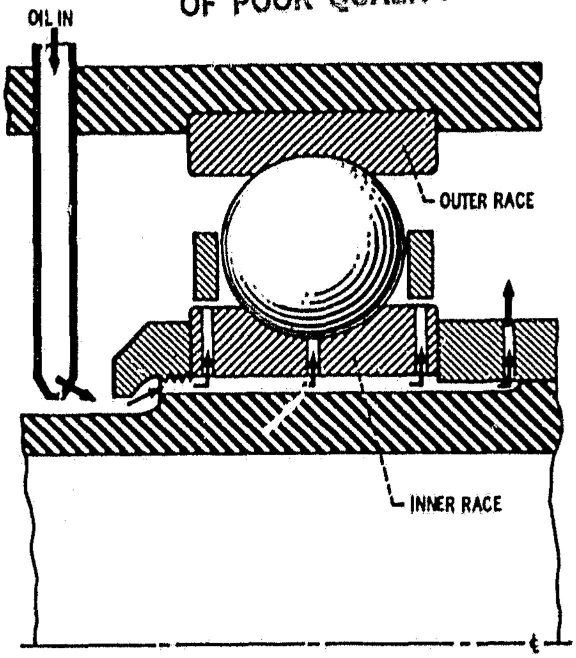
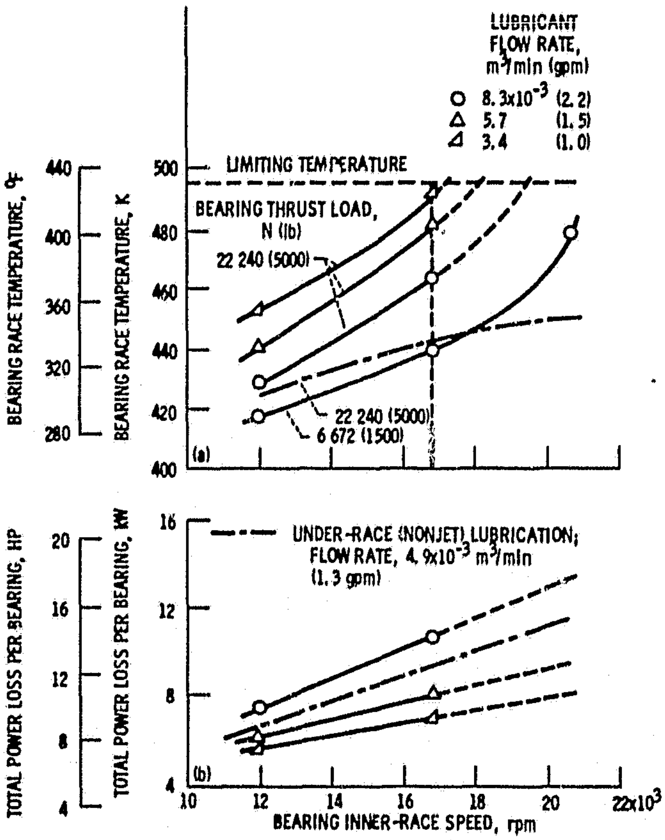


Figure 8. - Under-race lubrication.



(a) Temperature.
(b) Power loss.

Figure 9. - Bearing inner-race temperature and power loss are a function of speed for varying thrust loads and lubricant flow rates for inner-land riding cage with jet lubrication. Bearing type, 120-mm-bore angular-contact ball bearings; lubricant jet, dual orifice; number of jets, 2 per bearing; oil inlet temperature, 349 K (250° F); contact angle, 20°.