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ADVANCED DESIGN CONCEPTS FOR HIGH SPEED BEARINGS

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ABSTRACT

Advanced rolling-element bearing technology has been developed that has enabled bearing operation to temperatures of 600° F in an inerted environment. Longer fatigue lives than with present day bearing lubricants have been obtained. More than 13 times the AFBMA-predicted (catalog) life was obtained with bearings using AISI M-50 ball and race materials, an AMS 4892 nickel-base steel cage material, and a synthetic paraffinic lubricant at temperatures from 400° to 600° F.

A ball bearing design with 50-percent ball weight reduction has been run successfully for short time periods at DN values to 3 million. This concept offers promise of extending bearing fatigue life at high DN values.

INTRODUCTION

Advances in airbreathing turbojet engines have dictated that bearing materials and lubricants operate at higher temperatures, higher speeds and higher loads. The first generation supersonic transport (SST) turbine engine mainshaft bearings will operate at a bearing temperature of 425° F and a maximum speed of 1.3 million DN. (DN is a bearing speed parameter and is equal to the product of the bearing bore in millimeters and the shaft speed in rpm.) It is anticipated that more advanced engine designs may require bearings to operate at temperatures between 500° and 550° F and speeds of approximately 2 million DN. Projection of these trends to the 1990's would predict bearing temperatures to 600° F and bearing speeds of 3 to 4 million DN producing higher bearing operating stresses.

The objective of this presentation is to summarize the research results in advanced high-speed, high-temperature bearing technology. The principal requirements necessary to insure the success of a high-speed, high-temperature bearing design with adequate fatigue life are (1) the development of a high-temperature lubricant with sufficient elastohydrodynamic (EHD) film forming capability and good thermal stability at temperatures to 600° F; (2) the selection of a cage material with a good wear life and a race and ball material with high hardness and good compressive strength at elevated temperatures; and (3) an internal bearing design that will minimize the decrease in fatigue life resulting from ball centrifugal loading during high speed operation. The research results reported herein were obtained from experimental programs conducted at the NASA Lewis Research Center or from programs performed by contractor personnel under technical direction from Lewis. The work encompasses nearly 200,000 hours of bearing testing in the temperature range from 400° to 600° F.

HIGH-TEMPERATURE LUBRICANT SELECTION

Effects of Elastohydrodynamic Lubrication

In order to assure proper bearing operation at elevated temperatures, a lubricant film is required to separate the contacting components. This is illustrated in figure 1. In figure 1(a) a ball is in dry contact with the surface. Both the surface and the ball deform under the force which is placed on the ball due to the elastic properties of the material. With the ball rolling over the surface under load as a lubricant is introduced, a film is formed between the two surfaces (fig. 1(b)). The film is referred to as an elastohydrodynamic (EHD) film, and is formed as a result of the elastic deformation of the material and the hydrodynamic action of the lubricant. This film, which is generally dependent on lubricant base stock and viscosity, is in the order of 5 to 10 millionths of an inch thick at elevated temperatures.[1]

When a sufficient EHD film is present, rolling element bearings will not usually be subjected to early failures as a result of surface distress. They will fail from rolling-element fatigue. Fatigue usually manifests itself, in the early stages, as a shallow spall with a diameter about the same as the contact width. A typical fatigue spall of a bearing inner race is shown in figure 2.

As atmospheric viscosity of a particular lubricant is increased, rolling-element bearing fatigue life also increases. If the lubricant pressure-viscosity coefficient is increased by changing the lubricant base stock, longer fatigue life can be obtained for a given lubricant at atmospheric conditions. It has become generally accepted that fatigue life increases with increases in viscosity, pressure-viscosity coefficient, or rotational speed. These factors imply increasing bearing fatigue life with increasing EHD film thickness.

High Temperature Failure Modes

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 eleven lubricants in a low oxygen environment.[2] These lubricants, their base stock, and kinematic viscosities at three temperatures are listed in table I. Bearing test conditions included speeds from 20,000 to 45,000 rpm, maximum Hertz stresses at the inner race ranging from 189,000 to 347,000 psi, and outer race temperatures from 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 (see fig. 2) normally experienced under more conventional temperature conditions. An example of a high temperature fatigue failure is shown in figure 3. A normal appearing race that had been run under full elastohydrodynamic lubrication conditions is shown in figure 4(a).

Surface glazing, which can be present without spalling occuring, is illustrated in figure 4(b). Continued operation results in superficial pitting as shown in figure 4(c) and/or wear of the rolling-element surfaces. This condition was present in all bearings run with the various lubricants given in table I except those run with the synthetic paraffinic oil with anti-wear additive. The glazing was more severe with the bearings tested with the polyphenyl ethers, which were short lived. If operation of the bearing is continued under the condition shown in figure 4(c), spalling of the rolling-element surface will occur as illustrated in figure 3. Glazing phenomenon can be attributed to marginal elastohydrodynamic lubrication, high tangential forces can be induced which will tend to relocate the maximum shearing stresses closer to the surface. [3] Under these conditions, more shallow fatigue spalls would be expected than normally occur under full elastohydrodynamic conditions.

Another mode of failure at elevated temperatures is that of "smearing." This mode is illustrated in figure 5. Smearing manifests itself by gross metal transfer, plastic deformation and/or galling of the rolling-element surfaces. It is believed that smearing occurs when the condition is primarily that of boundary lubrication accompanied by a high degree of sliding in the ball-race contact. This mode of failure was dominant for those bearings which failed other than by fatigue spalling.

It is interesting to note that with the synthetic paraffinic oil with the anti-wear additive no glazing, superficial pitting, fatigue spalling, or wear occurred for all ten bearings tested. Each of these bearings was run for at least 180 hours. For the same lubricant without the additive under the same conditions (i.e., temperatures between 550° to 600° F) some glazing was observed. It is conceivable that anti-wear additives will reduce the tangential forces under marginal elastohydrodynamic conditions and thus increase bearing life.

Bearing Lives at 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 AISI M-50 consumable-electrode vacuum-melted (CVM) steel with a nominal Rockwell C hardness of 63 at room temperature.

Bearing test conditions were a thrust load of 5800 pounds which produced a maximum Hertz stress on the inner race of 323,000 psi, an outer-race temperature of 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 II. Ten-percent bearing fatigue lives from these tests are shown in figure 6. The ester fluid produced a life approximately 6 times that predicted by the Anti Friction Bearing Manufacturers Association (AFBMA) methods while the synthetic paraffinic oil produced a life more than 10 times the AFBMA life.

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

Bearing Lives at 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 425° F in air were fatigue tested with three lubricants at 600° F. 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 II.

Synthetic paraffinic oil. - The results of tests with this lubricant indicated there were no statistical differences in fatigue lives between bearings run in a low oxygen environment (less than 0.1 volume percent oxygen) at 600° F[4] and bearing run in air at 425° F. Metallurgical examination of the bearings indicated that failure was by classical rolling-element fatigue. Typical fatigue spalls on the balls and races were similar to that shown in figure 2. The fatigue spalls were subsurface in origin, initiating in the zone of resolved maximum shearing stress. An unfailed bearing run to suspension (500 hr of operation) with the synthetic paraffinic oil at 600° F is shown in figure 7. There was no apparent or measurable wear on the bearing surfaces at the 600° F operating temperature, however, there were some signs of surface glazing. This suggests that a 600° F some asperity contact of the mating surfaces occurred, 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 600° F in an air environment.[5], [6] Preliminary tests at a maximum Hertz stress of 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 600° F. As a result of these tests the maximum Hertz stress on the inner race was lowered to 295,000 psi and a series of 26 bearings was tested in an air environment.

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 hours, the average reduction in ball diameter was approximately 0.001 inch. 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, ten bearings ran for over 450 hours with miminal wear. The ball diameters on these bearings were reduced approximately 0.0003 inch. On all the bearings tested, glazing was present on the contacting surfaces, similar to that shown on the bearing race in figure 4(b). However, with the long-lived bearings, no micro-pitting of the surface accompanied the glazing.

Fluorocarbon fluid. - Preliminary tests with the fluorocarbon fluid at 600° F and at a maximum Hertz stress of 323,000 psi under a low oxygen environment (less than 0.1 volume percent) produced considerable ball wear and/or surface distress.[5],[6] As with the polyphenyl ether the maximum Hertz stress on the inner race was lowered to 295,000 psi.

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 0.0005 to 0.002 inch within 500 hours of operation. Conversely, there were some tests, which were terminated at 500 hours, exhibiting extremely good surfaces with no evidence of incipient fatigue failure or measurable wear.

Based upon its chemical make-up, the fluorocarbon fluid has the greatest potential as an extremely high-temperature lubricant. It exhibits a number of properties which make it extremely attractive to engine designers, chief among which is its inherent fire safe operation. (See table II.) The corrosive aspect might tend to make it difficult to apply in existing lubricating systems, although in advanced engines, the lubricating system can be designed with new materials which will accommodate this corrosive characteristic.

Two deficiencies of the fluorocarbon fluid as a high temperature lubricant appear to be its high density and low thermal conductivity. Values of these properties at 500° F are given in table II. As a result, additional lubricant cooling is needed to overcome an overheating problem. Hence in order to accommodate this fluid in a turbine engine it may be necessary to re-evaluate the engine cooling system, or to make changes in the basic bearing design to decrease heat generation.

The rolling-element fatigue lives of the 120-mm ball bearings at 600° F with the three lubricants are summarized in figure 8. The basis for comparison is the AFBMA predicted (catalog) life. Of the 26 bearings tested with the 5P4E polyphenyl ether, only two failed by fatigue. This ia 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. The 10-percent life with the fluorocarbon fluid was approximately three times that predicted by AFBMA. Under a low-oxygen environment, the synthetic paraffinic oil gave a 10-percent fatigue life more than 13 times the AFBMA-predicted life.

BEARING MATERIALS AND SELECTION

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 air-melted in electric furnaces, 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 350° F. Much effort has gone into developing steel alloys suitable for higher temperatures. The addition of elements such as molybdenum, tungsten, chromium, and vanadium promote the retention of hardness at elevated temperatures. Other materials which promote hardness retention are aluminum and silicon.

Several bearing alloys are available for operation between 350 and 750° F.[7] The most common of these is AISI M-50. In most turbojet applications, this material is used exclusively. The material has the capability of maintaining a hardness above Rockwell C 58 to temperatures of approximately 600° F.

Three bearing materials were investigated in rolling-element fatigue tests with 120-mm ball bearings at 600° F using a synthetic paraffinic oil in a low oxygen environment. These materials were AISI M-50, WB-49 and AISI M-1.[8] AISI M-50 is a martensitic high-speed tool steel, which has been used in critical bearing applications for the past decade. The steel was developed primarily for use as a high strength, high wear resistant tool steel. The material is produced by the consumable-electrode vacuum melting (CVM) process and has good through hardenability. The fatigue results with the 120-mm ball bearings made from AISI M-50 are summarized in table III. 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 results obtained with the 120-mm ball bearings made from AISI M-1 are summarized in table III and are compared in figure 9 with the AISI M-50 bearings under the same operating conditions.

WB-49 is a material developed specifically for high temperature bearing applications.[9] It contains considerably more alloying elements than either the M-50 or the M-1 material. The 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.[10] As a result, balls made from WB-49 had extremely short fatigue lives. Fatigue results with the WB-49 bearings are summarized in table III and are compared in figure 9 with the 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 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.

High Temperature Cage Materials

Cages are a much more severe problem in small-bore bearings than in larger ones. In extreme-speed applications of small-bore bearings, it is frequently necessary to use a silver-plated, semihard, tool-steel retainer rather than a bronze, and oil-mist lubrication rather than recirculating oil to reduce churning losses. In large-bore bearings, retainer failure is much less common than in small-bore bearings.

At temperatures above 500° F, tests have indicated that bearing cage wear can be a limiting factor in the operation of bearings under the severe lubrication conditions encountered at these elevated temperatures. Therefore, in addition to the race and the rolling-element material, careful consideration must be given to the choice of cage material.

In conventional rolling-element bearings, both metallic and nonmetallic cages have found widespread use. Under normal temperatures, for nonaerospace applications, practically all the roller bearings and a large percentage of the ball bearings in use have been equipped wither with stamped cages of low-carbon steel or with machined cages of iron-silicon bronze or lead brass. Precision bearings, such as those used for aerospace applications, are usually equipped with cages machined from copper alloys or nonmetallic phenolic materials. In some applications, where marginal lubrication exists during operation, such as at high temperatures, silver plating on the bronze has been used. Phenolic materials are limited to temperatures of approximately 275° F, while copper-base alloys are suitable for operation to approximately 600° F. Above 600° F some success has been obtained with low-carbon steel or cast-iron cages, but, generally, the most successful high-temperature cages have been nickel-base alloys.[11] One of the nickel-base alloys used was AMS 4892 (S-Monel).

Other materials which have shown promise are high-temperature plastics which exhibit low friction and wear characteristics, high-alloy steels capable of maintaining their hot hardness at elevated temperatures, and stainless steels.

For the temperature range of 500° to 700° F six materials were investigated as potential high temperature cage materials.[12]

Results of 30-minute wear tests at a test temperature of 500° F with a naphthenic mineral oil as the lubricant for six materials are shown in figure 10(a). At this set of conditions, four materials show promise of operating for extended periods of time: AISI M-1, S-Monel, modified AISI 440C stainless steel, and a basic polyimide polymer. Tests were also conducted at 700° F with the same materials but with the 5P4E polyphenyl ether as the lubricant. The results of these tests are shown in figure 10(b). At this condition, the AISI M-1, S-Monel, and AISI 440C stainless-steel materials show the greatest promise. The high wear with the polyimide is not totally unexpected inasmuch as thermal degradation of this polymer begins at 700° F.

In both the 500° and the 700° F tests, the AISI M-1 and S-Monel materials exhibited the least amount of wear relative to the other materials. It can be concluded that AISI M-1 and S-Monel of Rockwell A hardnesses 81 and 67, respectively, are potential cage materials for elevated-temperature bearing applications.

Additional tests indicate that for the AISI M-1 and S-Monel materials, cage wear decreases with increasing hardness at both 500° and 700° F. However, for small differences in hardness of the modified AISI 440C stainless-steel materials, there was no significant difference in wear. Operating temperature will, of course, decrease material hot hardness. The effect of increasing temperature from 500° to 700° F with M-1 and S-Monel cage materials run with the naphthenic mineral oil as the lubricant show that wear will increase two to five times, depending on the material and its heat treatment. However, the Rockwell A hardness for M-1 and S-Monel decrease approximately 1 and 0.5 point, respectively, because of the increase in temperature from 500° to 700° F. These differences in hardness are not sufficient to account for the marked increase in wear. It can be concluded that, with these materials, temperature affects the amount of wear through its effect on the lubrication process.

From these results it can also be concluded that, in general, potential high-temperature cage materials of the types reported herein should be heat-treated to their maximum room-temperature hardness, while sufficient ductility to prevent cracking is maintained. In application, however, the rolling-element material should be somewhat harder than the cage material to prevent damage to the rolling elements. Wear resistant platings or coatings may also be used to reduce cage wear.

SPEED EFFECTS ON BEARING LIFE

Effect of Reduced Ball Mass

When ball bearings are operated at high speed the contact stress at the outer race is increased due to the centrifugal force on the balls. As a result of increased stress, bearing life is decreased. This effect is illustrated in figure 11. The life of a 150-mm bore angular contact ball bearing at two loads, 1000 pounds and 5000 pounds, is shown as a function of speed. As the speed is increased from 2 to 4 million DN, life decreases drastically for both loads. For the 1000 pound load, life decreases on the order of 50 to 1 while for 5000 pound load, on the order of 10 to 1. There are several possible approaches to solve this problem.

The first and most obvious approach is to optimize the bearing internal geometry for maximum life using a high-speed, ball bearing dynamics computer program.[13, 14] This will yield the optimum ball diameter and number, race curvatures, and contact angle. However, this approach will not yield more than a small, incremental improvement in life over that of bearings now in use. As a result, less conservative approaches must be considered.

Another approach is to reduce the mass of the ball. A 50-percent reduction in ball-weight, while maintaining the ball diameter constant, can theoretically result in a significant increase in life relative to life with a solid ball at DN values greater than 3 million. The effects of decreasing ball weight 50 percent in a 150-mm bore ball bearing are illustrated in figure 12. For the 5000 and 1000 pound loads, the life can be increased more than 3 times at the higher DN values simply by reducing the ball mass 50 percent.

Spherically Hollow Balls

One means of reducing the ball mass is to use hollow balls. Spherically hollow balls are fabricated by forging two hemispherical shells, matching and joining the shells, and finishing and heat treating the hollow balls by methods similar to those used in the manufacture of conventional solid balls.

Hollow balls of 11/16-inch diameter with 0.060-inch wall thickness were fabricated and joined by electron-beam welding and fitted to 75-mm bore ball bearings. The bearings were operated at 500 and 1000 pound thrust loads at speeds up to 18,000 rpm (1.35 million DN.[15] A super-refined naphthenic mineral oil was used as the lubricant in an air-oil mist with flow rates ranging from 0.01 to 0.07 pound per minute.

Post-test inspection of the bearings after a few hours running showed extensive damage to the hollow balls with many spalls on the outer surfaces. Several balls from one bearing were sectioned through the weld and polished to determine the extent and source of the failures. A cross-section of a hollow ball is shown in figure 13. A bead on the inner diameter is formed by the electron-beam weld. It is speculated that this bead acted as a notch or stress raiser in the ball wall and initiated flexure fatigue failures after operating the bearing for only 13 hours. A typical flexure fatigue failure is shown in figure 14. In figure 14(a) two cracks can be seen originating at the weld bead and propogating toward the outer surface of the ball. The crack shown in figure 14(b) has also originated at the weld bead, but has propagated into a crack network and a fatigue spall at the ball outer surface.

Balls from other bearings that were run under similar conditions were also examined. Some showed spalls on the outer surface; some showed no apparent damage. All these balls were sectioned and polished, and all revealed cracks originating in the vicinity of the weld bead at the inner surface. From the examination of these hollow balls, it was concluded that they all had failed in flexure due to a stress concentration in the region of the weld at the ball inner surface.

The use of hollow balls can still be advantageous at higher bearing DN values, provided the following problems can be overcome: (1) use a wall thickness such that flexure is minimized and yet mass reduction is appreciable, (2) control the weld penetration around the periphery of the ball, and (3) maintain a uniform wall thickness. Unless these problems are solved, hollow balls will have an unbalance and a different stiffness at the weld. Under dynamic conditions, these factors adversely affect the lives of the balls.

Drilled Balls

Another method of reducing ball mass is to machine a concentric hole through the ball. The amount of mass reduction equal to that of a thin-wall spherically hollow ball can be achieved with several advantages. Using drilled ball alleviates possible problems of ball unbalance, because the hole concentricity can be maintained very accurately. Additionally, a very smooth finish can be achieved on the inner surface, without the irregularities present in the area of the weld.

Tests were conducted with 75-mm bore ball bearings using 11/16-inch diameter drilled balls.[16] The bearings were operated at a thrust load of 500 pounds at speeds to 28,000 rpm (2.1 million DN) with air-oil mist or oil jet lubrication. The drilled balls were made by electric-discharge-machining (EDM) with a 0.42-inch diameter hole through the center of a solid ball to effect a 50-percent weight reduction. A finished ball is illustrated in figure 15(a). Pins through the center of each ball pocket at the pitch diameter of bearing (fig. 15(b)) were used to retain the balls in the bearing. The pins also restrained the edge of the hole from contacting the race groove while the bearing was operated under a given thrust load.

Two 75-mm ball bearings operated successfully over a range of conditions without damage or excessive wear. One bearing was still operating satisfactorily after 107 hours accumulated running time, with 65 hours above 1.5 million DN.

In another study, [17] four 125-mm bore ball bearings with 0.8125-inch diameter drilled balls were operated under 2500 pound thrust load at speeds from 8000 to 24,000 rpm (1.0 to 3.0 million DN) in two separate tests. Four bearings with conventional solid balls were operated under similar test conditions for comparative purposes. Special modifications were made to all bearings that provided for adequate lubrication sufficient cooling with a type II ester oil at speeds to 3 million DN.

Both the solid and drilled ball bearings operated satisfactorily at the extreme test conditions for several hours running time without damage or significant wear. One of these drilled ball bearings is shown in figure 16. The components of this bearing were in very good condition after 4.7 hours of operation. The results of these experimental programs indicate that the drilled ball bearing concept shows a great deal of promise for high speed bearing applications.

SUMMARY

Advanced rolling-element bearing technology has been developed by the NASA Lewis Research Center. This work has enabled bearing operation to temperatures of 600° F in an inert environment with longer fatigue lives and lower bearing component wear than present day bearing lubricants. These results have been substantiated by nearly 200,000 hours of bearing testing at temperatures from 400° to 600° F.

Additionally, a bearing design has been successfully demonstrated for short time periods at 3 million DN. This concept offers promise of extending bearing fatigue life at high DN values. The following results are emphasized in this paper:

1. Bearing 10-percent fatigue life was more than 10 times the rated AFBMA (catalog) life with a synthetic paraffinic oil in air at 425° F. By comparison an advanced ester lubricant produced a fatigue life approximately 6 times the AFBMA value in bearings operating under the same conditions. When operating in a low oxygen environment at 600° F, the synthetic paraffinic oil with antifoam and antiwear additives produced a fatigue life more than 13 times the rated AFBMA life. At the 600° F temperature the synthetic paraffinic lubricant had sufficiently thick EHD films to prevent any surface distress or appreciable wear in all bearing components.

2. Comparison of bearing fatigue lives using AISI M-50, AISI M-1, and WB 49 ball and race materials at 600° F, showed that life with M-50 steel was more than twice that with the M-1 material. Bearing life with WB-49 material was less than the AFBMA (catalog) life and therefore had to be derated.

3. In cage material screening tests at 500° and 700° F, AISI M-1 and AMS 4892 (S-Monel) materials exhibited the least relative wear among six candidate materials.

4. A drilled ball bearing with 50-percent ball weight reduction ran successfully for 4.7 hours at speeds to 3 million DN. Modifications were made to the bearings that provided for adequate lubrication and cooling without any measurable bearing wear at these extreme operating conditions.

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TABLE I. - KINEMATIC VISCOSITIES

OF SCREENING LUBRICANTS

Lubricant type	Base stock	Kinematic viscosity, cs[a]		
		100° F	210° F	[b]500° F
Modified polyphenyl ether	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 II. - TEST LUBRICANT PROPERTIES

Property	Lubricant description				
	Advanced ester	Synthetic paraffinic oil		Fluoro-carbon fluid	5P4E Poly-phenyl 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 -					
100° F	29.0	61.0	443.3	298.3	358.0
210° F	5.4	8.9	39.7	29.8	13.0
400° F	1.5	1.9	5.8	4.6	2.1
600° F	-----	----	2.2[a]	1.8[a]	0.9[a]
Flash point, $^{\circ}$F	500	510	515	None	540
Fire point, $^{\circ}$F	-----	575	600	None	660
Autoignition temperature, $^{\circ}$F	830	700	805	None	1135
Pour point, $^{\circ}$F	-40	-55	-35	-30	40
Volatility (6.5 hr at 500° F), wt.%	2.0[b]	-----	14.2	18.0	8.5
Specific heat at 500° F, Btu/(hr)(ft)($^{\circ}$F)	0.59[b]	----	0.70	0.32	0.53
Thermal conductivity at 500° F, Btu/(hr)(ft)($^{\circ}$F)	-----	-----	70×10^{-3}	52×10^{-3}	78×10^{-3}
Specific gravity at 500° F	0.84	0.71	0.71	1.51	1.01

[a]Extrapolated value.

[b]At 400° F.

TABLE III. - FATIGUE-LIFE RESULTS FOR 120-MM BORE ANGULAR-CONTACT BALL BEARINGS MADE FROM THREE HIGH-TEMPERATURE BEARING STEELS

(Thrust load, 5800 pounds; speed, 12,000 rpm; outer-race temperature, 600° F)

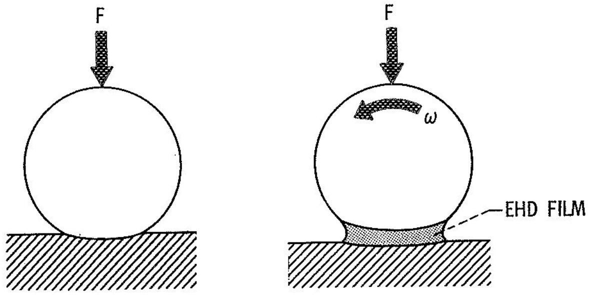
Material	Experimental life, millions of inner-race revolutions		Weibull slope	Failure index (a)	Confidence number at 10-percent life level (b)	AFBMA predicted 10-percent (catalog) life, millions of inner-race revolutions	Ratio of experimental 10-percent life to AFBMA-predicted life
	10-percent life	50-percent life					
AISI M-50	182	513	1.8	6 out of 26	---	13.4	~13.6
AISI M-1	89	2331	0.6	6 out of 26	67	13.4	~6.7
WB-49[c]	6	26	1.3	28 out of 30	>99	13.4	~0.4

[a]Number of fatigue failures out of number of bearings tested.

[b]Percentage of time that 10-percent life obtained with AISI M-50 bearings will have the same relation to the 10-percent life of the bearings made from other material.

[c]Bearings had AISI M-1 balls.

E-6144

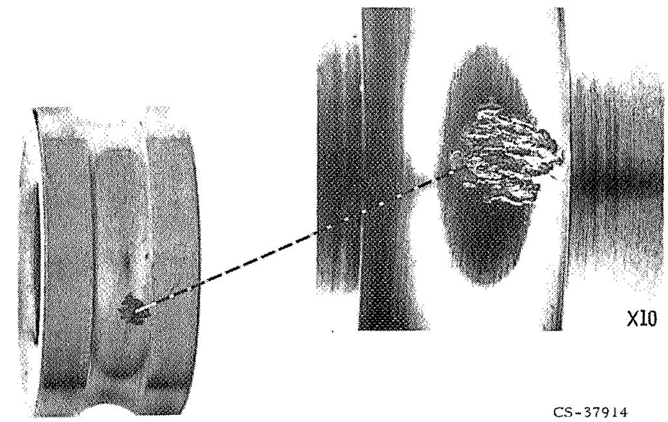


(a) Dry contact.

(b) Lubricated contact.

Figure 1. - Elastohydrodynamic lubrication.

CS-56856



CS-37914

Figure 2. - Typical fatigue spall.

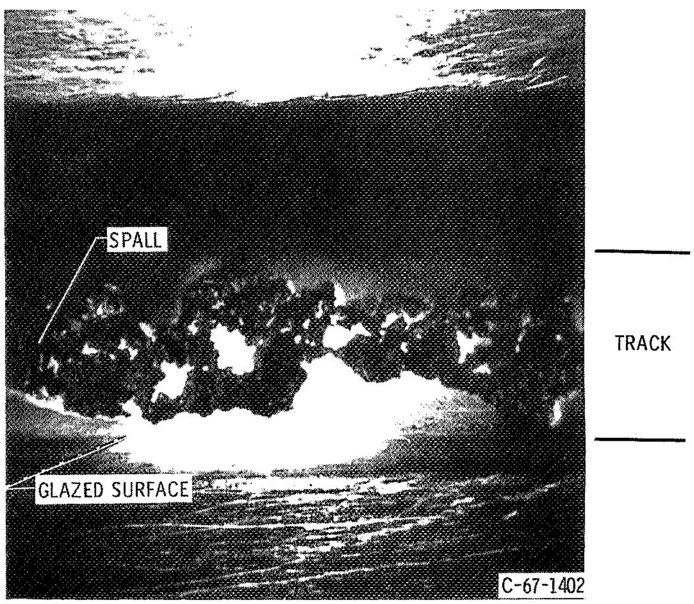


Figure 3. - Representative high-temperature fatigue spall.

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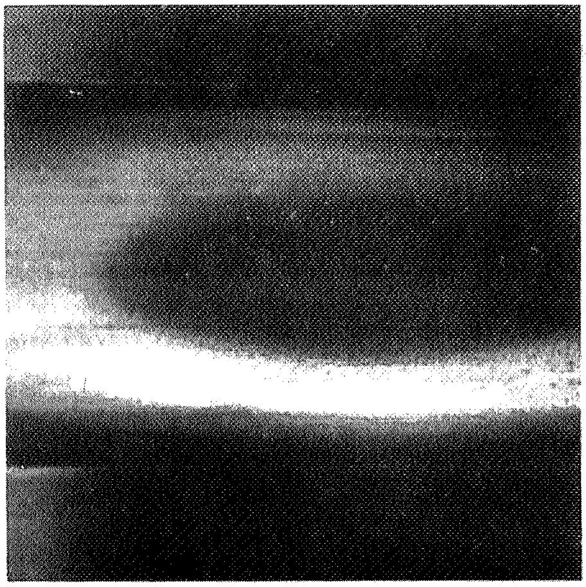
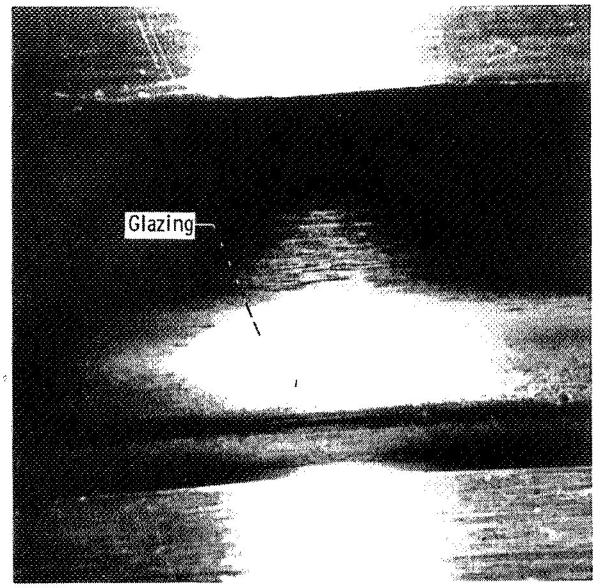
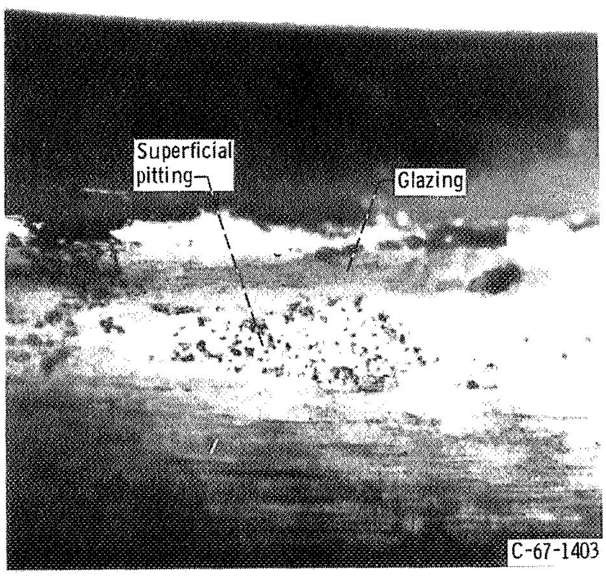


(a) Normal race appearance after being run with full elastohydrodynamic lubrication.



(b) Race appearance after glazing.



(c) Race appearance after glazing and superficial pitting.

Figure 4. - Race appearance representative of bearings run at elevated temperatures.

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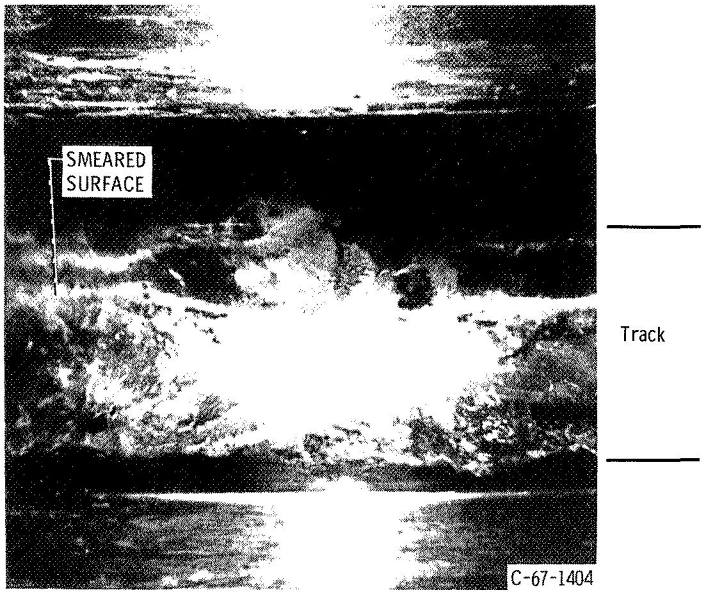

Figure 5. - Representative running track showing smeared surface.

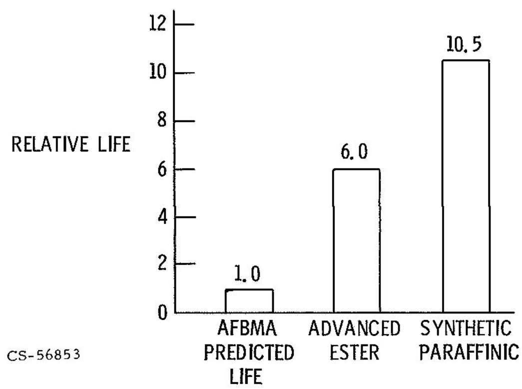

Figure 6. - Bearing fatigue life at 425° F in air.

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Figure 7. - Unfailed 120-mm angular contact ball bearing run for 500 hours at 600° F with synthetic paraffinic oil in a low oxygen environment.

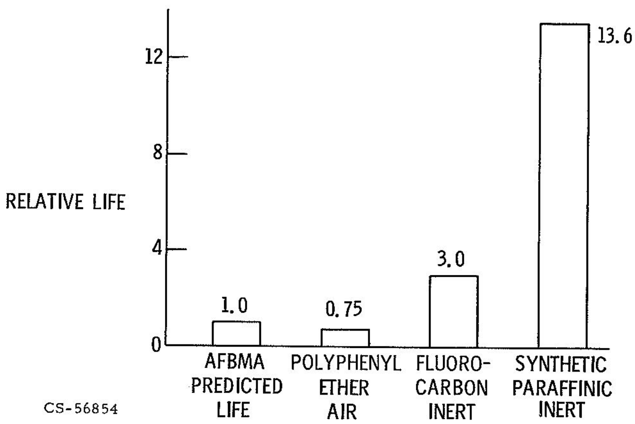

Figure 8. - Bearing fatigue life at 600° F.

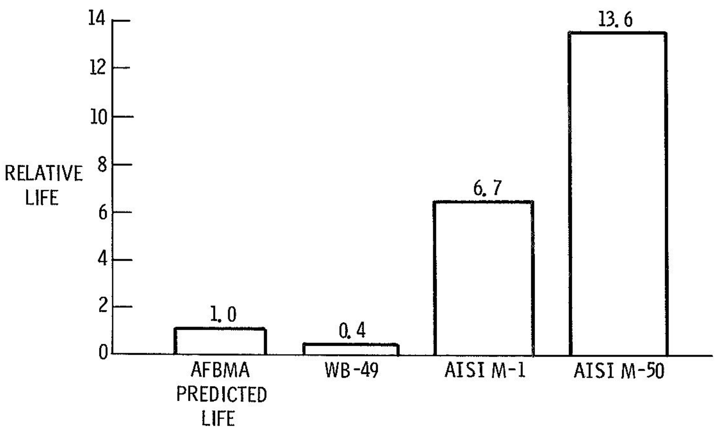

Figure 9. - Bearing life with three steels at 600° F with synthetic paraffinic lubricant in low oxygen environment.

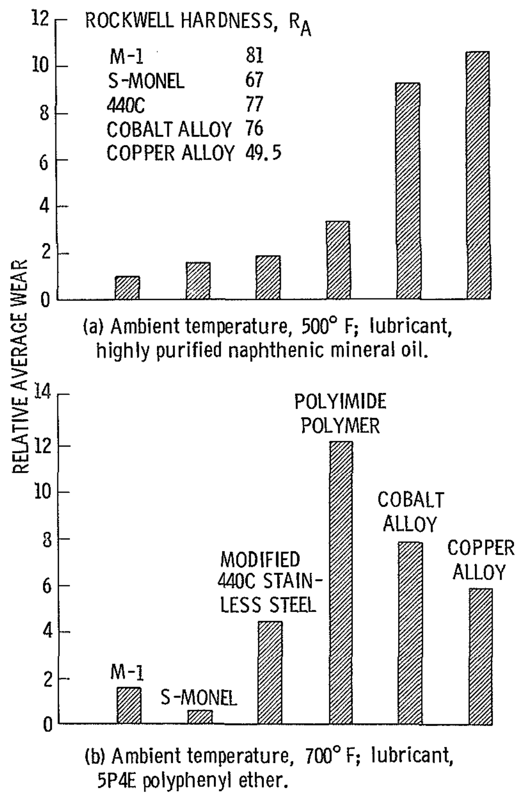

Figure 10. - Wear of various cage materials in an inert environment for 30 minutes duration.

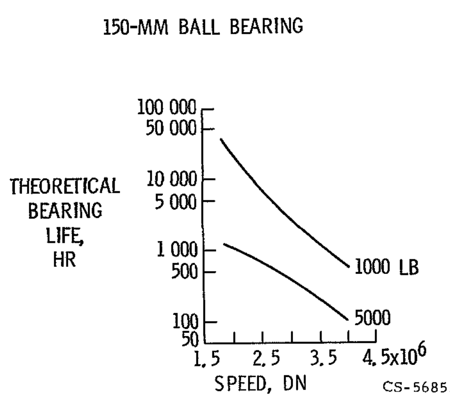

Figure 11. - Effect of speed on bearing life at two thrust loads.

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150-MM BALL BEARING

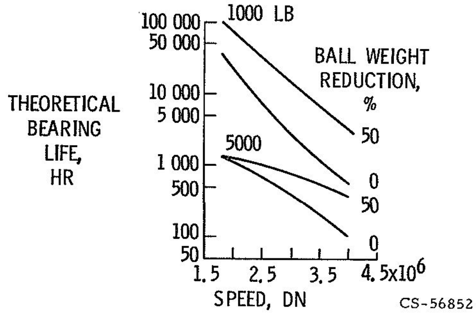

Figure 12. - Effect of a 50-percent ball weight reduction on bearing life at high speed.

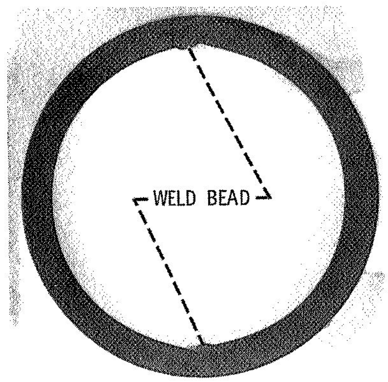

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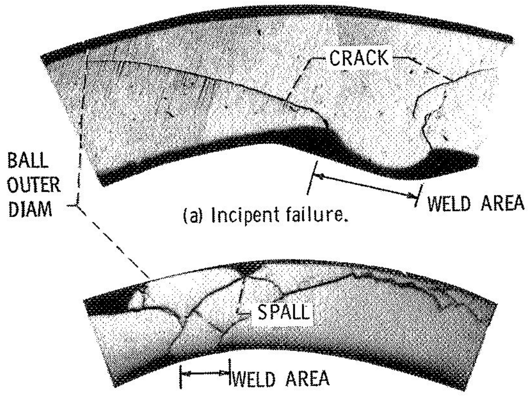
Figure 13. - Cross section of an electron beam welded hollow ball.

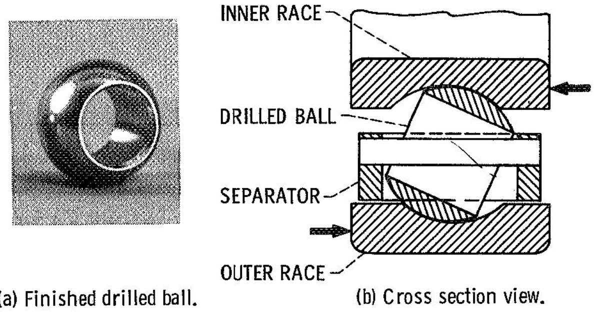


(a) Incipent failure.

CS-56864 (b) Crack network and spall.

Figure 14. - Flexure failures in an electron beam welded hollow ball.

E-6144



(a) Finished drilled ball. (b) Cross section view.

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Figure 15. - Drilled ball bearing, 75-mm bore.

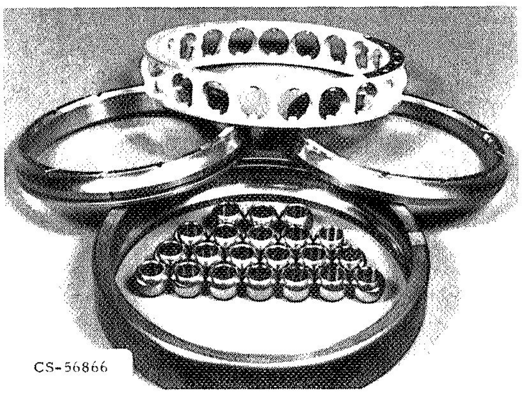

Figure 16. - Drilled ball bearing, 125-mm bore, after 4.7 hours of operation.